srivenkateshwaraa
College of Engineering & Technology
(Approved by AICTE, New Delhi & Affiliated to Pondicherry University, Puducherry)
13-A, Pondy - Villupuram Main Road, Ariyur, Puducherry - 605 102.
ASPIRE TO EXCEL

ISO 21001

## DEPARTMENT OF MECHANICAL ENGINEERING

## SEVENTH SEMESTER

## SUBJECT

# MET-73 Refrigeration, Air Conditioning and Cryogenic Engineering

**PREPARED BY**

**Mr. V. KARTHIKEYAN, M.Tech,**
**ASSISTANT PROFESSOR,**
**Department of Mechanical Engineering**

# UNIT – I

# REFRIGERATION

## INTRODUCTION

Refrigeration is the Process of Producing and maintaining a body temperature lower than that of the surroundings. i.e atmosphere. According to ASREA, it is defined as the science of providing and maintaining temperatures below that of surroundings. It can also be defined as the process of removing heat from a low temperature level and rejecting it at a relatively higher temperature level. It is nothing but removal of heat from the body. In order to maintain the body temperature lower than the surrounding, heat must be transferred from cold body to the surroundings at high temperature. Since heat cannot naturally flow from cold body to hot body, external energy will have to be supplied to perform this operation. The device used for this purpose is called refrigerating system or mechanical refrigerator of simply refrigerator. The system, which is kept at lower temperature, is termed refrigerated system. Nowadays many human workplaces and factories are air conditioned by using refrigeration system. Not only work places but also perishable goods are maintained at their required temperature to store it for a long time by refrigeration system. This device operates on a reversed power cycle. i.e. heat engine. The medium used to carry the heat from a low temperature level to high temperature level is called a “Refrigerant’.

## PRINCIPLE OF REFRIGERATION

Refrigeration is defined as the production of temperature lower than those of the surrounding and maintains the lower temperature within the boundary of a given space. The effect has been accomplished by non cyclic processes such as the melting of ice or sublimation of solid carbon dioxide. However, refrigeration effect is usually produced by transferring heat from a low temperature source to a high temperature source by spending mechanical work. To produce this effect requires certain machinery; hence, the method is called mechanical refrigeration. The working media of such machines are called refrigerants.

## METHODS OF REFRIGERATION

The refrigeration effect may be produced by bringing the substance to be cooled in direct or indirect contact with cooling medium such as ice. The common methods of refrigeration are as follows.

### 1. Art of Ice making by Nocturnal Cooling

The ice refrigeration systems consist of an insulated cabinet equipped with a tray or tank at the top, for holding blocks of ice pieces. Shelves for food are located below the ice compartment. Cold air flows downward from ice compartment and cools the food on the shelves below. Air returns from the bottom of the cabinet up, the sides and back of the cabinet which is warmer, flows over the ice, and again flows down over the shelves to be cooled.

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

#### *Direct method and indirect method*

a. ***Direct Method*:** In the Direct method, the ice is kept in the same compartment with the articles to be cooled . Ice is placed at elevated position and cools the surrounding air. Cold air due to high density descends below displacing warmer air towards the ice. In the storage space cold air is warmed by absorption of heat from the products to be cooled. And rises through the storage space due to light density. It returns to the ice space. A circulation of air is maintained so long as air cooled on ice space and the cycle is repeated. Direct method of ice refrigeration is shown in fig. This method is used in hotels for cooling the drinks.

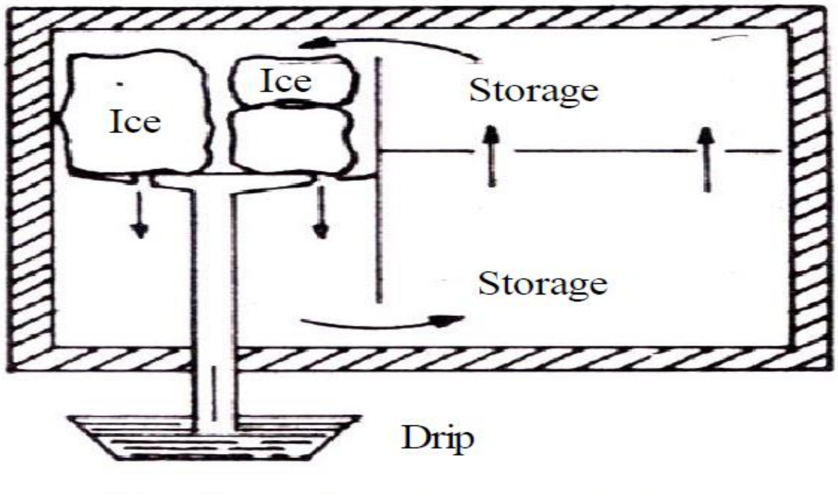


**Figure 1** Ice Refrigeration (Direct Method)

### *b. Indirect Method*

In the indirect method, the ice is isolated from the storage space and cooling of the articles is carried by circulation of brine. The ice is supplied to compartment D where it surrounds the vertical tubes containing brine.

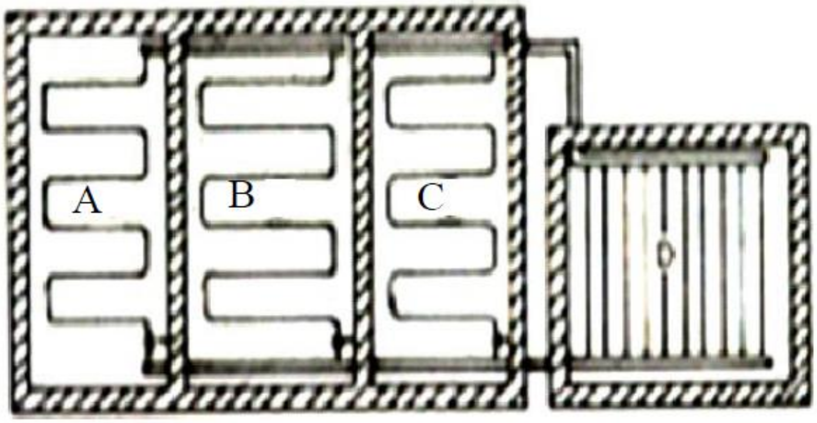


**Figure 2** Ice Refrigeration (In Direct Method)

Cold brine is circulated through the storage space A,B and C to cool the articles, and finally the brine returns to ice compartment D. This cycle is repeated. This system is more suitable for large plants and no danger of food infection by impure ice.

### 3. Dry Ice Refrigeration

Dry ice is the solidified form of CO2. It evaporates directly from solid to vapour without liquid phase. This phenomenon is called sublimation. In this, the dry ice in the form of flakes or slabs Is placed on the cartons containing food stuffs. When dry ice sublimates, it will absorb heat from food stuff in the cartons and thus keeps them in a frozen condition. This is used for preservation of frozen foods and ice creams in storage and transportation. It has twice the heat absorbing capacity of ice refrigeration. However its cost is high.

### 4. Evaporative Cooling

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

### 5. Gas Throttling Refrigeration

In the gas throttling refrigeration process, there is no change in enthalpy and also for a perfect gas, there is no change in temperature. However, for actual gases there is a substantial change, usually a decrease in temperature. This temperature drop depends upon the Joule- Thomas coefficient, the pressure drop, and the initial state of the gas. High pressure gas is throttled through porous plug into the space to be cooled and escapes outside after absorbing heat from the space. This produces the refrigeration effect in the space or the product kept in the space.

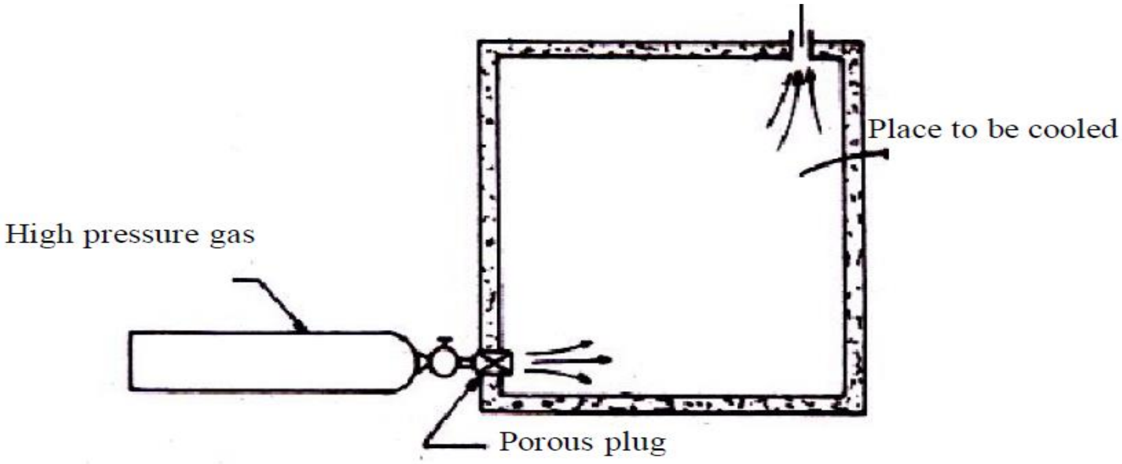


**Figure 3** Gas Throttling Refrigeration

## UNITS OF REFRIGERATION

Quantity of heat removed from the body per unit time is very important in the design of refrigerating system. The amount of heat removed is expressed in ton of refrigeration (TR). This is also called capacity of refrigerating system. It is defined as the quantity of heat removed in of heat is to be removed from one ton of water at 0˚C to freeze it to ice at 0˚C in one day, i.e. 24 hours.

## COEFFICIENT OF PERFORMANCE

Performance of a heat engine is measured by a term known as efficiency. Similarly, the performance of a refrigeration system is measured by a term known as coefficient of performance (COP). It is defined as the ratio of desired effect (quantity of heat removed from a cold body) to the net work done on the system.

A part of heat supplied to the engine ($Q_s$) is converted into useful work ($W_E$) and the remaining part of the heat is rejected ($Q_R$) to the surrounding (sink). According to the 1st law of thermodynamics,

$$W_E = Q_S - Q_R$$

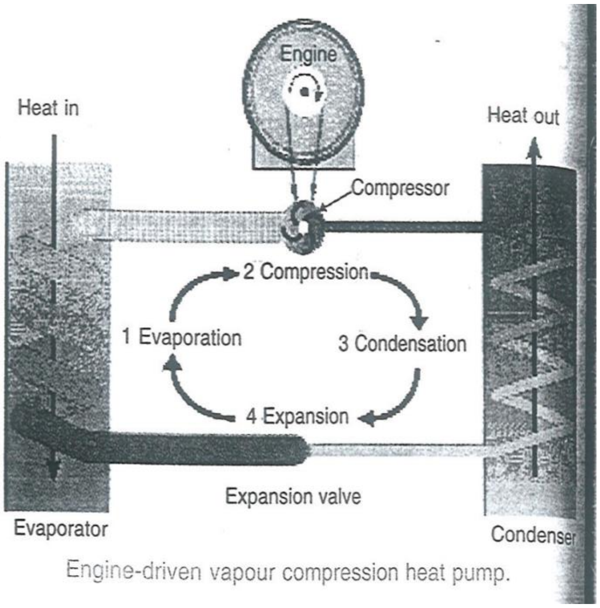
Efficiency of heat engine, $\eta_E = \dfrac{NETworkdone}{Heat\,\mathrm{sup}\,plied} = \dfrac{W_E}{Q_S} = \dfrac{Q_S - Q_R}{Q_S} = 1 - \dfrac{Q_R}{Q_S}$

## SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEMS

A vapors compression refrigeration system is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant, is used. It condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The refrigerants, usually, used for this purpose are ammonia (NH), carbon dioxide ($CO_2$) and sulphur dioxide ($SO_2$). The refrigerant used, does not leave the system, but is circulated

---

- Since low pressure vapor refrigerant from the evaporator is changed into high pressure vapor refrigerant in the compressor, therefore it is named as vapor compression refrigeration system throughout the system alternately condensing and evaporating, the refrigerant absorbs its latent heat from the brine* (salt water ) which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapors compression refrigeration system is therefore a latent heat from the brine and delivers it to the cooler.



Engine-driven vapour compression heat pump.

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

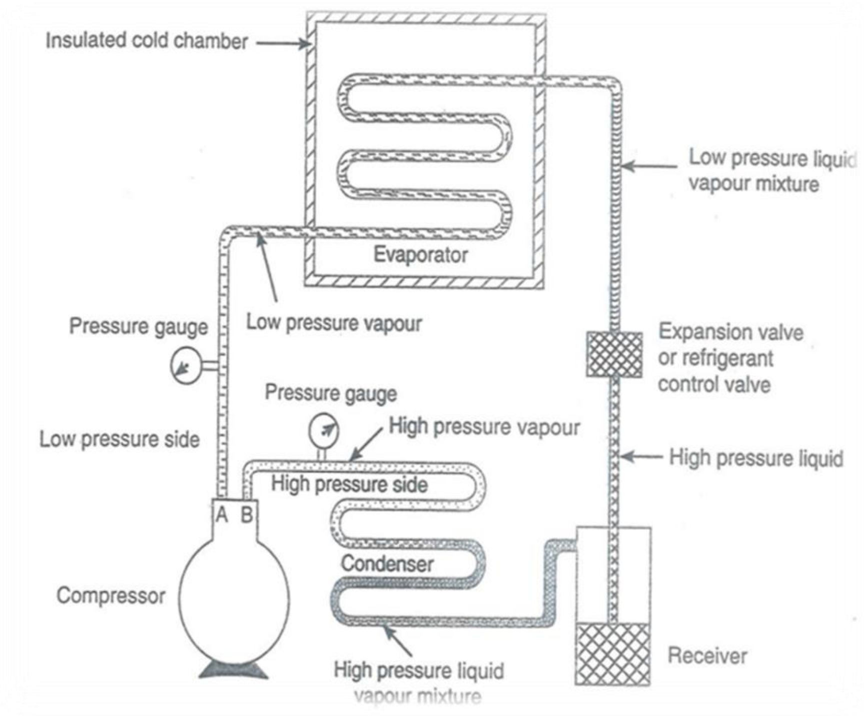


**Figure 4** simple Vapour compression refrigeration systems

1. **Compressor**. The low pressure and temperature vapor refrigerant from evaporation is drowning into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. The high pressure and temperature vapor refrigerant is discharged into the condenser through the delivery or discharge valve B.
2. **Condenser.** The condenser or cooler consist of coils of pipe in which the high pressure and temperature vapor refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.
3. **Receiver.** The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver form where it is supplied to the evaporator through the expansion valve or refrigerant control valve.
4. **Expansion valve.** It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the grater potion is vaporized in the evaporator at the low pressure and temperature.

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

## PRESSURE-ENTHALPY (P-H) CHART

The most convenient chart for studding the behavior of a refrigerant is the p-h char, in which the vertical ordinates represent pressure and horizontal ordinates represent enthalpy (i.e. total heat). A typical chart is shown in Fig. 5, in which a few important lines of the complete chart are drowned. The saturated liquids line and situated vapor line merge into one another at the critical point. A saturated liquid is one which has a temperature equal to the saturation temperature corresponding to its pressure. The space to the left of the saturated liquid line will; therefore, be sub-cooled liquids region. The space between the liquids and the vapor lines is called wet vapor region and to the right of the saturated vapor line is a superheated vapor region

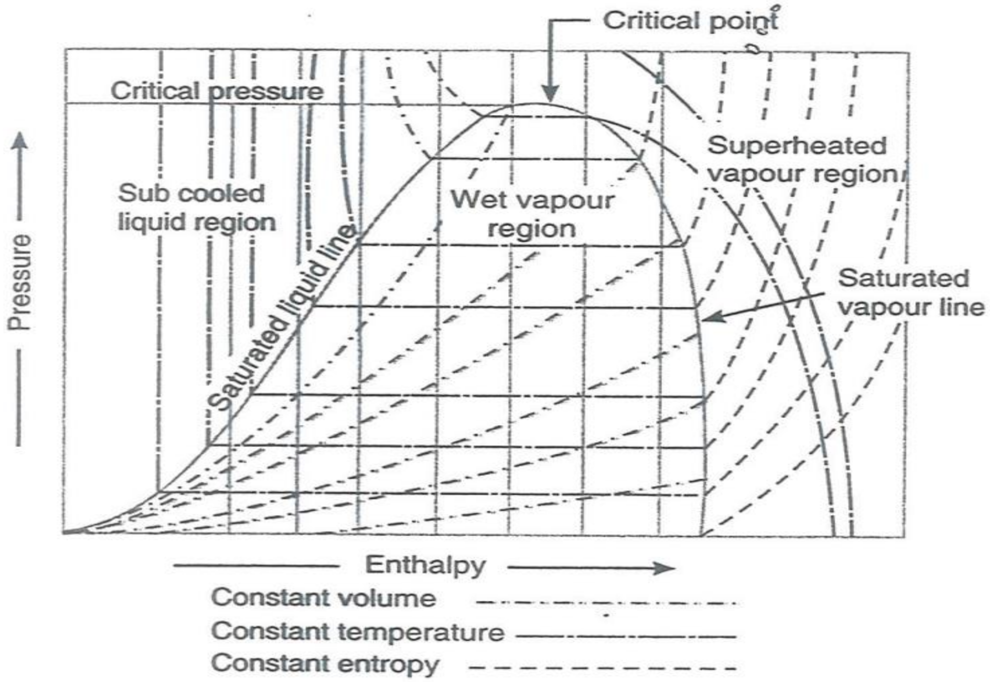


**Figure 5** Pressure –enthalpy (p-h) Chart

## TYPES OF VAPOR COMPRESSION CYCLES

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

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

### 1. Theoretical Vapor Compression Cycle with Dry Saturated vapor after compression

A vapour compressing cycle with dry started vapour after compressions is shown on T-s and p-h diagrams in Fig 6. (a) and (b) respectively. At point 1,let $T_1$ $p_1$ and $s_1$ be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycles are as follows:

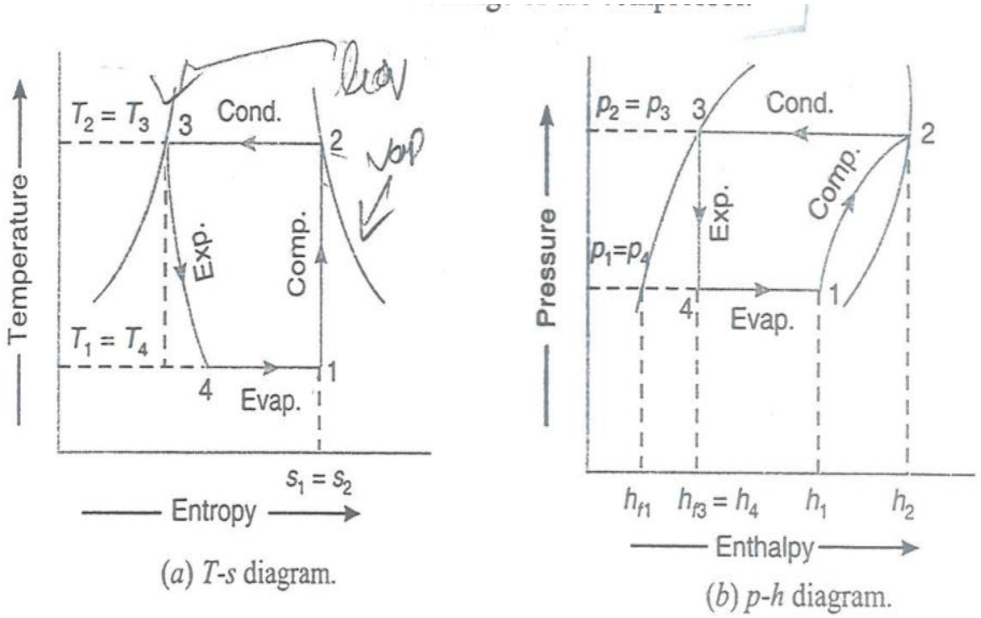


**Figure 6** Theoretical vapour compression cycle with dry saturated vapour after compression

1. Compassion process. The vapour reliant at low pressure $p_1$ and temperature $T_1$ is compressed is entropic ally to dry saturated vapour as shown by the vertical line 1-2 on$T_1$ is and by the cove 1-12 on p-h diagram. The pressure and temperature rise from $p_1$ to $p_1$ and $T_1$ to $T_2$ respectively.

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

    $$W = h_2 - h_1$$

    $h_1$ = Enthalpy of vapour refrigerant at temperature $T_1$

    $h_2$=Enthalpy of the vapor refrigerant at temperature $T_2$

2. **Condensing process.** The high pressure and temperature vapor refrigerant from the compressor is passed through the condensed $T_{2'}$ as shown by the horizontal line 2-3 on $T\text{-}_s$ and p-h diagrams. The vapor refrigerant is changed into liquied refrigerant. The refrigerant, where passing through the condenser, given its latent heat to the surrounding condensing medium.
3. **Expansion process.** The liquid refrigerant at pressure $p_3=p_2$ and temperature $T_3=T_2$ is expanded by * throttling process through the expansion valve to a low pressure $p_4=p_1$ and temperature $T_4=T_{1'}$ as shown by the curve 3-4 on $T\text{-}_s$ diagram and by the vertical line 3-4 on through the expansion value, but the greater portion is vapored in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

**4. Vaporizing process.** The liquid-vapour mixture of the refrigerant at pressure $p_4=p_1$ and temperature $T_4$=T1is evaporated and changed into vapor refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporates 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-vapor refrigerant during evaporation per kg of refrigeration is given by

$$R_E = h_1 - h_4 = h_1 - h_{f3} \,.....(\because h_{f3} = h_4)$$

Where $h_{f3}$ = Sensible heat at temperature $T_{3'}$i.e., enthalpy of liquid

Refrigerant leaving the condenser.

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

$\therefore$ Coefficient of performance,

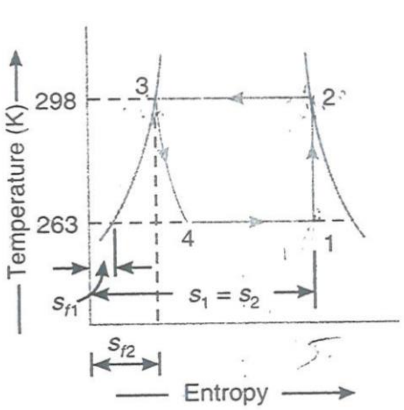
$$c.o.p = \frac{refigeratingeffect}{Workdone} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**PROBLEMS**

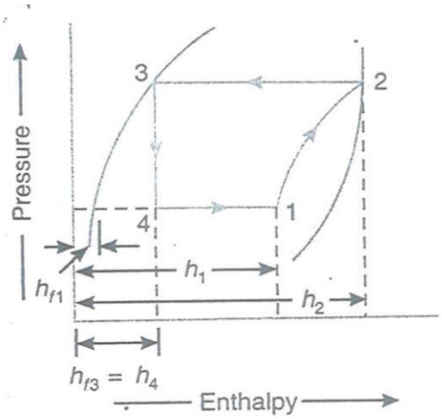
**1. The temperature limits of an ammonia refrigerating system are 20ºC and -10ºC. If the gas is dry at the end of compression, calculate the coefficient of performance of the cycle assuming no under cooling of the liquid ammonia. Use the following table for properties of ammonia:**

| **Temperature (ºC)** | **Liquid heat (kJ/kg)** | **Latent heat (kJ/kg)** | **Liquid entropy (KJ/Kg K)** |
|---|---|---|---|
| **25** | **298.9** | **1166.94** | **1.1242** |
| **-10** | **135.37** | **1297.68** | **0.5443** |

**Solution.** Given: $T_2=T_3=25ºC=25+273=298$ K; $T_1=T_4=$ -10ºC =-10+273=263 K; $h_3=h_4=298.9$ KJ/Kg; $h_{fg2}=1166.94$ KJ/Kg; $S_{f2}=1.1242$ Kj/Kg K; $h_{f1}$ =135.37KJ/Kg ;$h_{fg1}=1297.68$ KJ/Kg;$S_{f1}=0.5443$ KJ/KgK



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



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

The T-Sand p-h diagram is shown in above figure respectively.

Let $x_1$=Dryness fraction at point 1.

We know that entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$$= 0.5443 + 4.934 x_1 \qquad \text{….. (i)}$$

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{h_{fg2}}{T_2} = 1.1242 + \frac{1166.94}{298} = 5.04 \quad \text{…..(ii)}$$

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$$0.5443 + 4.934 x_1 = 5.04 or x_1 = 0.91$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x1 h_{fg1} = 135.37 + 0.91 \times 1297.68 = 1316.26 KJ / Kg$$

And enthalpy at point 2, $h_2 = h_{f2} + h_{fg2} = 298.9 + 1166.94 = 1465.84 KJ / Kg$

$\therefore$ Coefficient of performance of the cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26} = 6.8 Ans.$$

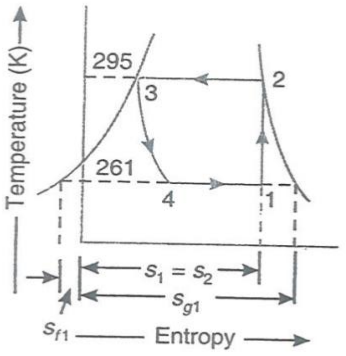
**2. A vapor compression refrigerant work between the pressure limits of 60 bars and 25 bars. The working fluid is just dry at the end of compression and there is no under cooling of the liquid before the expansion valve. Determine: 1. C.O.P of the cycle; and 2. Capacity of the refrigerant if the fluid flow is at the rate of 5Kg/min.**

| **Pressure (bar)** | **Saturation temperature(K)** | **Enthalpy (KJ/Kg)** | | **Entropy (KJ/KgK)** | |
|---|---|---|---|---|---|
| | | **Liquid** | **Vapor** | **Liquid** | **Vapor** |
| **60** | **295** | **151.96** | **293.29** | **0.554** | **1.0332** |
| **25** | **261** | **56.32** | **322.58** | **0.226** | **1.2464** |

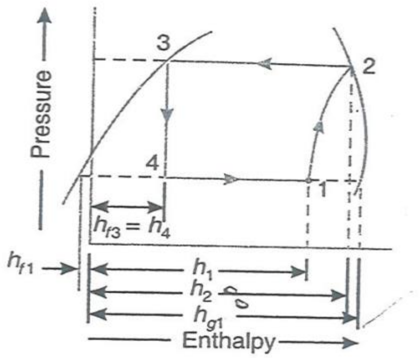
**Solution. Given: $P_2=p_3=60$ bar; $p_1=p_4=25$ bar; $T_2=T_3=295$ K$T_1=T_4$261K;$h_{f3}=h_4=151.96$ KJ/Kg K;$S_{f1}=0.226$ KJ/KgK; $S_{g2}=S_2=1.0332$ KJ/Kg K; $S_{g1}=1.2464$ KJ/KgK**

1. **C.O.P. of the cycle**

The T-s and p-h diagrams are shown in Fig 4.5 (a) and (b respectively.



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



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

Let $x_1$=Dryness fraction of the vapor refrigerant entering the compressor at point 1.

We know that entropy at point 1.

$$s_1 = s_{f1} + x_1 s_{fg1} = s_{f1} + x_1(s_{g1} - s_{f1}).....(\because s_{g1} = s_{f1} + s_{fg1})$$

$$= 0.226 + x_1(1.2464 - 0.226) = 0.226 + 1.0204x_1....(i)$$

And entropy at point 2, $s_2 = s_{g2} = 1.0332 KJ / KgK......(Given)...(ii)$

Since the entropy at point 1 is equal to entropy at point 2, therefore equaling equations (i) and (ii)

$$0.226+1.0204X_1=1.0332 \qquad \text{or} \quad x_1=0.791$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = h_{f1} + x_1(h_{g1} - h_{f1})....(\because h_{g1} = g_{f1} + h_{fg1})$$

$$=56.32+0.791(322.58-56.32)=266.93\text{KJ/Kg}$$

$\therefore$ C.O.P of the cycle

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.36 Ans.$$

2. **Capacity of the refrigerator**

We know that the heat extracted or refrigerating effect produced per kg of refrigerant

$$= h_1 - h_{f3} = 266.93 - 151.96 = 114.97 KJ / Kg$$

Since the fluid flow is at the rate of 5kg/min, therefore total heat extracted

$$= 5 \times 114.97 = 574.85 kj / kg$$

$\therefore$ Capacity of the refrigerator

$$= \frac{574.85}{210} = 2.74 TRAns. \text{…… } (\therefore \text{ 1TR=210kj/min})$$

3. **28 tons of ice from and at 0ºCis produced per day in an ammonia refrigerator. The temperature range in the compressor is from 25ºC to -15ºC. The vapor is dry and saturated at the end of compression and an expansion valve is used. There is no liquid sub cooling. Assuming actual C.O.P of 62% of the theoretical. Calculate the power required to drive the compressor. Following properties of ammonia are given:**

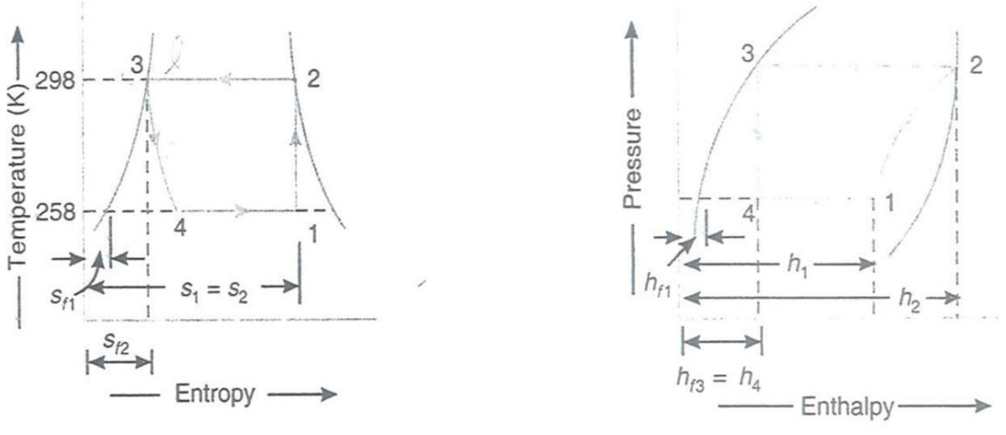
| Temperature<br>0°C | Enthalpy (kJ/Kg) | | Entropy (kj/kg K) | |
|---|---|---|---|---|
| | Liquid | Vapor | Liquid | Vapor |
| 25<br>-15 | 298.9<br>112.34 | 1465.84<br>1426.54 | 1.1242<br>0.4572 | 5.0391<br>5.5490 |

**Take latent heat of ice =335 kJ/kg.**

**Solution.** Given: Ice produced =28t/day; $T_2$=$T_3$=25ºC =25+273=298K; $T_1$=$T_4$=-15ºC=-15+273=258K; $h_{f3}$=$h_{4=}$298.9 kJ/kg ; $h_{f1}$=112.34 kJ/kg; $h_{g2}$=$h_2$=1465.84 kj/kg; $h_{g1}$=1426.54kj/kg; *$s_{f2}$=1.1242kj/kg K; $S_{f1}$=0.4572 kj/kg K; $S_{g2}$=$s_2$=5.0391 kJ/kg; $s_{g1}$=5.5490 kJ/kg K.

The T-s and p-h diagrams are shown in below figure respectively.

First of all let us find the dryness fraction ($x_1$) of the vapor refrigerant entering the compressor at point 1.



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

We know that entropy at point 1,

$$s_1 = s_{f1} + s_{fg1} = s_{fg1} = x_1(s_{g1} - s_{f1}) \qquad \ldots(\therefore s_{g1} = s_{f1} + s_{fg1})$$

$$=0.4572+ x_1 (5.5490\text{-}0.4572)$$

$$= 0.4572+5.0918\, x_1 \qquad (kj / kgK.....(Given)....(ii)$$

And entropy at point 2,

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$$0.4572+5.0918\,x_1=5.0391 \quad \text{or} \quad x_1=0.9$$

We know that enthalpy at point 1,

$$h_1=h_{f1}+x_1h_{fg1}=h_{f1}+x_1(h_{g1}-h_{f1}) \;\ldots.(\because h_{g1}=h_{f1}+h_{fg1})$$

=112.34+0.9(1426.54-112.34)=1295.12 kJ/kg

$\therefore$ Theoretical C.O.P = $\dfrac{h_1-h_{f3}}{h_2-h_1}=\dfrac{1295.12-298.9}{1465.84-1295.12}=\dfrac{996.22}{170.72}==5.835$

Since actual C.O.P is 62%of theoretical C.O.P therefore

Actual C.O.P= $=0.62\times 5.835=3.618$

We know that ice produced from and at 0ºC

$$=28t/day=\frac{28\times 1000}{24\times 3600}=0.324kg/s$$

Latent heat of ice = 335KJ/kg

$\therefore$ Refrigeration effect produced

$$=0.324\times 335=108.54kj/s$$

We know that actual C.O.P

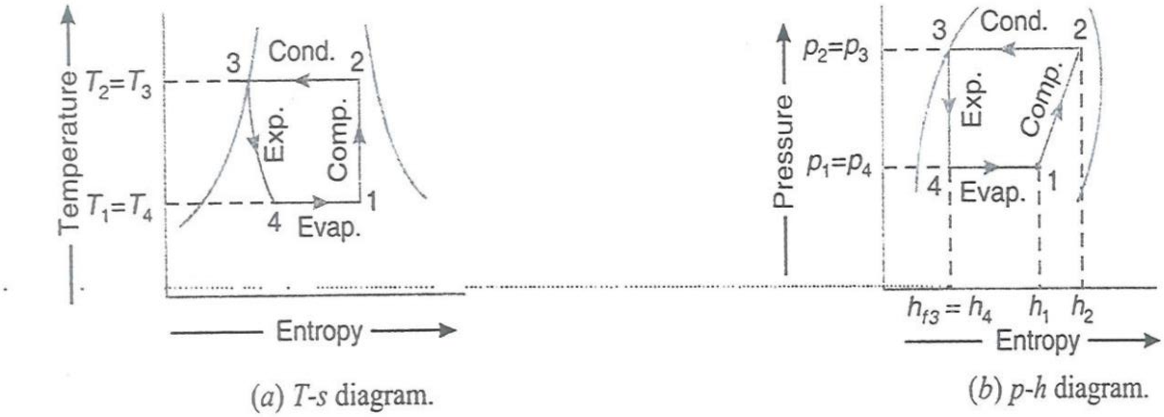
$$3.618=\frac{refrigerationeffect}{Workdone}=\frac{108.54}{Workdone}$$

$\therefore$ Work done or power required driving the compressor

$$=\frac{108.54}{3.618}=30\,\text{kJ/s} \quad \text{or KW ans.}$$

## 2. Theoretical Vapor Compression cycle with wet Vapor after compression

A vapor compression cycle with wet vapor after compression is shown on T-s and p-h diagrams I Fig 7 (a) and (b) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at point 1 and 2.



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

**Figure 7** Theoretical vapour compression cycle with wet vapour after compression

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

$$C.O.P = \frac{refrigeratingeffect}{workdine} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**4.** Find the theoretical C.O.P for a $CO_2$machine working between the temperature range of 25ºC and -5ºC. The dryness fraction of $CO_2$ gas during the suction stroke is 0.6. Following properties of $CO_2$ are given:

| **Temperature ºC** | **Liquid** | | **Vapor** | | **Latent heat KJ/Kg** |
|---|---|---|---|---|---|
| | **Enthalpy KJ/Kg K** | **Entropy KJ/Kg K** | **Enthalpy KJ/Kg** | **Entropy KJ/Kg K** | |
| 25 | 164.77 | 0.5978 | 282.23 | 0.9918 | 117.46 |
| -5 | 72.57 | 0.2862 | 321.33 | 1.2146 | 248.76 |

**Solution.** Given $T_2=T_3=25ºC = 25+273=298$ K; $T_1=T_4= -5ºC =-5+273=268$ K; $x_1=0.6$;$h_{f3}$ $h_{f2}=164.77$ kJ/kg; $h_1=h_{f4}=72.57$ kJ/kg ; $S_{f2}=0.5978$ kJ/kg K;$s_{f1}=0.2862$ kJ/kg K; $h_{2'}=282.23$ kj/kg; $h_{1'}=321.33$ kj/kg ; *$s_{2'}=0.9918$ kj/kg K; *$s_{1'}=1.2146$ kj/kg $h_{fg2}=117.46$ kj/kg ;$h_{fg1}=248.76$ kj/kg

The T-s and p-h diagram are shown in Figure.

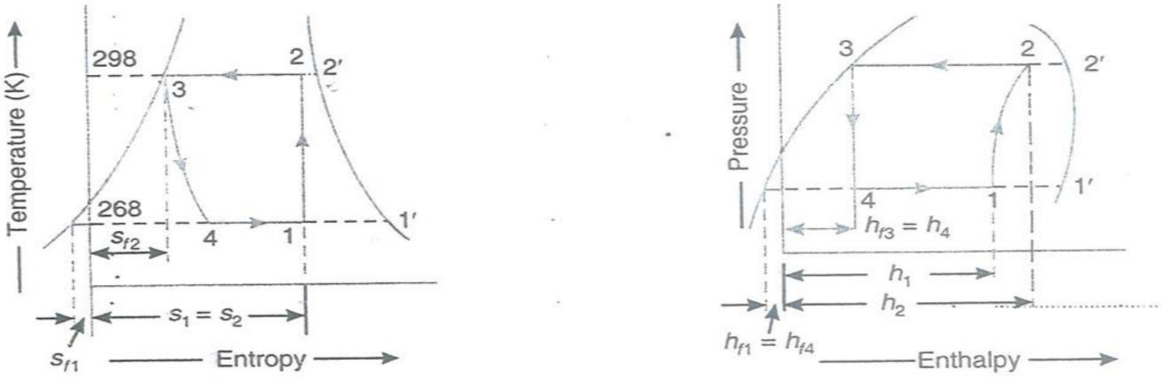
First of all, let us find the dryness fraction at point 2,i.e $x_2$,. We know that th entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{fg1}}{T_1} = 0.2862 + \frac{0.6\times 248.76}{268} = 0.8431....(i)$$

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2 h_{fg2}}{T_2} = 0.5978 + \frac{x_2 \times 117.46}{268}$$

$$=0.5978+0.3941\, x_2 \qquad ....(ii)$$



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

Since the entropy at point 1($s_1$) is equal to trophy at point 2($s_2$), therefore equaling equations (i) and (ii),

$$0.8431=0.5978+0.3941\, x_2 \quad \text{or} \quad x_2 =0.622$$

We knows that enthalpy at point 1,

$$h_1 = h_{f1} + x_1 h_{fg1} = 72.57 + 0.6\times 248.76 = 221.83 kj/kg$$

And enthalpy at point 2, $h_2 = h_{f2} + x_2 h_{fg2} = 164.77 + 0.622 \times 117.76 = 237.83 kj/kg$

$$\therefore \text{Theretical C.O.P.} = = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{221.83 - 164.77}{237.83 - 221.83} = \frac{57.06}{16} = 357\,Ans.$$

**5. An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of -10$^oC$ and 30$^oC$. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the fluid leaving the condenser is at 30$^oC$. Assuming actual C.O.P. as 60% of the theoretical, calculate the kilograms of ice produced per kW hour at 0$^oC$ from water at 10$^oC$. Latent heat of ice is 335 kj/kg. Ammonia has the following properties:**

| Temperature $^oC$ | Liquid heat ($h_1$) kj/kg $^oC$ | Latent heat ($h_{f8}$) kj/kg | Liquid entropy ($S_f$) | Total entropy of dry saturated vapour |
|---|---|---|---|---|
| 30 | 323.08 | 1145.80 | 1.2037 | 4.9842 |
| -10 | 135.37 | 1297.68 | 0.5443 | 5.4770 |

**Solution:** Given: $T_1 = T_4$ = -10$^o$C = -10 + 273 = 263 K: $T_2 = T_3$=30$^o$C+273=303K ; $x_2$ = 0395 ; $h_{f3} = h_{f2}$ = 323.08 kj/kg ; $h_{f1} = h_{f4}$ = 135.37 kl/kg ; $h_{f82}$ = 1145.8 kj/kg ; $h_{f81}$ = 1297.68 kj/kg, $s_{f2}$ = 1.2037 ; $s_{f1}$ = 0.5443 ‘ $*s_{2'}$ = 4.9842 , $*s_{1'}$ = 5.4770

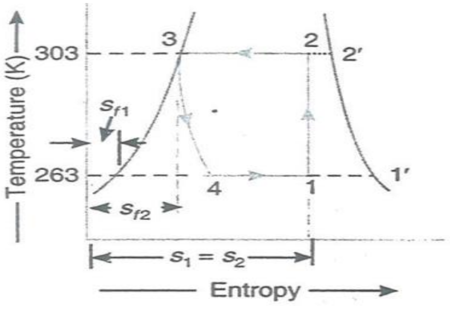
The T-s and p-h diagrams are shown in Figure respectively.

Let $x_1$ = Dryness fraction at point 1.

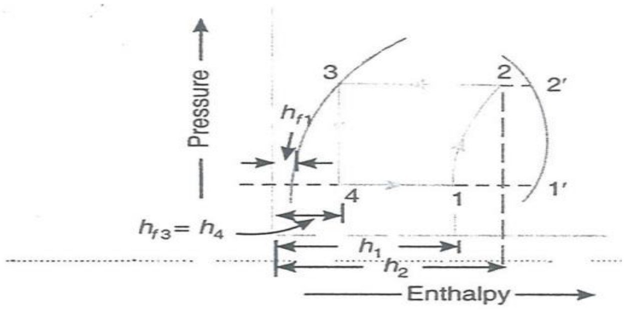
We know that entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1 h_{f81}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$$= 0.5443 + 4.934\, x_1 \qquad \text{..... (i)}$$



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



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

Similarly, entropy at point 2,

$$s_1 = s_{f2} + \frac{x_2 h_{f8^2}}{T_2} = 1.2037 + \frac{0.95 \times 1145.8}{303} = 4.796 \qquad \text{...(ii)}$$

Since the entropy at point 1 ($s_1$) is equal to entropy at point 2 ($s_2$), therefore equating equations (i) and (ii)

$0.5443 + 4.934\, x_1 = 4.796$ or x1 = 0.86

∴ Enthalpy at point 1, $h_1 = h_{f1}$ + x1 $h_{fg1}$ = 135.37 + 0.86 x 1297.68 = 1251.4 kj/kg

and enthalpy at point2, h2 = $h_{f2}$ + x2 $h_{f2}$ = 323.08 + 0.95 x 1145.8 = 1411.6 kj/kg

We know that theoretical C.O.P.

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1251.4 - 323.08}{1411.6 - 1251.4} = 5.8$$

∴ Actual C.O.P. = 0.6x5.8=3.48

Work to be spent corresponding to 1kW hour,

W = 3600 kJ

∴ Actual heat extracted or refrigeration effect produced per kW hour

=W x Actual C.O.P. = 3600 x 3.48 = 12 528 kJ

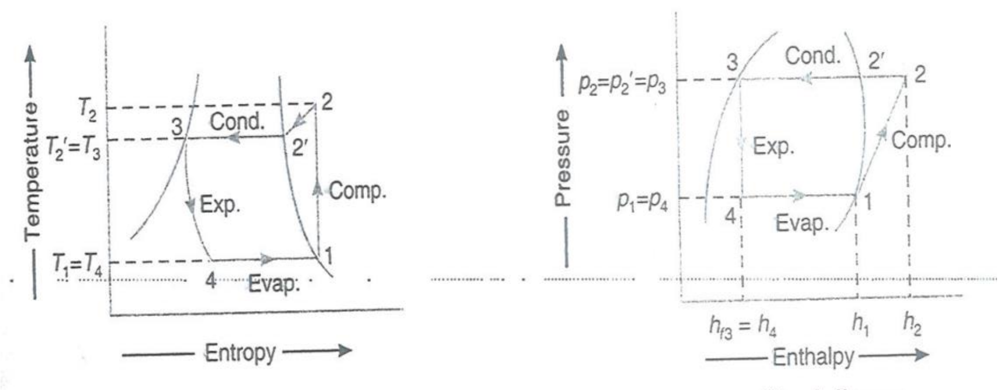
We know that heat extracted from 1kg or water at 10ºC for the formation of 1kg of ice at 0ºC

= 1x4 4.187 x 10 + 335 = 376.87 kJ

∴ Amount of ice produced

$$= \frac{12528}{376.87} = 33.2 kg / kWhourAns.$$

### 3. Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(a) T-s diagram.

(b) p-h diagram.

**Figure 8** Theoretical vapour compression cycle with Superheated after compression

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

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

$$C.O.P. = \frac{\mathrm{Re}\, frigeratingeffect}{workdone} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

A little consideration will show that the superheating increases the refrigerating effect and the amount of work done in the compressor. Since the increase in refrigerating effect is less as compared to the increase in work done, therefore, the net effect of superheating is to have low coefficient of performance.

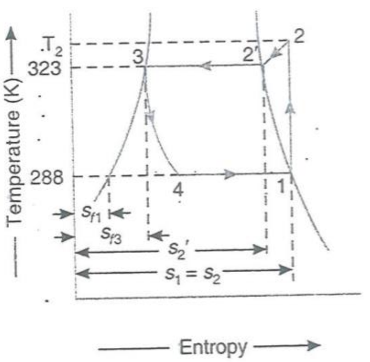
**6. A simple refrigerant 134a (tetraflurocthane) heat pump for space hearing, operates between temperature limits of 15$^{o}$C and 50$^{o}$C. The heat required to be pumped is 100MJ/h. Determine: 1. The dryness fraction of refrigerant entering the evaporator; 2.The discharge temperature assuming the specific heat of vapour as 0.996 kJ/kg K; 3. The theoretical piston displacement of the compressor; 4. The theoretical power of the compressor; and 5. The C.O.P.**

**The specific volume of refrigerant 134a saturated vapour at 15$^{o}$C is 0.04185 m3/kg. The other relevant properties of R-134a are given below:**

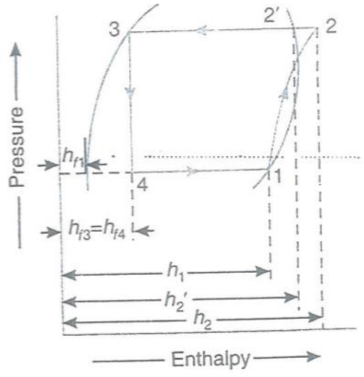
| **Saturation temperature (°C)** | **Pressure (bar)** | **Specific enthalpy (kJ/kg)** | | **Specific entopy (kJ/kg K)** | |
|---|---|---|---|---|---|
| | | **Liquid** | **Vapour** | **Liquid** | **Vapour** |
| 15 | 4.887 | 220.26 | 413.6 | 1.0729 | 1.7439 |
| 50 | 13.18 | 271.97 | 430.4 | 1.2410 | 1.7310 |

**Solution:** Given: $T1 = T4 = 15^oC = 15 + 273 = 288\ K$ ; $T2 = T3 = 50^oC = 50 + 273 = 323\ K$ ; $Q = 100\ MJ/h = 100 \times 10^3 kJ/h$; $c_p = 0.996\ kJ/kg\ K$ ; $v_1 = 0.04185\ m^3/kg$; $h_{f1} = 220.26\ kJ/kg$ ; $h_{f3} = h_4 = 271.97\ kJ/kg$ ; $h_1 = 413.6\ kJ/kg$ ; $h_2 = 430.4\ kJ/kg$ ; $s_{f1} = 1.0729\ kJ/kg\ K$ ; $s_1 = s_2 = 1.7439\ kJ/kg\ K$ ; $s_{f3} = 1.2410\ kJ/kg\ K$ ; $s_2 = 1.7312\ kJ/kg\ K$

The *T*-*s* and *p*-*h* diagrams are shown in Figure



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



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

***1. Dryness fraction of refrigerant entering the evaporator***

We know that dryness fraction of refrigerant entering the evaporator *i.e.* at point 4,

$$x_4 = \frac{h_4 - h_{f1}}{h_1 - h_{f1}} = \frac{271.97 - 220.26}{413.6 - 220.26} = \frac{51.71}{193.34} = 0.2675\, Ans.$$

***2. Discharge temperature***

Let $T_2$= Discharge temperature.

We know that entropy at discharge *i.e.* at point 2,

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

$$1.7439 = 1.7312 + 2.3 \times 0.996 \log\left(\frac{T_2}{T_{2'}}\right)$$

$$\log\left(\frac{T2}{T2'}\right)=\frac{1.7436-1.7312}{2.3\times 0.996}=0.00554$$

$$\frac{T_2}{T_{2'}}=1.0128 \qquad \text{...(Taking antilog of 0.005 54)}$$

$$T2=T2'\times 1.0128=323\times 1.0128=327.13K=54.13^{\circ}C Ans.$$

***3. Theoretical piston displacement of the compressor***

We know that enthalpy at discharge *i.e.* at point 2,

$$h_2=h_{2'}+c_p(T_2-T_{2'})$$

$$=430.4+0.996\,(327.13-323)=434.5\ kJ/kg$$

And mass flow rate of the refrigerant,

$$m_R=\frac{Q}{h_2-h_{f3}}=\frac{100\times 10^3}{434.5-271.97}=615.3kg/h= \qquad 10.254\ \text{kg/min}$$

∵ Theoretical piston displacement of the compressor

$=m_R x v_1=10.254 \times 0.4185 = 4.59\ m^3/min$ Ans.

**4. *Theoretical power of the compressor***

We know that work done by the compressor

$=m_R(h_2-h_1) = 10.254\,(434.5 - 413.6) = 214.3$ kJ/min

∵ Power of the compressor = 214.3/60 = 3.57 kJ/s or kW Ans.

**5. *C.O.P.***

We know that $$C.O.P.=\frac{h_1-h_{f3}}{h_2-h_1}=\frac{413.6-271.97}{434.5-413.6}=\frac{141.63}{20.9}6.8Ans.$$

**7. A refrigeration machine using R-12 as refrigerant operates between the pressures 2.5 bar and 9 bar. The compression is isentropic and there is no undercooling in the condenser.**

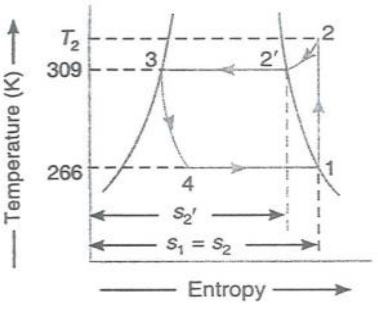
**The vapour is in dry saturated condition at the beginning of the compression Estimate the theoretical coefficient of performance. If the actual coefficient of performance is 0.65 of theoretical value, calculate the net cooling produced per hour. The refrigerant flow is 5 kg per minute. Properties of refrigerant are**

| Pressure, bar | Saturation temperature, C | Enthalpy, kJ/kg | | Entropy of saturated vapour, kj/kg K |
|---|---|---|---|---|
| | | Liquid | Vapour | |
| 9.0 | 36 | 70.55 | 201.8 | 0.6836 |
| 2.5 | -7 | 29.62 | 184.5 | 0.7001 |

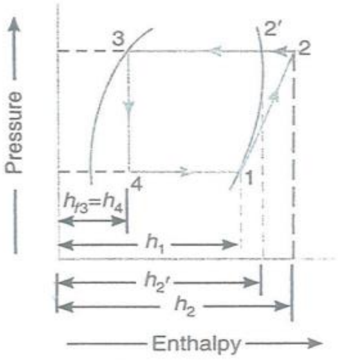
*Take Cp for superheated vapour at 9 bar as 0.64 kj/kg K.*

***Solution.****Given :* $T_{2'} = T_3 = 36C = 36 + 273 = 309$ K ; $T_1 = T_4 = -7C = 7 + 273 = 266$ K l; $(C.O.P.)_{actual} = 0.65\ (C.O.P.)_{th}$ ;*m = 5kg / min ;* $h_{f3} = h_4 = 70.55$kJ/kg ; *$h_{f1} = h_{f4} = 293.62$kJ/kg ; $h_2 = 201.8$ kJ/jkg; $h_1 = 184.5$ kJ/kg; $s_2 = 0.6836$ kJ/kg K; $s_1 = s_2 = 0.7001$ kJ/kg K; $c_p = 0.64$ kJ/jkg K

The *T*-s and *p-h* diagrams are shown in Figure



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



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

***Theoretical coefficient of performance***

First of all, let us find the temperature at point 2 ($T_2$)

We know that entropy at point 2,

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

$$0.7001 = 0.6836 + 2.3 \times 0.64 \log\left(\frac{T_2}{309}\right)$$

$$\log\left(\frac{T_2}{309}\right) = \frac{0.7001 - 0.6836}{2.3 \times 0.64} = 0.0112$$

$$\left(\frac{T_2}{309}\right) = 1.026$$ ...(Taking antilog of 0.0112)

∵ $T_2 = 1.026 \times 309 = 317$ K

We know that enthalpy of superheated vapour at point 2,

$$h_2 = h_{2'} + c_p(T_2 - T_{2'})$$

$$=201.8 + 0.64\ (317-309) = 206.92\ \text{kJ/kg.}$$

∵ Theoretical coefficient of performance,

$$(C.O.P.)_{th} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1845 - 7055}{206.92 - 184.5} = 5.1 Ans.$$

*Net cooling produced per hour*

We also know that actual C.O.P. of the machine.

$$(C.O.P)_{actual} = 0.65 \times (C.O.P.)_{th} = 0.65 \times 5.1 = 3.315$$

and actual work done, $w_{actual} = h_2 - h_1 = 206.92 - 184.5 = 22.42$ kJ/kg

We know that net cooling (or refrigerating effect) produced per kg of refrigerant

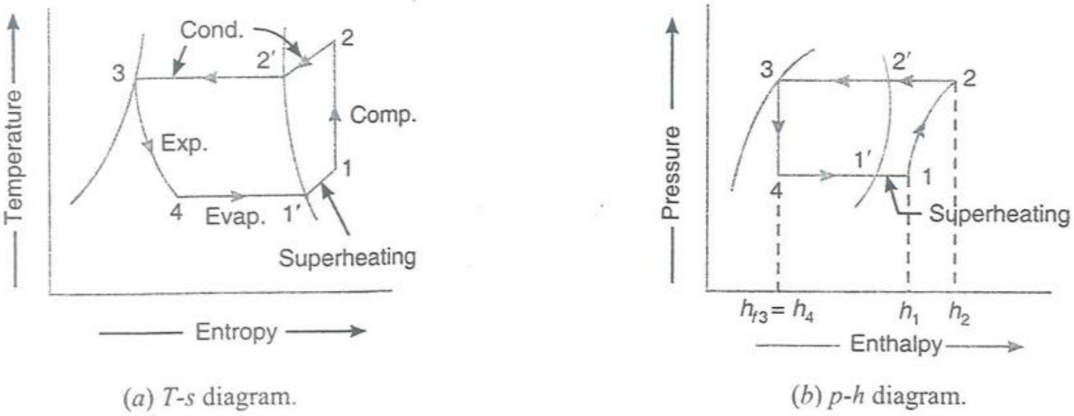
$$= w_{actual} = \times (C.O.P.)_{actual} = 22.42 \times 3.315 = 74.3\ \text{kJ/kg}$$

∵ Net cooling produced per hour

$$= m \times 74.3 = 5 \times 74.3 = 371.5\ \text{kJ.min}$$

$$= \frac{375.5}{210} = 1.77 TR Ans. \qquad ...(\because 1\ \text{TR} = 210\ \text{kJ/min})$$

### 4. Theoretical Vapour Compression Cycle with Superheated Vapour before compression



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

**Figure 9** Theoretical vapour compression cycle with Superheated before compression

A Vapour compression cycle with superheated vapour before compression is shown on T-s and p-h diagrams in Fig. 9 (a) and (b) respectively. In this cycle, the evaporation starts at point 4 and continues upto point 1' ,when it is dry saturated. The vapour is now superheated before entering the compressor upto the point 1.

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

$$\text{C.O.P} = \frac{\text{Re } frigeratingeffect}{workdone} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

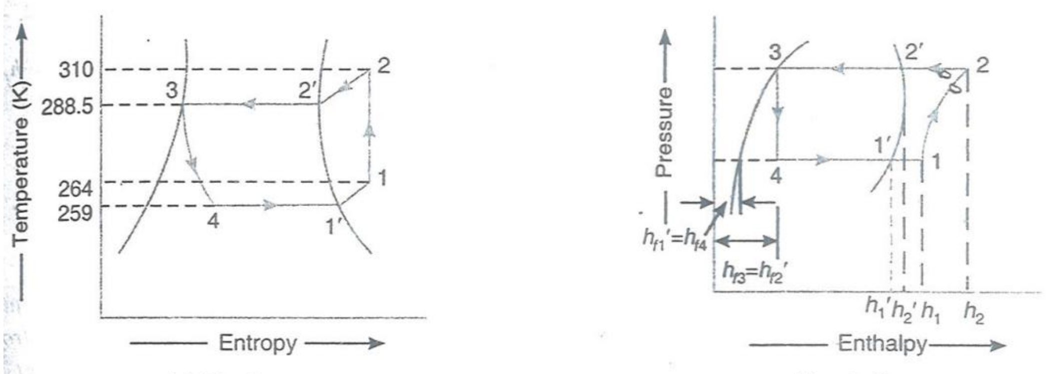
**8. A vapour compression refrigeration plant works between pressure limits of 5.3 bar and 2.1 bar. The vapour is superheated at the end of compression. its temperature being performance of the plant. Use the data given below:**

| **Pressure , bar** | **Saturation temperature, $^\circ C$** | **Liquid heat, kJ/kg** | **Latent heat, kJ/kg** |
|---|---|---|---|
| **5.3** | **15.5** | **56.15** | **144.9** |
| **2.1** | **-14.0** | **25.12** | **158.7** |

**Solution.** Given: $P_2$=5.3 bar; $p_1$=2.1 bar; $T_2$=37$^\circ C$=37+273=310K; $T_1$-$T_{1'}$=5$^\circ C$; $C_P$=0.63 kJ/kg K; $T_{2'}$=15.5$^\circ C$=15.5+273=288.5k; $T_{1'}$=-14$^\circ C$=14+273=259k; $h_{f3}$=$h_{f2}$=56.15kJ/kg; $h_{f1'}$=25.12 kJ/kg; $h_{fg2}$=144.9kJ/kg; $h_{fg1'}$=158.7 kJ/kg

the T-s and p-h diagrams are shown in Figure

we know that enthalpy of vapour at point 1,



(a) T-s diagram.

(b) p-h diagram.

$$h_1 = h_{1'} + C_p(T_1 - T_{1'}) = (h_{f1'} + h_{fg1'}) + C_p(T_1 - T_{1'})$$

$$= (25.12 + 158.7) + 0.63 \times 5 = 186.97 \text{ kJ/kg}$$

Similarly, enthalpy of vapour at point 2,

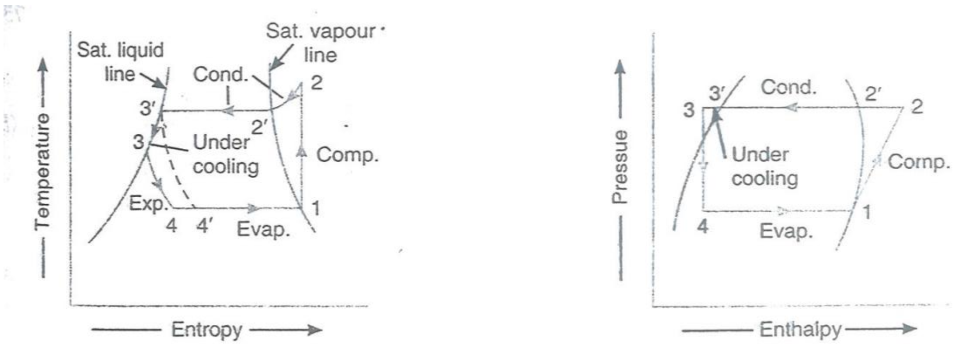
$$h_2 = h_{2'} + C_p(T_2 - T_{2'}) = (h_{f2'} + h_{fg2'}) + C_p(T_2 - T_{2'})$$

$$= (56.15 + 144.9) + 0.63\,(310 - 288.5) = 214.6 \text{ kJ/kg}$$

$\therefore$ Coefficient of performance of the plant,

$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{186.97 - 56.15}{214.6 - 186.97} = \frac{130.82}{27.63} = 4.735 \text{ Ans.}$$

## 5. Theoretical vapour compression cycle with under cooling or subcooling of Refrigerant



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

**Figure10** Theoretical vapour compression cycle with under cooling or sub cooling of Refrigerant

Sometimes, the refrigerant, after condensation process 2'-3', is cooled below the saturation temperature ($T_3$') before expansion by throttling. such a process is called under cooling or sub cooling of the refrigerant and is generally done along the liquid line as shown in Figure 10 (a) and (b). The ultimate effect of the under cooling is to increase the value of coefficient of performance under the same set of conditions.

The process of under cooling is generally brought about by circulating more quantity of cooling water through the condenser or by using water colder than the main circulating water. Sometimes, this process is also brought about by employing a heat exchanger. In actual practice. The refrigerant is superheated after compression and under cooled before throttling, as shown in

Fig. 10 (a) and (b). A little consideration will show, that the refrigerating effect is increased by adopting both the superheating and under cooling process as compared to a cycle without them, which is shown by dotted lines in Fig. 10 (a).I n this case, the refrigerating effect or heat absorbed or extracted,

$$R_E=h_1-h_4=h_1-h_{f3} \qquad \text{--------}(\because h_4=h_{f3})$$

$$w = h_2-h_1$$

$$\text{C.O.P} = \frac{\text{Re } frigeratingeffect}{Workdone} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Note: The value of $h_{f3}$ may be found out from the relation,

$$h_{f3}=h_{f3'}-C_p \text{ x Degree of undercooling}$$

**8. A vapour compression refrigerator uses R-12 as refrigerant and the liquid evaporates in the evaporator at - 15 $^\circ C$. The temperature of this refrigerant at the delivery from the compressor is 15 $^\circ C$ when the vapour is condensed at 10 $^\circ C$. find the coefficient of performance if (i) there is no under cooling, and (ii) the liquid is cooled by 5 $^\circ C$ before expansion by throttling. Take specific heat at constant pressure for the superheated vapour as 0.64 kJ/kg K and that for liquid as 0.94 kJ/kg K. the other properties of refrigerant are as follows:**

| **Temperature in $^\circ C$** | **Enthalpy in kJ/kg** | | **Specific entropy in kJ/kg K** | |
|---|---|---|---|---|
| | **Liquid** | **vapour** | **Liquid** | **Vapour** |
| **-15** | **22.3** | **180.88** | **0.0904** | **0.7051** |
| **+10** | **45.4** | **191.76** | **0.1750** | **0.6921** |

**Solution.** Given: $T_1=T_4=-15\ ^\circ C=-15+273=258K$; $T_2=15\ ^\circ C=15+273=288K$; $T_2'=10\ ^\circ C=10+273=283K$; $C_{pv}=0.64$ kJ/kg K; $C_{PL}=0.94$ kJ/kg K; $h_{f1}=22.3$ kJ/kg; $h_{f3}'=45.4$ kJ/kg; $h_{1'}=180.88$ kJ/kg; $h_2' = 191.76$ kJ/kg; $S_{f1}0.0904$ kJ/kg K; $*S_{f3}=0.1750$kJ/kg K; $S_{g1}=0.7051$ kJ/kg K; $S_2' = 0.6921$ kJ/kg K

(i) Coefficient of performance if there is no undercooling

The T-s and p-h diagrams, when there is no undercooling, are shown I fig. 4.19 (A) and (b) respectively.

Let $x_1$= Dryness fraction of the refrigerant at point 1.

we know that entropy at point 1,

$S_1=S_{f1}+x_1S_{fg1}=S_{f1}+x_1(S_{g1}-S_{f1}) \qquad$ ----($\because S_{g1}=S_{f1}+S_{fg1}$)

$=0.0904+x_1(0.7051-0.0904)=0.0904+0.6147\ x_1$ -----------------(i)

and entropy at point 2,

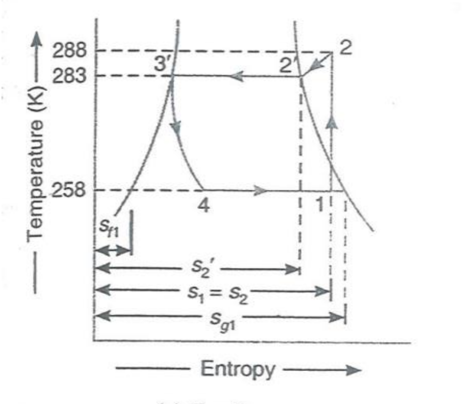
$$S_2=S_2'+2.3\ C_{pv}\log\frac{T_2}{T_2'}$$

$$= 0.06921+2.3 \times 0.64 \log \frac{288}{283}$$

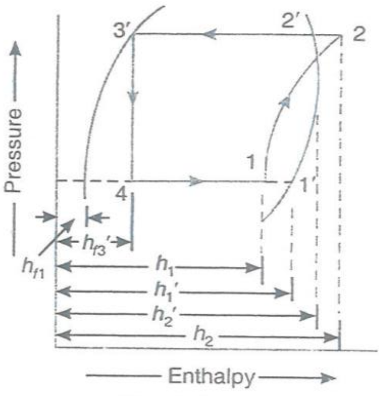
$=0.06921+2.3 \times 0.64 \times 0.0077 = 0.7034$ --------(ii)

Since the entropy at point 1 is equal to entropy at point 2, therefore equating equations (i) and (ii),

$0.0904+0.6147\ x_1=0.7034$ or $x_1=0.997$



(a) T-s diagram.



(b) p-h diagram.

We know that the enthalpy at point 1,

$h_1=h_{f1}+x_1h_{fg1}=h_{f1}+x_1(h_{g1}-h_{f1})$

$=22.3+0.997\ (180.88-22.3)=180.4$ kJ/kg -------($\because h_{g1}-h_1'$)

and enthalpy at point 2,

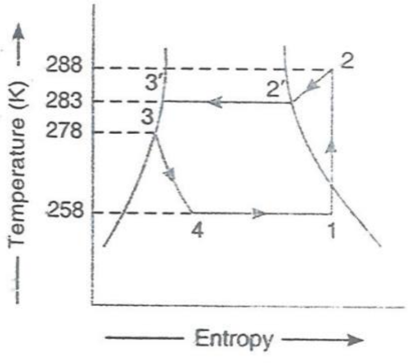
$h_2 = h_2' + C_{PV}\ (T_2 - T_2')$

$=191.76+0.64\ (288-283)=194.96$ kJ/kg

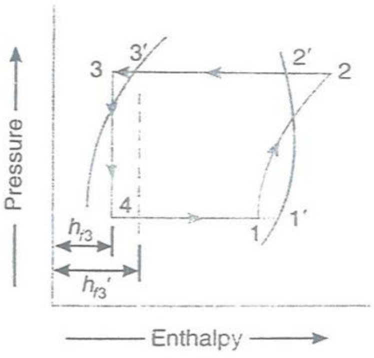
$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{180.4 - 45.4}{194.96 - 180.4} = 9.27 \text{ Ans.}$$

(ii) Coefficient of performance when there is an under cooling of 5 °C

The T-s and p-h diagrams, when there is an under cooling of 5 °C, are shown in Figure



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



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

We know that enthalpy of liquid refrigerant at point 3,

$h_{f3} = h_{f3}' - C_{pl}$ x Degree of under cooling

$= 45.5 - 0.94 \times 5 = 40.7$ kJ/kg

$$\text{C.O.P} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{180.4 - 40.7}{194.96 - 180.4} = 9.59 \text{ Ans.}$$

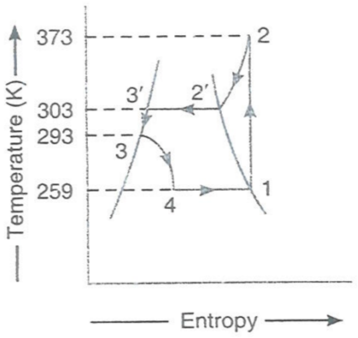
**9. A simple $NH_3$ Vapour compression system has compressor with piston displacement of 2 $m^3$/min, a condenser pressure of 12 bar and evaporator pressure of 2.5 bar. The liquid is sub-cooled to 20 °C by soldering the liquid line to suction line. The temperature of vapour leaving the compressor is 100 °C, heat rejected to compressor cooling water is 5000kJ/hour. and volumetric efficiency of compressor is 0.8.**

**Compute: Capacity: Indicated power; and C.O.P of the system**.

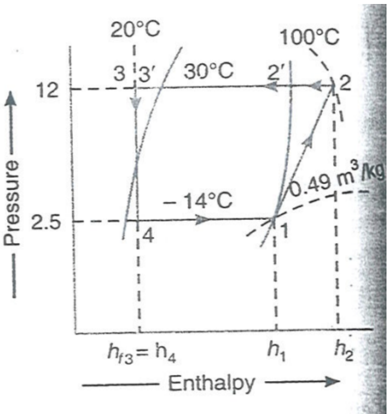
**Solution:** Given: $V_p$=2 $m^3$/min; p2 = p2’ = p3 = 12 bar ; p1 = p4 = 2.5 bar T3 = $20^\circ C$ = 20 + 273 = 293 K ; $T_2$ = $100^\circ C$=100+273=373K; $\eta_v$=0.8

Capacity of the system

The T-s and p-h diagrams are shown in Figure



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



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

From p-h diagram, we find that the evaporating temperature corresponding t 2.5 bas is $T_1$=$T_4$=-$14^\circ C$=-14+273 = 259K

Condensing temperature corresponding to 12 bar is

$T_2$’=$T_3$’=$30^\circ C$=30+273=303K

Specific volume of dry saturated vapour at 2.5 bar (i.e at point 1).

$V_1$=0.49 $m^3$/kg

Enthalpy of dry saturated vapour at point 1,

$h_1$=1428 kJ/kg

Enthalpy of superheated vapour at point 2,

$h_2$=1630 kJ/kg

and enthalpy of sub-cooled liquid at $20^\circ C$ at point 3,

$h_{f3}=h_4=270$ kJ/kg

$m_R$= Mass flow of the refrigerant in kg/min.

We know that piston displacement,

$$V_p=\frac{m_R \times v_1}{\eta_v} \text{ or } m_R=\frac{v_p \times \eta_v}{v_1}=\frac{2\times 0.8}{0.49}=3.265 \text{ kg/min}$$

We know that refrigerating effect per kg of refrigerant

$=h_1-h_{f3}=1428-270=1158$kJ/kg

and total refrigerating effect $=m_R(h_1-h_2)= 3.265(1428-270)=3781$ kJ/min

$\therefore$ Capacity of the system = 3781/210 = 18 TR Ans.

Indicated power of the system

We know that work done during compression of the refrigerant

$= m_R(h_2-h_1)=3.265\ (1630-1428)=659.53$kJ/min

Heat rejected to compressor cooling water

=5000 kJ/h=5000/60=83.33 kJ/min …… (Given)

$\therefore$ Total work done by the system

=659.53 + 83.33 = 742.86 kJ/min

and indicated power of the system

=742.86/60=12.38kW Ans.

COP of the system ,We know that C.O.P of the system

$$=\frac{Total refrigerating effect}{Total work done}=\frac{3781}{742.86}=5.1 \text{ Ans}$$

## COMPOUND VAPOUR COMPRESSION REFRIGERATION SYSTEMS

## INTRODUCTION

Compression refrigeration system in which the low pressure vapour refrigerant from the evaporator is compressed in a single stage (or a single compressor) and then delivered to a condenser at a high pressure. But sometimes, the vapour refrigerant is required to be delivered at a very high pressure as in the case of low temperature refrigerating systems. In such cases either we should compress the vapor refrigerant by employing a single stage compressor with a very high pressure ratio between the condenser and evaporator or compress it in two or more compressors placed in series. The compression carried out in two or more compressors is called compound or multistage compression.

In vapour compression refrigeration systems, the major operating cost is the energy input to the system in the form of mechanical work. Thus any method of increasing coefficient of performance is advantageous so long as it does not involve too heavy an increase in other operating expenses, as well as initial plant cost and consequent maintenance.

Since the coefficient of performance of a refrigeration system is the ratio of refrigerating effect to the compression work, therefore the coefficient of performance can be increased either by increasing the refrigerating effect or by decreasing the compression work. A little consideration will show that in a vapour compression system, the compression work is greatly reduced if the refrigerant is compressed very close to the saturated vapour line. This can be achieved by compressing the refrigerant in more stages with intermediate inter cooling. But it is economical only where the pressure ratio is considerable as would be the case when very low evaporator temperatures are desired or when high condenser temperature may be required. The compound compression is generally economical in large plants.

## ADVANTAGES OF COMPOUND (OR MULTI-STAGE) VAPOR COMPRESSION WITH INTERCOOLER

Following are the main advantages of compound or multi-stage compression over single stage compression:

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

## TYPES OF COMPOUND VAPOUR COMPRESSION WITH INTERCOOLER

In compound compression vapour refrigeration systems, the superheated vapour refrigerant leaving the first stage of compression is cooled by suitable method before being fed to the second stage of compression and so on. Such type of cooling the refrigerant is called inter cooling. Though there are many types of compound compression with intercoolers, yet the following are important from the subject point of view:

**1.** Two stage compression with liquid intercooler.
**2.** Two stage compression with water intercooler.
**3.** Two stage compression with water intercooler, liquid sub cooler and liquid flash chamber.
**4.** Two stage compression with water intercooler, liquid sub cooler and flash intercooler.
**5.** Three stage compression with flash chambers.
**6.** Three stage compression with water intercoolers.
**7.** Three stage compression with flash intercoolers..

## TWO STAGE COMPRESSION WITH LIQUID INTERCOOLER

The arrangement of a two stage compression with liquid intercooler is shown in figure and the corresponding p-h diagram is shown in figure

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

1. First of all, draw a horizontal pressure line representing the evaporator pressure $p_e$ (or suction pressure of low pressure compressor) which intersects the saturated vapour line

at point 1. At this point, the saturated vapour is supplied to the low pressure compressor. Let, at point 1, the enthalpy of the saturated vapour is h1 and entropy s.

2. The saturated vapour refrigerant admitted at point 1 is compressed isentropic ally in the low pressure compressor and delivers the refrigerant in a superheated state. The pressure rises from pe to p2. The curve 1-2 represents the isentropic compression in the low pressure compressor. In order to obtain point 2, draw a line from point 1, with entropy equal to along the constant entropy line intersecting the intermediate pressure line at point 2. Let enthalpy at this point is h.
3. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is cooled (or desuperheated) at constant pressure $p_2=p_3$ in a liquid intercooler by the liquid refrigerant from the condenser. The refrigerant leaving the liquid intercooler is in saturated vapour state. The line 2-3 represents the cooling or desupertheating process. Let the enthalpy and entropy at point 3 is h and s respectively.
4. The dry saturated vapour refrigerant now supplied to high pressure compressor where it is compressed isentropic ally from intermediate or intercooler pressure ps to condenser pressure. The curve 3-4 represents the isentropic compression in the high pressure compressor. The point 4 on the p-h diagram is obtained by drawing a line of entropy equal to along the constant entropy line as shown in fig.11 (b). let the enthalpy of superheated vapor refrigerant at point 4 is $h_4$.

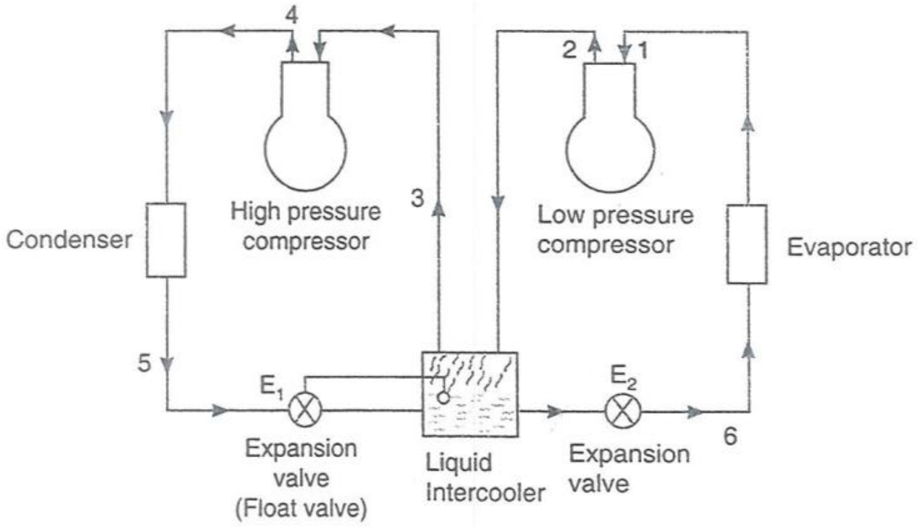


**Figure11** Two stage compression with liquid intercooler

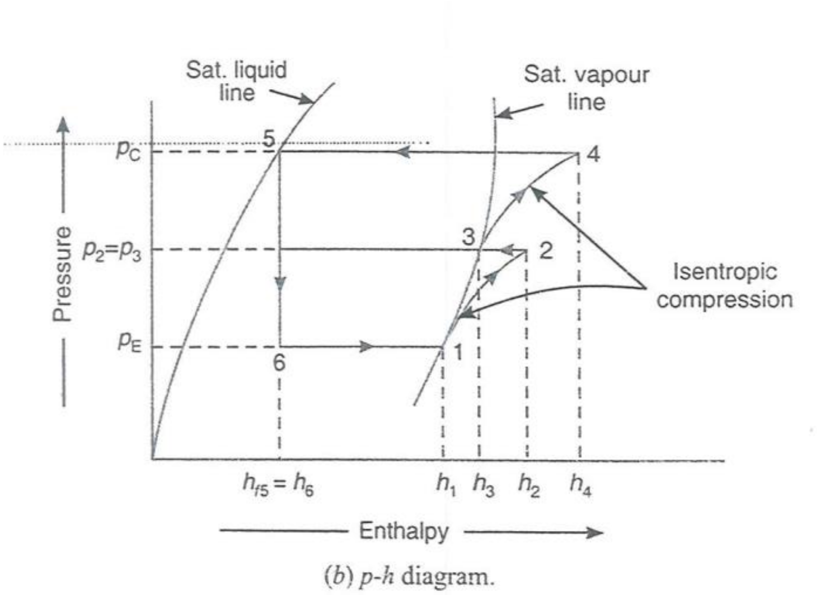


**Figure12** P-h Diagram of Two stage compression with liquid intercooler

5. The superheated vapor refrigerant leaving the high pressure compressor at point 4 is now passed through the condenser at constant pressure pc as shown by a horizontal line 4-5. The condensing process 4-5 changes the state of refrigerant from superheated vapour to saturated liquid.
6. The high pressure saturated liquid refrigerant from the condenser is passed to the intercooler where some of liquid refrigerant evaporates in desupertheating the superheated vapour refrigerant from the low pressure compressor. In order to make up for the liquid evaporated i.e. to maintain a constant liquid level, an expansion value E which acts as a float value is provided.
7. The liquid refrigerant from the intercooler is first expansion value $E_2$ and then evaporated in the evaporator to saturated vapour condition as shown in fig. 5.1(b).

$M_1$ = mass of refrigerant passing through the evaporator
(or low pressure compressor) in kg/min, and
$M_2$ = mass of refrigerant passing through the condenser
(or high pressure compressor) in kg/min.

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

$$M_1 = m_2 - m_1$$

The value of m2 may be obtained by considering the thermal equilibrium for the liquid intercooler as shown in fig. 10, i.e.

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

$$M_2h_{f5}+m_1h_2=m_1h_6+m_2h_3$$

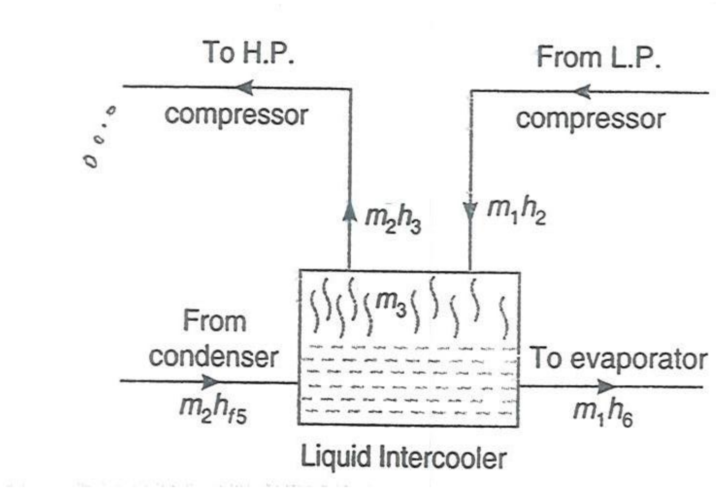


**Figure13** Thermal equilibrium for liquid intercooler

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

and mass of liquid refrigerant evaporated in the intercooler,

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

We know that refrigerating effect,

$$R_E = m_1(h_1 - h_f) = m_1(h_1 - h_{f5}) = 210QkJ / \min$$

where $Q$ is the load on the evaporator in tonne of refrigeration.

Total workdone in both the compressors.

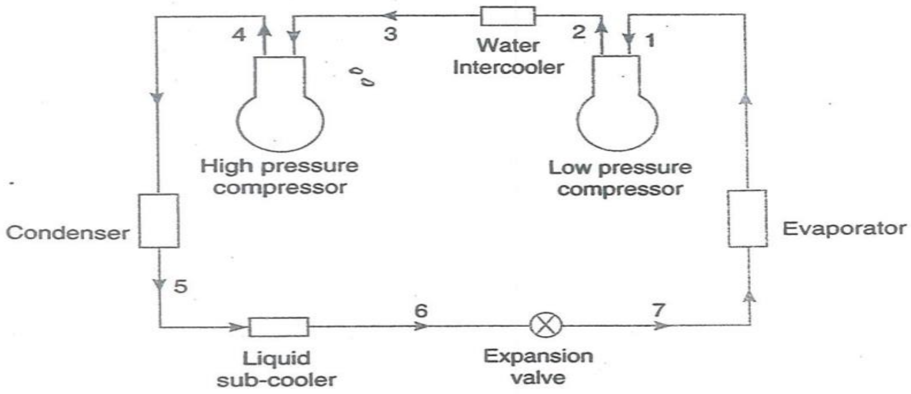
$$P = \frac{m_1(h_2 - h_1) + m_2(h_4 - h_3)}{60} kW$$

and C.O.P. of the system $= \dfrac{R_E}{W} = \dfrac{m1(h_1 - h_{f5})}{m_1(h_2 - h_1) + m_2(h_4 - h_3)} \dfrac{210Q}{P \times 60}$

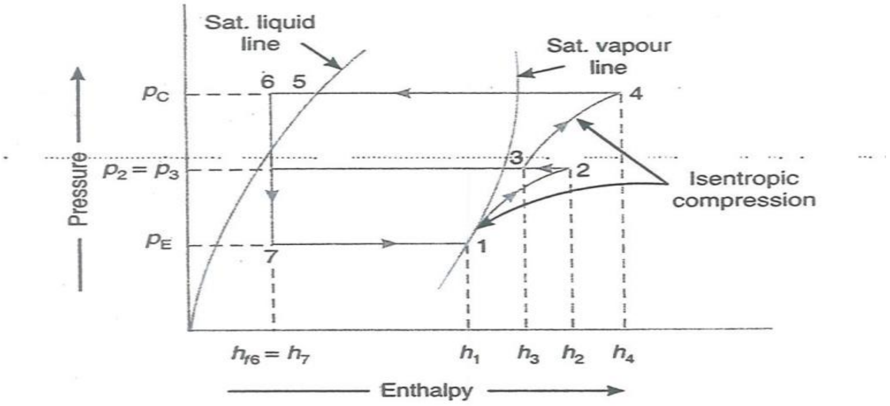
## TWO STAGE COMPRESSION WITH WATER INTERCOOLER AND LIQUID SUB-COOLER

The arrangement of a two-stage compression with water intercooler and liquid sub-cooler is shown in fig14 (a). the corresponding p-h diagram is shown in fig. 14 (b). the various processes in this system are as follows:

1. The saturated vapour refrigerant at the evaporator pressure $p_E$ is admitted to low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropic ally from the evaporator pressure $p_e$ to the water intercooler pressure $p_2$, as shown by the curve 1-2 in fig14 (b).
2. The refrigerant leaving the low pressure compressor at point 2 is in superheated state. This superheated vapour refrigerant is now passed through the water intercooler at constant pressure, in order to reduce the degree of superheat. The line 2-3 represents the water intercooling or desuperheating process.
3. The refrigerant leaving the water intercooler at point 3 (which is still in the superheated state) is compressed isentropic ally in the high pressure compressor to the condenser pressure pe. The curve 3-4 shows the isentropic compression in high pressure compressor.
4. The discharge from the high pressure compressor is now passed through the condenser which changes the state of refrigerant from superheated vapour to saturated liquid as shown by process 4-5.
5. The temperature of the saturated liquid refrigerant is further reduced by passing it through a liquid sub-cooler as shown by process 5-6.
6. The liquid refrigerant from the sub-cooler is now expanded in an expansion value (process 6-7) before being sent to the evaporator for evaporation (process 7-1).



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



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

**Figure14** Two stage compression with water intercooler and liquid sub-cooler

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

Let $Q$ = Load on the evaporator in tonnes of refrigeration.

$\therefore$ Mass of refrigerant passing through the evaporator (or passing through the L.P. compressor),

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

Since the mass of refriger5ant passing through the compressors is same, therefore, total work done in both the compressors,

W=Work done in L.P. compressor + Work done in H.P. compressor

$$= m(h_2 - h_1) + m(h_4 - h_3) = m[(h_2 - h_1) + (h_4 - h_3)]$$

$\therefore$ Power required to drive the system.

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

We know that refrigerating effect,

$$R_E = m(h_1 - h_{f6}) = 210QkJ/\min$$

$$\therefore \text{C.O.P. of the system } = \frac{\mathrm{R}_E}{W} = \frac{m(h_1 - h_{f6})}{m[(h_2 - h_1) + (h_4 - h_3)]} = \frac{210Q}{P \times 60}$$

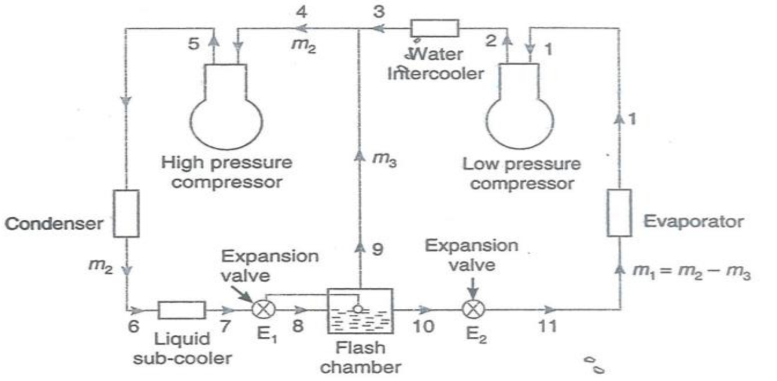
## TWO STAGE COMPRESSION WITH WATER INTERCOOLER, LIQUID SUB-COOLER AND LIQUID FLASH CHAMBER

The arrangement of a two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber is shown in fig. 15(a). the corresponding p-h diagram is shown in fig. 15(b). the various processes, in this system, are as follows:

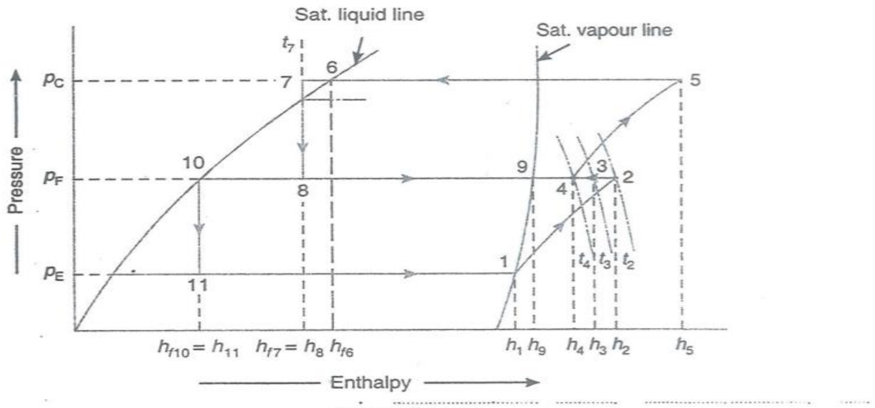
1. The saturated vapour refrigerant at the evaporator pressure $p_e$ is admitted to low pressure compressor at point 1. In this compressor, the refrigerant is compressed isentropically from evaporator pressure $p_e$ to water intercooler (or flash chamber) pressure $p_f$ as shown by the curve 1-2 in fig. 15 (b).
2. The superheated vapour refrigerant leaving the low pressure compressor at point 2 is now passed through the water intercooler at constant pressure pf in order to reduce the degree of superheat (i.e, from temperature $t_2$ to $t_3$). The line 2-3 represents the water intercooling or de-superheating process.
3. The superheated vapour refrigerant leaving the water intercooler at point 3 is mixed with the vapour refrigerant supplied by the flash chamber at point 9. The condition of

refrigerant after mixing is shown by point 4 which is in superheated state. Let the temperature at this point is $t_4$.

4. The superheated vapour refrigerant admitted at point 4 to the high pressure compressor is compressed isentropically from the intercooler or flash chamber pressure $p_f$ to condenser pressure $p_c$ as shown by the curve 4-5. The temperature rises from $t_4$ to $t_5$.
5. The superheated vapour leaving the high pressure compressor at pressure pc is passed through a condenser at constant pressure as shown by a horizontal line 5-6. The condensing process 5-6 changes the state of refrigerant from superheated vapour to saturated liquid.
6. The saturated liquid refrigerant from the condenser is now cooled in liquid sub-cooler to a temperature, say $t_7$. The line 6-7 represents a sub-cooling process.



Two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber.



(b) p-h diagram.

**Figure15** Two stage compression with water intercooler, liquid sub-cooler and liquid flash chamber

7. The liquid refrigerant leaving the sub-cooler at pressure $p_c$ is expanded in an expansion value $E_1$ to a pressure equal to the flash chamber pressure pf as shown by vertical line 7-8. The expanded refrigerant which is a mixture of vapour and liquid refrigerants is admitted to a flash chamber at point 8. The flash chamber separates the vapour and liquid refrigerants at pressure. The vapour refrigerant from the flash chamber at point 9 is mixed with the refrigerant from the water intercooler. The liquid refrigerant from the flash chamber at point 10 is further expanded in an expansion value $E_2$ as shown by the vertical line 10-11.
8. The liquid refrigerant leaving the expansion value $E_2$ is evaporated in the evaporator at the evaporator pressure pe (usually 2 bars) as shown by the horizontal line 11-1 in fig. 5.7 (b)

Let $M_2$= mass of refrigerant passing through the condenser or high Pressure compressor), and
$M_3$=mass of vapor refrigerant formed in the flash Chamber.

∴ Mass of refrigerant passing through the evaporator (or low pressure compressor.)

$$M_1=m_2-m_3$$

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

$$m1=\frac{210Q}{h_1-h_{11}}=\frac{210Q}{h_1-h_{f10}}kg/\min \qquad \ldots(\because h_{11}=h_{f10})$$

Now let us consider the thermal equilibrium of the flash chamber. Since the flash chamber is an insulated vessel, therefore there is no heat exchange between the flash chamber and atmosphere. In other words, the heat taken and given by the flash chamber are same. Mathematically.

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

or $$m_2h_8=m_3h_9+m_1h_{f10}$$

$$=m_3h_9+(m_2-m_3)h_{f10} \qquad \ldots(\because m_1=m_2-m_3)$$

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

$$m3 = m2\left(\frac{h_8 - h_{f10}}{h_9 - h_{f10}}\right) = m_2\left(\frac{h_{f7} - h_{f10}}{h_9 - h_{f10}}\right) \qquad \ldots(\because h_8 - h_{f7})\ldots(i)$$

The vapour refrigerant from the water intercooler (represented by point 3) is mixed with vapour refrigerant m$_3$ from the flash chamber (represented by point 9) at the same pressure before entering the high pressure compressor. The enthalpy of the mixed refrigerant (represented by point 4) may be calculated by using the equation.

$$m_2 h_4 = m_3 h_9 + m_1 h_3$$

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

We know that refrigerating effect of the system.

$$R_E = m_1(h_1 - h_{11}) = 210Qkj / \min$$

Work done in low pressure compressor, $W_L = m_1(h_2 - h_1)$

Work done in high pressure compressor, $W_H = m_2(h_5 - h_4)$

Total workdone in both the compressors,

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

$\therefore$ Power required to drive the system.

$$P = \frac{M_1(h_2 - h_1) + m_2(h_5 - h_4)}{60} kW$$

and C.O.P. of the system $= \dfrac{\mathrm{R}_E}{W} = \dfrac{m_1(h_1 - h_{11})}{m_1(h_2 - h1) + m_2(h_5 - h_4)} = \dfrac{210Q}{P \times 60}$

## REFRIGERANTS

## INTRODUCTION

The refrigerant is a heat carrying medium which during their cycle (i.e comparison, condensation, evaporation) in the refrigeration system absorbs heat from allow temperature system and discards the eat so absorbed to higher temperature system.

The natural ice and mixture of ice and salt were the first refrigerants. In 1834, edger ammonia, sulphur dioxide, methyl chloride and carbon dioxide came into use as refrigerants in compression cycle refrigeration machines. Most of the early refrigerant materials have been discarded for safety reasons or for lack of chemical or thermal stability. In the present days, many new refrigerants including halo-carbon compounds, hydro-carbon compounds are used for air –condition and refrigeration applications.

The suitability of a refrigerant for a certain application is determined by its physical thermodynamic, chemical properties and by various practical factors. There is no one refrigerant which can be used for all types of applications i.e there is no ideal refrigerant. It one refrigerant has certain good advantages, it will have some disadvantages also. Hence, a refrigerant is chosen which has greater advantages and less disadvantages.

## DESIRABLE PROPERTIES OF AN IDEAL REFRIGERANT

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

1. Low boiling and freezing point,
2. High critical pressure and temperature,
3. High latent heat of vaporization,
4. Low specific heat of liquid, and high specific heat of vapour,
5. Low specific volume of vapour,
6. High thermal conductivity,
7. Non-corrosive to metal,
8. Non-flammable and non-explosive,
9. Non-toxic
10. Low cost,
11. Easily and regularly available,
12. Easy to liquefy at moderate pressure and temperature,
13. Easy to locating leaks by our or suitable indicator,
14. Mixes well with oil,
15. High coefficient of performance, end
16. Ozone friendly.

## CLASSIFICATION OF REFRIGERANTS

The refrigerants may, broadly, be classified in the following two group:

1. Primary refrigerants and 2. Secondary refrigerant

The refrigerants which directly take part in the refrigeration system are called primary refrigerants whereas the refrigerants which are first cooled by primacy refrigerants and then used for cooling purposes as known as secondary refrigerants.

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

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

These above mentioned refrigerants are discussed, in detail, in the following pages.

## HALO-CARBON REFRIGERANTS

The American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) identifies 42 halo-carbon compounds as refrigerants, but only a few of them are commonly used. The following table gives some of the commonly used halo-carbon refrigerants:

1. R-11, Trichloro-monofluoro-methane (CCl3F). the R-11 is a synthetic chemical product which can be used as a refrigerant. It is stable, non-flammable and one- toxic. It is considered to be low-pressure refrigerant. It has a low side pressure of 0.202 bar at - 15°C and high side pressure of 1.2606 bar at 30°C. The latent heat at - 15°C is 195 kJ/kg. The boiling point at atmospheric pressure is 23.77°C. Due to its low operating pressures, this refrigerant is exclusively used in large centrifugal compressor stems of 200 TR and above. The leaks may be detected by using a soap solution, a halide torch or by using an electronic detector.

   R-11 is often used by service technicians as a fishing agent for cleaning the internal parts of a refrigerator compressor when overhauling systems. It is useful after aa system had a motor burn our or after it has great deal of moisture in the system. By flushing moisture from the system with R-11, evacuation time is shortened. R-11 is one of the safest cleaning solvents that can be used for this purpose. The cylinder colour code for R-11 is orange.

2. R-12, Dichloro-diflluoro-methane ($CCl_2F_2$). The R-12 is a very popular refrigerant. It is a colourlesss, almost odorless liquid with boiling point of -29°C at atmospheric pressure. It is non-toxic, non-corrosive, non-irritating and non-flammable. It has a relatively how latent heat value which is an advantage is small refrigerating machines. The large amount of

refrigerant circulated will permit the use of less sensitive and more positive operating and regulating mechanisms. It operates at allow but positive head and back pressure and with a good volumetric efficiency. This refrigerant is used in many different types of industrial and commercial applications such as refrigerators, freezers, water coolers, room and window air-conditioning units etc. Its principal use is found in reciprocating and rotary compressor. But its use in centrifugal compressor for large commercial air-conditioning is increasing.

R-12 has a pressure of 0.82 bar at - 15°C and a pressure of 6.4 bar at 30°C. the latent heat of R-12 at - 15°C is 159 kJ/kg. the leak may be detected by soap solution, halide torch or an electronic leak detector. Water is only slightly soluble in R-12. At -18°C it will hold six parts per million by mass. The solution formed is very slightly corrosive to any of the common metals used in refrigerator construction. The addition of mineral oil to the refrigerant has no effect upon the corrosive action.

R-12 is more critical as to its moisture content when compared to R-22 and R-502. It is soluble in oil down to- 68°C. the oil will begin to separate at this temperature and due to its lightness than the refrigerant, it will collect on the surface of the liquid refrigerant. The refrigerant an available in a variety of cylinder sixe and the cylinder colour code is white.

3. R-13, Monochloro-trifluore-methane ($CClF_3$). The R-13 has a boiling temperature of 81.4°C at atmospheric pressure and a critical temperature of + 28.8°C. This refrigerant is used for the low temperature side of cascade systems. It is suitable with reciprocating compressors.
4. R-14, Carbontetrafluoride ($CF_4$). The R-14 has a boiling temperature of - 128°C at atmospheric pressure and critical temperature of -45.5°C. It serves as an ultra-low temperature refrigerant for use in cascade systems.
5. R-21, Dichloro-monofluoro-methane ($CHCl_2F$). The R-21 has a boiling temperature of +9°C at atmospheric pressure. It has found its principal use in centrifugal compressor systems for relatively high temperate refrigeration requirements.
6. R-22, Monochloro-difluoro-methane ($CHClF_2$). The R-22 is a man-made refrigerant developed for refrigeration installations that need low evaporating temperatures, as in fast freezing unit which maintain a temperature of -29°C to -40°CC. it has also been successfully used in air-conditioning units and in household refrigerators. It is use with reciprocating and centrifugal compressors. It is not necessary to use R-22 at below atmospheric pressures in order to obtain the low temperatures.

The boiling point of R-22 is -41°C at atmospheric pressure. It has a latent heat of 216.5 kJ/kg at -15°C. The normal head pressure at 30°C is 10.88 bar. These refrigerants stable and are non –toxic, non-corrosive, non-irritating and non-flammable. The evaporator pressure of this refrigerant at -15°C is 1.92 bar. Since water mixes better with R-22 than

R-12 by a ratio of 3 to 1, therefore driers (dessicants0 should be used to remove most of the moisture to keep water to a minimum. This refrigerant has good solubility in oil sown to -9°C. However, the oil remains flow enough to flow down the suction line at temperatures as low as -40°C. The oil will begin to separate at this point. Since oil is lighter, therefore it will collect on the surface of the liquid refrigerant. The leaks may be deiced with a sop solution, a halide torch or with an electronic leak detector. The cylinder colour code for R-2 is green.

7. R-30, Methylene chloride ($CH_2Cl_2$). The R-30 is a clear, water-white liquid with sweet, non-irritating odour similar to that of chloroform. It has a boiling point of 9.8°C at atmospheric pressure. It is non-flammable, non-explosives and non-toxic. Due to its high boiling point, this refrigerant may be stored in cloded cans instead of in compressed gas cylinders. This high and low side of refrigeration system using R-30 operates under a vacuum. Since the volume of vapour at suction conditions is very high, therefore the use of R-30 is restricted to rotary or centrifugal compressor. This refrigerant was extensively used for air conducting of theatres auditorium, and office buildings. Now –a days, the refrigerant R-11 is used in place of R-30.

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

8. R-40, Methyl-chloride (CH3Cl). The R-40 is colorless liquid with a faint, wet, and non-irritating odour. Its boiling pint at atmospheric pressure is -23.7°C and the usual condenser pressure is 5 to 6.8 bar. The latent heat of vaporization at - 15°C is 423.5kJ/kg. it is flammable and explosive when mixed with air in concentrations from 8.1 to 17.2 per cent. This regieranct is non-corrosive in its pure state, but it becomes corrosive in the presence of moisture. Aluminum zinc and magnesium alloys should never be used with thus refrigerant as they will cored considerably and pollute the lubricating oil. since the refrigerant R-40 is a solvent for many materials used in ordinary refrigeration compressor, therefore rubber and gaskets containing rubber and gaskets containing rubber should never be used. However, synthetic rubber is not affected byR-40. Thus metallic asbsstor-fibre gaskets containing insoluble binders should be used. The mineral oils are soluble in this refrigerant to a small extent.

   This refrigerant has been used in domestic units with both reciprocating and rotary compressors and in commercial units with reciprocating compressors up to approximately 10 TR capacities. The leaks with R-40 may be detected by soap solution or electronic leak detector.
9. R-100, Ethyl chloride ($C_2H_5Cl$). The R-100 is a colure less liquid and in many respites it is similar to R-40 (Methyl Chloride) but with low operating pressures. It has b a boiling point of 13.1°C at atmospheric pressure. It is both toxic and flammable. Due to it low operating pressure, it is not used in refrigerating equipment.

10. R-113, Tricholor-trifluoro-etane ($CC_{l2}FCClF_2$ or $C_2ClE_3$). The R-113 has a boiling point of 47.6°C at atmospheric pressure. It is used in commercial and industrial air-conditioning with centrifugal compressor systems. Since this refrigerant has the advantage of remaining liquid at room temperatures and pressures, therefore it can be carried in sealed tins rather than cylinders.

11. R-114, Dichloro-tetrafluoro-ethane ($CClF_2$ or $C_2Cl_3F_3$). The R-114 has a oblong point of 3.6°C at atmospheric pressure. At-15°C,it evaporates at pressure of 0.54 bar boiling and at +30°C it condenses at a pressure of 1.5 bar. Its latent heat of vaporization at -15°C is 143kJ/kg. it is non-toxic, non-explosive and non corrosive even in the presence of water. It is used in fractional power household refrigerating system and drinking water coolers employing rotor vane type compressors.

12. R-123., Dichloro- trifluore –methane ($CF_3CHCl_2$). The R-123 is a potential substitute to R-11. It has about 4.3°C higher boiling postman R-11. It is, therefore, a tower pressure replacement for R-11, thus having larger specific volume of suction vapourt. Hence, its use results in 10 to 15%reduion in capacity, it used in capacity, if used in existing R-11 centrifugal compressors.

13. R-134 ,Tetrafluoro-ethane (CF3CH2F0. The R134a is considered to be the most preferred substitute for refrigerant R-12. Its boiling point is -26.15 °C which is quite close to the boiling point of R-12 which is -29°C at atmospheric pressures. Since the refrigerant R-134 has no chlorine atom, there for this refrigerant has zero * ozone depleting potential (ODP) and has 74% less global warming potential (GWP) as compared to R-12. It has lower suction pressure and large suction vapor volume sis not soluble in mineral oil (polyester based) is used. Care should be taken to prevent moisture from getting into their figuration system. For use in existing R-12 reciprocating compressors, it would require either an average increases in compressors speed of 5 to 8% or an equivalent increases in cylidervalum. Since the moledecules of R-134a are smaller than R-12, therefore Avery sensitive leak detector is used to detect leaks. .

    14.R-152a, Difluoro- enhance (CH3CHF2). The R-15a has similar characteristic as R 134a except that R-152 has slight vacuum in the evaporator at -25°C and the discharge temperature is higher because of its high value of the ratio of specific heats (y)

## AZEOTROPE REFRIGERANTS

The term“Azeotrope’ refers to stabel mixture of refrigerants whose vapour and liquid phases retain identical compositions over a wide range of temperatures. However, the these mixtures, usually, have properties that differ from either of the components. Some of the Azeotrope are given is the following table.

**Table 1 Azeotrope refrigerants**

| Refrigerant number | Azetrope mixing refrigerant | Chemical formula |
|---|---|---|
| R-500 | 73.8% R-12 and 26.2% R-152 | $CCL_2F_2/CH_3CHF_2$ |
| R-502 | 48.8% R-22 and 51.2% R-115 | $CHCIF_2/CCIF_2CF_3$ |
| R-503 | 40.1% R-23 and 59.9% R-13 | $CHF_3/CCLF_3$ |
| R-504 | 48.2% R-32 and 51.8% R-115 | $CHF_2F_2/CCLF_2CF_3$ |

These refrigerants are discussed, in discussed, in detail, as below:

1. R-500. The R-500 is an Azeotrope mixture of 73.8% R-12 (CCl2F2) and 26.2% R-152 (CH2CHF2). It is non-flammable, low in toxicity and non- corrosive. It is used in both industrial and metrical applications but only in systems enteric oration compressors. It has a fairly constant vapor presser temperature curve which is different from the vaporizing curves for either R-152a or R-12. This refrigerant officers about 20% greater refrigerating capacity than R-12 for the same size of motor when used for the same purpose. The evaporator pressure of this refrigerant is 1.37 bar at-15°C is 192 kJ/Kg. it can be used whenever a higher capacity that that obtained with R-12 is needed. The sedulity of water in R-500 is highly critical. It has fairly high salinity with oil. The leakage may be erected by using soap solution, a halide. I this fairly high solubituyt with oil. The leakage may be detected by using soap solution, a halide torch, an electronic teak detector or a cored tracing agent. The servicing refrigeration using that refrigerant does not present any unusual problem. Water is quite soluble in the regieranct. It necessary to keep moisture out of the system by careful dehydration and by using dryers. The cylinder colour code for this refrigerant is yellow.
2. R-502. The R-502 is an azeotopic-misture of 48.8% R-22 ($CHCLF_2$) and d51.2% R-115 ($CCLF_2CF_3$). It is non-flammable, non-corrosive, practically non-toxic liquid. It is a good refrigerant of obtaining medium and low temperature. It is suitable where temperatures from -18°C to -15°C are needed. It is often used in frozen food lockers, frozen food processing plant frozen food displeases and it storage in its for frozen foods and ice-cream. It is only used with reciprocating compressors. The boiling point of this refrigerant at atmospheric pressures is -46°C. its evaporating pressure at -15°C is 2.48 bar and the condensing pressure at 30°C is 12.6. its latent het at-29°C is 168.6 kJ/Kg.The R-502 combines many of the good properties of R-12 and R-22. It gives a machine capacity equal to that of R-22 with just about the condemn sing temperature of a system using R-12. Since this refrigerant has a relatively low condensing pressure and temperature, therefore increases the light of compressor values and other parts. Better lubricating is possible because of the increases

viscosity of the oil at low condensing temperature. It is possible to eliminate liquid injection to cool the compressor because of the low condensing pressure. This refrigerant has all the qualities found in other halogenated (fluorocarbon) regirgernt. It is non-toxic. Non-flammable, non-irradiating, stable and non-corrosive. The leaks may be detected by soap solution, halide torch or electronic leak detector. It will hold 1.5 times more moisture at 18°C than R-12. It has fair solubility in oil above 82°C. Below this temperature, the oil tries to separate and tends to collect on the surface of the liquid refrigerant. However, oil is carried to the compressor at temperature down to -40°C. Special device are sometimes used to the oil to the compressors. The cylinder color code for this refrigerant is orchid.

3. R-503. This anAzeotrope mixture of 40.1% R-23 ($CHF_3$) and 59.9% of R-1 ($CCIF_3$). This is a non-flammable, non-corrosive, practically non-toxic liquid. Its temperature at atmospheric pressure is -88°C which is lower than either R-23 or R-13. It evaporating pressure at -15°C is 17.15 bar. Its critical temperature is 20°C and its critical pressure is 14.15 bar. This is a low temperature refrigerant and good for use in the low state of cascade systems which enquire temperatures in the range of -73°C to 87°C. The latent heat of vaporization at atmospheric pressure is 173 kJ/Kg. The leaks in R-503 system may be detected with the use of soap solution, a halide torch an electronic leak detector. This refrigerant wills hold more mister than some other temperature refrigerants. It may be not that all low temperature applications must have control devices. The oil does not circle are well a low temperatures. The cascade and other lower temperatures equipments are normally fleeted will oil separators and other devices of retuning the oil to the compressor the colder colure code for R-503 is aquamarine.

4. R-504. The R-504 is an azetropic mixture of 48.2% R-32 (CH2F2) and 51.8 % R-115 (CCIF2CF3). It non- flammable, non-corrosive and non-toxic.the boiling g temperature at atmospheric pressure is -57°c. its evaporating pressure at -15°C is 5.88 bar and its critical with the oil circulation. With the addition of 2 to 5% R-170 (ethane), the oil will be taken into the solution with the refrigerant and will circle through the system with it. The leaks in R-504 system may be easily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of -40°C to -62°C is desired. The cylinder color code for R-504 is ton. The leaks in R-504 systems may be classily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of -40°C to - 62°C is desired. The cylinder color code for R-504 is tan.

## INORGANIC REFRIGERANTS

The inorganic refrigerant swore exclusively used before the introduction of halo-carbon refrigerants. These regiments are still is use due to their inherent thermodynamic and physical properties. The vacuous inorganic regiments are given in the flowing table:

**Table 2 inorganic refrigerants**

| Refrigerant number | Chemical name | Chemical formula |
|---|---|---|
| R-171 | Ammonia | NH3 |
| R-729 | Air | -- |
| R-744 | Carbon dioxide | $CO_2$ |
| R-764 | Sulphur dioxide | $SO_2$ |
| R-118 | Water | $H_2O$ |

These refrigerants are discussed, in detail, as below:

1. R-717 (Ammonia). The R-717, i.e ammonia (NH3) is one the oldest and most widely used of l the reigigerants. Its greatest applications found in large and commercial reciprocating compression suttees where high toxicity is secondary. It is also widely in absorption systems. It is a chemical compound of nitrogen an hydrogen and under ordinary conditions, it is a colorless gas. Its boiling point at atmospheric pressure is-33.3°C and its melting point from the solid is -78°C. the low boiling point makes it possible to have regeneration at temperatures considerably below 0°C without using pressures below atmospheric in the evaporators. Its latent heat of vaporization at -15°C is 1315 kJ/Kg. thus, large refrigerating effects are possible with relatively small sized machinery. The condenser pressure at 30°C is 10.78 bar. The conddensersofrR-717 are usually of water cooled type. It is poisonous gas if inhaled in large quantities. In lesser quantities, it is irritating to the eyes, nose and throat. This regieranct is somewhat flammable and, when mixed with air the ratio of 16% to 25% gas by volume, will form an explosive mixture. The leaks of this refrigerant may be quickly and easily detected by the use of boring sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphite. This refrigerant attacks copper and bronzed in the presence of a little moisture but does not corrode iron or steel. It presents no special problems in connection temperatures are encountered. Since the refrigerant R-717 is lighter than oil, therefore, its separation does not rate any problem. The excess oil in the evaporator may be removed by opening a valve in the bottom of the evaporator. This refrigerant is used in large

compression machines using reciprocating compressors and in many absorption type systems. the use of this refrigerant is extend silvery foundin cold storage, warehouse plants, ice cram manufacture, ice manufacture, been manufacture, food freezing plants etc.

2. R-729 (Air). The dry air us used as gaseous refrigerant in some compression system, particularly in aircraft air conditioning systems, particularly in aircraft air-conditioning.
3. R-744 (Carbon dioxide). The principal refrigeration use of carbon dioxide is same as that of dry ice. It is non-toxic, non-irritating and non-flammable. The boiling point of this refrigerant is so extremely low (-73.6°C) that at -15°C, a pressure of well over 20.7 bar is enquired to pre cent its evaporation. At a condenser temperature of +30°C, a pressure of approximately 70 bar is required to liquefy the gas. Its critical temperature is 31°C and triple point is -56.6°C. due to its high operating pressure, the compressor of carbon dioxide refrigerator unities very small even for a comparatively large regenerating capacity, however, because of its low efficiency as compared to other common refrigerants, it is seldom in household units, but is used is some industrial applications and aboard ships.
4. R-764 (Sulphur dioxide). The refrigerants are produced by the combustion of sulphur in air. In the form years, it widely used in household and small commercial units. This boiling point of sulphur dioxide is -10°c at atmospheric pressure. The condensing pressure varies between 4.1 bar and 6.2 bar infer normal operating conditions. The latent heat of sulphur dioxide at -15°C is 396 kJ/kg. it is a very stable refrigerant with a high critical temperature and it is non-flammable and non-explosive. It has very unpleasant and irritating odour. This reigigerants is not injurious to food and is used commercially as a ripened and preservative of foods. It is, however, extremely injurious to flower, plants and shrubbery. The sulphur dioxide in its pure steins not corrosive, but when there is moister present, the mixture forms sulphurous accede which is corrosive to steel. Thus it is very important and that the moisture in the refrigerating system be held to a minimum. The sulphur dioxide does not mix readily withoil. Therefore, an oil light than that used with other refrigerants may be used in the compressors. The refrigerant in the evaporator with oil floating on be top has atedecy to have a higher boiling point than that corresponding to its pressure. The mourn evaporator resovercoe this by having the liquid introduced in such a way that the reigigerants is kept agitated while the unit is in operation. The leaks in the system with sulphur dioxide may be easily detected by means of soap solution or ammonia swab. A dense hate smoke forms when sulphur dioxide and ammonia fumes come in contact.
5. R-118 (water). The principle refrigeration use of water is as ice. The high freezing temperature of water limits its use in vapour compression systems. It is use as there fragrant vapour is some absorption systems and in systems with steam jet compressors.

**HYDRO-CARBON REFRIGERANTS**

Most of the hydro-carbon refrigerants are successfully used in industrial and commercial installations. They possess satisfactory thermodynamic properties but are highly flammable and explosive. The various hydra-carbon refrigerants are given in the following table

**Table 3 Hydro-carbon refrigerants**

| **Refrigerant number** | **Chemical name** | **Chemical formula** |
|---|---|---|
| R-170 | Ethane | $C_2 H_2$ |
| R-290 | Propane | $C_3H_3$ |
| R-600 | Butane | $C_4H_{10}$ |
| R-$600_a$ | Isobutene | $C_4H_{10}$ |
| R-1120 | Trichloroethylene | $C_2H_4Cl_3$ |
| R-1130 | Dichloroethylene | $C_2H_4Cl_2$ |
| R-1150 | Ethylene | $C_2H_4$ |
| R-1270 | Propylene | $C_3H_6$ |

## DESIGNATION SYSTEM FOR REFRIGERANTS

The refrigerants are internationally designated as ‘R’ followed by certain numbers such as R-11, R-12, R-114 etc. A refrigerant followed by a two-digit numbers indicates that a refrigerant is derived from methane base while three-digit number represents ethane bases. The numbers assigned to hydro-carbon and halo- and refrigerants have a special meaning. The first digest on the rightist the number of fluorine (F) atoms in the reigigerants. The second digit from the right is one more than the number of hydrogen (H) atoms present. The third digit from the right is one less than the number of carbon (C) atoms, but when this digit is zero, it is omitted. The general chemical formula for the refrigerant, either for methane or ethane base, is given as $C_mH_nCl_pF_q$ in which $n + p + q = 2m + 2$

Where $m$ = Number of carbon atoms,

$n$ = Number of hydrogen atoms,

$p$ = Number of chlorine atoms, and

$q$ = Number of fluorine atoms.

As discussed above, the number of the refrigerants is given by R9m -1) (n + 1) (q). Let us consider the following refrigerants to find its chemical formula and the number.

1. ***Dichloro-difluoro-methane***

We see that in this refrigerant
Number of chlorine atoms, $p = 2$
Number of fluorine atoms $q = 2$
And number of hydrogen atoms, $n= 0$

We know that $n + p + q = 2m +2$
$0 + 2 + 2 = 2m +2$ or $m = 1$

i.e Number of carbon atoms = 1

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

2. ***Dichloro-tetrafluoro-ethane***

We see that in this refrigerant
Number of chlorine atoms, $p = 2$
Number of fluorine atoms, $q = 4$
And number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m +2$
$0 + 2 + 4 = 2m + 2$ or $m=2$

i.e Number of carbon atoms = 2

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

3. ***Dichloro-trifluore-ethane***

We see that in this refrigerant
Number of chlorine atoms, $p = 2$
Number of fluorine atoms, $q = 3$
And number of hydrogen atoms, $n = 1$

We know that $n + p + q = 2m + 2$
$1 + 2 + 3 = 2m +2$ or $m =2$

i.e Number of carbon atoms =2

Thus the chemical formula for Dichloro – trifluore-ethane become $CHCL_2CF_3$ and the number of refrigerant becomes R(2-1) (1+1) (3) or R-123.

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

## SUBSTITUTES FOR CHLORO-FLUORO-CARBON (CFC) REFRIGERANTS

The most commonly used halo-carbon or organic refrigerants are the Chloro-fluoro derivatives of methane (CH4) and ethane ($C_2H_6$). The fully halogenated refrigerants with chlorine (CI) atom in their molecules are referred to as Chloro-fluoro-carbon (CFC) refrigerants. The refrigerants such as R-11, R-12,R-13,R-113,R-114 and R-115 are CFC refrigerants.

The refrigerants which contain hydrogen (H) and ethane ($C_2$ $H_6$). The fully halogenated refrigerants with chlorine (C1) atom in their molecules are referred to as Chloro-carbon (CFC) refrigerants. The refrigerants such as R-11, R-12, R-13, R-13, R-14 and are CFC refrigerants.

The refrigerants which contain hydrogen (H) atoms in their molecule along with chlorine (C1) and fluorine (F) atoms are referred as hydro-Chloro-flour-carbon (HCFC) refrigerants.

The refrigerants which contain no chlorine and fluorine atoms in their molecule are referred to as hydrocarbon (HC) refrigerants. The refrigerants such as R-290, R-600a are HC refrigerants.

It may be notes that the fluorine (F) atom in the molecule of the refrigerants make s them physiologically more favorable. The chlorine (CI) atom in the molecule of the refrigerants is considered to be responsible for the depletion of ozone layer in the upper atmosphere which allow harmful ultra-violet rays from the sun to penetrate through the atmosphere and reach the earth's surface coursing skin cancer. The Chloro-fluoro-carbon (FCF) refrigerants have been linked to the defection of the ozone layer. They have varying degree of *ozone depletion potential (ODP).

In addition to the ozone depletion effect on the environment, the halo-carbon refrigerants have a global warming effect, which may cause serious changes in the environment.

According to an international agreement (Montreal protocol, 1987), the use of halogenated CFC refrigerants that are considered to have high OFP (such as commonly used refrigerants R-11, R-12, R-113, R-114 andR-02) have been phased out. The refrigerant R-22 which is a hydro-chore-fluorne carbon (HCFC) refrigerant is not covered under the original Montreal Protocol as it ODP is one-twentieth of R-11 and R-12. But because of its GWP, it has to be phased out. Nevertheless, R-22 is found to be of greater use these days as it its being employed not only on its existing R-22 applications but also as substitue for R-11 in very large capacity air-conditioning.

The hydrocarbon (HC) and hydro-fluoro carbon (HFC) refrigerants or vide and alternative for fully halogenated CFC refrigerant. Since they contain no chlorine atom at all, therefore they have zero ODP. Even hydro-Chloro-fluoro carbon (HCFC) refrigerants which cantina some chlorines (CI) atoms, but in association with hydrogen) atoms, have much reduced ODP. However, the hydro-fluoro carbon(HFCs), because of their hydrogen (H) content, may be slightly flammable. The degree of flammability depends upon the number of H-atoms in the molecule. The pure hydrocarbons (HCs) are of course, high flammable:

At present, the following substitutes are available:

1. The HCFC refrigerant R-123 ($CF_3$ $CF_3CHCl_2$) in place of R-11 ($CCl_3F$).
2. The HFC refrigerant R-134a ($FG_3CH_2F$) and R-152a ($CH_3CHF_2$) in place of R-12.
3. The HFC refrigerant R-143a ($CH_3CH_3$) and R-125 ($CHF_2CF_3$) in place of R-502 (a mixture of R-22 and R-115).
4. The HC refrigerants, propane i.e R-290 ($C_3H_8$) and is isobutene i.e R-600a ($C_4H_{10}$) may also be used in place of R-12.

## COMPARISON OF REFRIGERANTS

There is no such refrigerant (i.e. ideal refrigerants.) which can be used all operating conditions. The characteristic of some refrigerants. make them suitable for use with reciprocating compressors and other refrigerants. best suited to centrifugal or rotary compressors. Therefore in order to select correct refrigerants. it is necessary that it should satiety those properties which make it ideal to be used for the application. We shall now discuss the thermodynamic, chemical and physical properties of some important refrigerants.

## THERMODYNAMIC PROPERTIES OF REFRIGERANTS

The thermodynamic properties of refrigerants are discussed, in detail, as follows:

**1.** Boiling temperature. The boiling temperature of the refrigerants. at atmospheric pressure should be low. It the bolting temperature of the refrigerants. is high at atmospheric pressure, the cool presser should be opened at high vacuum. The high boiling temperature refuses the capacity and operating cost of the system. The following table shows the boiling temperatures at atmospherics pressure of some commonly used refrigerants.

**Table 4 Boiling temperatures**

| Refrigerant | Boiling temperature (°C) at atmospheric pressure |
|---|---|
| R-11 | + 23.77 |
| R-12 | - 29 |
| R-21 | + 9 |
| R-22 | -14 |
| R-30 | + 39.8 |
| R-40 | -23.7 |

| | |
|---|---|
| R-113 | + 47.6 |
| R-114 | +3.6 |
| R-123 | + 27.85 |
| R-134a | -26.15+ |
| R-717 | -33.3 |
| R-744 | -73.6 |
| R-764 | -10 |

2. Freezing temperature. The freezing temperature of a refrigerant should be well below the operating exploratory temperature. Since the freezing temperature of most of the refrigerants are below- 35°C, therefore this property is taken into consideration only in low temperature operation. The following table shows the freezing temperatures of some common refrigerants.

**Table 5 Freezing temperature**

| Refrigerant | Freezing temperature (°C) |
|---|---|
| R-11 | -111 |
| R-12 | -157.5 |
| R-21 | -135 |
| R-22 | -160 |
| R-30 | -96.6 |
| R-40 | -97.5 |
| R-113 | -35 |
| R-114 | -94 |
| R134a | -101 |
| R-717 | -77.8 |
| R-744 | -56.7 |
| R-764 | -75.6 |

3. Evaporator and condenser pressure.
   Both the evaporation (low side) and condensing (high side) pressure should be positive (i.e above atmospheric) and it should be as bear to the atmospheric pressure as possible. The positive pressures are necessary in order to prevent leakage of air and moisture into the refrigerating system. It also permits easier dictation of leaks. Too high evaporating and condensing pressures (above atsmpsheric) would require stronger refrigerating equipment (i.e. compressors, evaportoator and condenser) resulting higher initial coast. The following table shows the evaporating and condensing pressure, and compression ratio for various refrigerant when operating on the standard cycle of -15°C evaporator temperature and +30°C condenser temperature.

4. **Critical temperature and pressure.** The critical temperature of a refrigerant the highest temperature at which it can be condensed to a liquid, regardless of a higher pressure. It should be above the highest condensing temperature that might be encountered. It the critical temlperarue of results. The following table shows the critical temperature, the excessive power consumption refrigerants. The critical temperature for most of the common reigigerants is well above the normal condensing temperature with the exception of carbon dioxide (R-744) whose critical temperature is 31°C.
5. **Coefficient of performance and power requirements.**
   For an ideal refrigerant operating between -15°C evaporator and 30°C condenser temperature, the theoretical coefficient of performance for the reversed Carnot cycle is 5.74. the following table shows thecaules of theoretical coefficient ofperformance and power per tonne of refrigeration for some common refrigerants operating between -15°CC evaporator temperature and 30°C condenser temperature.
6. **Latent of heat of vaporization.**
   A refrigerant should have a high latent heat of vapors ton at the evaporator temperature. The high latent heat results high refrigerating effect per kg of regirgernat circulated with reduces the mass of refrigerant to be circulated pep tonne of refrigerating. Table 8.10 shows the refrigerating effect for the common refrigerants operating between -15°C evaporator temperarue and 30°C condenser temperature. It also shows the lateen theta, mass of refrigerant circulated per tonne of refrigeration and the volume of the liquid refrigerant per tonne of reiteration.
7. **Specific volume.**
   The specific volume of the refrigerant vapour at evaporator temperature (i.e volume of suction vapour to the compressor) indicated the theoretical displacement of the compressor. The reciprocating compressor are used with referents having high pressures and low volumes of the suction vapour. The centrifugal or turbo compressors are used with refrigerants having low pressure and high volumes of the suction vapour. The rotary compressor are used with refrigerants having intermediate pressures and volumes of the suction vapour. Table 8.11 shows the specific volume of the refrigerant vapor and the optical piston displacements for various refrigerants.

## CHEMICAL PROPERTIES OF REFRIGERANTS

1. **Flammability.** We have already discussed that hydro-carbon refrigerant such as ethane, propane etc, are highly flammable. Ammonia is also somewhat flammable and becomes explosive when mixed with air the radio of 16 to25 per cent of gas by volume. He halo-carbon refrigerants are neither flammable nor explosive.
2. **Toxicity.** The toxicity of referent may be prime or secondary importance, depending upon the application. Some non-toxic refrigerants. (i.e all fluorocarbon refrigerants) when mixed with certain portage of air certain percentage of air become toxic

3. **Solubility of water.** Water is only slightly soluble in R-12. At -18°C, it will hold six parts per million by weight. The solution formed is very slightly corrosive to any of the common medial. The solubility of water with R-22 is by a ratio of 3 to 1. It more water is preset than can be dissolved by the refrigerant, the ice will be formed which chokes the expansion valve or capillary tube used for throttling the system. This may be avoided by the proper dehydration of the refrigerating unit before charging and by the use of sepia gel drier of the liquid line. Ammonia is highly soluble in water. Due to this reason, a wetted cloth is put at the point of leak to avoid harm to the persons working in ammonia refrigerating plants.
4. **Miscibility.** The ability of a refrigerant to mix with oil is called miscibility. This property of refrigerant is considered be to secondary factor in the selection of are grievant. The degree of miscibility depends upon the temperature of the oil and presser of the refrigerating vapour. The dioxide and methyl chloride are relatively non-miscible.
4. **Effect on perishable materials.** The refrigerants used in cold stronger plant and in domestic refrigerators should be such that in case of leaker, it should have no effect on the perishable materials. The Freon group of refrigerants has no effect upon dairy products, meats vegetables, flowers and ford. There will be no changeling colour, tests or texture of the material when exposed to Freon. Methyl chloride vapors have no effect upon furs, flowers, eating, foods or drinking beverages. Sulphur dioxide destroys flowers, plant and furs, but it does not affect foods. Ammonia dissolves easily in water and becomes alkaline in nature. Since most fruits and vegetables are acidic in nature, therefore ammonia reacts with these products and spills the taste.

## PHYSICAL PROPERTIES OF REFRIGERANTS

1. **Stability and Inertness.** An ideal refrigerant should not decompose at any temperature normally encountered in there refrigerating system. It should not form higher boiling point liquids or solid substance through polymerization. Some refrigerants disinter grate forming non-condensable gases which causes high condensing pressure and vapour lock. The disintegration of refrigerant may be due to reaction with melts. In order to avoid this, a refrigerant should be inert with respect all materials used in refrigents system. The freon group of refrigerants is stable up to a temperature of 535°C. Above this temperature, it decomposes and forms corrosive and poisonous products. The freon refrigerants not used with rubber gaskets as if acts as a solvent with rubber, since sulphur dioxide does not decompose below 1645°C, therefore are One of the most stable refrigerants.
2. **Corrosive property.**

The corrosive property of a refrigerant must be taken into consideration while selecting the refrigerants. The freon group of refrigerants are non- corrosive with practically all metals. Ammonia is used only with iron or steel. Sulphur dioxides is non-corrosive to al metals in the absence of water because suphur dioxide reacts with water and forms sulphuric acid.

3. Viscosity. The refrigerant in the liquid and vapour states should have low viscosity. The low viscosity of the refrigerant is desirable because the pressure drops in passing through liquid and such tin lines are small. The heat transfer through condenser and evaporator is improved at low viscosities. The following table shows the victories (in centipoises) at atsmpsheric pressure for the common refrigerants.

4. **Thermal conductivity.**The refrigerant in the liquid and vapour states should have high thermal conductivity. This property is required in finding the health transfer coefficients in evaporators and condensers. Table 8.14 shows the thermal conductivities of common refrigerants.
5. **Dielectric strength**. The dialectic strength of a refrigerant is important in hermetically sealed units in which the eclectic motor is exposed the refrigerant. The relative dielectric strength of the refrigerant is the ratio of the dielectric strength of nitrogen and the refrigerant vapour mixture shows the alive dielectric strengths of common refrigerants.

6. Leakage tendency. The leakage tendency of a refrigerant should be low. It there is leakage of refrigerants, it should be easily detectable. The leakage occurs due to opening in the joints of flaws in material, used for constructing. Since the fluorocarbon refrigerants are collarless, therefore, their leakage will increase the operating cost. The ammonia leakage is easily detected due to its pungent odour. The leakage of fluorocarbon refrigerants may be detected by soap solution, a halide torch or and electronic leak detector. The latter is generally used in big refrigerating plants. The ammonia leakage is detected by using burring sulphur candle which in the presence of ammonia forms while fumes of ammonium sulphite.
7. **Cost.**The cost of refrigerant is not so important in small refrigerating units but is very important in high capacity refrigerating systems like industrial and commercial. The ammonia, being the cheapest, is widely used in large industrial plants such as cold storage and ice plants. The refrigerant R-22 is costlier then refrigerant R-12. The cost of losses due to leakage is also important.

## SECONDARY REFRIGERANTS – BRINES

Brines are secondary referents and are generally used where temperatures are required to be maintained below the freezing point of water i.e 0°C. In case the temperature involved is above the freezing point of water (0°C), then water is commonly used as a secondary refrigerant.

Brine is a solution of salt in water. It may be noted that when salt is mixed in water, than the freezing temperature of the solution becomes lower than the of the water. This is due to the fact that the salt while dissolving in water takes of it its latent health form the solution and cods it bellwether freezing point of water. The mass of the salt in resolution expressed as the percentage of the mass of the solution is known as concentration of the solution. As he concentration of the solution increases, it's the salt is increased beyond a certain point, the freezing point decreases. But it the concentration of the salt is increased beyond a certain point, the freezing point increases instead of decreasing. A which the freezing point decreases. But if the concentration of the salt is increase beyond a certain point, the freezing point increases of decreasing. The point at which the freezing temperature is minimum, is known aesthetic temperature and the conception at this points known as eutectic temperature and the concentration at this point is known as eutectic concentration. Brine used in a particular application should have a concentration fort which the freezing point of the brined is at last 5°C to 8°C lower than the than brine temperature required.

The brine commonly used are calcium chloride ($CaCl_2$), sodium chloride i.e common salt (NaCl) and glycols such as ethylene glycol, propylene glycol etc.

The calcium chloride brine has the eutectic temperature of -55°C at salt concentration of 30% by mass. This brine is primarily used where temperatures below- 18°C are required. It is generally used in industrial process cooling and product freezing. The chief disadvantages of calcium chloride brine are its dehydrating effect and its tendency to impart a bitter taste to food products.

The sodium chloride brine has the eutectics temperature of -21.1°C at salt concentration of23% by mass. This brine is used in chilling and freeing of meat and fish.

Both of the above two brines are corrosive in nature for metallic containers which put limitation on their use. Also the thermal properties of the above tow brines are less satisfactory.

Other water soluble compounds known as antifreeze are also used for decreasing the freezing point of water for certain refrigeration used. Ethylene and propylene's glycol have a number of good properties. Since they are non-corrosive and non-elecolytic even in the in the presence of water, therefore, these brines are most extensively used as antifreeze elements. The following table shows typical applications of various brines.

# UNIT II

# OTHER REFRIGERATION SYSTEMS

## 2.1 VAPOUR ABSORPTION REFRIGERATION SYSTEMS

### 2.10 Introduction

The vapour absorption refrigeration system is one of the oldest methods of producing refrigerating effect. The principle of vapour absorption was first discovered by Michael Faraday in 1824 while performing a set of experiments to liquefy certain gases. The first vapour absorption refrigeration machine was developed by a French scientist, Ferdinand Carre, in 1860. This system may be used in both the domestic and large industrial refrigerating plants. The refrigerant, commonly used in a vapour absorption system, is ammonia.

The vapour absorption system uses heat energy, instead of mechanical energy as in vapour compression systems, in order to change the conditions of the refrigerant required for the operation of the refrigeration cycle. We have discussed in the previous chapters that the function of a compressor, in a vapour compression system, is to withdraw the vapour refrigerant from the evaporator. It then raises its temperature and pressure higher than the cooling agent in the condenser so that the higher pressure vapors can reject heat in the condenser. The liquid refrigerant the condenser is now ready to the evaporator conditions again.

In the vapour absorption system, the compressor is replaced by an absorber, a pump, a generator and a pressure reducing value. These components in vapour absorption system perform the same function as that of a compressor in vapour compression system in this system, the vapour refrigerant from the evaporator is drawn into an absorber where it is absorbed by the weak solution of the refrigerant forming a strong solution. This strong solution is pumped to the generator where it is heated by some external source. During the heating process, the vapour refrigerant is driven off by the solution and enters into the condenser where it is liquefied. The liquid refrigerant then flows into the evaporator and thus the cycle is completed.

### 2.11 Simple vapour absorption system

The simple vapour absorption system, as shown in fig.2.1, consists of an absorber, a pump, a generator and a pressure reducing value to replace the compressor of vapour compression system. The other components of the system are condenser, receiver, expansion value and evaporator as in the vapour compression system.

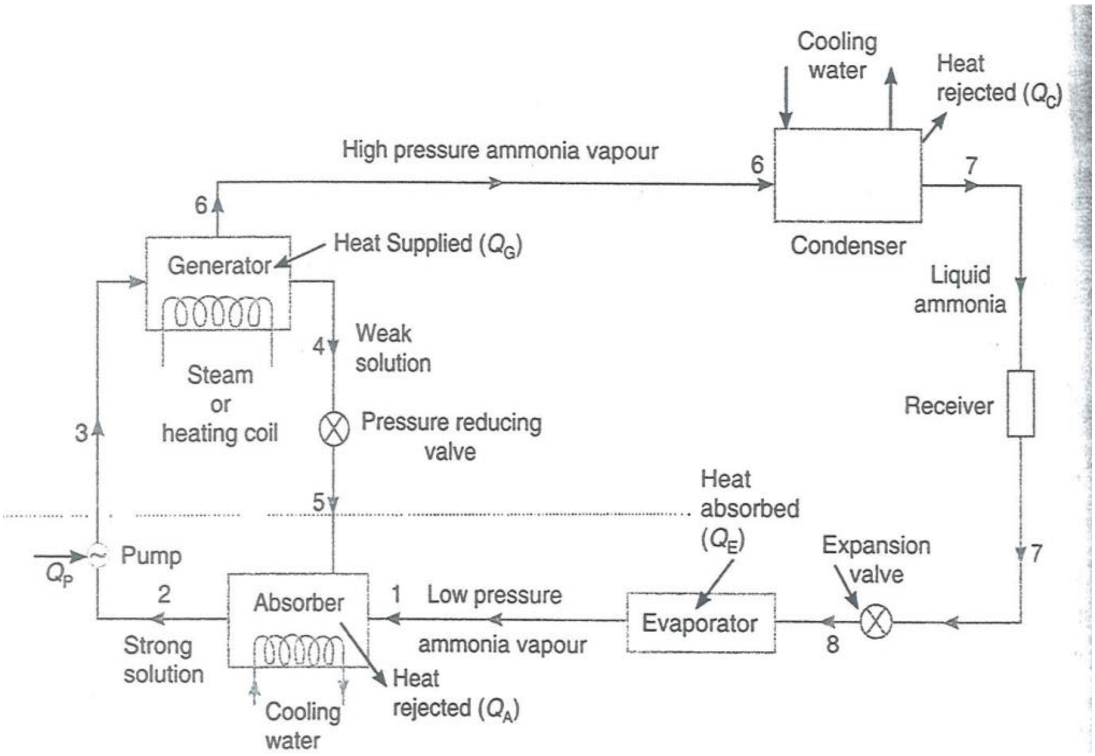


**Figure 2.1** Simple vapour absorption system

In this system, the low pressure ammonia vapour leaving the evaporator entres the absorber where it is absorbed by the cold water in the absorber. The water has the ability to absorb very large quantities of ammonia vapour and the solution, thus formed, is known as aqua-ammonia. The absorption of ammonia vapour in water lowers the pressure in the absorber which in turn draws more ammonia vapour from the evaporator and thus raises the temperature of solution. Some form of cooling arrangement (usually water cooling) is employed in the absorber to remove the heat of solution evolved there. This is necessary in order to increase the absorption capacity of water, because at higher temperature water absorbs less ammonia vapour. The strong solution thus formed in the absorber is pumped to the generator by the liquid pump. The pump increase the pressure of the solution up to 10 bar.

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

### 2.12 Practical vapour absorption system

The simple absorption system as discussed in the previous article is not very economical. In order to make the system more practical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in fig. 2.2. These accessories help to performance and working of the plant, as discussed below:

1. **Analyser.** When ammonia is vaporized in the generator, some water is also vaporized and will flow into the condenser, along with the ammonia vapours in the simple system. If these unwanted water particles are not removed before entering into the condenser, they will enter into the expansion value where they freeze and choke the pipeline. In order to remove these unwanted particles flowing to the condenser, an analyser is used. The analyser may be built as an intergral part of the generator or made as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier are introduced at the top of the analyser and flow downward over the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator. The vapour is cooled and most of the water vapour condenses, so that mainly ammonia vapour (approximately 99%) leaves the top of the analyser. Since the aqua is heated by the vapour, less external heat is required in the generator.
2. **Rectifier.** In case the water vapours are not completely removed in the analyser, a closed type vapour cooler called rectifier (also known as dehydrator) is used. It is generally water cooled and may be of the double pipe, shell and coil or shell and tube type. Its function is to cool further the ammonia vapours leaving the analyser so that remaining water vapours are condensed. Thus, only dry or anhydrous ammonia vapours flow to the condenser. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

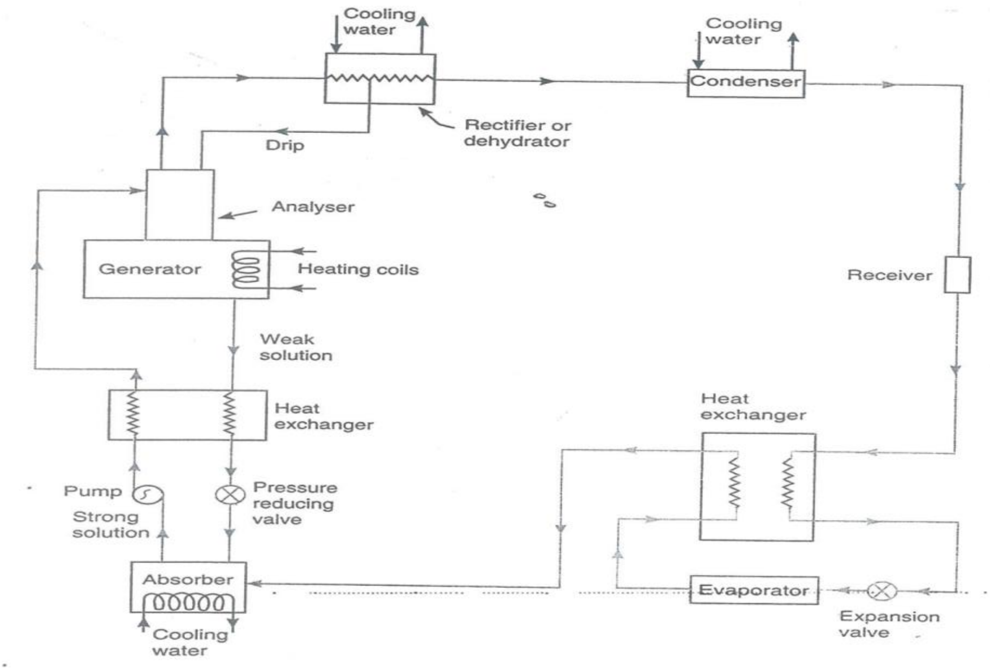


**Figure 2.2** Practical vapour absorption systems

3. **Heat exchangers.** The heat exchanger provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to the generator and the amount of cooling required for the absorber. Thus the economy of the plant increases. The heat exchanger provided between the condenser and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub-cooled by the low temperature ammonia vapour from the evaporator as shown in fig. 2.2. this sub-cooled liquid is now passed to the expansion value and then to the evaporator.

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

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

## 2.13 Thermodynamic requirements of refrigerant-absorbent mixture

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

1. Solubility requirement. The refrigerant should have more than Raoult's law solubility in the adsorbent so that a strong solution, highly rich in the refrigerant, is formed in the absorber by the absorption of the refrigerant vapour.
2. Boiling points requirement. There should be a large difference in the normal boiling points of the two substances, at least 200˚C, so that the absorbent exerts negligible vapour pressure at the generator temperature. Thus, almost absorbent-free refrigerant is boiled off from the generator and the absorbent alone returns to the absorber.

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

   (a) It should have low viscosity to minimize pump work.
   (b) It should have low freezing point.
   (c) It should have good chemical reactions of all kinds such as decomposition; polymerization, corrosion etc. are to be avoided.

## 2.14 Properties of ideal refrigerant-absorbent combination

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

1. The refrigerant should have high affinity for the absorber at low temperature and less affinity at high temperature.

2. The combination should have high degree of negative deviation from Raoult's law.

3. The mixture should have low specific heat and low viscosity.

4. The mixture (solution) should be non-corrosive.

5. The mixture should have a small heat.

6. The mixture should have low freezing point.

7. There should be a large difference in normal boiling points of the refrigerants and the absorbent.

(a) Ammonia-water combination; and (b) lithium-bromide water combination.

The ammonia-water absorption system finds a significant place in large tonnage industrial applications

## 2.15 Comparison of refrigerant-liquid absorbent combination (say $NH_3$-water) with refrigerant-solid absorbent combination (say $NH_3$-$CaCl_2$)

The solid absorbent system has the following two main advantages:

1. The amount of refrigerant cycled in relation to the amount of absorbent is larger for the solid absorbent than for the liquid absorbent. Therefore, in solid absorbent system, the thermal capacity of the salt contributes little, and hence heat required in the generator is less as compared to that for water in the aqua-ammonia system.

2. The solid absorbent system is extremely robust, not only in mechanical sence, but also with respect to adverse operating conditions. The performance is remarkably insensitive to changes in both condensing and evaporating temperatures. However the solid-absorption cycle has one major drawback. The heat of reaction is large compared with those found for liquid absorbent, approximately twice the latent heat of vaporization.

However the C.O.P. for solid absorption cycle is still higher than that of liquid absorption cycle, particularly when the difference between the condensing temperature and the evaporating temperature is large.

The solid-absorbent combination is ideally suited for intermittent operation on solar energy.

## 2.16 Advantages of vapour absorption refrigeration system over vapour compression refrigeration system.

Following are the advantages of vapour absorption system over vapour compression system.

1. In the vapour absorption system, the only moving part of the entire system is a pump which has a small motor. Thus, the operation on this system is essentially quiet and is subjected to little wear.

   The vapour compression system of the same capacity has more wear, tear and noise due to moving parts of the compressor.

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

## 2.17 Coefficient of performance of an ideal vapour absorption refrigeration system

(a) The heat ($Q_G$) is given to the refrigerant in the generator.
(b) The heat ($Q_C$) is discharged to the atmosphere or cooling water from the condenser and absorber.
(c) The heat ($Q_E$) is absorbed by the refrigerant in the evaporator, and
(d) The heat ($Q_P$) is added to the refrigerant due to pump work.

Neglecting the heat due to pump work (Qp), we have according to First Law of Thermodynamics,

$$Q_c = Q_G + Q_E$$

Let $T_G$ = Temperature at which heat ($Q_G$) is given to the generator,

$TC$ = Temperature at which heat ($Q_C$) is discharged to atmosphere or cooling water from the condenser and absorber, and

$TE$ = Temperature at which heat ($QE$) is absorbed in the evaporator.

Since the vapour absorption system can be considered as a perfectly reversible system, therefore the initial entropy of the system must be equal to the entropy of the system after the change in its condition.

$$\frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_C}{T_C} \qquad \text{... (ii)}$$

$$= \frac{Q_G + Q_E}{T_C}$$

$$= \frac{Q_G}{T_G} - \frac{Q_G}{T_C} = \frac{Q_E}{T_C} - \frac{Q_E}{T_E}$$

$$Q_G \left( \frac{T_C - T_G}{T_G \times T_C} \right) = Q_E \left( \frac{T_E - T_C}{T_C \times T_E} \right)$$

$$Q_G = Q_E \left[ \frac{T_E - T_C}{T_C \times T_E} \right] \left[ \frac{T_G \times T_C}{T_C - T_G} \right]$$

$$= Q_E \left[ \frac{T_C - T_E}{T_C \times T_E} \right] \left[ \frac{T_G \times T_C}{T_G - T_C} \right]$$

$$= Q_E \left[ \frac{T_C - T_E}{T_E} \right] \left[ \frac{T_G}{T_G - T_C} \right]$$

Maximum coefficient of performance of the system is given by

$$(C.O.P)_{\max} = \frac{Q_E}{Q_G} = \frac{Q_E}{Q_E \left( \frac{T_C - T_E}{T_E} \right) \left( \frac{T_G}{T_{G-} T_C} \right)}$$

$$= \left( \frac{T_E}{T_C - T_E} \right) \left( \frac{T_G - T_C}{T_G} \right)$$

It may be noted that,

1. The expression $\frac{T_E}{T_C - T_E}$ is the C.O.P. of a Carnot refrigerator working between the temperature limits of $T_E$ and $T_C$
2. The expression $\frac{T_G - T_C}{T_G}$ is the efficiency of a Carnot engine working between the temperature limits of $T_G$ and $T_C$

Thus an ideal vapour absorption refrigeration system may be regarded as a combination of a Carnot engine and a Carnot refrigerator to produce the desired refrigeration effects as shown in Fig.2.3.

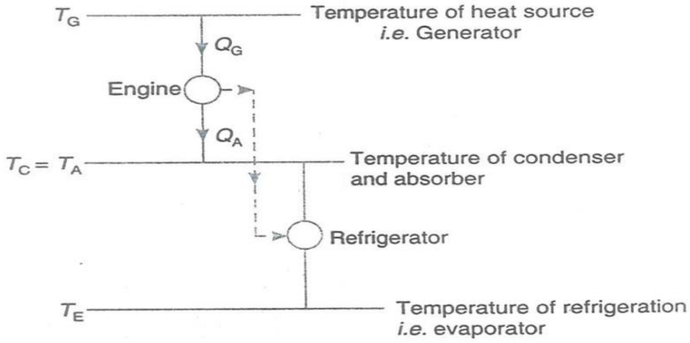


**Figure 2.3** Representation of vapour absorption refrigeration system

The maximum C.O.P. may be written as

$$(C.O.P.)_{\max} = (C.O.P)_{carnot} \times \eta_{carnot}$$

In case the heat is discharged at different temperatures in condenser and absorber, then

$$(C.O.P.)_{\max} = \left[ \frac{T_E}{T_C - T_E} \right] \left[ \frac{T_G - T_A}{T_G} \right]$$

## 2.18 Domestic Electrolux (ammonia hydrogen) refrigerator

The domestic absorption type refrigerator was invented by two Swedish engineers, Carl munters and baltzer von platan, in 1925 while they were studying for their undergraduate course of royal institute of technology in Stockholm. The idea was first developed by the ‘Electrolux company’ of Luton, England.

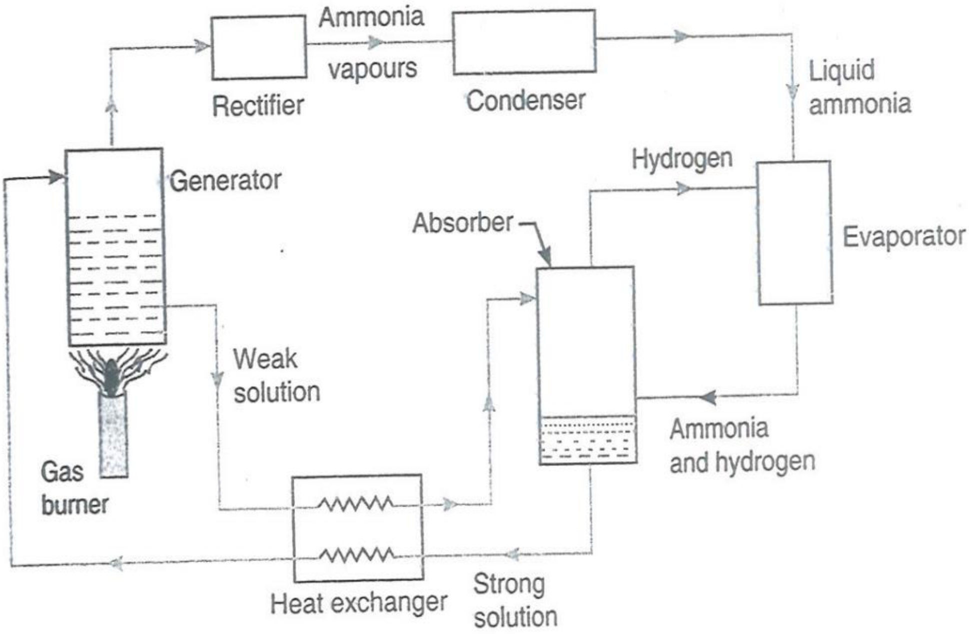


**Figure 2.4** Domestic Electrolux type refrigerator

This type of refrigerator is also called three-fluid absorption system. The main purpose of this system is to eliminate the pump so that in the absence of moving parts, the machine becomes noiseless. The three fluids used in this system are ammonia is used as a refrigerant because it possesses not of the desirable properties. It is toxic, but due to absence of moving parts, there is very little chance for the leakage and the total amount of refrigerant used is small. The hydrogen, being the lightest gas, is used to increase the rate of evaporation of the liquid ammonia passing through the evaporator. The hydrogen is also non-corrosive and insoluble in water. This is used in the low-pressure side of the system. The water is used as a solvent because it has the ability to absorb ammonia readily.

The strong ammonia solution from the absorber through heat exchanger is heated in the generator by applying heat from an external source, usually a gas burner. During this heating process, ammonia vapours are removed from the solution and passed to the condenser. A rectifier or a water separator fitted before the condenser removes water vapour carried with the ammonia vapours, so they dry ammonia vapours are supplied to the condenser. These water vapours, if not removed, will enter into the evaporator causing freezing and choking of the machine. The hot weak solution left behind in the generator flows to the absorber through the heat exchanger. This hot weak solution while passing through the exchanger is cooled. The heat removed by the weak solution is utilized in raising the temperature of strong solution passing through the heat exchanger. In this way, the absorption is accelerated and the improvement in the performance of a plant is achieved.

The ammonia vapours in the condenser are condensed by using external cooling souce. The liquid refrigerant leaving the condenser flows under gravity to the evaporator where it meets the hydrogen gas. The hydrogen gas which is being fed to the evaporator permits the liquid ammonia to evaporation at a low pressure and temperature according to Dalton's principle. During the process of evaporation the ammonia absorbs latent heat from the refrigerated space and thus produces cooling effect.

The mixture of ammonia vapour and hydrogen is passed to the absorber where ammonia absorbed in water while the hydrogen rises to the top and flow a back to the evaporator. This completes the cycle. The coefficient of performance of this refrigerator is given by:

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

## 2.19 Lithium Bromide Absorption Refrigeration System

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

Fig. 2.5 shows a lithium bromide vapour absorption system. In this system, the absorber and the evaporator are place in one shell which operates at the same low pressure of the systems. The generator and condenser are placed in another shell which operates at the same high pressure of the system. The principle of operations of this system is discussed below:

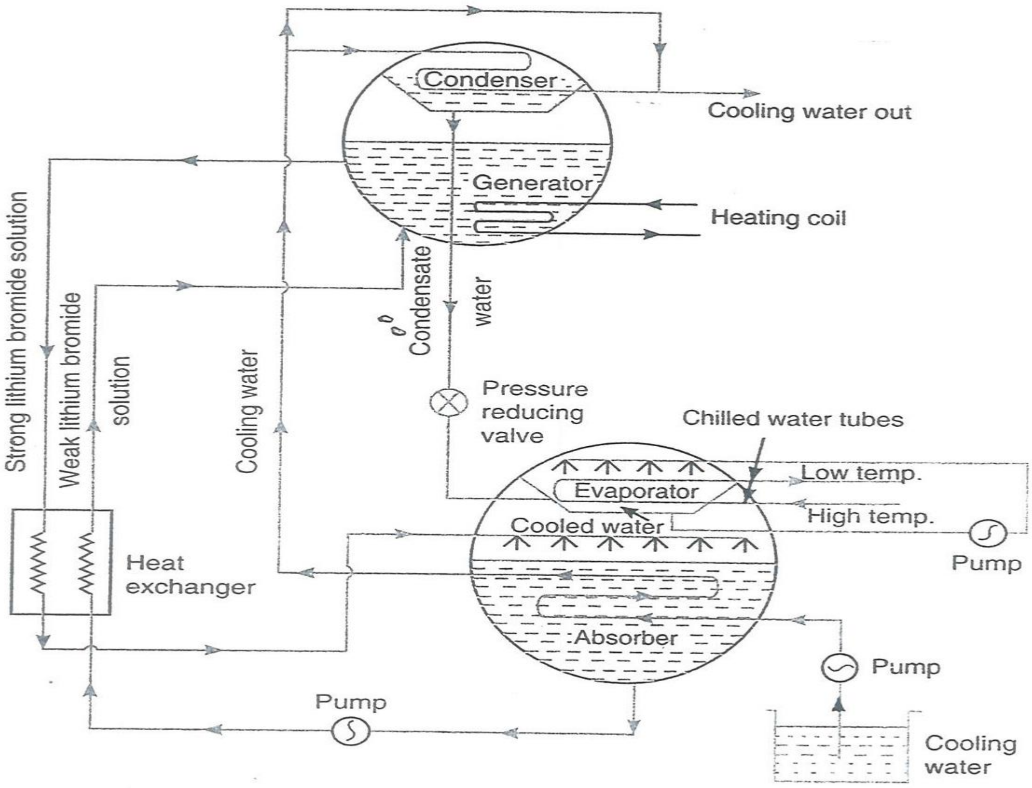


**Figure 2.5** Lithium-Bromide absorption refrigeration system

## 2.2 STEAM JET REFRIGERATION SYSTEM

### 2.20 Introduction

The stem jet refrigeration system (also known as ejector refrigeration system) is one of the oldest methods of producing refrigerating effort. The basic components of this system are an evaporator, components device, a condenser, and a refrigerant control device. This system employs a steam ejector or booster (instead of mechanical compressor) to compress the refrigerant to the required condenser pressure level. In this system, water is used as the refrigerant. Since the freezing point of water is 0ºC. The stem jet refrigeration system is widely used in food processing plants for pre cooling of vegetables and concentrating fruit juices, gas plants, paper mills, breweries etc.

### 2.21 Principle of Steam Jet Refrigeration System

The boiling point of a liquid changes with change in external pressure. In normal conditions, pressure exerted on the surface of any liquid is the atmospheric pressure. If this atmospheric pressure is reduced on the surface of a liquid, by some means, then the liquid will star boiling at lower temperature, because of reduced pressure. This basic principle of boiling of liquid at lower temperature by reducing the pressure on its surface is used in steam jet refrigeration system.

The boiling point of pure water at standard atmospheric pressure of 760 mm of Hg (1.013 bar) is 100ºC. It may be noted that water boils at 12ºC if the pressure on the surface of water is kept at 0.014 bar and at 7ºC if the pressure on the surface of water is 0.01 bar. The reduced pressure on the surface of water is maintained by throttling the steam through the jets or nozzles.

### 2.22 Water as a Refrigerant

We have already discussed that water is used as a reagent is steam jet refrigeration system, and the cooling effect is produced by the continuous vaporization of a part of water in the evaporator at reduced pressure. When water is to be chilled from 10ºC to 5ºC, at least one per cent of water flowing through the evaporator must be vaporized.

Let m= Mass of water in the evaporator in kg,

S = Specific heat of water = = 4.2kj/kg ºC,

$H_{fg}$ = Latent heat of vaporization of water at some reduced pressure in KJ/kg, and

QR = Heat removed from the water kJ/Kg.

Consider that one per cent of m kg of water is evaporated by throttling the steam through the nozzle at some reduced pressure (say at a pressure of 0.085 bar). Thus, the total heat removed by this one per cent of evaporated water.

$$= \frac{m}{100} \times h_{f8}$$

- Fall in temperature of remaining water will be

$$\Delta T_F = \frac{q_R}{\left(m - \frac{m}{100}\right)s} = \frac{h_{fg}}{\left(m - \frac{m}{100}\right)s} \ldots (\because q_8 = h_{fg})$$

Now for a mass, m = 100Kg and $h_{fg}$ = 2400.5 kJ/kg at some reduced pressure (at a pressure of 0.085 bar, from steam tables).

$$\Delta T_F = \frac{2400.5}{\left(100 - \frac{100}{100}\right)4.2} = 5.77^{\circ}C$$

### 2.23 Working of steam Jet Refrigeration System

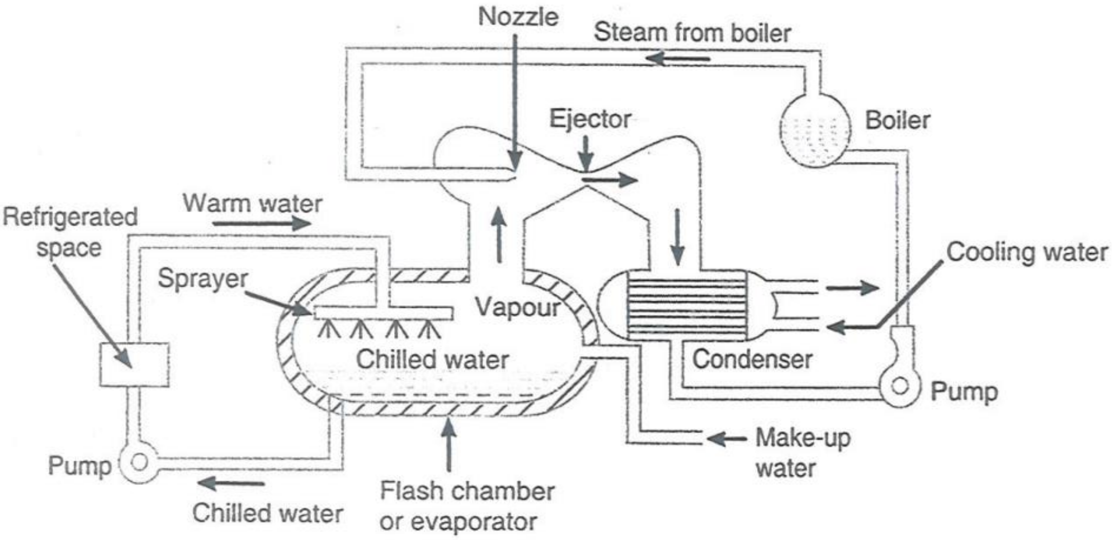


**Figure 2.5** Steam jet refrigeration system

The main components of the steam jet refrigeration system, as shown in Fig. 2.5, are the flash chamber or evaporator, steam nozzles, ejector and condenser.

The flash chamber or evaporator is a large vessel and is heavy insulated to avoid the rise in temperature of water due to high ambient temperature. It is fitted with perforated pipes for spraying water. The warm water coming out of the refrigerated space is sprayed into the flash water chamber where some of which is converted into vapors after absorbing the latent heat, thereby cooling the rest of water.

The high pressure steam from the boiler is passed through the steam nozzles thereby increasing its velocity. This high velocity steam in the ejector would entrain the water vapors from the flash chamber which would result in further formation of vapors. The mixture of steam and water vapor passes through the venture-tube of the ejector and gets compressed. The temperature and pressure of the mixture rises considerably and fed to the water cooled condenser where it gets condensed. The condensate is again fed to the boiler as feed water. A constant water level is maintained in the flash chamber and any loss of water due to evaporation is made up from the make-up water line.

### 2.24 Steam Ejector

The steam ejector is one of the important components of a steam jet refrigeration system. It is used to compress the water vapors' coming out of the flash chamber. It uses the energy of fast moving jet of steam to entrain the vapors' from the flash chamber and then compress it. The essential components of a steam ejector are shown in Fig. 2.6.

The high pressure steam from the boiler (generally called primary fluid or motive steam) expands while flowing through the convergent divergent nozzle. The expansion causes a very low pressure and increases steam velocity. The steam attains very high velocities in the range of 1000m/s to 1350 m/s. The nozzles are designed for lowest operating pressure ratio between nozzle throat and exit. The nozzle pressure ratio of less than 200are undesirable because of poor ejector efficiency when operating at low steam pressure.

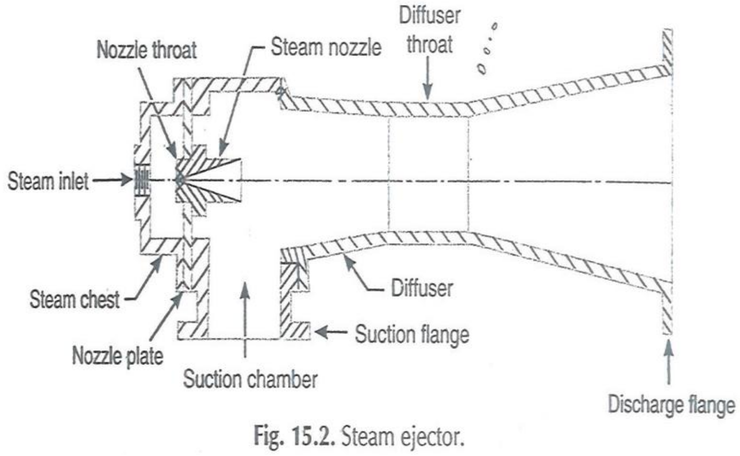
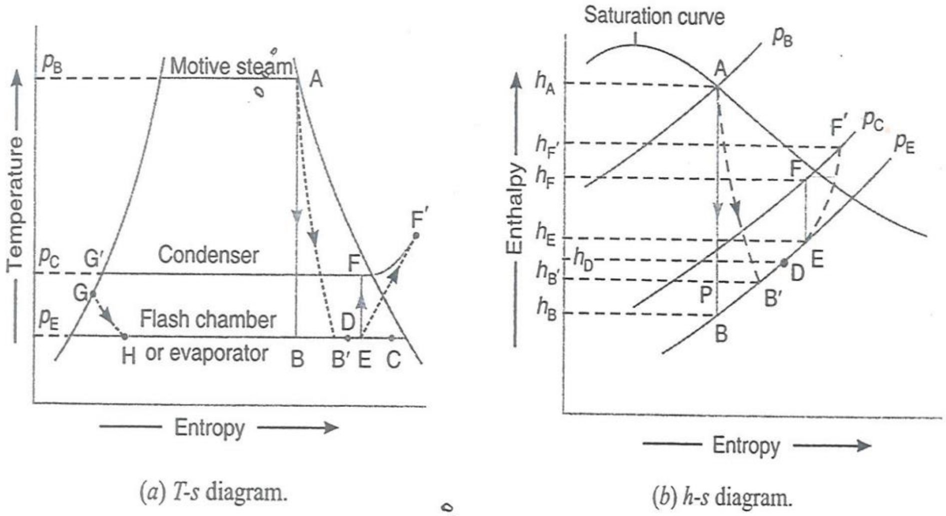


**Figure 2.6** Steam ejector

The water vapor from the flash chamber are entrained by the high velocity steam and both are mixed in the mixing is complete. This supersonic steam gets a normal shock, in the constant area throat of the diffuser. This results in the rise of pressure and subsonic flow. The function of the diverging portion of the differ user is to recover the velocity head as pressure head by gradually reducing the velocity.

### 2.25 Analysis of steam jet refrigeration system

The temperature- energy (T-s) and enthalpy-entropy (h-s) diagrams for a steam jet refrigeration system are shown in Fig. 2.7 (a) and (b) respectively.



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

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

**Figure 2.7** T-S and H-S diagram

The point A represents the initial condition of the motive steam before passing through the nozzle and the point B is the final condition of the steam, assuming isentropic expansion. The point C represents the initial condition of the water vapor in the flash chamber or evaporator and the point E is the condition of the mixture of high velocity steam from the nozzle and the entrained water vapor before compression. Assuming isentropic compression, the final condition of the mixture discharged to the condenser is represented by point F. The condition of motive steam just before mixing with the water vapor is shown at point D. The make-up water is supplied at point G whose temperature is slightly lower than the condenser temperature and is throttled to point H in the flash chamber.

## 2.26 Efficiencies used in steam Jet Refrigeration System

The various efficiency used in steam jet refrigeration system are discussed below:

1. **Nozzle efficiency.** It is defined as the ratio of actual enthalpy drop to the isentropic enthalpy drop of the motive steam passing through the nozzle. Mathematically, nozzle efficiency,

$$\eta_N = \frac{Actualenthalpydrop}{Isentropicenthalpydrop} = \frac{AP}{AB} = \frac{h_A - h_B}{h_A - h_B}$$

The nozzle efficiency may vary from 85 to 90 per cent.

2. **Entrainment efficiency.** The water vapor formed in the flash chamber or evaporator comes out with a very low velocity as compared to the velocity of the steam (V) coming out of the nozzle which is given by

$$*V = \sqrt{2000(h_A - h_{B'})} = 44.72\sqrt{h_A - h_{B'}}$$

The expression ($h_A$-$h_B$')represents the kinetic energy of the motive steam. This kinetic energy gives the required momentum to the water vapors coming out of the flash chamber by the high velocity steam is called entrainment of vapor. During the entrainment of water vapor from the flash chamber, the motive steam loses some of its kinetic energy. This process of entrainment is inefficient and part of the original motive force available for compression is reduced. This is taken into consideration by a factor known as entrainment efficiency. Mathematically, entrainment efficiency,

$$\eta_E = \frac{h_A - h_D}{h_A - h_{B'}}$$

The entrainment efficiency may be taken as 65 percent.

**3. Compression efficiency.** It is defined as the ratio of the isentropic enthalpy increase to the actual enthalpy increase required for the compression of the mixture of motive steam and the water vapours, in the diffuser. Mathematically, compression efficiency.

$$\eta_C = \frac{Isentropicenthaplyincrease}{Actualenthalpyincrease} = \frac{h_F - h_E}{h_{F'} - h_E}$$

The compression efficiency may be taken as 75 to 80 per cent.

## 2.27 Mass of Motive Steam Required

According to the law of conservation of energy, the available energy for compression must be equal to the energy required for compression

Let $m_S$ = Mass of motive steam supplied in kg/min,

$m_v$ = Mass of water vapours formed from the flash chamber or evaporator in kg/min

m = Mass of the mixture for compression in kg/min = $m_s + m_v$

We know that available energy for compression

$$= ms(h_A - h_D)$$

and energy required for compression

$$= m(h_{F'} - h_E) = (m_s + m_v)(h_{C'} - h_E)$$

Now according to law of conservation of energy.

$$m_s(h_A - h_D) = (m_s + m_v)(h_{F'} - h_E)$$

We have already discussed that the nozzle efficiency.

$$\eta_N = \frac{h_A - h_{B'}}{h_A - h_B} \text{ or } h_A - h_{B'} = \eta_N (h_A - h_B) \text{ ... (iv)}$$

Entrainment efficiency

$$\eta_E = \frac{h_A - h_D}{h_A - h_{B'}} \text{ or } h_A - h_D = \eta_E (h_A - h_{B'}) \text{...(v)}$$

and compression efficiency,

$$\eta_C = \frac{h_F - h_E}{h_{F'} - h_E} \text{ or } h_{F'} - h_E = \frac{h_F - h_E}{\eta_C} \quad \text{...(vi)}$$

Substituting the value of $(h_A - h_D)$ and $(h_{F'} - h_E)$ from equations (v) and (vi) in equation (iii), we have

$$m_s \times \eta_E = (h_A - h_{B'}) = (m_s + m_v)\left(\frac{h_F - h_E}{\eta_C}\right)$$

$$m_s \times \eta_E \times \eta_N (h_A - h_B) = (m_s + m_v)\left(\frac{h_F - h_E}{\eta_C}\right)$$ ... [From equation (iv)]

$$m_s \times \eta_E \times \eta_N \times \eta_C (h_A - h_B) = m_s (h_F - h_E) + m_v (h_F - h_E)$$
$$m_s [(h_A - h_B)\eta_N \eta_E \eta_C - (h_F - h_E)] = m_v (h_F - h_E)$$

**or** $$\frac{m_s}{m_v} = \frac{(h_F - h_E)}{(h_A - h_B)\eta_N \eta_C - (h_F - h_E)}$$

where $\frac{m_s}{m_v} =$ Mass of motive steam required per kg of water vapour produced in the flash chamber

The make-up water is supplied at point G whose temperature is slightly lower than the condenser temperature and is throttled to point *H* in the flash chamber and leaves it corresponding to the condition at point C. Since the enthalpy of water at point *G* is equal to the enthalpy of water at point *H*, therefore for each kg of water vapour formed heat absorbed is ($h_C$-$h_{fg}$)kJ/kg. In other words, net refrigerating effect,

$$R_E = m_v (h_C - h_{fg}) kJ / \min$$

If *Q* tonnes of refrigeration is the refrigerating load, then the heat absorbed or net refrigerating effect,

$$R_E = 210 Q kJ / \min$$

From the above expressions, we find that the mass of water vapour formed.

$$m_v = \frac{210Q}{h_C - h_{fg}} kg / \min$$

Since one kg of water vapour requires $m_s$ kg of motive steam, therefore, Mass of motive steam required per *Q* tonne of refrigerating load

= Mass of water vapour per minute

X Motive steam required per kg of vapour

= $m_v$ X $m_s$

$$= \frac{210Q}{h_C - h_{fg}} m_s$$

$$C.O.P. = \frac{m_v(h_c - h_{fg})}{m_s(h_A - h_{fg})}$$

## 2.28 Advantages and Disadvantages of Steam Jet Refrigeration System

Following are the advantages and disadvantages of a steam jet refrigeration system:

**Advantages:**

1. It is simple in construction and rigidly designed.
2. It is a vibration- free system as pumps are the only moving parts.
3. It has low maintenance cost, low production cost and high reliability.
4. It has relatively less plant mass (kg/TR). Hence, there are now a number of air- conditioning application ranging up to 300 TR in capacity as well as many industrial applications of even larger size.
5. It uses water as a refrigerant. Water is very safe to use as it is non-poisonous and non-inflammable.
6. This system has an ability to adjust quickly to load variations.
7. The running cost of this system is quite low.

**Disadvantages:**

1. The system is not suitable for water temperature below4$^{o}$C.
2. For proper functioning of this system, maintenance of high vacuum in the evaporator is necessary. This is done by direct vaporization to produce chilled water which is usually limited as tremendous volume of vapor is to be handled.

## 2.3 THERMO ELECTRIC REFRIGERATION SYSTEM

## 2.30 Introduction

Thermoelectric refrigeration owes its origin to the discovery of Seebeck and Peltier effects in t 1821. Seebeck found that if two dissimilar metals are joined at two junctions, one at high temperature and other at low temperature, current was produced. Peltier in 1834 observed that if current was passed through two dissimilar metals joined at two junctions, one was cooled and the other was heated. In 1838, Lenz used both effects to freeze water and thermoelectric refrigeration was born. Further development occurred in 1930 when semiconductors were discovered.

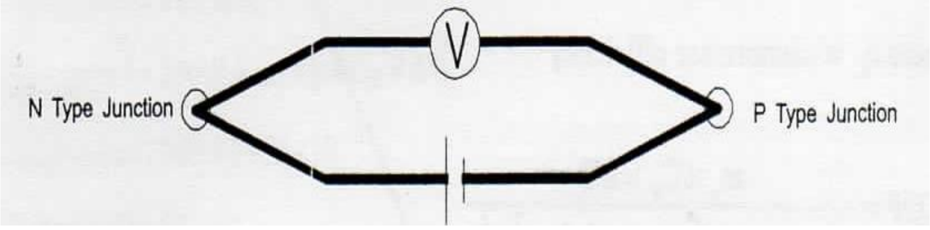


**Figure 2.8 Thermoelectric Refrigeration Set-up**

### 2.31 Applications of this refrigeration system are:

1) Portable refrigerator
2) Water coolers
3) Space apparatus
4) Blood analyzers
5) CCD cameras
6) Laser diodes

The typical thermoelectric system contains thin ceramic wafers with a series of P and-N doped bismuth-telluride semiconductor materials sandwiched between them. The ceramic material adds rigidity and necessary insulation. N type material has excess electrons while P type material has deficiency of electrons. The P and N materials are joined as shown in fig. 5. The thermoelectric couples are electrically in series and thermally parallel. As electrons move from P type to N type material through an electrical connector, electrons jump to a higher energy state, absorbing energy from the cold side. The electrons then flow from N type to P type through electrical connector, thereby releasing energy to the hot side. Thus, one junction is hot while the other is cool. Thermoelectric system can thus be used to heat or cool depending on current direction.

### 2.32 Analysis of Thermoelectric System

Three parameters are important. They are

i) Hot surface temperature ($T_h$)

ii) Cold surface temperature (Tc)

iii) Heat load to be absorbed at cold side (Qc)

Let Ta = ambient temperature, $Q_h$ = heat released to hot side

$\therefore T_h = Ta + KQ_h$, where,

K = thermal resistance of path

$\therefore Q_h = Qc + P$

Where P = electrical power required for producing thermoelectric effect = V x I.

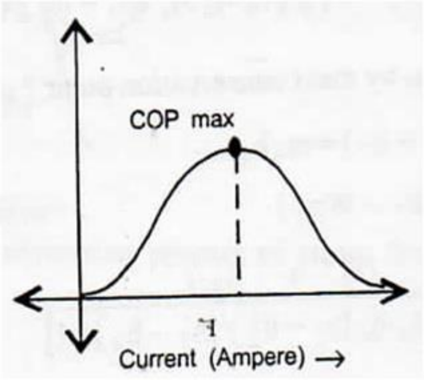
Temperature difference across the thermoelectric system is $\Delta T = Th - Tc$

Figure shows variation in COP of system with current. The system COP can be calculated as follows:

$$\therefore \mathrm{COP}_{\max} = \frac{T_C}{T_h - T_C}$$

$$\therefore \mathrm{COP} = \frac{\left(ZT_C^{2}/2 - T_h + T_C\right)}{\left(Z\ T_h T_C\right)}$$

$$\text{Where } Z = \text{figure of merit} = \frac{\alpha^2}{UR}$$



Where a = thermoelectric power or See back coefficient

U = overall conductance

R = overall resistance Current *I*

### 2.33 Advantages of Thermoelectric Refrigeration

1) Absence of moving parts, so no vibration problems

2) Long life and no wear and tear

3) Load can be controlled by adjusting current.

4) Very compact and portable

5) Light weight

6) No leakage problems; no refrigerant required

7) Can be used for year-round air-conditioning

8) Design and manufacture are simple.

9) It can be operated in any position, vertical or horizontal.

10) Suitable for production of cooling suit

### 2.34 Disadvantages of Thermoelectric Refrigeration

1) Unavailability of suitable material with high figure of merit 'Z'

2) Costly as initial expenditure high

3) Running cost is high.

4) Lower COP of the order of 0.1 to 0.2

## 2.4 AIR REFRIGERATION SYSTEM

### 2.40 Introduction

In an air refrigeration cycle, the air is used as a refrigerant. In olden days, air was widely used in commercial applications because of its availability at free of cost. Since air does not change its phase i.e. remains gaseous throughout the cycle, therefore the heat carrying capacity per kg of air is very small as compared to vapour absorbing systems. The air-cycle refrigeration systems, as originally designed and installed, are now practically obsolete because of their low coefficient of performance and high power requirements. However, this system continues to be favored for air refrigeration because of the low weight and volume of the equipment. The basic elements of an air cycle refrigeration system are the compressor, the cooler or heat exchanger, the expander and the refrigerator.

Before discussing the air refrigeration cycle, we should first know about the unit of refrigeration, coefficient of performance of a refrigerator and the difference between the heat engine, a refrigerator and a heat pump.

### 2.41 Difference Between a Heat Engine, refrigerator and Heat Pump

In a heat engine, as shown in Fig.2.9 (a), the heat supplied to the engine, is converted into useful work. If $Q_2$ is the heat supplied to the engine and $Q_1$ is the heat rejected from the engine, then the net work done by the engine is given by

$$W_E = Q_2 - Q_1$$

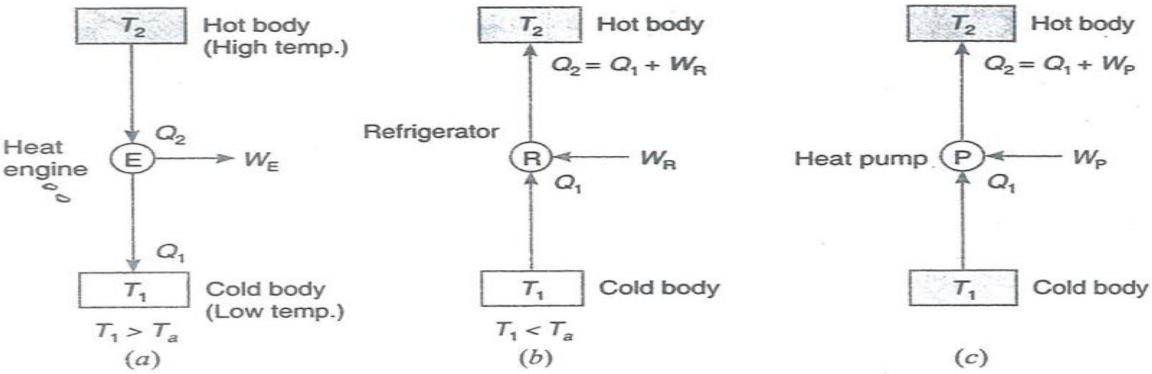


**Figure 2.9 Difference between a heat engine, refrigerator and heat pump.**

The performance of a heat engine is expressed by its efficiency. We know that the efficiency or coefficient of performance of an engine.

$$\eta_E or (C.O.P.)_E = \frac{Workdone}{Heat \sup plied} = \frac{W_E}{Q_2} = \frac{Q_2 - Q_1}{Q_2}$$

A refrigerator as shown in Fig. 2.9 (b), is reversed heat engine which either cool or maintain the temperature of a body ($t_1$) lower than the atmospheric temperature ($t_a$). This is done by extracting the heat ($Q_1$) from a cold body and delivering it to a hot body ($Q_2$). In doing so, work $W_R$ is required to be done on the system. According to First Law if Thermodynamics,

$$W_R = Q_2 - Q_1$$

The performance of a refrigerator is expressed by the ratio of amount of heat taken from the cold body ($Q_1$) to the amount of work required to be done on the system ($W_R$). This ratio is called coefficient of performance. Mathematically, coefficient of performance of a refrigerator,

$$(C.O.P.)_R = \frac{Q_1}{W_R} = \frac{Q_1}{Q_2 - Q_1}$$

Any refrigerating system is a heat pump as shown in Fig.2.9 (c), which extracts heat ($Q_1$) from a cold body and delivers it, to a hot body. Thus there is no difference between the cycle of operations of a heat pump and a refrigerator. The main difference between the two is in their operating temperatures. A refrigerator works between the cold body temperature ($T_1$) and the atmospheric temperature ($T_a$). A refrigerator used for cooling in summer can be used as heat pump for heating in winter.

In the similar way, as discussed for refrigerator, we have

$$W_p = Q_2 - Q_1$$

The performance of a heat pump is expressed by the ratio of the amount of heat delivered to the hot body ($Q_2$) to the amount of work required to be done on the system( $W_p$). This ratio is called coefficient of performance of energy performance ratio (E.P.R.) if a heat pump. Mathematically, coefficient of performance if energy performance ratio of a heat pumps.

$$(C.O.P.)_P or E.P.R. = \frac{Q_2}{W_P} = \frac{Q_2}{Q_2 - Q_1}$$

$$= \frac{Q_1}{Q_2 - Q_1} + = (C.O.P.)_R + 1$$

From above we see that the C.O.P. may be less than one or greater than one depending on the type of refrigeration system used. But the C.O.P. of a heat pump is always greater than one.

### 2.42 Open Air Refrigeration Cycle

In an open air refrigeration cycle, the air is directly led to the space to be cooled (i.e. a refrigerator), allowed to circulate through the cooler and then returned to the compressor to start another cycle. Since the air is supplied to the refrigerator at atmospheric pressure, therefore, volume of air handled by the compressor and expander is large. Thus the size of compressor and expender should be large. Another disadvantage of the open cycle system is that the moisture is regularly carried away by the air circulated through the cooled space. This leads to the formation of frost at the end of expansion process and clog the line. Thus in an open cycle system, a drier should be used.

### 2.43 Closed or Dense Air Refrigeration Cycle

In a closed or dense air refrigeration cycle, the air is passed through the pipes and component parts of the system at all times. The air, in this system, is used for absorbing heat from the other fluid (say brine) and this cooled brine is circulated into the space to be cooled. The air in the closed system does not come in contact directly with the space to be cooled.

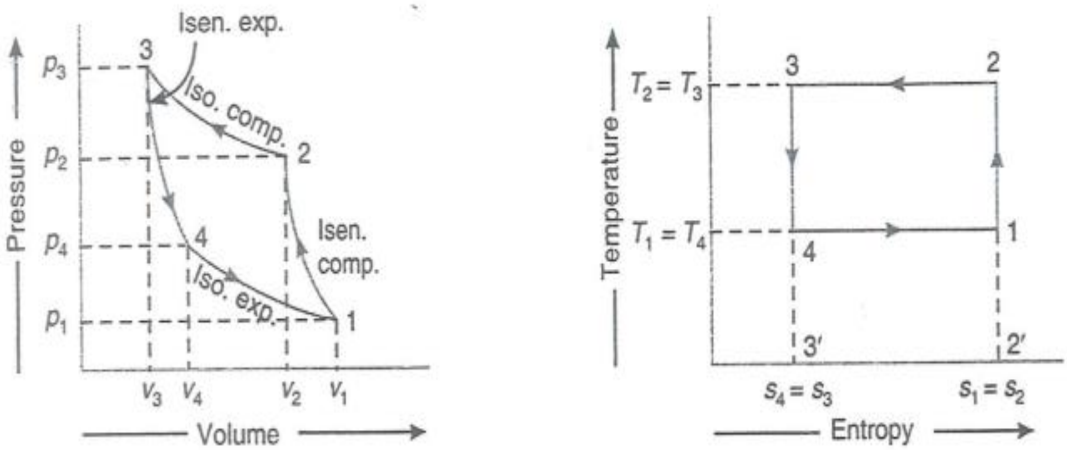
The closed air refrigeration cycle has the following thermodynamic advantages:

1. Since it can work at a suction pressure higher than that of atmospheric pressure, therefore the volume of air handled by the compressor and expender are smaller as compared to an open air refrigeration cycle system.
2. The operating pressure ratio can be reduced, which results in higher coefficient of performance.

### 2.44 Air Refrigerator Working on Reversed Carnot Cycle

In refrigerating systems, the Carnot cycle considered is the reversed Carnot cycle. We know that a heat engine working on Carnot cycle has the highest possible efficiency. Similarly, a refrigerating system working on the reversed Carnot cycle will have the maximum possible coefficient of performance. We also know that is not possible to make an engine working on the Carnot cycle. Similarly, it is also not possible to make a refrigerating machine working on the reversed Carnot cycle. However, it is used as the ultimate standard of comparison.

A reversed Carnot cycle, using air as working medium (or refrigerant) is shown on p-v and T-s diagrams in fig. 2.10(a) and (b) respectively. At point 1, let $p_1$, v1, T1 be the pressure, volume and temperature of air respectively.



(a) p-v diagram.

(b) T-s diagram.

**Figure 2.10 Reversed Carnot cycle**

The four processes of the cycle are as follows:

1. Isentropic compression process. The air is compressed isentropic ally as shown by the curve 1-2 on p-v and T-s diagrams. During this process, the pressure of air increases from $p_1$ to $p_2$ specific volume decreases from $v_1$ to $v_2$. And temperature increase from $T_1$ to $T_2$. We know that the heat rejected by the air during isothermal compression per kg of air,
2. Isentropic expansion process. The air is now expanded isothermally (i.e. at constant temperature, $T_2 = T_3$) as shown by the curve 2-3 on p-v and T-s diagrams. During this process, the pressure of air during isothermal compression per kg of air

$$Q_R = q_{2-3} = \text{Area } 2\text{-}3\text{-}3'\text{-}2'$$

$$= T_3 (s_2 - s_3) = T_2 (s_2 - s_3)$$

3. ***Isentropic expansion process.*** The air is now expanded isentropic ally as shown by the curve 3-4 on p-v and T-s diagrams. The pressure of air decreases from $p_3$ to $p_4$ specific volume increases from $v_3$ to $v_4$ and the temperature decreases from $T_3$ to $T_4$. We know that during isentropic expansion, no heat is absorbed or rejected by the air.
4. ***Isothermal expansion process.*** The air is now expanded isothermally (i.e. at constant temperature, $T_4=T_1$) as shown by the curve 4-1 on p-v and T-s diagrams. The pressure of air decreases from $P_4$ to $P_1$, and specific volume increases from $v_4$ to $v_1$. We know that the heat absorbed by the by the air (or heat extracted from the cold body) during isothermal expansion per kg or air,

$$Q_A = q_{4-1} = \text{Area } 4\text{-}1\text{-}2'\text{-}3'$$

$$= T_4 (s_1 - s_4) = T_4 (s_2 - s_3) = T_1 (s_2 - s_3) = T_1 (s_2 - s_3) = T_1 (s_2 - s_3)$$

We know that work done during the cycle per kg of air

$$W_R = {}^* \text{Heat rejected} - \text{Heat absorbed} = Q_R - q_A = q_{2-3} - q_{4-1}$$

$$= T_2 (s_2 - s_3) - T_1(s_2 - s_3) = (T_2 - T_1)(s_2 - s_3)$$

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

$$(C.O.P.)_R = \frac{Heatabsorbed}{Workdone} = \frac{q_A}{q_R - q_A} = \frac{q_4 - 1}{q_{2-3} - q_{4-1}}$$

$$= \frac{T_1(S_2 - S_3)}{(T_2 - T_1)(s_2 - s_3)} = \frac{T_1}{T_2 - T_1}$$

Though the reversed Carnot cycle is the most efficient between the fixed temperature limits, ye no refrigerator has been made using this cycle. This is due to the reason that the is entropic processes of the cycle require high speed of which the isothermal processed require an extremely low speed. This variation in speed of air is not practicable.

Note: We have already discussed that C.O.P of a heat pump,

$$(C.O.P)_P = (C.O.P)_R + 1 = \frac{T_1}{T_2 - T_1} + 1 = \frac{T_1}{T_2 - T_1}$$

and C.O.P or efficiency of a heat engine.

$$(C.O.P)_E = \frac{W_R}{q_R} = \frac{(T_2 - T_1)(S_2 - S_3)}{T_2(s_2 - s_3)} = \frac{T_2 - T_1}{T_2} = \frac{1}{(C.O.P)_P}$$

**PROBLEMS**

**2.1. A cycle Carnot refrigeration absorbs heat at 270 K and rejects it at 300 K.**

1. **Calculate the coefficient of performance of this refrigeration cycle.**
2. **If the cycle is absorbing 1130 kJ/min at 270 K, how many KJ of work is required per second?**
3. **It the Carnot heat pump operates between the same temperatures as the above refrigeration cycle, what is the coefficient of performance?**
4. **How many K will the heat pump deliver at 300 K if it absorbs 1130 KJ/min at 270 K.**

**Solution:** Given: $T_1$=270 K; $T_2$ = 300K

1. Coefficient of performance of Carnot refrigeration cycle

We know that coefficient of performance of Carnot refrigeration cycle,

$$(C.O.P)_R = \frac{T_1}{T_2 - T_1} = \frac{270}{300 - 270} = 9 \text{ Ans.}$$

2. Work required per second

Let Heat absorbed at 270K (i.e. $T_1$)

$Q_1$=1130 kJ/min=18.83 kJ/s -------(Given)

We know that $(C.O.P)_R = \frac{Q_1}{W_R}$ OR $9 = \frac{18.83}{W_R}$

$W_R$ =2.1 kJ/s Ans.

3. Coefficient of performance of Carnot heat pump

We know that coefficient of performance of a carnot heat pump,

$$(C.O.P)_P = \frac{T_2}{T_2 - T_1} = \frac{300}{300 - 270} = 10 \text{ Ans.}$$

4. Heat delivered by heat pump at 300K

Let $Q_2$=Heat delivered by heat pump at 300K
Heat absorbed at 270K (i.e. $T_1$),
$Q_1$ =1130 kJ/min
We know that

$$(\text{C.O.P})_P = \frac{Q_2}{Q_2 - Q_1} \quad \text{OR} \quad 10 = \frac{Q_2}{Q_2 - 1130}$$

$10Q_2 - 11300 = Q_2$ OR $Q_2 = 1256$ =kJ/min Ans.

### 2.45 Air refrigerator Working on a Bell-Coleman Cycle (or Reversed Brayton or Joule Cycle

As Bell-Coleman air refrigeration machined was developed by Bell-Coleman and Light Foot by reversing the Joule's air cycle. It was one of the earliest types of refrigerators used in ships carrying frozen meat. Fig. 2.11 show a schemata diagram of such a machine which consists of compressor, a cooler, an expander and a refrigerator.

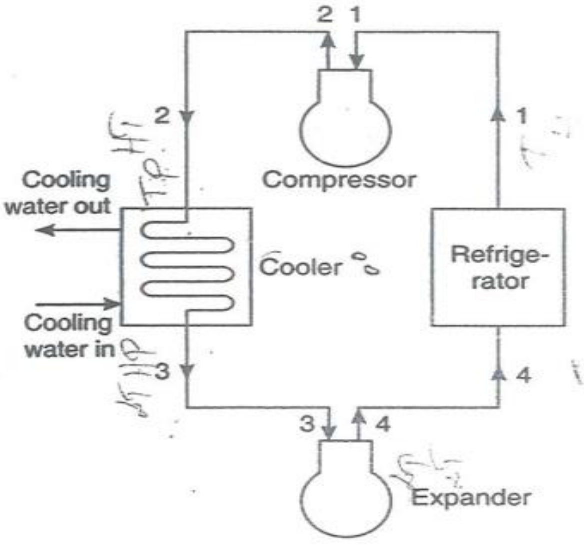


**Figure 2.11** Open cycle air Bell-Coleman Refrigerator

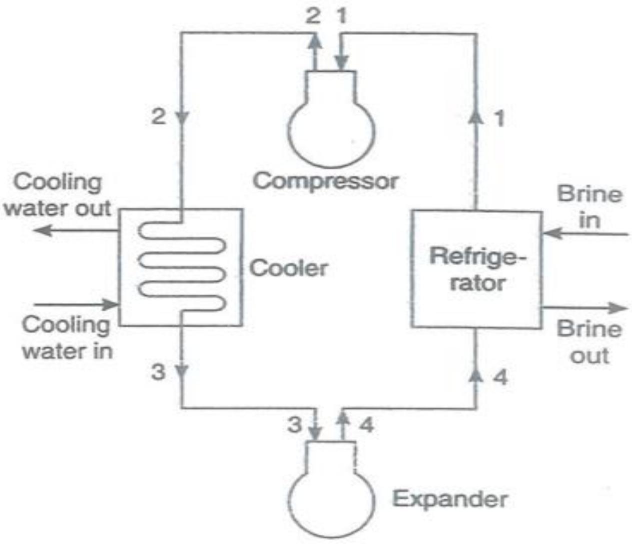
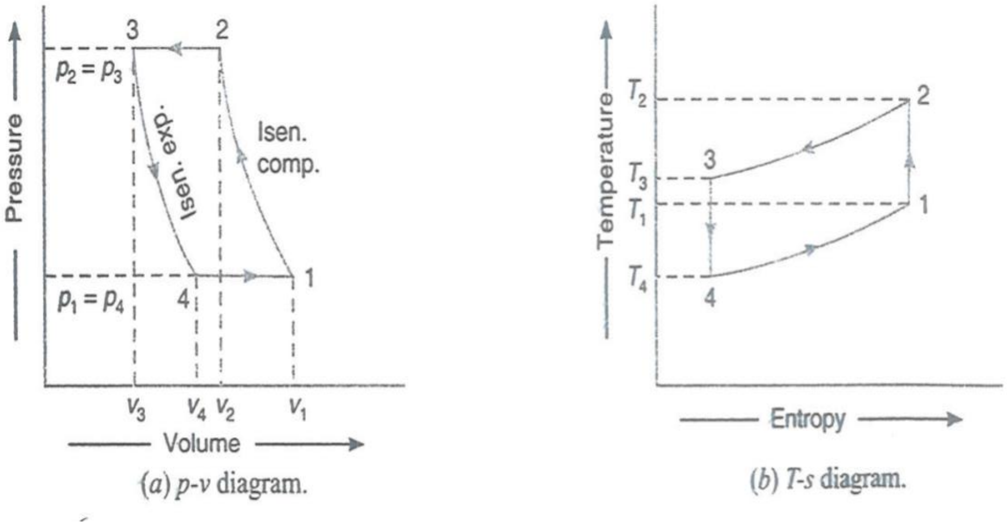


**Figure 2.13** Closed cycle air Bell-Coleman Refrigerator

The Bell-Coleman cycle (also known as revered Brayton or Joule cycle) is a modification of reversed Carnot cycle. The cycle is shown on P-v and T-s diagrams in Fig. 2.13 (a) and (b). at point 1, let $p_1$,$v_1$ and $T_1$ be the pressure, volume and importune of air respectively. The four processes of the cycle are as follows:

1. Isentropic compression process. The cold air from the refrigerator is drawn into the compressor cylinder where it is compressed isentropic ally in the compressor as shown by the curve 1-2 on p-v and T-s diagrams. During the compression stroke, both the pressure and temperature increases and the specific volume of air at delivery from compressor reduce from $V_1$ to $V_2$. We know that during isentropic compression process, no heat is absorbed or rejected by the air.



(a) p-v diagram.

(b) T-s diagram.

**Figure 2.14** p-v and T-s Diagram

2. Constant pressure cooling process. The warm air from the compressor is now passed into the cooler where it is cooled at constant pressure $p_3$ (equal to $p_2$), reducing the temperature from $T_2$ to $T_3$ (The temperature of cooling water) as shown by the curve 2-3 on p-v and T-s diagrams. The specific volume also reduces from V2 to $V_3$. We know that heat rejected by the air during constant pressure per kg of air.

   $q_A=q_{4-1}=C_P(T_1-T_4)$

   We know that work done during the cycle per kg of air

   = Heat rejected – Heat absorbed $=q_R - q_A$

   $=C_p(T_2-T_3)-C_p(T_1-T_4)$

   $\therefore$ Coefficient of performance,

$$\text{C.O.P} = \frac{Heatabsorbed}{Workdone} = \frac{q_A}{q_R - q_A} = \frac{c_p(T_1 - T_4)}{c_p(T_2 - T_3) - C_P(T_1 - T_4)}$$

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

$$= \frac{T_4\left[\frac{T_1}{T_4} - 1\right]}{T_3\left[\frac{T_2}{T_3} - 1\right] - T_4\left[\frac{T_1}{T_4} - 1\right]} \qquad \text{-------------(i)}$$

We know that for isentropic compression process 1-2,

$$\frac{T_2}{T_1} = \left[\frac{p_2}{p_1}\right]^{\frac{\gamma-1}{\gamma}} \qquad \text{------------------------(ii)}$$

Similarly, for isentropic expansion process 3-4,

$$\frac{T_3}{T_4} = \left[\frac{p_3}{p_4}\right]^{\frac{\gamma-1}{\gamma}} \qquad \text{------------------(iii)}$$

Since, $p_2 = p_3$ and $p_1 = p_4$, therefore from equations (ii) and (iii)

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ OR } \frac{T_2}{T_3} = \frac{T_1}{T_4} \qquad \text{----------------(iv)}$$

Now substituting these values in equation (i), we get

$$\text{C.O.P} = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1}$$

$$= \frac{1}{\left[\frac{p_3}{p_4}\right]^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left[\frac{p_2}{p_1}\right]^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left[r_p\right]\frac{\gamma-1}{\gamma} - 1} \qquad \text{-----------(v)}$$

$r_p$ =Compression or expansion ratio = $\frac{p_2}{p_1} = \frac{p_3}{p_4}$

Where

Sometimes, the compression and expansion processes take place according to the law $pv^n$ =Constant. In such a case, the C.O.P is obtained from the fundamentals as discussed below:

We know that work done by the compressor during the process 1-2 per kg of air,

$$W_C = \frac{n}{n-1}\left(p_2v_2 - p_1v_1\right) = \frac{n}{n-1}\left(RT_2 - RT_1\right) ......(\because pv = RT)$$

and work done by the expander during the process 3-4 per kg of air.

$$W_E = \frac{n}{n-1}\left(p_3v_3 - p_4v_4\right) = \frac{n}{n-1}\left(RT_3 - RT_4\right)$$

$\therefore$ Net work done during the cycle per kg of air.

w= $w_c - w_E = \frac{n}{n-1}\times R\left[\left(T_2 - T_1\right)-\left(T_3 - T_4\right)\right]$

We also know that heat absorbed during constant pressure process 4-1,
=$C_p$ $(T_1$-$T_4)$

C.O.P = $\frac{Heatabsorbed}{Workdone} = \frac{q_A}{w} = \frac{c_p(T_1 - T_4)}{\frac{n}{n-1}\times R\left[\left(T_2 - T_1\right)-\left(T_3 - T_4\right)\right]}$ ---------(vi)

We know that R=$C_P$-$C_V$ = $C_V$ $\left(\gamma - 1\right)$

Substituting the value of R in equation (vi),

$$\text{C.O.P} = \frac{C_P\left(T_1 - T_4\right)}{\frac{n}{n-1}\times C_V\left(\gamma-1\right)\left[\left(T_2 - T_1\right)-\left(T_3 - T_4\right)\right]}$$

$$= \frac{\gamma\left(T_1 - T_4\right)}{\frac{n}{n-1}\times\left(\gamma-1\right)\left[\left(T_2 - T_1\right)-\left(T_3 - T_4\right)\right]} \because \frac{C_P}{C_V} = \gamma$$

$$= \frac{T_1 - T_4}{\frac{n}{n-1}\times\frac{\left(\gamma-1\right)}{\gamma}\left[\left(T_2 - T_3\right)-\left(T_1 - T_4\right)\right]} \quad \text{--------(vii)}$$

$$\frac{T_2}{T_1} = \left[\frac{p_2}{p_1}\right]^{\frac{n-1}{n}} \text{ and } \frac{T_3}{T_4} = \left[\frac{p_3}{p_4}\right]^{\frac{n-1}{n}}$$

3. For isentropic compression or expansion, n = $\gamma$, Therefore, the equation (vii) may be written as

$$\text{C.O.P} = \frac{T_1 - T_4}{\left(T_2 - T_3\right)-\left(T_1 - T_4\right)}$$

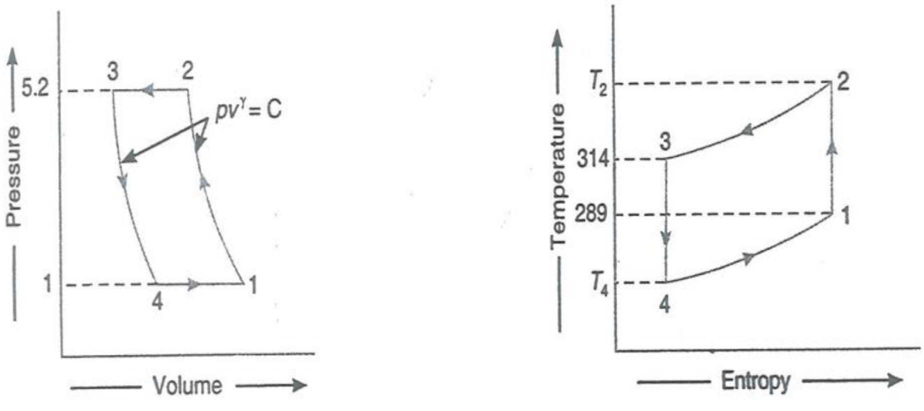
**Example 2.2 A refrigerating machine of 6 tonnes capacity working on Bell-Coleman cycle has an upper limit of pressure of 5.2 bar. The pressure and temperature at the start of compression are 1 bar and 16$^\circ c$ respectively. The compressed air is cooled at constant pressure to a temperature of 41$^\circ c$, enters the expansion cylinder, assuming both expansion and compression processes to be isentropic with $\gamma$ =1.4, Calculate:**

1. **Coefficient of performance;**

2. **Quantity of air in circulation per minute.**
3. **Piston displacement of compressor and expander;**
4. **Bore of compressor and expansion cylinders. The unit runs at 240 r.p.m and is double acting. Stroke length is 200 mm; and**
5. **Power required to drive the unit.**
   **For air, take $\gamma$ =1.4, and $C_p$=1.003 kJ/kg K**

**Solution :** Given: Q=6 TR=6x210=1260kJ/min; $p_2=p_3$=5.2 bar; $p_1=p_4$=1 bar =$1\times10^5$ N/m$^{2;}$ $T_1=16^\circ c$ =16+273=289 K; $T_3=41^\circ c$ =41+273=314k; $\gamma$ =1.4,

The Bell-Coleman cycle on p-v and T-s diagrams is shown in Fig. 2.13 (a) and (b) respectively.



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

1. **Coeffi**

   Let $T_2$ and $T_4$ =Temperature at the end of compression and expansion respectively. The compression and expansion are isentropic (i.e. $pv^\gamma$ =C) and $\gamma$ for air = 1.4.

   We know that

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left[\frac{5.2}{1}\right]^{\frac{1.4-1}{1.4}} = (5.2)^{0.286} = 1.6$$

$$T_2 = T_1 \times 1.6 = 289 \times 1.6 = 462.4K$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left[\frac{5.2}{1}\right]^{\frac{1.4-1}{1.4}} = (5.2)^{0.286} = 1.6$$

$$\therefore T_4 = T_3 / 1.6 = 314 \times 1.6 = 196.25K$$

We know that coefficient of performance,

C.O.P = $\dfrac{T_4}{T_3 - T_4} = \dfrac{196.25}{314 - 196.25}$ =1.674 Ans.

2. Quantity of air in circulation per minute

Let $m_a$=Mass of air in circulation in kg per minute.
We know that heat extracted from the refrigerating machine (or refrigerating effect produced) per kg or air

$=C_p(T_1-T_4)=1.003\,(289-196.25) = 93$ kJ/kg

and refrigerating capacity of the machine

=6 TR=6 x 210 = 1260 kJ/min

$\therefore$ Mass of air in circulation

$m_a=1260/93=13.548$ kg/min Ans.

3. Piston displacement of compressor and expander
Let $V_1$ and $V_4$ =Piston displacement per minute of compressor and expander respectively.
We know that characteristic gas constant,

$$*R_a=C_P\left(\frac{\gamma-1}{\gamma}\right)=1.003\left(\frac{1.4-1}{1.4}\right)=0.287\text{kJ/kg K} =287 \text{ J/kg K}$$

We also know that

$P_1V_1=m_a R_aT_1$

$$V_1=\frac{m_aR_aT_1}{P_1}=\frac{13.548\times287\times289}{1\times10^5}=11.237m^3/\min \text{ Ans.}$$

4. Bore of compressor and expansion cylinders
Let D and d = Bore of compressor and expansion cylinder in metres, respectively:
N= Speed of the unit = 240 r.p.m -----(Given)
L = Length of stroke = 200 mm = 0.2 m -----(Given)

We know that piston displacement of compressor cylinder,

$$V_1=\left[\frac{\pi}{4}\times D^2\times L\times 2\right]N \quad \text{--------}(\because \text{ of double acting})$$

$$1-\frac{1}{\gamma}=\frac{R_a}{C_P} \qquad \text{--------}\left(\because \frac{C_V}{C_P}=\frac{1}{\gamma}\right)$$

$$R_a=C_P\left[\frac{\gamma-1}{\gamma}\right]$$

$$11.237=\left[\frac{\pi}{4}\times D^2\times0.2\times2\right]240=75.4D^2$$

$\because D^2=11.237/75.4 = 0.149$ or $D=0.386$m = 386 mm Ans.
Similarly piston displacement of expansion cylinder,

$$V_4=\left[\frac{\pi}{4}\times d^2\times 2\times 2\right]N$$

$$7.63=\left[\frac{\pi}{4}\times d^2\times 0.2\times 2\right]240=75.4\ d^2$$

$d^2=7.63/75.4=0.1012$ or $d=0.318$m=318mm Ans.

5. Power required to drive the unit

We know that heat absorbed during the constant pressure process 2-3

$$=m_a c_p(T_1-T_4)= 13.548 \times 1.003\ (289-196.25)=1260\ \text{kJ/min}$$

$$\therefore \text{ Work done per minute}=\frac{Heatabsorbed}{C.O.P}=\frac{1260}{1.674}=752.7kJ/\min$$

and power required to drive the unit

$$=752.7/60=12.54 \text{ kJ/s or kW Ans.}$$

$$\text{We know that C.O.P}=\frac{\text{Re}\ frigeratingcapacityorheatabsorbed(Q)}{Workdone}$$

$$1.674=\frac{6\times 210}{workdone}$$

$\therefore$ Work done = 6 x 210/1.674=752.7 kJ/min

and power required = 752.7/60=12.54 kJ/s or kW Ans.

## 2.5 VORTEX-TUBE REFRIGERATION

### 2.50 Introduction

This method of refrigeration was first proposed by Ranque in 1931. Prof. Hilsch also came up with its design. Thus the vortex tube, which is the central part of this system, was named Ranque— Hilsch Tube. Lot of research has been done at HT Bombay under the guidance of Prof. Parulekar on vortex-tube refrigeration.

Temperatures as low as - 50°C were obtained. If this system was used along with conventional system like vapour compression, temperatures up to - 100°C were achieved.
Refer to Fig.1 for construction of vortex tube.

The vortex tube, as shown in figure has an air compressor. The air is compressed and cooled in a heat exchanger. After cooling, it expands in a nozzle, which is fitted tangentially to the vortex tube. The vortex tube has a throttle valve at one end and the other end is kept open. The gas or air enters the vortex tube and is separated in two streams. The cold air stream exits from end 'A' and the hot air stream exits from end 'B' after passing through the throttle valve.

Thus vortex tube is a device, which produces cooling at one end and heating at the other end.

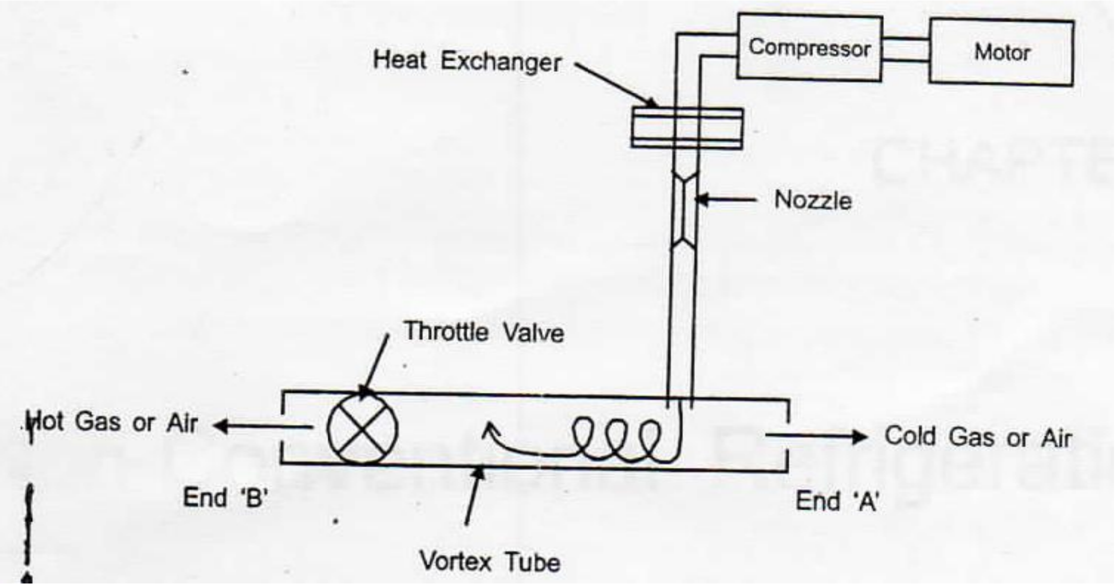


**Figure 2.15** Vortex Tube

### 2.51 Theory of Operation and Analysis

When compressed air expands through the nozzle, swirl motion is created. The axial component of velocity is less than the tangential component. Air moves as a free vortex from nozzle plane to valve end. As the vortex reaches near the valve, it reaches a stagnation state, where kinetic energy is converted to pressure energy. At this point, there is reversal of flow due to high stagnation pressure. The reversed flow mixes with forward moving vortex, causing rotational motion. During this turbulent rotation, energy is pumped from the cold core to the outer layer making it hot. Thus, there is a cold core surrounded by concentric hot layer. These streams are separated and hot air exits at one end and cold air at the other end.

If m. = mass at inlet; $m_c$ = mass of cold air

$m_h$ = mass of hot air $\therefore \mu$ = cold air fra $\frac{m_c}{m_i}$ =

mass of air at exit = $m_e = m_e + m_h$

and $m_i h_i = m_e h_e + m_h h_h$

We can also write the balance as

$m_e(T_i - T_e) + m_h (T_h - T_i) = (m_i - m_e)(m_h - m_e)$

$\therefore \mu = (T_h - T_i) / [(T_h - T_i) + (T_i - T_c)] = \Delta T_h / (\Delta T_h + \Delta T_c)$

Where $\Delta T_h = T_h$ - Ti & $\Delta T_c = T_i - T_c$

Reversible temperature drop = $\Delta T_c' = T_i - T_c'$

$$\therefore \Delta T_c' = T_i \left[ 1 - (P_a / P_i)^{\frac{\gamma - 1}{\gamma}} \right]$$

Where $P_a$ = atmospheric pressure and $P_i$ = inlet pressure

$$\therefore \Delta T_{relative} = \frac{\Delta T_C}{\Delta T/C} = \frac{\Delta T_C}{T_i} \Big/ \left[ 1 - (P_a / P_i)^{\frac{\gamma - 1}{\gamma}} \right]$$

∴ Where $\Delta T_{relative}$ is the relative temperature drop

$$\therefore \eta = \frac{\text{Actual Cooling}}{\text{Ideal Cooling}} = \frac{m_c \times C_P \times \Delta T_C}{m_i \times C_P \times \Delta T_C'}$$

$$\therefore \eta = \mu \Delta T_{rel} = \mu \Delta T_C \Big/ T_i \left[1-\left(P_a/P_i\right)^{\frac{\gamma-1}{\gamma}}\right]$$

∴ Energy supplied or work required to compress air is given by

$$W = m_i C_P T_a \left[\left(P_a/P_i\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \Big/ \eta_C$$

Where $\eta_c$ compressor efficiency

$$\therefore COP = \frac{m_c \times C_P \times \Delta T_C}{m_i \times C_P \times T_a \left[\left(P_a/P_i\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \Big/ \eta_C}$$

$$\therefore COP = \left(T_i/T_a\right) \times \eta_C \times \eta \times \left(\frac{P_a}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$$

For perfect heat exchanger

$$\therefore COP = \eta \times \eta_c \times \left(\frac{P_a}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$$

For isothermal compression

$$COP_{isothermal} = \frac{\gamma-1}{\gamma} \eta \times \eta_c \times \left[\frac{1-\left(P_i/P_a\right)^{\frac{\gamma-1}{\gamma}}}{l_n\left(P_i/P_a\right)}\right]$$

## 2.52 Characteristics of Vortex Tube

Figure 2, 3 4 show some characteristics of a vortex tube.

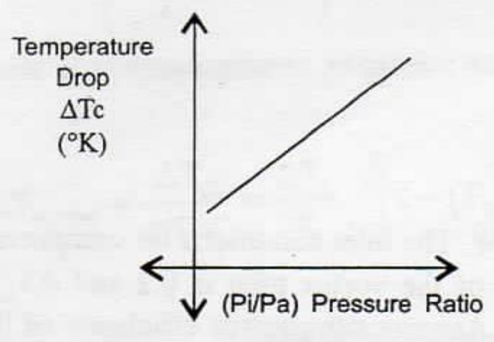


**Figure 2.16: ΔTc *Vs* Pi / Pa**

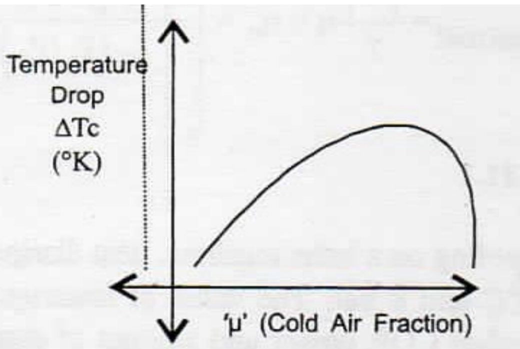


**Figure 2.17 ΔTc *Vs* '$\mu$'**

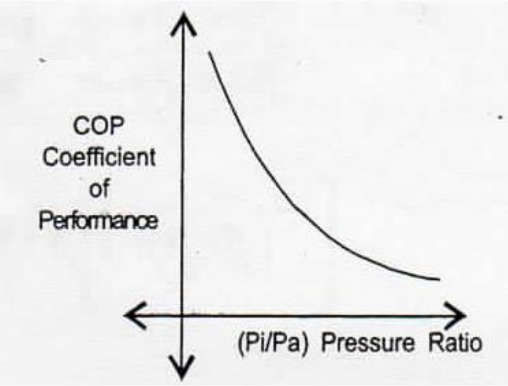


**Figure 2.18 COP *Vs* Pi/Pa**

If we plot the temperature drop ΔTc = (Ti-Tc) Vs the pressure ratio, we observe a linear relationship. As pressure ratio increases, temperature drop increases. If we plot the temperature drop against the cold air

Fraction $\mu = \frac{m_C}{m_i}$, we observe a bell shaped curve. As ΔTc increases, $\mu$ initially increases, reaches a maximum and then decreases. The COP reduces with increasing pressure ratio in an exponential manner.

## 2.53 Advantages of Vortex Tube

(i) Leakages are insignificant as air is the refrigerant.
(ii) No moving parts, thus no maintenance
(iii) Longer life
(iv) Simple design and function
(v) Light in weight and compact
(vi) Initial investment is less
(vii) Operation is simple and no skilled manpower required

## 2.54 Disadvantages of Vortex Tube

i) Poor COP
ii) Limitation on size of unit, not suitable for large capacity
iii) Compressor results in noisy operation.

## 2.55 Applications of Vortex Tube

i) Cooling of cutting tools
ii) Air suits for mines
iii) Condensation of natural gases
iv) Turbine blade cooling
v) Aircraft systems

## 2.6 PULSE TUBE REFRIG RATION

### 2.60 Introduction

The idea of pulse tube was first proposed by Gifford and Longsworth in 1961. The effects of sudden expansion and release of refrigerant gas are employed to get refrigeration effect. Refer to figure 7 for the circuit of pulse-tube refrigeration.

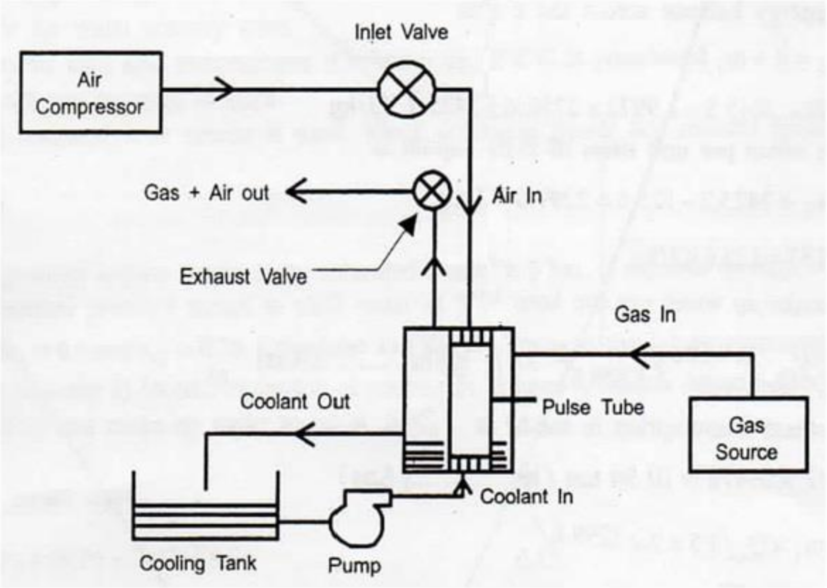


**Figure 2.19** Pulse-Tube Refrigeration

### 2.61 Working

The apparatus consists of a high-pressure gas source, which supplies gas at temperature equal to the ambient temperature to the base of the pulse tube. Now compressed air is supplied from the air compressor. The compressed air enters the vortex tube through a valve. Further, compressed gas is allowed to enter the pulse tube. Now, the compressed gas acts on an imaginary piston i.e. the separating air column. Thus, gas inside the pulse tube gets compressed resulting in an increase in the temperature in the pulse tube from top to bottom. The maximum temperature is at the base. A coolant is circulated to remove heat from the gas, which is now compressed at the base. The coolant rejects heat absorbed to a coolant tank and is pumped back to the pulse tube. Due to cooling by the coolant, gas temperature reduces. Now, the supply of high-pressure gas is stopped and inlet valve is closed. The exhaust valve now opens and gas is exhausted lowering temperature inside the tube. A detailed sketch of the pulse tube is shown below in figure .8 and temperature profile is shown in figure 2.19

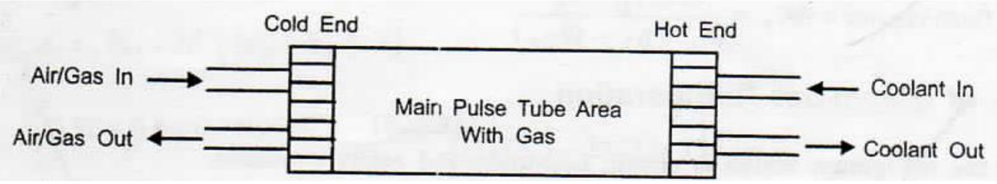


**Figure 2.20** Pulse Tube

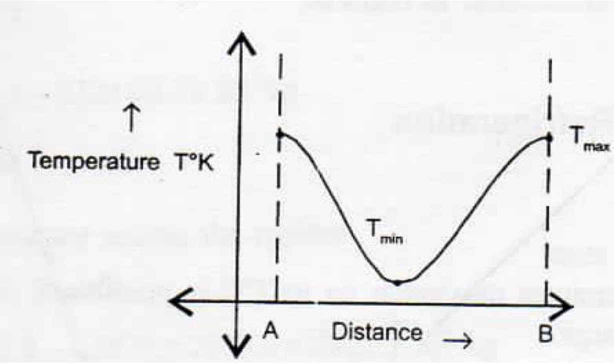


**Figure 2.21** Temperature profile

This lowered temperature causes the refrigerating effect. The air leaving the pulse tube also becomes cooler. It is possible to use this cool air to cool down hot compressed air entering the pulse tube with the help of a heat exchanger. This further lowers the temperature of the pulse tube. Temperature as low as 190°K is achieved by pulse tube. Thus, pulse tube is a preferred choice if cooling is to be done in areas with low power generation capacities.

### 2.62 Advantages of Pulse Tube

1) Less moving parts, so no maintenance and fewer losses
2) Low power consumption
3) Multi staging is possible.
4) Longer life
5) No lubrication required
6) Can be used instead of cryogenic cycles

### 2.63 Disadvantages

1) Poor coefficient of performance (COP)
2) COP depends on length of tube
3) Tubes costly to manufacture

## 2.7 ADIABATIC DEMAGNETISATION

### 2.70 Introduction

All substances exhibit some magnetic properties and are classified as “ferromagnetic” “diamagnetic” or “Paramagnetic”.

When paramagnetic salts are pre-cooled to low temperatures, motion of molecules is reduced to minimum. The molecules act as individual magnets aligning themselves to a strong electromagnetic field. During this process, heat is released. If this heat is removed and then magnetic field is removed, there will be adiabatic cooling.

### 2.71 Working

For adiabatic demagnetization, the apparatus is as shown in figure

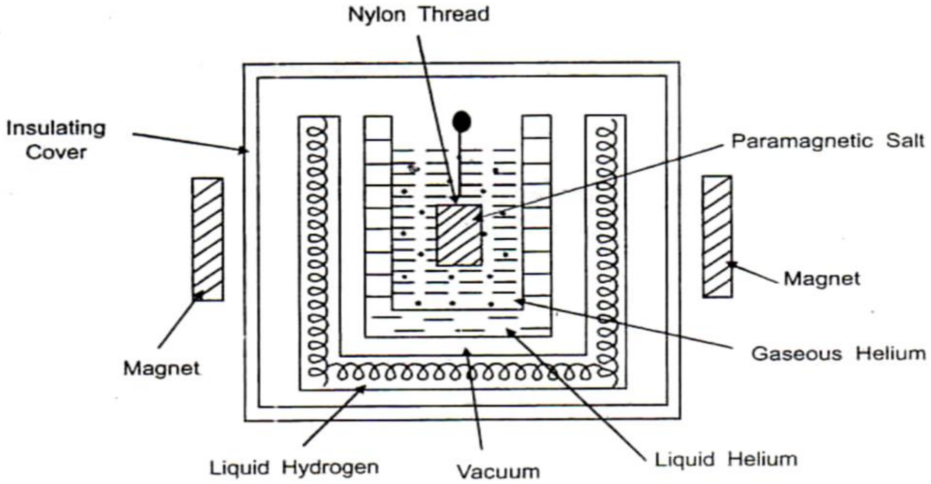


**Figure 2.12** Adiabatic Demagnetization

The paramagnetic salt is suspended by a thread in a tube containing low-pressure gaseous helium surrounded by a bath of liquid helium to provide thermal communication. The liquid helium bath is cooled down by pumping to low pressures. Thus temperature of salt approaches temperature of liquid helium.

Now, the magnetic field is turned on causing heating of the salt. The heat is conducted from salt to helium bath and salt again cools down. Now, helium gas is removed and salt is isolated. The magnetic field is turned off. The temperature of salt now decreases further due to "adiabatic demagnetization". Very low temperatures are achieved by this method. Approximately 0.001°K is also possible. The strength of the magnetic field is roughly 80 kilogauss and paramagnetic salt commonly used is ferric alum.

### 2.72 Advantages

1) Useful to produce very low temperatures 2) No moving parts 3) Ease of operation

### 2.73 Disadvantages

1) Expensive
2) Difficult to implement in practice; still on laboratory scale
3) Leakage of helium is a problem.

# UNIT-III

# PSYCHROMETRY

## 3.1 INTRODUCTION

The psychrometric is that branch of engineering science which deals with the study of moist air i.e., dry air mixed with water vapour or humidity. It also includes the study of behavior of dry air and water vapour mixture under various sets of conditions. Though the earth's atmosphere is a mixture of gases including nitrogen ($N_2$), oxygen ($O_2$), argon (Ar) and carbon dioxide ($CO_2$), yet for the purpose of psychrometric, it is considered to be a mixture of dry air and water vapour only.

## 3.2 PSYCHOMETRIC TERMS

Though there are many psychometric terms, yet the following are important from the subject point of view :

1. ***Dry air***. The pure dry air is a mixture of a number of gases such as nitrogen, oxygen, carbon dioxide, hydrogen, argon, neon, helium etc. But the nitrogen and oxygen have the major portion of the combination. The dry air is considered to have the composition as given in the following table:

**Table .1 Composition of dry air**

| ***S.No.*** | ***Constituent*** | ***By volume*** | ***By mass*** | ***Molecular Mass*** |
|---|---|---|---|---|
| 1 | Nitrogen ($N_2$) | 78.03% | 75.47% | 28 |
| 2 | Oxygen ($O_2$) | 20.99% | 23.19% | 32 |
| 3 | Argon (Ar) | 0.94% | 1.29% | 40 |
| 4 | Carbon dioxide ($CO_2$) | 0.03% | 0.05% | 44 |
| 5 | Hydrogen ($H_2$) | 0.01% | - | 2 |

The molecular mass of dry air is taken as 28.966 and the gas constant of air ($R_a$) is equal 0.287 kJ / kg K or 287 J/kg K.

The molecular mass of water vapour is taken as 18.016 and the gas constant for water vapour (k) is equal to 0.461-kJ/kg K or 461 J/kg K.

Notes: (a) The pure dry air does not ordinarily exist in nature because it always contains some water vapout

(b) The term air, wherever used in this text, means dry air containing moisture in the vapour form.

(c) Both dry air and water vapour can be considered as perfect gases because both exist in the atmosphere at low pressure. Thus all the perfect gas terms can be applied to them individually.

(d) The density of dry air is taken as 1.293 kg/m3 at pressure 1.0135 bar or 101.35 1(11/m2 and at temperature 0°C (273 K).

2. ***Moist air***. It is a mixture of dry air and water vapour. The amount of water vapour present. in the air depends upon the absolute pressure and temperature of the mixture.

3. ***Saturated air***. It is mixture of dry air and water vapour, when the air has diffused the maximum amount of water vapour into it. The water vapours, usually, occur in the form of superheated steam as an invisible gas. However, when the saturated air is cooled, the water vapour in the air starts condensing, and the same may be visible in the form of moist, fog or condensation on cold surfaces.

4. ***Degree of saturation***. It is the ratio of actual mass of water vapour in a unit mass of dry air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature.

5. ***Humidity***. It is the mass of water vapour present in 1 kg of dry air, and is generally expressed in terms of gram per kg of dry air (g / kg of dry air). It is also called specific humidity or humidity ratio.

6. ***Absolute humidity***. It is the mass of water vapour present in 1 m3 of dry air, and is generally expressed in terms of gram per cubic metre of dry air (g /m3 of dry air). It is also expressed in terms of grains per cubic metre of dry air. Mathematically, one kg of water vapour is equal to 15 430 grains.

7. ***Relative humidity***. It is the ratio of actual mass of water vapour in a given_volume of moist air to the mass of water vapour in the same volume of saturated air at the same temperature and pressure. It is briefly written as RH.

8. ***Dry bulb temperature***. It is the temperature of air recorded by a thermometer, when it is not affected by the moisture present in the air. The dry bulb temperature (briefly written as DBT) is generally denoted by $t_d$ or $t_{db}$.

9. ***Wet bulb temperature***. It is the temperature of air recorded by a thermometer, when its bulb is surrounded by a wet cloth exposed to the air. Such a thermometer is called *wet bulb thermometer. The wet bulb temperature (briefly written as WBT) is generally denoted by $t_w$ or $t_{wb}$.

10. ***Wet bulb depression***. It is the difference between dry bulb temperature and wet bulb temperature at any point. The wet bulb depression indicates relative humidity of the air.

11. ***Dew point temperature***. It is the temperature of air recorded by a thermometer, when the moisture (water vapour) present in it begins to condense. In other words, the dew point temperature is the saturation temperature ($t_{sat}$). corresponding to the partial pressure of water vapour ($P_v$) It is, usually, denoted by $t_{dp}$. Since $p_v$. is very small, therefore the saturation temperature by water vapour at pv is also low (less than the atmospheric or dry bulb temperature). Thus the water vapour in air exists in the superheated state and the moist air containing moisture in such a form (i.e., superheated state) is said to be unsaturated air. This condition is shown by point A on temperature-entropy (T-s) diagram as shown in Fig.1. When the partial pressure of water vapour (Pv) is equal to the saturation pressure (Ps) the water vapour is in dry condition and the air will be saturated air

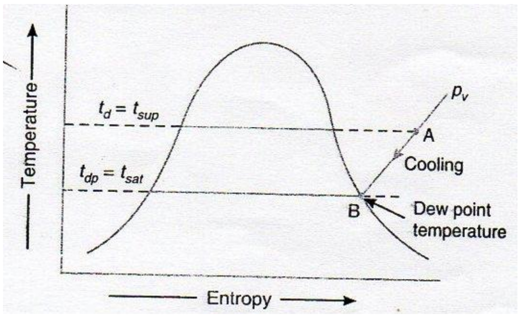


Fig.1. T-s diagram

If a sample of unsaturated air, containing superheated water vapour, is cooled at constant pressure, the partial pressure (pr) of each constituent remains constant until the water vapour reaches the saturated state as shown by point B in Fig.1. At this point 8, the first drop of dew will' be formed and hence the temperature at point B is called dew paint temperature. Further cooling will cause condensation of water vapour.

From the above we see that the dew point temperature is the temperature at which the water vapour begins to condense.

Note: For saturated air, the dry bulb temperature, wet bulb temperature and dew point temperature is same.

12. ***Dew point depression***. It is the difference between the dry bulb temperature and dew point temperature of air.

13. ***Psychrometer***. There are many types of psychrometers, but the sling psychrometer, as shown in Fig..2, is widely used. It consists of a dry bulb thermometer and a wet bulb thermometer mounted side by side in a protective case that is attached to a handle by a swivel connection so that the case can be easily rotated. The dry bulb thermometer is directly exposed to air and measures the actual temperature of the air. The bulb of the wet bulb thermometer is covered; by a wick thoroughly wetted by distilled water. The temperature measured by this wick covered bulb of a thermometer is the temperature of liquid water in the wick and is called wet Nib j temperature.

The sling psychrometer is rotated in the air for approximately one minute after which HO readings from both the thermometers are taken. This process is repeated several times to assure': that the lowest possible wet bulb temperature is recorded.

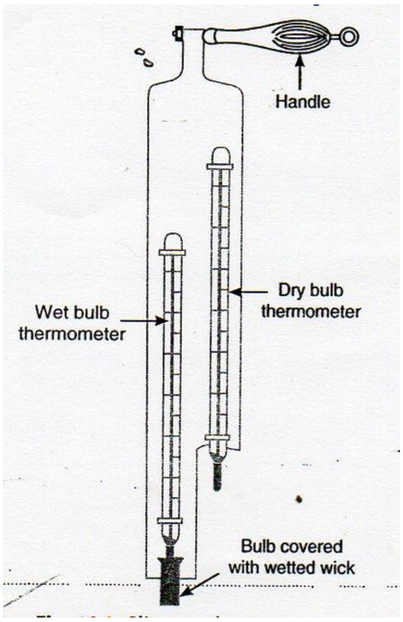


Fig.2, Sling psychrometer

### 3.3 DALTON'S LAW OF PARTIAL PRESSURES

It states, The total pressure exerted by the mixture of air and water vapour is equal to the sum of the pressures, which each constituent Fould exert, if it occupied the same space by itself. In other words, the total pressure exerted by air and water vapour mixture is equal to the barometric pressure. Mathematically, barometric pressure of the mixture,

$$P_b = P_a + P_v,$$

where $P_a$ = Partial pressure of dry air, and

$P_v$= Partial pressure of water vapour.

### 3.4 PSYCHROMETRIC RELATIONS

We have already discussed some psychrometric terms in Art. These terms have some relations between one another. The following psychrometric relations are important from the subject point of view:

1. ***Specific humidity***, humidity ratio or moisture content. It is the mass of water vapour present in 1 kg of dry air (in the air-vapour mixture) and is generally expressed in g /kg of dry air. It may also be defined as the ratio of mass of water vapour to the mass of dry air in a given volume of the air-vapour mixture.

Let $P_a$, $V_a$, $Ta$, $m_a$ and $R_a$ = Pressure, volume, absolute temperature, mass and gas constant respectively for dry air, and

$P_v$, $V_v$, $m_v$ and $R_v$ = Corresponding values for the water vapour.

Assuming that the dry air and water vapour behave as perfect gases, we have for dry air,

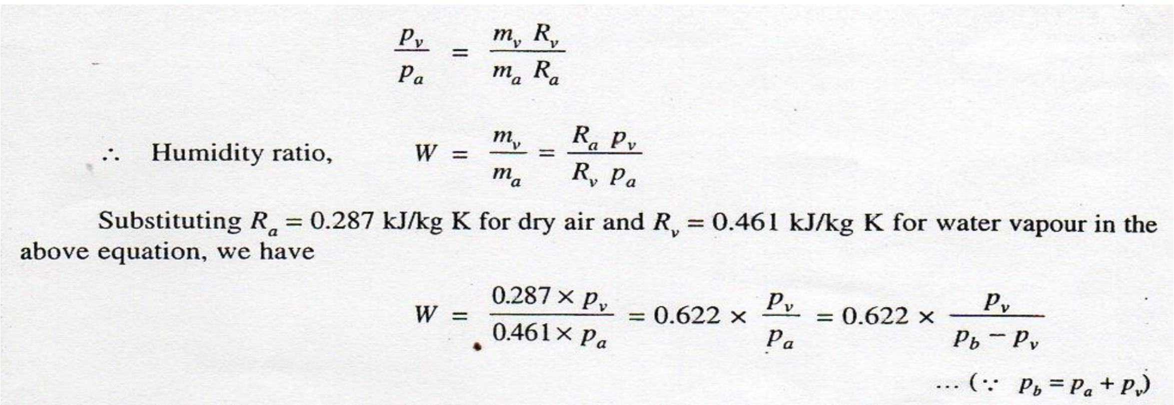
$$P_a v_a = m_a R_a T_a$$

and for water vapour, $P_v v_v = m_v R_v T_v$,

Also $v_a = v_v$

and $T_a = T_v, = T_d$ ... (where Td is dry bulb temperature)

From equations (i) and (ii), we have



$$\frac{p_v}{p_a} = \frac{m_v R_v}{m_a R_a}$$

∴ Humidity ratio, $W = \frac{m_v}{m_a} = \frac{R_a p_v}{R_v p_a}$

Substituting $R_a = 0.287$ kJ/kg K for dry air and $R_v = 0.461$ kJ/kg K for water vapour in the above equation, we have

$$W = \frac{0.287 \times p_v}{0.461 \times p_a} = 0.622 \times \frac{p_v}{p_a} = 0.622 \times \frac{p_v}{p_b - p_v}$$

$$\ldots (\because \; p_b = p_a + p_v)$$

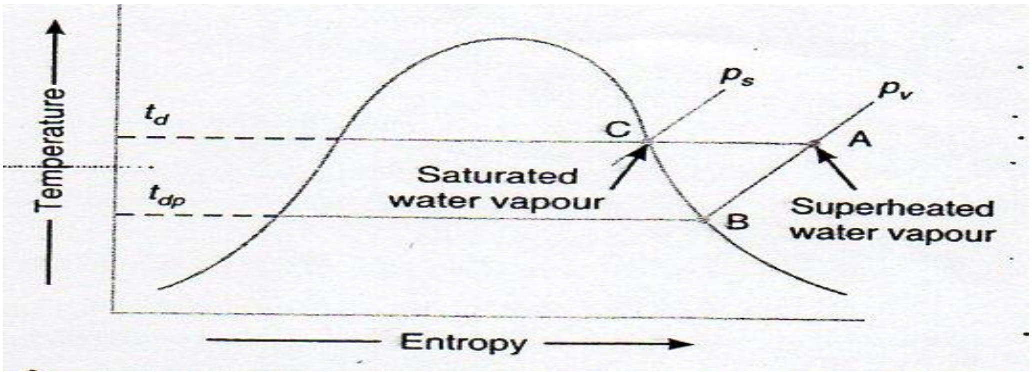


Fig.3 T-s diagram

Consider unsaturated air containing superheated vapour at dry bulb temperature $t_d$ and partial pressure $p_v$ as shown by point A on the *T-s* diagram in Fig. 3. If water is added into this unsaturated air, the water will evaporate which will increase the moisture content (specific humidity) of the air and the partial pressure $p_v$ increases. This will continue until the water vapour becomes saturated at that temperature, as shown by point C in Fig.3, and there will be more evaporation of water. The partial pressure $p_v$, increases to the saturation pressure $p_s$ and it is maximum partial pressure of water vapour at temperature $t_d$. The air containing moisture in such a state (point *C*) is called saturated air.

For saturated air (i.e. when the air is holding maximum amount of water vapour), the humidity ratio or maximum specific humidity,

$$W_s = W_{max} = 0.622 \times \frac{p_s}{p_b - p_s}$$

where $P_s$ = Partial pressure of air corresponding to saturation temperature (i.e. dry bulb temperature td).

2. ***Degree of saturation or percentage humidity***. We have already discussed that the degree of saturation is the ratio of vapour in a unit mass of water air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature (dry bulb temperature), it may be defined as the ratio of actual specific humidity to the specific humidity of saturated air at the same dry bulb temperature. It is, usually, denoted by $\mu$. Mathematically, degree of saturation,

$$\mu = \frac{W}{W_s} = \frac{\dfrac{0.622\, p_v}{p_b - p_v}}{\dfrac{0.622\, p_s}{p_b - p_s}} = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right) = \frac{p_v}{p_s}\left[\frac{1 - \dfrac{p_s}{p_b}}{1 - \dfrac{p_v}{p_b}}\right]$$

Notes: (a) The partial pressure of saturated air ($P_s$) is obtained from the steam tables corresponding to dry bulb temperature $t_d$.

(b) If the relative humidity,∅) = $Pv / Ps$ is equal to zero, then the humidity ratio, W = 0, i.e. for dry air, $\mu = 0$.

(c) If the relative humidity, ∅) $Pv / Ps$ is equal to 1, then W = Ws and $\mu = 1$. Thus p. varies between 0 and 1.

3. ***Relative humidity***. We have already discussed that the relative humidity is the ratio of actual mass of water vapour ($m_v$) in a given volume of moist air to the mass of water vapour ($m_s$) in the same volume of saturated air at the same temperature and pressure. It is usually denoted by ∅. Mathematically, relative humidity,

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

Let $p_v$, $v_v$ , $T_v$ , $m_v$ and $R_v$ = Pressure, volume, temperature, mass and gas constant respectively for

water vapour in actual conditions, and

$p_s$, $v_s$, $T_s$, $m_s$ and $Rs$ = Corresponding values for water vapour in saturated air.

We know that for water vapour in actual conditions,

$$P_v\, v_v = m_v\, R_v\, T_v \qquad \text{.....(i)}$$

Similarly, for water vapour in saturated air,

$$Ps\, vs = ms\, Rs\, Ts \qquad \text{...(ii)}$$

According to the definitions,

$$v_v = v_s$$

$$Tv = Ts$$

Also $Rv = Rs = 0.461$ kJ/kg K

∴ From equations (i) and (ii), relative humidity,

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

Thus, the relative humidity may also be defined as the ratio of actual partial pressure of water vapour in moist air at a given temperature (dry bulb temperature) to the saturation pressure of water vapour (or partial pressure of water vapour in saturated air) at the same temperature.

The relative humidity may also be obtained as discussed below:

We know that degree of saturation,

$$\mu = \frac{p_v}{p_s}\left[\frac{1-\dfrac{p_s}{p_b}}{1-\dfrac{p_v}{p_b}}\right] = \phi\left[\frac{1-\dfrac{p_s}{p_b}}{1-\phi\times\dfrac{p_s}{p_b}}\right] \qquad \cdots \left(\because\ \phi = \frac{p_v}{p_s}\right)$$

$$\phi = \frac{\mu}{1-(1-\mu)\dfrac{p_s}{p_b}}$$

4. ***Pressure of water vapour***. According to Carrier's equation, the partial pressure of water vapours,

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

Where $p_w$, = Saturation pressure corresponding to wet bulb temperature (from steam tables),

$P_b$ = Barometric pressure,

$t_d$ = Dry bulb temperature, and

$t_w$ = Wet bulb temperature.

5.***Vapour density or absolute humidity***. We have already discussed that the vapour density or absolute humidity is the mass of water vapour present in 1 $m^3$ of dry air.

Let $v_v$ = Volume of water vapour in $m^3$/kg of dry air at its partial pressure,

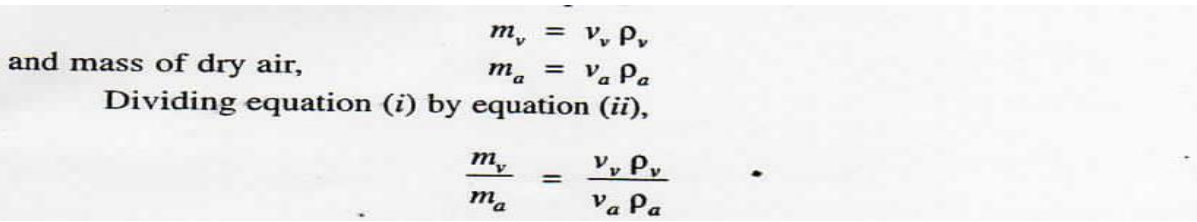
$v_a$ = Volume of dry air in $m^3$/kg of dry air at its partial pressure,

$\rho_v$, = Density of water vapour in kg/$m^3$ corresponding to its partial pressure and dry bulb

temperature $t_d$, and

$\rho_a$ = Density of dry air in kg/$m^3$ of dry air.

We know that mass of water vapour,

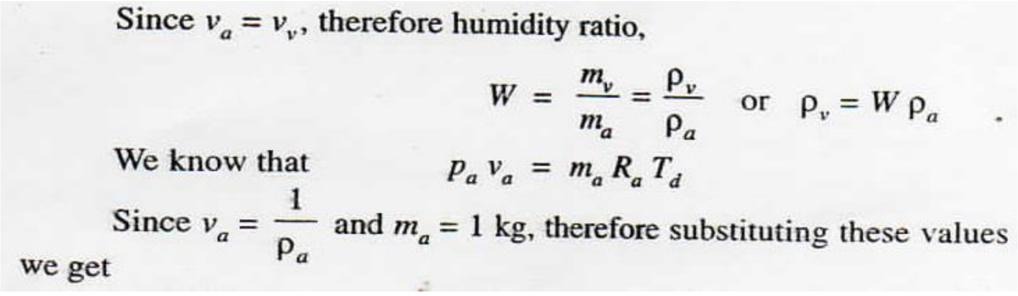


$$m_v = v_v \rho_v$$

and mass of dry air, $m_a = v_a \rho_a$

Dividing equation (*i*) by equation (*ii*),

$$\frac{m_v}{m_a} = \frac{v_v \rho_v}{v_a \rho_a}$$

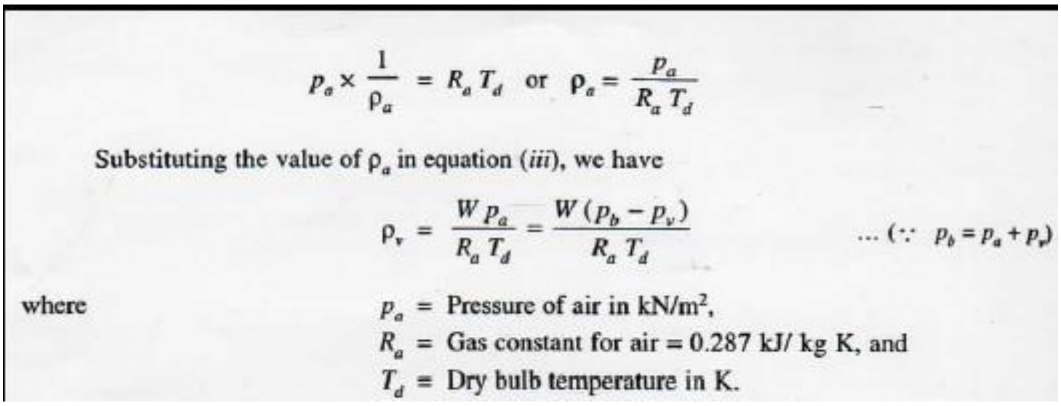


Since $v_a = v_v$, therefore humidity ratio,

$$W = \frac{m_v}{m_a} = \frac{\rho_v}{\rho_a} \quad \text{or} \quad \rho_v = W \rho_a$$

We know that $p_a v_a = m_a R_a T_d$

Since $v_a = \frac{1}{\rho_a}$ and $m_a$ = 1 kg, therefore substituting these values we get



$$p_a \times \frac{1}{\rho_a} = R_a T_d \quad \text{or} \quad \rho_a = \frac{p_a}{R_a T_d}$$

Substituting the value of $\rho_a$ in equation (*iii*), we have

$$\rho_v = \frac{W p_a}{R_a T_d} = \frac{W(p_b - p_v)}{R_a T_d} \qquad \ldots (\because \; p_b = p_a + p_v)$$

where $p_a$ = Pressure of air in kN/$m^2$,

$R_a$ = Gas constant for air = 0.287 kJ/ kg K, and

$T_d$ = Dry bulb temperature in K.

**Example.1. The readings from a sling psychrometer are as follows ry bulb temperature = 30° C ; Barometer reading 740mm of Hg Using steam tables, determine : I. Dew point temperature ; 2. Relative humidity ; 3. Specific humidity ; 4. Degree of-saturation ; 5. Vapour density ; and 6. Enthalpy of mixture per kg of dry air.**

Solution given: $t_d$ = 30°C ; $t_{w.}$ = 20°C ; $P_4$= 740 mm of Hg

**1.Dew point temperature**

First of all, let us find the partial pressure of water vapour ($P_v$).

From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 20° C is

$$P_w = 0.023\ 37 \text{ bar}$$

We know that barometric pressure,

ph = 740 mm of Hg ... (Given)

= 740 x 133.3 = 98 642 N/m$^2$ … (∵ mm of Hg = 133.3 N/m$^2$)

= 0.986 42 bar …. ∵1 bar = 10$^5$ N/m$^2$)

∴ Partial pressure of water vapour,

$$\begin{aligned} p_v &= p_w - \frac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44\, t_w} \\ &= 0.023\ 37 - \frac{(0.986\ 42 - 0.02337)(30 - 20)}{1544 - 1.44 \times 20} \\ &= 0.023\ 37 - 0.006\ 36 = 0.017\ 01 \text{ bar} \end{aligned}$$

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour ($P_v$), therefore from steam tables, we find that corresponding to pressure 0.017 01 bar, the dew point temperature is

$t_{dp}$ = 15°C Ans

***2. Relative humidity***

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30°C is

$$Ps = 0.042\ 42 \text{ bar}$$

We know the relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{0.01701}{0.04242} = 0.40 \text{ or } 40\% \quad \text{Ans.}$$

**3.** ***Specific humidity***

We know that specific humidity,

$$W = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622 \times 0.01701}{0.98642 - 0.01701}$$

$$= \frac{0.01058}{0.96941} = 0.010\,914 \text{ kg/kg of dry air}$$

$$= 10.914 \text{ g/kg of dry air} \quad \text{Ans.}$$

**4.** ***Degree of saturation***

We know that specific humidity of saturated air,

$$W_s = \frac{0.622\, p_s}{p_b - p_s} = \frac{0.622 \times 0.04242}{0.98642 - 0.04242}$$

$$= \frac{0.02638}{0.944} = 0.027\,945 \text{ kg/kg of dry air}$$

We know that degree of saturation,

$$\mu = \frac{W}{W_s} = \frac{0.010\,914}{0.027\,945} = 0.391 \text{ or } 39.1\% \quad \text{Ans.}$$

**Note :** The degree of saturation ($\mu$) may also be calculated from the following relation :

$$\mu = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right)$$

$$= \frac{0.01701}{0.04242}\left[\frac{0.98642 - 0.04242}{0.98642 - 0.01701}\right]$$

$$= 0.391 \text{ or } 39.1\% \quad \text{Ans.}$$

**5.** ***Vapour density***

We know that vapour density,

$$\rho_v = \frac{W\,(p_b - p_v)}{R_a\, T_d} = \frac{0.010\,914\,(0.98642 - 0.01701)\,10^5}{287\,(273 + 30)}$$

$$= 0.012\,16 \text{ kg/m}^3 \text{ of dry air} \quad \text{Ans.}$$

**6.** ***Enthalpy of mixture per kg of dry air***

From steam tables, we find that the latent heat of vaporisation of water at dew point temperature of 15°C is

$$h_{fgdp} = 2466.1 \text{ kJ/kg}$$

∴ Enthalpy of mixture per kg of dry air,

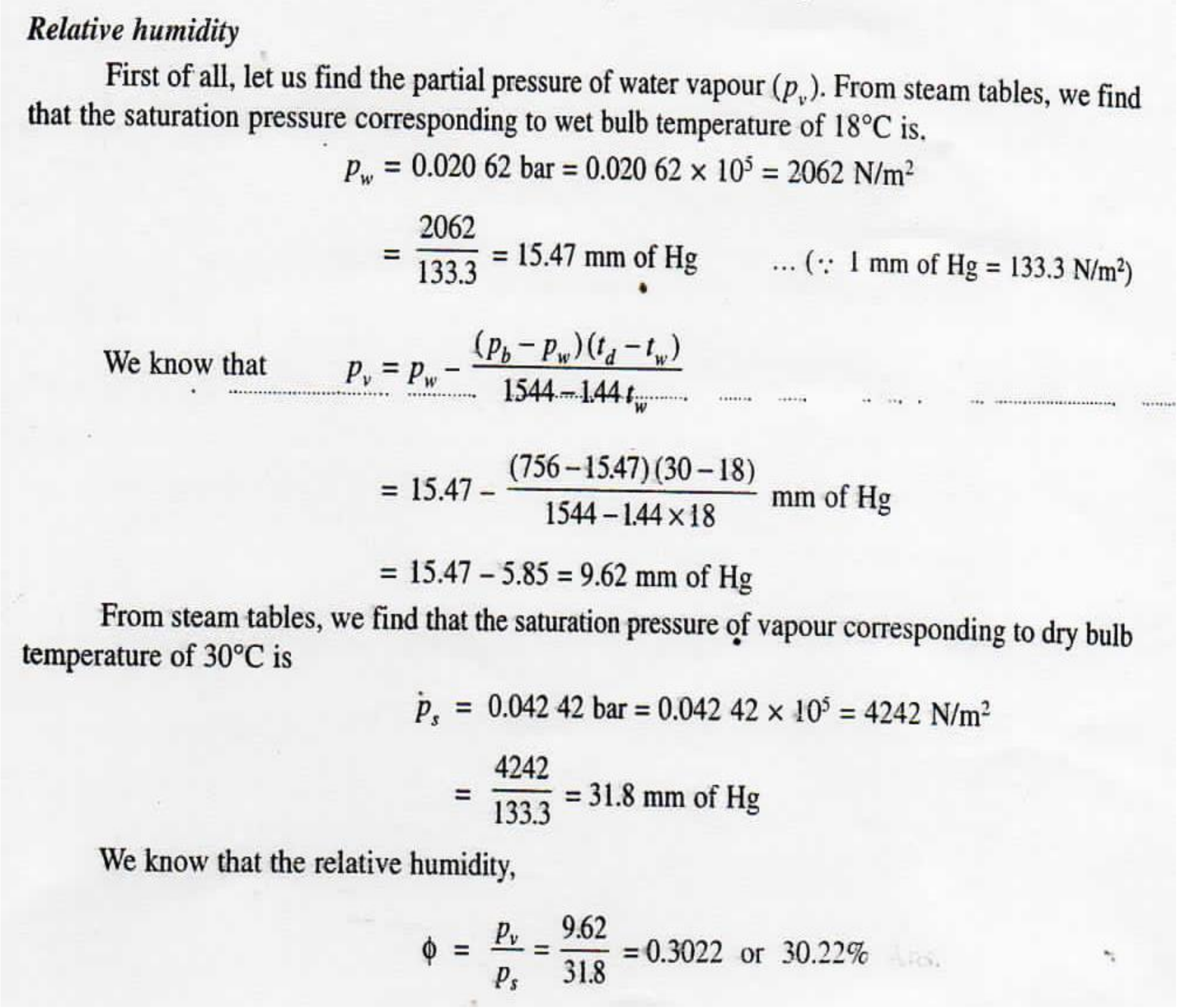
$$h = 1.022\, t_d + W\,[h_{fgdp} + 2.3\, t_{dp}]$$

$$= 1.022 \times 30 + 0.010\,914\,[2466.1 + 2.3 \times 15]$$

$$= 30.66 + 27.29 = 57.95 \text{ kJ/kg of dry air} \quad \text{Ans.}$$

**Example.2: On a particular day, the atmospheric air was found to have a dry bulb temperature of 30℃ and a wet bulb temperature of 18℃. The barometric pressure was observed to b 756mm of Hg. Using the tables of psychrometric properties of air, determine the relative humidity, the specific humidity, the dew point temperature, the enthalpy of air per kg of dry air and the volume of mixture per kg of dry air.**

Solution: Given: $t_d = 30°C$; $t_w - 18°C$; $P_b = 756\ mm$ of $Hg$



***Relative humidity***

First of all, let us find the partial pressure of water vapour ($p_v$). From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 18°C is.

$$p_w = 0.020\,62 \text{ bar} = 0.020\,62 \times 10^5 = 2062 \text{ N/m}^2$$

$$= \frac{2062}{133.3} = 15.47 \text{ mm of Hg} \qquad \ldots (\because 1 \text{ mm of Hg} = 133.3 \text{ N/m}^2)$$

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

$$= 15.47 - \frac{(756 - 15.47)(30 - 18)}{1544 - 1.44 \times 18} \text{ mm of Hg}$$

$$= 15.47 - 5.85 = 9.62 \text{ mm of Hg}$$

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30°C is

$$p_s = 0.042\,42 \text{ bar} = 0.042\,42 \times 10^5 = 4242 \text{ N/m}^2$$

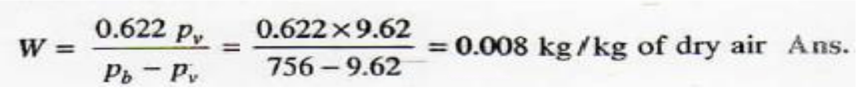
$$= \frac{4242}{133.3} = 31.8 \text{ mm of Hg}$$

We know that the relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{9.62}{31.8} = 0.3022 \text{ or } 30.22\% \text{ Ans.}$$

***Specific humidity***

We know that specific humidity,



$$W = \frac{0.622\,p_v}{p_b - p_v} = \frac{0.622 \times 9.62}{756 - 9.62} = \mathbf{0.008 \text{ kg/kg of dry air}} \text{ Ans.}$$

***Dew point temperature***

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour ($P_v$), therefore from steam tables, we find that corresponding to 9.62 mm of Hg or 9.62 x 133.3 = 1282.3 N/m$^2$ = 0.012 823 bar, the dew point temperature is,

$$t_{dp} = 10.6° \text{ C Ans.}$$

***Enthalpy of air per kg of dry air***

From steam tables, we also find that latent heat of vaporization of water at dew point temperature of 10.6°C,

$$h_{\text{fgdp}} = 2476.5 \text{ kJ/kg}$$

We know that enthalpy of air per kg of dry air,

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

$$= 1.022 \times 30 + 0.008\,(2476.5 + 2.3 \times 10.6)$$

$$= 30.66 + 20 = 50.66 \text{ kJ/kg of dry air Ans.}$$

***Volume of the mixture per kg of dry air***

From psychrometric tables, we find that specific volume of the dry air at 760 mm of Hg and 30°C dry bulb temperature is 0.8585 m$^3$/kg of dry air. We know that one kg of dry air at a partial pressure of (756 — 9.62) mm of Hg occupies the same volume as W = 0.008 kg of vapour at its partial pressure of 9.62 mm of Hg. Moreover, the mixture occupies the same volume but at a total pressure of 756 mm of Hg.

∴ Volume of the mixture (v) at a dry bulb temperature of 30°C and a pressure of 9.62 mm of Hg

= Volume of 1 kg of dry air ($v_a$) at a pressure of ( 756 — 9.62 ) or 746.38 mm of Hg

$$= 0.8585 \times \frac{760}{746.38} = 0.8741 \text{ kg/kg of dry air Ans.}$$

Note : The volume of mixture per kg of dry air may be calculated as discussed below :

We know that $$v = v_a = \frac{R_a\, T_d}{p_a}$$

where $R_a$ = Gas constant for air = 287 J/kg K

$T_d$ = Dry bulb temperature in K

= 30 + 273 = 303 K, and

$p_a$ = Pressure of air in N/m$^2$

$= P_b - P_v = 756 - 9.62 = 746.38$ mm of Hg

$= 746.38 \times 133.3 = 994\ 92$ N/m$^2$

Substituting the values in the above equation,

$$v = \frac{287 \times 303}{994\,92} = 0.8741 \text{ m}^3\text{/kg of dry air} \quad \text{Ans.}$$

**Example.3. The humidity ratio of atmospheric air at 28°C dry bulb temperature and 760 mm of mercury is 0.016 kg / kg of dry air. Determine: 1. partial pressure of Water vapour; 2.relative humidity; 3. dew point temperature; 4. specific enthalpy; and 5. vapour density**.

Solution: Given: $t_d = 28°$C ; $P_b = 760$ mm of Hg ; W = 0.016 kg/ kg of dry air

***1.Partial pressure of water vapour***

Let $P_v$ = Partial pressure of water vapour.
We know that humidity ratio (W),

$$0.016 = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622\, p_v}{760 - p_v}$$

$12.16 - 0.016\, P_v = 0.622\, P_v$ or $0.638\, P_v = 12.16$
$P_v = 12.16/0.638 = 19.06$ mm of Hg
$= 19.06 \times 133.3 = 2540.6$ N/m$^2$ Ans.

***2. Relative humidity***

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 28`C is

$P_s = 0.03778$ bar $= 3778$ N/m$^2$

∴ Relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{2540.6}{3778} = 0.672 \text{ or } 67.2\% \quad \text{Ans.}$$

***3. Dew point temperature***

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour ($P_v$), therefore from steam tables, we find that corresponding to a pressure of 2540.6 N/m$^2$ (0.025406 bar), the dew point temperature is,

$t_{dp} = 21.1°$ C Ans.

### *4. Specific enthalpy*

From steam tables, latent heat of vaporization of water corresponding to a dew point temperature of 21.1° C,

$$h_{fgdp} = 2451.76 \text{ kJ/kg}$$

We know that specific enthalpy.

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

$$= 1.022 \times 28 + 0.016\,(2451.76 + 2.3 \times 21\_1)$$

$$= 28.62 + 40 - 68.62 \text{ kJ/kg of dry air Ans.}$$

### *5. Vapour density*

We know that vapour density,

$$\rho_v = \frac{W\,(p_b - p_v)'}{R_a\, T_d} = \frac{0.016\,(760 - 19.06)\,133.3}{287\,(273 + 28)}$$

$$= 0.0183 \text{ kg/m}^3 \text{ of dry air.}$$

## 3.5 THERMODYNAMIC WET BULB TEMPERATURE OR ADIABATIC SATURATION TEMPERATURE

The thermodynamic wet bulb temperature or adiabatic saturation temperature is the temperature at which the air can be brought to saturation state, adiabatically, by the evaporation of water into the flowing air.

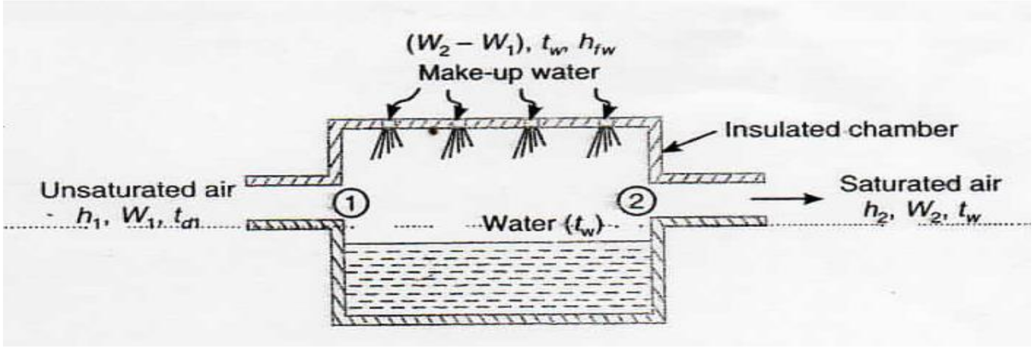


Fig.4 Adiabatic saturation of air.

The equipment used for the adiabatic saturation of air, in its simplest form, consists of an insulated chamber containing adequate quantity of water. There is also an arrangement for extra water (known as make-up water) to flow into the chamber from its top, as shown in Fig.4.

Let the unsaturated air enters the chamber at section 1. As the air passes through the chamber over a long sheet of water, the water evaporates which is carried with the flowing stream of air, and the specific humidity of the air increases. The make-up water is added to the chamber at this temperature to make the water level constant. Both the air and water are cooled as the evaporation takes place. This process continues until the energy transferred from the air to the water is equal to the energy required to vaporize the water. When steady conditions are reached, the air flowing at section 2 is saturated with water vapour. The temperature of the saturated air at section 2 is known as *thermodynamic wet bulb temperature or adiabatic saturation temperature.*

The adiabatic saturation process can be represented on *T-s* diagram as shown by the curve 1-2 in Fig.5.

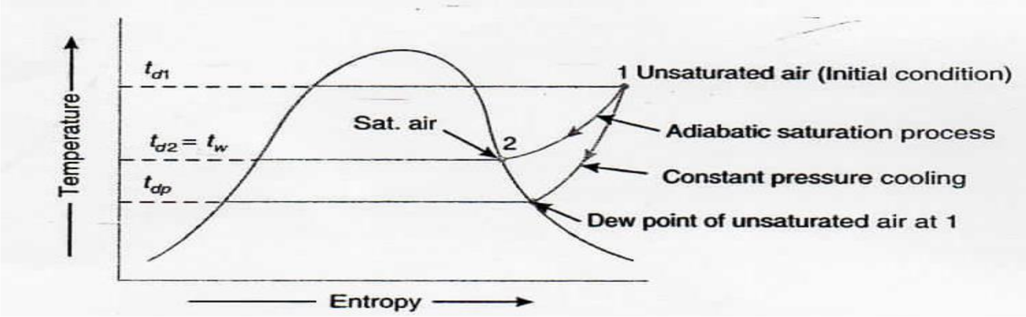


Fig.5. T-s diagram for adiabatic saturation process

During the adiabatic saturation process, the partial pressure of vapour increases, although the total ressure of the air-vapour mixture. The unsaturated air initially at dry bulb temperature $t_{d2}$, is coo e adiabatically to dry bulb temperature td, which is equal to the adiabatic saturation temperature $t_w$. It may be noted that the adiabatic saturation temperature is taken equal to the wet bulb temperature for all practical purposes.

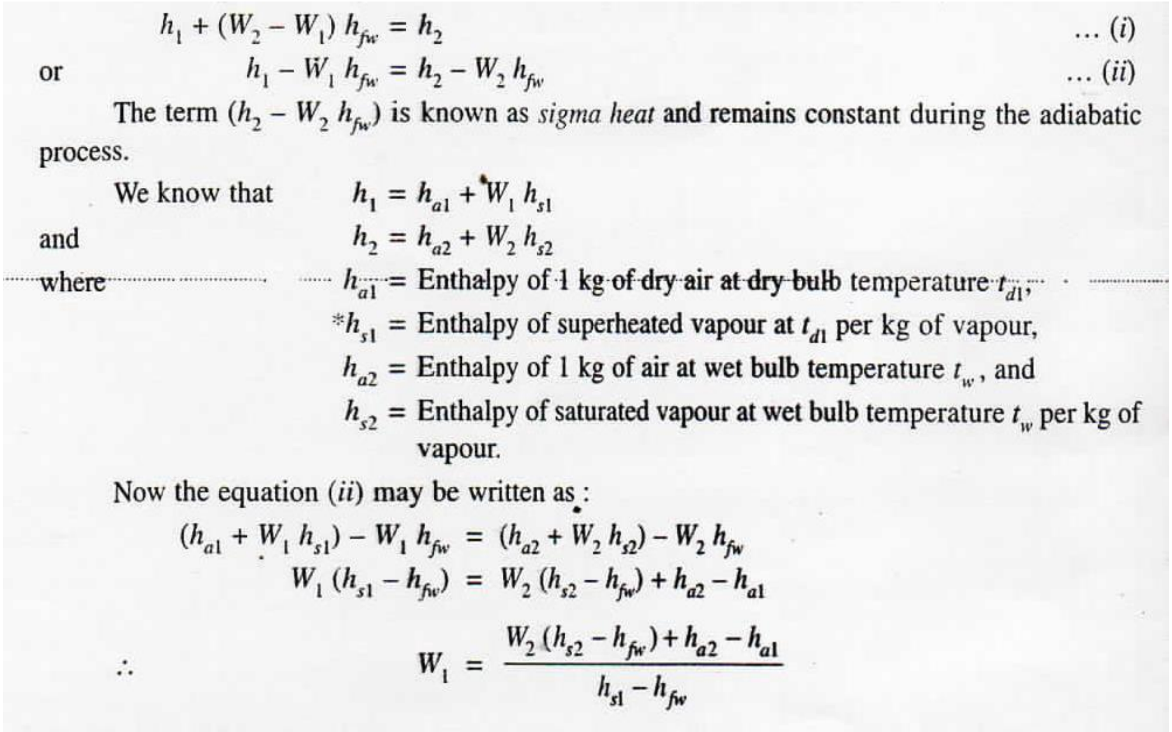
Let $h_1$ = Enthalpy of unsaturated air at section 1,

$W_1$ = Specific humidity of air at section 1,

$h_2, W_2$ = Corresponding values of saturated air at section 2, and

$h_{fw}$= Sensible heat of water at adiabatic saturation temperature.

Balancing the enthalpies of air at inlet and outlet (i.e. at sections 1 and 2),



$$h_1 + (W_2 - W_1)\, h_{fw} = h_2 \qquad \dots (i)$$

or

$$h_1 - W_1\, h_{fw} = h_2 - W_2\, h_{fw} \qquad \dots (ii)$$

The term $(h_2 - W_2\, h_{fw})$ is known as *sigma heat* and remains constant during the adiabatic process.

We know that $h_1 = h_{a1} + W_1\, h_{s1}$

and $h_2 = h_{a2} + W_2\, h_{s2}$

where $h_{a1}$ = Enthalpy of 1 kg of dry air at dry bulb temperature $t_{d1}$,

$^*h_{s1}$ = Enthalpy of superheated vapour at $t_{d1}$ per kg of vapour,

$h_{a2}$ = Enthalpy of 1 kg of air at wet bulb temperature $t_w$, and

$h_{s2}$ = Enthalpy of saturated vapour at wet bulb temperature $t_w$ per kg of vapour.

Now the equation $(ii)$ may be written as :

$$(h_{a1} + W_1\, h_{s1}) - W_1\, h_{fw} = (h_{a2} + W_2\, h_{s2}) - W_2\, h_{fw}$$

$$W_1\,(h_{s1} - h_{fw}) = W_2\,(h_{s2} - h_{fw}) + h_{a2} - h_{a1}$$

$$\therefore \qquad W_1 = \frac{W_2\,(h_{s2} - h_{fw}) + h_{a2} - h_{a1}}{h_{s1} - h_{fw}}$$

## 3.6 PSYCHROMETRIC CHART

It is a graphical representation of the various thermodynamic properties of moist air. The psychrometric chart is very useful for finding out the properties of air (which are required in the field of air conditioning) and eliminate lot of calculations. There is a slight variation in the charts prepared by different air-conditioning manufactures but basically they are all alike. The psychrometric chart is normally drawn for standard atmospheric pressure of 760 mm of Hg (or 1.01325 bar).

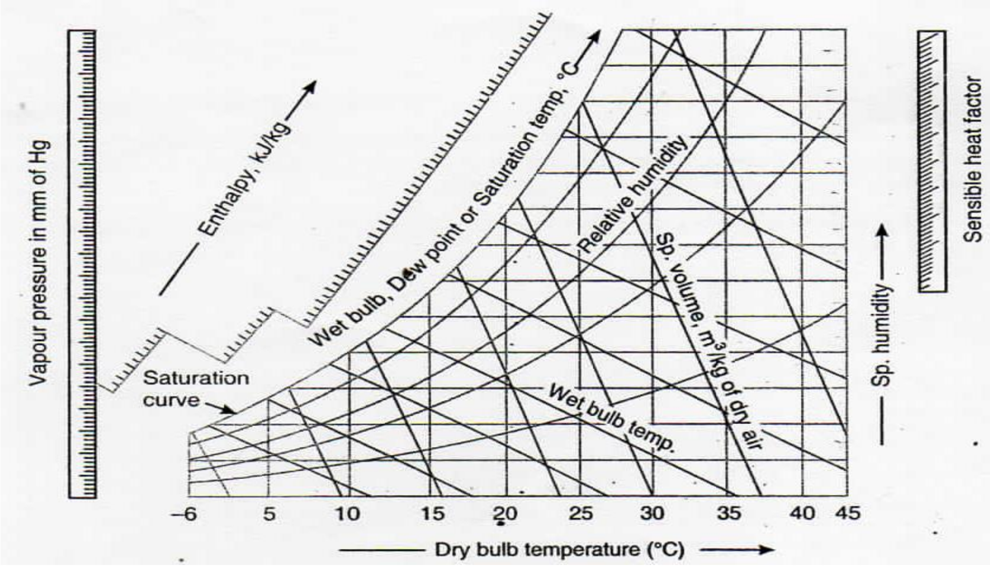


Fig. 6 Psychrometric chart.

In a psychrometric chart, dry bulb temperature is taken as abscissa and specific humidity i.e. moisture contents as ordinate, as shown in Fig. 6. Now the saturation curve is drawn by plotting the various saturation points at corresponding dry bulb temperatures. The saturation curve represents 100% relative humidity at various dry bulb temperatures. It also represents the wet bulb and dew point temperatures.

Though the psychrometric chart has a number of details, yet the following lines are important frpm the subject point of view :

1. ***Dry bulb temperature lines***. The dry bulb temperature lines are vertical i.e. parallel to the ordinate and uniformly spaced as shown in Fig. 7. Generally the temperature range of these lines on psychrometric chart is from - 6° C to 45° C. The dry bulb temperature lines are drawn with difference of every 5°C and up to the saturation curve as shown in the figure. The values of dry bulb temperatures are also shown on the saturation curve.

2. ***Specific humidity or moisture content lines***. The specific humidity (moisture content) lines are horizontal i.e. parallel to the abscissa and are also uniformly spaced as shown in Fig. 16.8. Generally, moisture content range of these lines on psychrometric chart is from 0 to 30 g / kg of dry air (or from 0 to 0.030 kg / kg of dry air). The moisture content lines are drawn with a difference of every 1 g (or 0.001 kg) and up to the saturation curve as shown in the figure.

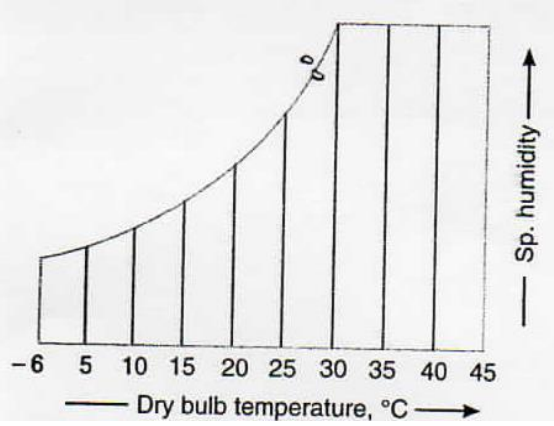


Fig.7. Dry bulb temperature lines.

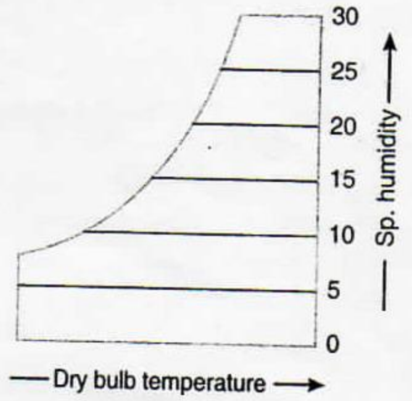


Fig. 8. Specific humidity lines.

3. ***Dew point temperature lines***. The dew point temperature lines are horizontal i.e. parallel to the abscissa and non-uniformly spaced as shown in Fig. 16.9. At any point on the saturation curve, the dry bulb and dew point temperatures are equal.

The values of dew point temperatures are generally given along the saturation curve of the chart as shown in the figure.

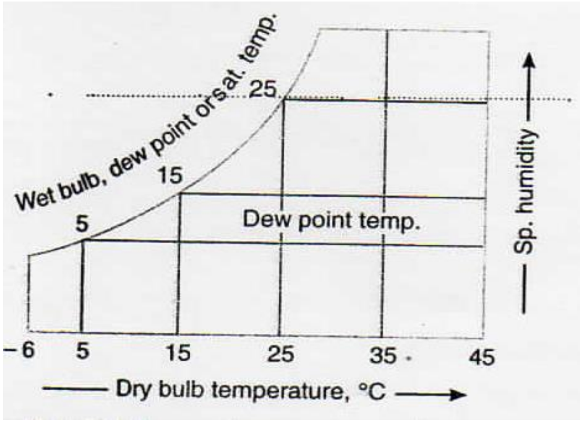


Fig. 9 Dew point temperature lines.

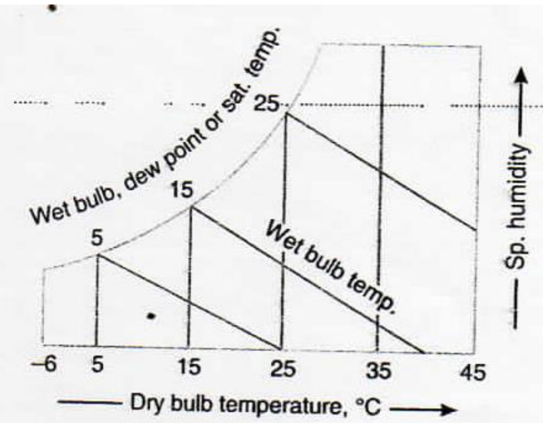


Fig.10 Wet bulb temperature lines.

4. ***Wet bulb temperature lines***. The wet bulb temperature lines are inclined straight lines and non-uniformly spaced as shown in Fig.10. At any point on the saturation curve, the dry bulb and wet bulb temperatures are equal.

The values of wet bulb temperatures are generally given along the saturation curve of the chart as shown in the figure.

5. ***Enthalpy (total heat) lines***. The enthalpy (or total heat) lines are inclined straight lines and uniformly spaced as shown in Fig.11. These lines are parallel to the wet bulb temperature lines, and are drawn up to the saturation curve. Some of these lines coincide with the wet bulb temperature lines also.

The values of total enthalpy are given on a scale above the saturation curve as shown in the figure.

6. ***Specific volume lines.*** The specific volume lines are obliquely inclined straight lines and uniformly spaced as shown in Fig.12. These lines are drawn up to the saturation curve. The values of volume lines are generally given at the base of the chart.

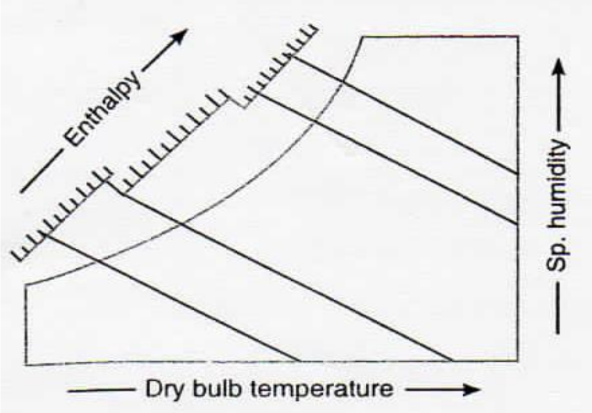


Fig. 11. Enthalpy lines.

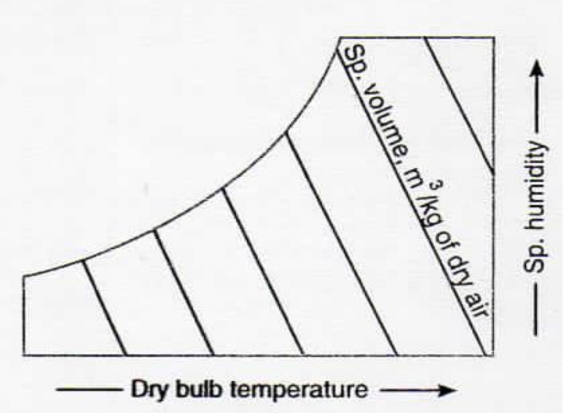
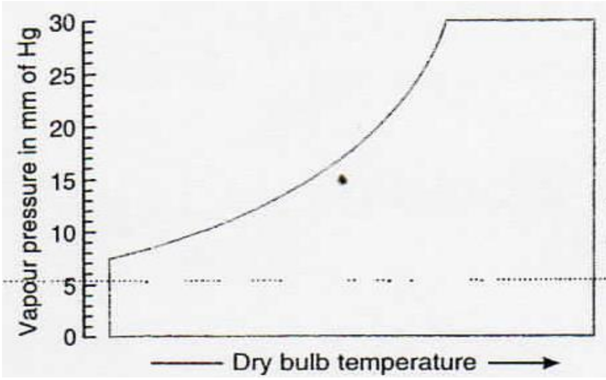


Fig. 12. Specific volume lines.

7. ***Vapour pressure lines***. The vapour pressure lines are horizontal and uniformly spaced. Generally, the vapour pressure lines are not drawn in the main chart. But a scale showing vapour pressure in mm of Hg is given on the extreme left side of the chart as shown in Fig.13.



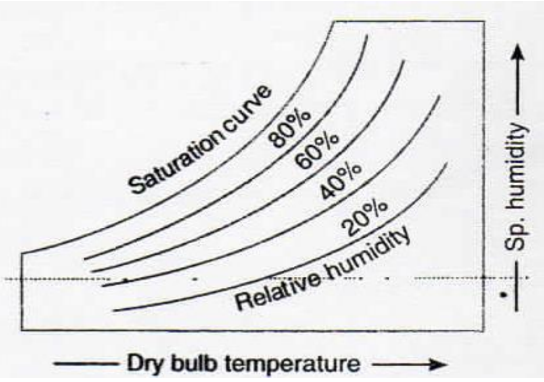


Fig. 13. Vapour pressure lines. Fig. 14. Relative humidity lines.

8. ***Relative humidity lines***. The relative humidity lines are curved lines and follow the saturation curve. Generally, these lines are drawn with values 10%, 20%, 30% etc. and up to 100%. The saturation curve represents 100% relative humidity. The values of relative humidity lines are generally given along the lines themselves as shown in Fig. 14.

## 3.7 PSYCHROMETRIC PROCESSES

The various psychrometric processes involved in air conditioning to vary the psychrometric properties of air according to the requirement are as follows:

1. Sensible heating, 2. Sensible cooling, 3. Humidification and dehumidification, 4. Cooling and adiabatic humidification, 5. Cooling and humidification by water injection, 6. Heating and humidification, 7. Humidification by steam injection, 8. Adiabatic chemical dehumidification, 9. Adiabatic mixing of air streams.

We shall now discuss these psychrometric processes, in detail, in the following pages.

### 3.71 Sensible Heating

The heating of air, without any-change in its specific humidity, is known as sensible heating. Let air at temperature td, passes over a heating coil of temperature $t_{d3}$, as shown in Fig. 15 (a). It may be noted that the temperature of air leaving the heating coil ($t_{d2}$) will be less than $t_{d3}$. The process of sensible heating, on the psychrometric chart, is shown by a horizontal line 1-2 extending from left to right as shown in Fig.15 (b). The point 3 represents the surface temperature of the heating coil.

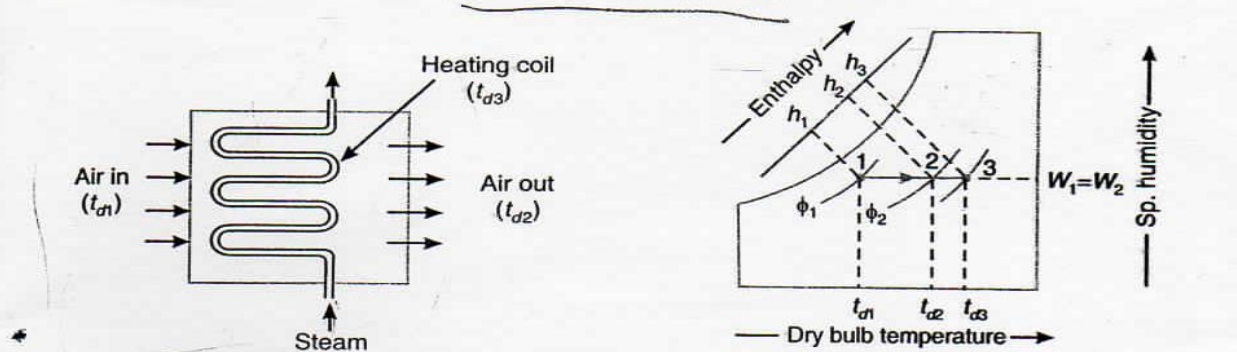
The heat absorbed by the air during sensible heating may be obtained from the psychrometric chart by the enthalpy difference ($h_2$ - $h_1$) as shown in Fig. 15 (b). It may be noted that the specific humidity during the sensible heating remains constant (i.e. $W_1 = W_2$).

The dry bulb temperature increases from $t_{d1}$, to $t_{d2}$ and relative humidity reduces from $\emptyset_1$, to $\emptyset_2$ as shown in Fig. 15 (b). The amount of heat added during sensible heating may also be obtained from the relation:

$$\text{Heat added,} \quad q = h_2 - h_1$$
$$= c_{pa}(t_{d2} - t_{d1}) + W\,c_{ps}(t_{d2} - t_{d1})$$
$$= (c_{pa} + W\,c_{ps})(t_{d2} - t_{d1}) = c_{pm}(t_{d2} - t_{d1})$$

The term $(c_{pa} + W\,c_{ps})$ is called *humid specific heat* $(c_{pm})$ and its value is taken as 1.022 kJ /kg K.

$$\therefore \quad \text{Heat added,} \quad q = 1.022\,(t_{d2} - t_{d1}) \text{ kJ/kg}$$



(*a*) Psychrometric process. (*b*) Psychrometric chart.

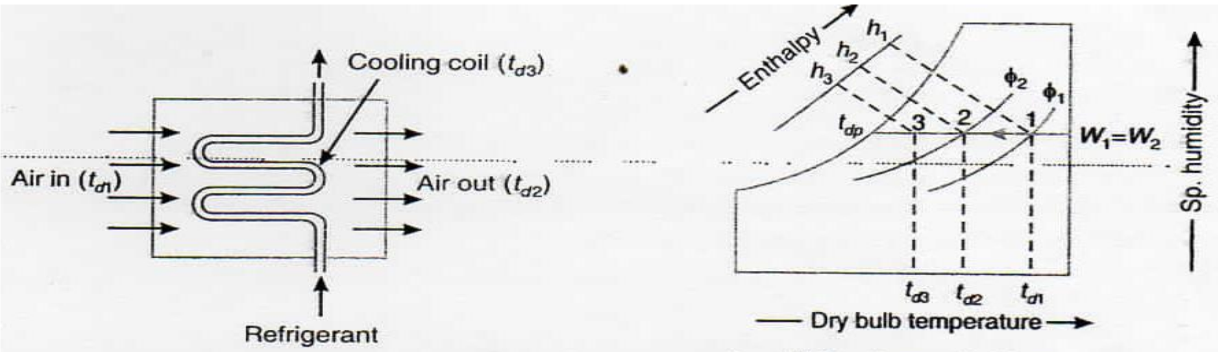
Fig.15 Sensible heating

Notes: 1. For sensible heating, steam or hot water is passed through the heating coil. The heating coil may be electric resistance coil.

2. The sensible heating of moist air can be done to any desired temperature.

### *3.72 Sensible Cooling*

The cooling of air without any change in its specific humidity, is known as sensible cooling. Let air at temperature rd, passes over a cooling coil of temperature $t_{d3}$ as shown in Fig. 16 (a). It may be noted that the temperature of air leaving the cooling coil ($t_{d2}$) will be more than $t_{d3}$. The process of sensible cooling, on the psychrometric chart, is shown by a horizontal line 1-2 extending from right to left as shown in Fig. 16



(*a*) Psychrometric process. (*b*) Psychrometric chart.

Fig. 16 Sensible cooling.

The heat rejected by air during sensible cooling may be obtained from the psychrometric chart by the enthalpy difference (h1 — h,) as shown in Fig. 16(b).

It may be noted that the specific humidity during the sensible cooling remains constant (i.e.$W_1 = W_2$). The dry bulb temperature reduces from $t_{d1}$ to $t_{d2}$ and relative humidity increases from $\emptyset_1$ to $\emptyset_2$ as shown in Fig. 16(b). The amount of heat rejected during sensible cooling may also be obtained from the relation:

Heat rejected, $q = h_1 - h_2$

$$= C_{pa} (t_{d1} - t_{d2}) + W\, C_{ps}(t_{d1} - t_{d2})$$

$$= (C_{pa} + W\, C_{ps})\, (t_{d1} - t_{d2}) = C_{pm}\, (t_{d1} - t_{d2})$$

The term ($C_{pa} + W\, C_{ps}$) is called humid specific heat ($C_{pm}$) and its value is taken as 1.022 kJ /kg K.

∴ Heat rejected, $q = 1.022\, (t_{d1} - t_{d2})$ kJ/kg

For air conditioning purposes, the sensible heat per minute is given as

$$SH = m_a\, C_{pm}\, \Delta t = v\rho\, C_{pm}\, \Delta t \text{ kJ/min} \qquad \ldots (\because m = v\rho)$$

where v = Rate of dry air flowing in $m^3$/min,

$\rho$ = Density of moist air at 20° C and 50% relative humidity

= 1.2 kg / m3 of dry air,

$C_{pm}$ = Humid specific heat = 1.022 kJ /kg K, and

$\Delta t = t_{d1} - t_{d2}$ = Difference of dry bulb temperatures between the entering and leaving conditions of air in ° C.

Substituting the values of p and cp,,, in the above expression, we get

$$SH = v \times 1.2 \times 1.022 \times \Delta t = 1.2264\, v \times \Delta t \text{ kJ/min}$$

$$= \frac{1.2264\, v \times \Delta t}{60} = 0.020\,44\, v \times \Delta t \text{ kJ/s or kW} \qquad \ldots (\because 1 \text{ kJ/s} = 1 \text{ kW})$$

### 3.73 By-pass Factor of Heating and Cooling Coil

The temperature of the air corning out of the apparatus ($t_{d2}$) will be less than *$t_{d3}$ in case the coil is a heating coil and more than $t_{d3}$ in case the coil is a cooling coil.

Let 1 kg of air at temperature $t_{d1}$ is passed over the coil having its temperature (i.e. coil surface temperature) $t_{d3}$ as shown in Fig. 17.

A little consideration will show that when air passes over a coil, some of it (say *x* kg) just by-passes unaffected while the remaining (1 - *x*) kg comes in direct contact with the coil. This by-pass process of air is measured in terms of a by-pass factor. The amount of air that by-passes or the by-pass factor depends upon the following factors :

1. The number of fins provided in a unit length i.e. the pitch of the cooling coil fins ;
2. The number of rows in a coil in the direction of flow; and
3. The velocity of flow of air.

It may be noted that the by-pass factor of a cooling coil decreases with decrease in fin spacing and increase in number of rows.

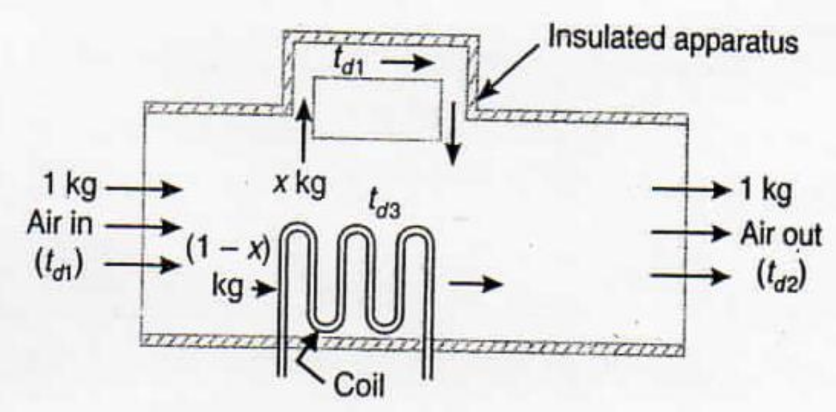


Fig.17. By-pass factor

Balancing the enthalpies, we get

$$x\,c_{pm}\,t_{d1} + (1-x)\,c_{pm}\,t_{d3} = 1 \times c_{pm}\,t_{d2} \qquad \text{... ( where } c_{pm} = \text{Specific humid heat)}$$

or

$$x\,(t_{d3} - t_{d1}) = t_{d3} - t_{d2}$$

$$\therefore \qquad x = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

where *x* is called the *by-pass factor* of the coil and is generally written as *BPF*. Therefore, by-pass factor for heating coil,

$$BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

Similarly, *by-pass factor for cooling coil,

$$BPF = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

The by-pass factor for heating or cooling coil may also be obtained as discussed below :

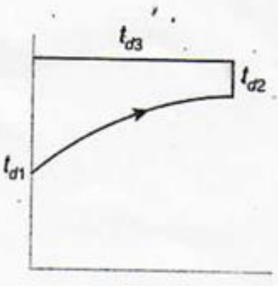
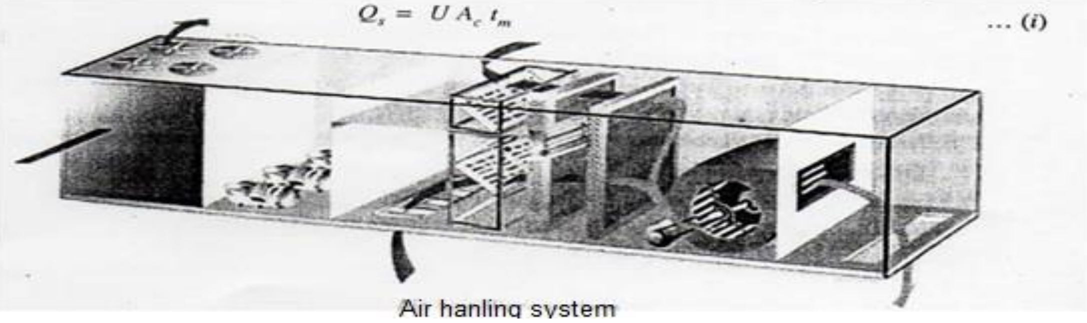


**Fig. 18.**

Let the air passes over a heating coil. Since the temperature distribution of air passing through the heating coil is as shown in Fig.18. therefore sensible heat given out by the coil.

$$Q_s = U A_c t_m \quad \ldots (i)$$



Air hanling system

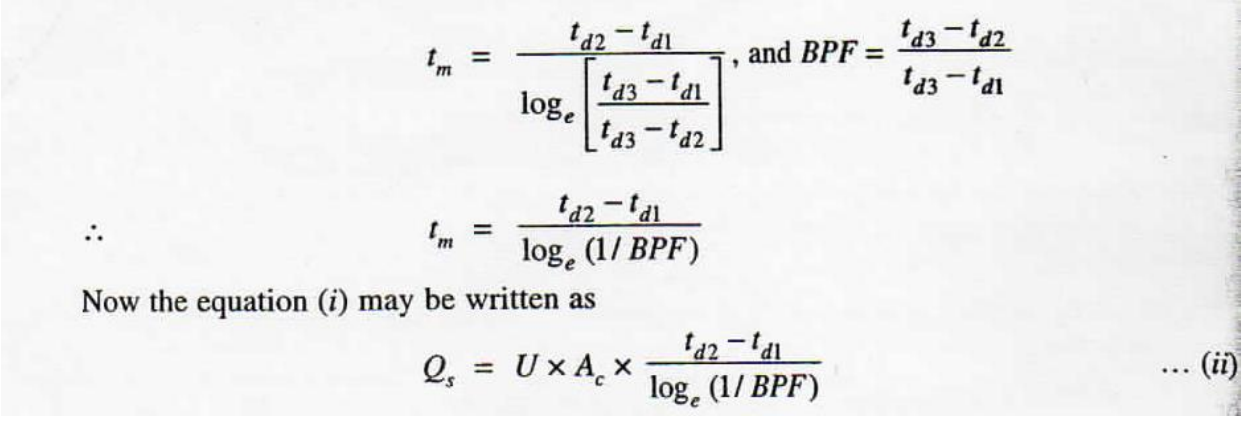
where

$U$ = Overall heat transfer coefficient,

$A_c$ = Surface area of the coil, and

$t_m$ = Logarithmic mean temperature difference.

We know that logarithmic mean temperature difference,

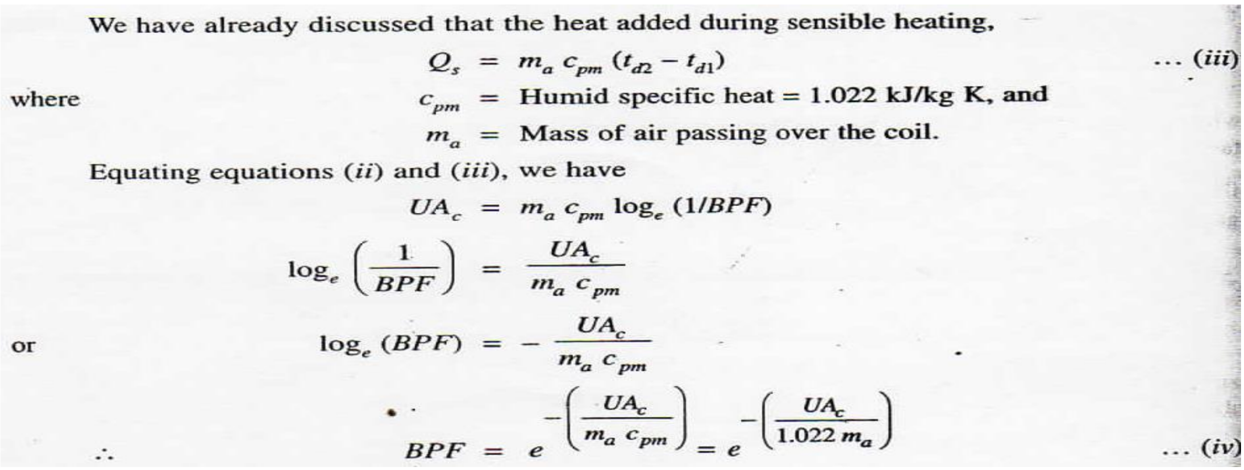


$$t_m = \frac{t_{d2} - t_{d1}}{\log_e \left[\frac{t_{d3} - t_{d1}}{t_{d3} - t_{d2}}\right]}, \text{ and } BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

$$\therefore \quad t_m = \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)}$$

Now the equation (*i*) may be written as

$$Q_s = U \times A_c \times \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)} \quad \ldots (ii)$$



We have already discussed that the heat added during sensible heating,

$$Q_s = m_a \, c_{pm} \, (t_{d2} - t_{d1}) \quad \ldots (iii)$$

where $c_{pm}$ = Humid specific heat = 1.022 kJ/kg K, and

$m_a$ = Mass of air passing over the coil.

Equating equations (*ii*) and (*iii*), we have

$$UA_c = m_a \, c_{pm} \log_e (1/BPF)$$

$$\log_e \left(\frac{1}{BPF}\right) = \frac{UA_c}{m_a \, c_{pm}}$$

or

$$\log_e (BPF) = -\frac{UA_c}{m_a \, c_{pm}}$$

$$\therefore \quad BPF = e^{-\left(\frac{UA_c}{m_a \, c_{pm}}\right)} = e^{-\left(\frac{UA_c}{1.022 \, m_a}\right)} \quad \ldots (iv)$$

Proceeding in the same• way as discussed above, we can derive the equation (iv) for a cooling coil.

Note: The performance of a heating or cooling coil is measured in terms of a by-pass factor. A coil with low by-pass factor has better performance.

### 3.74 Efficiency of Heating and Cooling Coils

The term (1 - BPF) is known as efficiency of coil or contact factor.

∴ Efficiency of the heating coil,

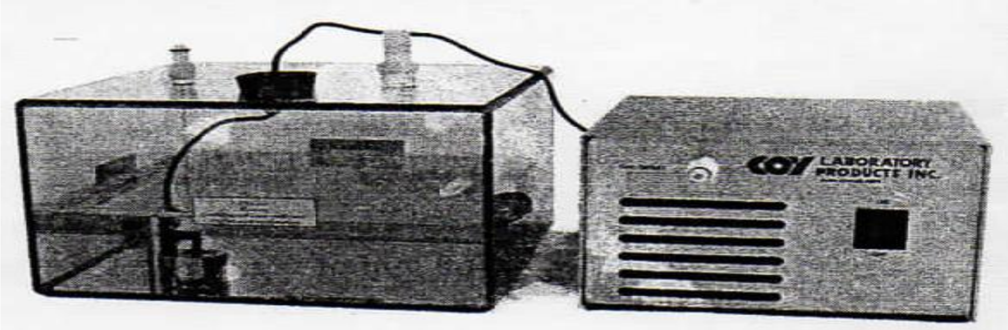
$$\eta_H = 1 - BPF = 1 - \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}} = \frac{t_{d2} - t_{d1}}{t_{d3} - t_{d1}}$$

Similarly, efficiency of the cooling coil,

$$\eta_C = 1 - \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}} = \frac{t_{d1} - t_{d2}}{t_{d1} - t_{d3}}$$

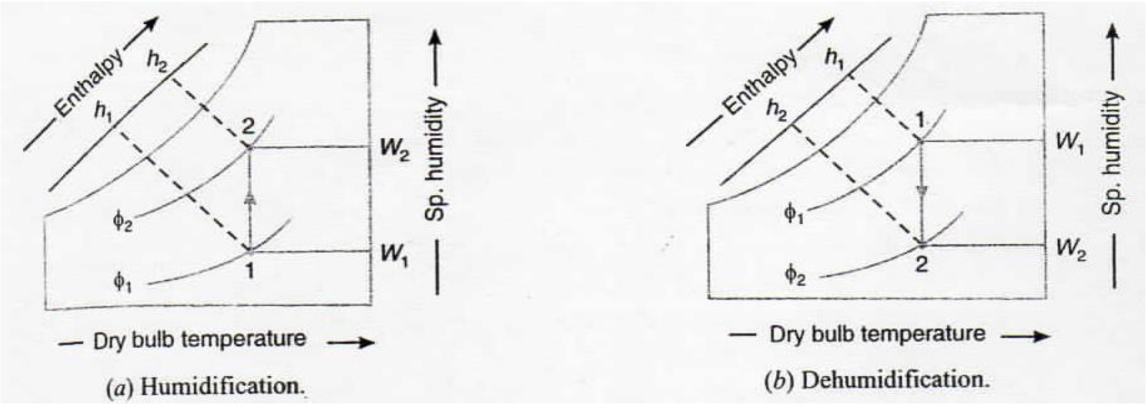
### 3.75 Humidification and Dehumidification

The addition of moisture to the air, without change in its dry bulb temperature, is known as *humidification*. Similarly, removal of moisture from the air, without change in its dry bulb temperature, is known as *dehumidification*. The heat added during humidification process and heat removed during dehumidification process is shown on the psychrometric chart in Fig. 19 (a) and (b) respectively.



Ultrasonic humidification system

It may be noted that in humidification, the relative humidity increases from $\emptyset_1$ to $\emptyset_2$ and specific humidity also increases from $W_1$ to $W_2$ as shown in Fig. 19 (a). Similarly, in dehumidification, the relative humidity decreases from $\emptyset_1$ to $\emptyset_2$ and specific humidity also decreases from $W_1$ to $W_2$ as shown in Fig. 19 (b).

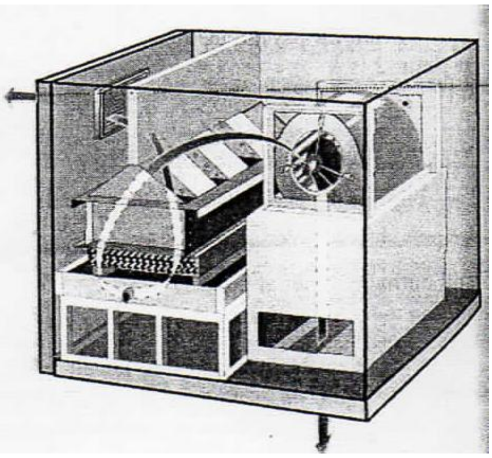


(*a*) Humidification.

(*b*) Dehumidification.

Fig. 19 Humidification and dehumidification

It may be noted that in humidification, change in enthalpy is shown by the intercept ($h_2$ - $h_1$) on the psychrometric chart. Since the dry bulb temperature of air during the humidification remains constant, therefore its sensible heat also remains constant. It is thus obvious that the change in enthalpy per kg of dry air due to the increased moisture content equal to ($W_2$ - $W_1$) kg per kg of dry air is considered to cause a latent heat transfer (LH). Mathematically,



Multiple small plate dehumidification system

LH = ($h_2$ - $h_1$) = $h_{fg}$ ($W_2$ - $W_1$) where $h_{fg}$ is the latent heat of vaporization at dry bulb temperature ($t_{dt}$).

Notes: 1. For dehumidification, the above equation may be written as:

$$\text{LH} = (h_1 - h_2) = h_{fg}\,(W_1 - W_2)$$

2. Absolute humidification and dehumidification processes are rarely found in practice. These are always accompanied by heating or cooling processes.

3. In air conditioning, the latent heat load per minute is given as

$$\text{LH} = m_a\,\Delta h = m_a\,h_{fg}\,\Delta W = v\,\rho\,h_{fg}\,\Delta W \qquad \ldots (\because m_a = v\,\rho)$$

where v = Rate of dry air flowing in $m^3$/min,

$\rho$ = Density of moist air = 1.2 kg/$m^3$ of dry air,

$h_{fg}$ = Latent heat of vaporization = 2500 kJ /kg, and

AW = Difference of specific humidity between the entering and leaving conditions of

air = ($W_2$ - $W_1$) for humidification and ($W_1$ – $W_2$) for dehumidification.

Substituting these values in the above expression, we get

$$LH = v \text{ x } 1.2 \text{ x } 2500 \text{ x } \Delta W = 3000 \text{ v x } \Delta W \text{ kJ/min}$$

$$= \frac{3000\, v \times \Delta W}{60} = 50\ v \times \Delta W \text{ kJ/s or kW}$$

### 3.8 Methods of Obtaining Humidification and Dehumidification

The humidification is achieved either by supplying or spraying steam or hot water or cold water into the air. The humidification may be obtained by the following two methods:

1. **Direct method**. In this method, the water is sprayed in a highly atomized state into the room to be air-conditioned. This method of obtaining humidification is not very effective.

2. **Indirect method**. In this method, the water is introduced into the air in the air-conditioning plant, with the help of an air-washer, as shown in Fig. 20. This -conditioned air is then supplied to the room to be air-conditioned. The air-washer humidification may be accomplished in the following three ways:

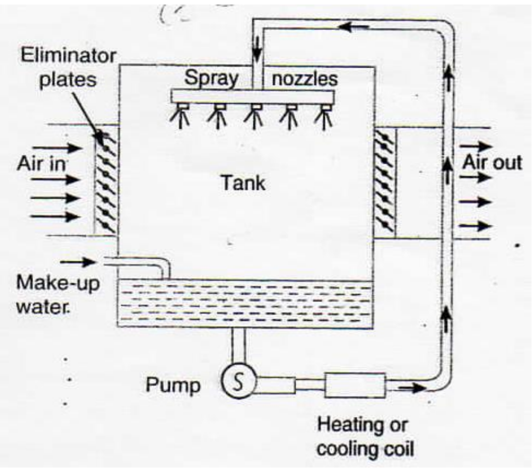


Fig. 20. Air-washer.

(a) by using re-circulated spray water without prior heating of air,

(b) by pre-heating the air and then washing if with re-circulated water, and

(c) by using heated spray water.

The dehumidification may be accomplished with the help of an air-washer or by using chemicals. In the air-washer system the outside or entering air is cooled below its dew point temperature so that it loses moisture by condensation. The moisture removal is also accomplished when the spray water is chilled water and its temperature is lower than the dew point temperature of the entering air. Since the air leaving the air-washer has its dry bulb temperature much below the desired temperature in the room, therefore a heating coil is placed after the air-washer. The dehumidification may also be achieved by using chemicals which have the capacity to absorb moisture in them. Two types of chemicals known as absorbents (such as calcium chloride) and adsorbents (such as silica gel and activated alumina) are commonly used for this purpose.

**Sensible Heat Factor**

As a matter of fact, the heat added during a psychrometric process may be split up into sensible heat and latent heat. The ratio of the *sensible heat to the total heat is known us *sensible heat factor* (briefly written as SHF) or *sensible heat ratio* (briefly written as SHR). Mathematically,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{SH + LH}$$

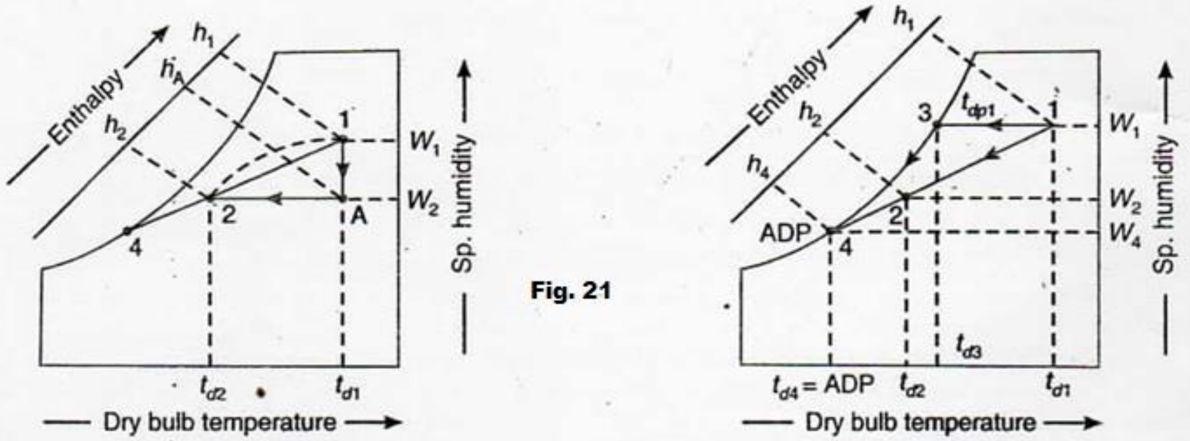
where SH = Sensible heat, and

LH = Latent heat.

The sensible heat factor scale is shown on the right hand side of the psychrometric chart.

### 3.9 Cooling and Dehumidification

This process is generally used in summer air conditioning to cool and dehumidify the air. The air is passed over a cooling coil or through a cold water spray. In this process, the dry bulb temperature as well as the specific humidity of air decreases. The final relative humidity of the air is generally higher than that of the entering air. The dehumidification of air is only possible wh6n the effective surface temperature of the cooling coil (i.e.$t_{d4}$) is less than the dew point temperature of the air entering the coil (i.e., $t_{dpt}$.). The effective surface temperature of the coil is known as *apparatus dew point* (briefly written as ADP). The cooling and dehumidification process is shown in Fig. 21.



(a) **Fig. 21** Cooling and dehumidification. (b)

$t_{d1}$ = Dry bulb temperature of air entering the coil,

$t_{dpl}$ = Dew point temperature of the entering air = $t_{d3}$ and

$t_{d4}$ = Effective surface temperature or ADP of the coil.

Under ideal conditions, the dry bulb temperature of the air leaving the cooling coil (i.e. $t_{d4}$) should be equal to the surface temperature of the cooling coil (i.e. ADP), but it is never possible due to inefficiency of the cooling coil. Therefore, the resulting condition of air coming out of the coil is shown by a point 2 on the straight line joining the points 1 and 4. The by-pass factor in this case is given by

Also

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d1} - t_{d4}} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

$$BPF = \frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$$

Actually, the cooling and dehumidification process follows the path as shown by a dotted curve in Fig. 21(a), but for the calculation of psychrometric properties, only end points are important. Thus the cooling and dehumidification process shown by a line 1-2 may be assumed to have followed a path 1-A (i.e. dehumidification) and A-2 (i.e. cooling) as shown in Fig. 21 (a). We see that the total heat removed from the air during the cooling and dehumidification process is

$$q = h_1 - h_2 = (h_1 - h_A) + (h_A - h_2) = LH + SH$$

where LH = $h_1 - h_A$ = Latent heat removed due to condensation of vapour of the reduced moisture content ($W_1 - W_2$), and

SH = $h_A - h_2$ = Sensible heat removed.

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{LH + SH} = \frac{h_A - h_2}{h_1 - h_2}$$

Note: The line 1-4 (i.e. the line joining the point of entering air and the apparatus dew point) in Fig. 21 (b) is known as sensible heat factor line.

**Example 1: In a cooling application, moist air enters a refrigeration coil at the rate of 100 kg of dry air per minute at 35° C and 50% RH. The apparatus dew point of coil is 5° C and by-pass factor is 0.15. Determine the outlet state of moist air and cooling capacity of coil in TR.**

Solution Given: $m_a = 100$ kg/min; $t_{dt} = 35°C$; $\emptyset = 50\%$; ADP = 5°C; BPF = 0.15

**Outlet state of moist air**

Let $t_{d2}$, and $\emptyset_2$ = Temperature and relative humidity of air leaving the cooling coil.

First of all, mark the initial condition of air, i.e. 35° C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 1, as shown in Fig. 22. From the psychrometric chart, we find that the dew point temperature of the entering air at point 1,

$$t_{dpt} = 23°C$$

Since the coil or apparatus dew point (ADP) is less than the dew point temperature of entering air, therefore it is a process of cooling and dehumidification.

We know that by-pass factor,

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d1} - t_{d4}} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

$$0.15 = \frac{t_{d2} - 5}{35 - 5} = \frac{t_{d2} - 5}{30}$$

$$t_{d2} = 0.15 \times 30 + 5 = 9.5°\text{C} \quad \text{Ans.}$$

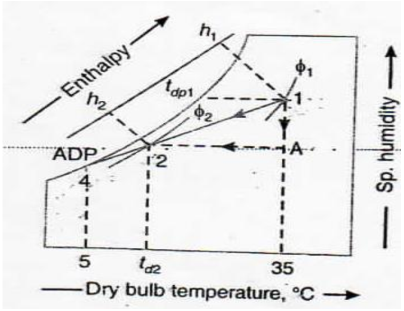


**Fig.22**

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature ($t_{d2}$,) of 9.5°Con the line 1-4 is $\emptyset_2 = 99\%$. Ans.

***Cooling capacity of the coil***

The resulting condition of the air coming out of the coil is shown by point 2, on the line joining the points 1 and 4, as shown in Fig. 22. The line 1-2 represents the cooling and dehumidification process which may be assumed to have followed the path 1-A (i.e. dehumidification) and A-2 (i.e. cooling). Now from the psychrometric chart, we find that enthalpy of entering air at point 1,

$h_1$= 81 kJ/kg of dry air

and enthalpy of air at point 2,

$h_2$= 28 kJ/kg of dry air

We know that cooling capacity of the coil

$= m_a(h_1 - h_2) = 100\ (81 - 28) = 5300$ kJ/min

= 5300/210 = 25.24 TR Ans. ….(∵ 1 TR = 210 kJ/min)

**Example 2. 39.6 m$^3$/min of a mixture of re-circulated room air and outdoor air enters cooling coil at 31°C dry bulb temperature and 18.5°C wet bulb temperature. The effective surface temperature of the coil is 4.4°C. The surface area of the coil is such as would give 12.5 kW of refrigeration with the given entering air state. Determine the dry and wet bulb temperatures of the air leaving the coil and the by-pass factor.**

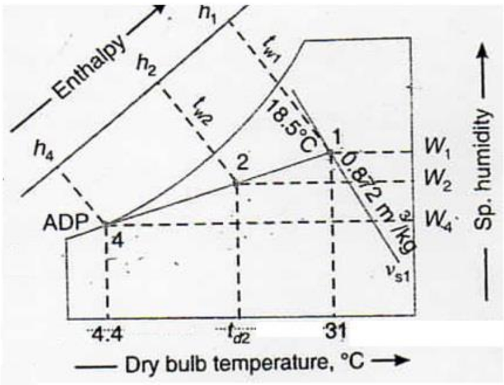
**Solution**: Given: $v_1$= 39.6 m$^3$/min; $t_{dt}$ = 31°C; $t_{wt}$ = I8.5°C; ADP= $t_{d4}$= 4.4°C; Q= 12.5 kW = 12.5 kJ/s = 12.5 x 60 kJ/min

**Dry and wet bulb temperature of the air leaving the coil**

Let $t_{d2}$ and $t_{w2}$ = Dry and wet bulb temperature of the air leaving the coil.

First of all, mark the initial condition of air, i.e. 31°C dry bulb temperature and 18.5°C wet bulb temperature on the psychrometric chart at point 1, as shown in Fig. 23. Now mark the effective surface temperature (ADP) of the coil at 4.4°C at point 4.

From the psychrometric chart, we find that enthalpy at point 1



$h_1$ = 52.5 kJ / kg of dry air

Enthalpy at point 4,

$h_4$ = 17.7kJ/kg of dry air

Specific humidity at point 1

$W_1$ = 0.0082 kg / kg of dry air

Specific humidity at point 4,

$W_4$= 0.00525 kg / kg of dry air

Specific volume at point

$v_{s1}$, = 0.872m$^3$/ kg

We know that mass flow rate of dry air at point 1,

$$m_a = \frac{v_1}{v_{s1}} = \frac{39.6}{0.872} = 44.41 \text{ kg/min}$$

and cooling capacity of the coil,

$$Q = m_a (h_1 - h_2)$$

or

$$h_1 - h_2 = \frac{Q}{m_a} = \frac{12.5 \times 60}{44.41} = 16.89 \text{ kJ / kg of dry air}$$

$$\therefore \quad h_2 = h_1 - 16.89 = 52.5 - 16.89 = 35.61 \text{ kJ / kg of dry air}$$

The equation for the condition line 1–2–4 is given as

$$\frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$$

$$\frac{W_2 - 0.005\,25}{0.0082 - 0.005\,25} = \frac{35.61 - 17.7}{52.5 - 17.7}$$

$$\therefore \quad W_2 = 0.006\,77 \text{ kg / kg of dry air}$$

Now plot point 2 on the psychrometric chart such as enthalpy, $h_2$ = 35.61 kJ/kg of dry air and specific humidity, $W_2$ = 0.00677 kg/kg of dry air. At point 2, we find that

$t_{d2}$ = 18.5°C; and $t_{w2}$ = 12.5°C Ans.

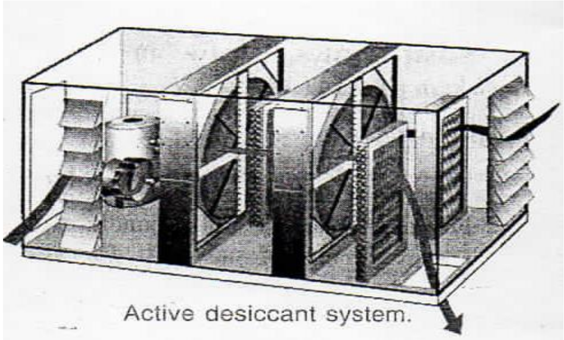
By-pass factor

We know that by-pass factor,

$$BPF = \frac{h_2 - h_4}{h_1 - h_4} = \frac{35.61 - 17.7}{52.5 - 17.7} = 0.5146 \text{ Ans.}$$

### 3.10 Heating and Humidification

This process is generally used in winter air conditioning to warm and humidify the air. It is the reverse process of cooling and -- dehumidification. When air is passed through a humidifier having spray water temperature higher than the dry bulb temperature of the entering air, the unsaturated air will reach the condition of saturation and thus the air becomes hot. The heat of vaporization of water is absorbed from the spray water itself and hence it gets cooled. In this way, the air becomes heated and humidified. The process of heating and humidification is shown by line 1-2 on the psychrometric chart as shown in Fig. 24.



The air enters at condition 1 and leaves at condition 2. In this process, the dry bulb temperature as well as specific humidity of air increases. The final relative humidity of the air can be lower or higher than that of the entering air.

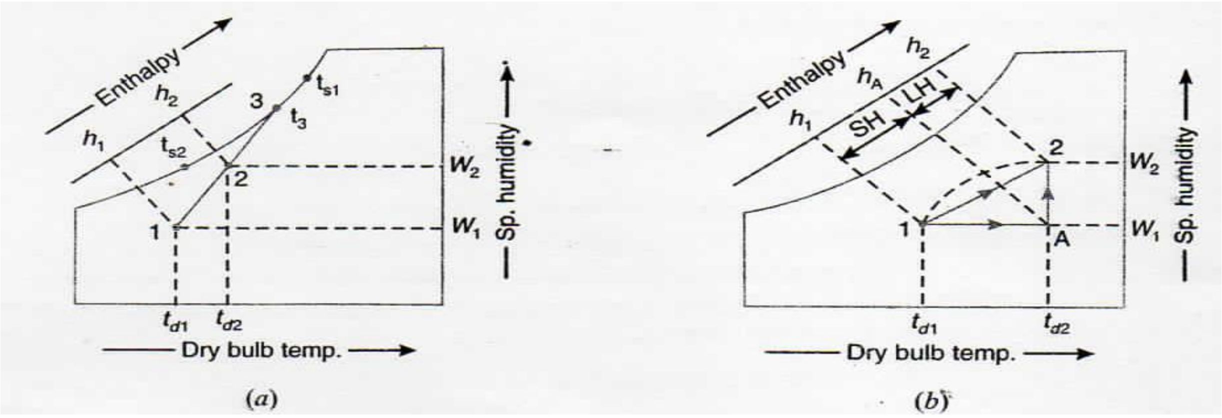


Fig.24 heating and humidification

Let $m_{w1}$ and $m_{w2}$ = Mass of spray water entering and leaving the humidifier in kg,

$h_{fw1}$ and $h_{fw2}$ = Enthalpy of spray water entering and leaving the humidifier in kJ/kg,

$W_1$ and $W_2$ = Specific humidity of the entering and leaving air in kg/kg of dry

$h_1$ and $h_2$ = Enthalpy of entering and leaving air in kJ/kg of dry air, and

$m_a$ = Mass of dry air entering in kg.

For mass balance of spray water,

$$(m_{w1} - m_{w2}) = m_a (W_2 - W_1)$$

$$m_{w2} = m_{w1} - m_a (W_2 - W_1) \quad \ldots. (i)$$

or and for enthalpy balance,

$$m_{w1}\, h_{fw1} = m_{w2}\, h_{fw2} = m_a(h_2 - h_1) \quad \ldots.(ii)$$

Substituting the value of mw2 from equation (i), we have

$$m_{w1}\, h_{fw1} - [\, m_{w1} - m_a (W_2 - W_1)]\, h_{fw2}$$

$$= m_a (h_2 - h_1)$$

$$\therefore \qquad h_2 - h_1 = \frac{m_{w1}}{m_a} (h_{fw1} - h_{fw2}) + (W_2 - W_1)\, h_{fw2}$$

The temperatures $t_{s1}$ and $t_{s2}$ shown in Fig. 24 (a) denote the temperatures of entering and leaving spray water respectively. The temperature 13 is the mean temperature of the spray water which the entering air may be assumed to approach.

Actually, the heating and humidification process follows the path as shown by dotted curve in Fig. 24(b), but for the calculation of psychrometric properties, only the end points are important. Thus, the heating and humidification process shown by a line 1-2 on the psychrometric chart may be assumed to have followed the path 1-A (i.e. heating) and A-2 (i.e. humidification), as shown in Fig. 24(b). We see that the total heat added to the air during heating and humidification is

$$q = h_2 – h_1 = (h_2 – h_1) + (h_A - h_i) = q_t + q_s$$

where $q_t = (h_2 - h_A)$ = Latent heat of vaporization of the increased moisture

content $(W_2 – W_1)$, and

$q_s = (h_A - h_i)$ = Sensible heat added

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{q_S}{q} = \frac{q_S}{q_S + q_L} = \frac{h_A - h_1}{h_2 - h_1}$$

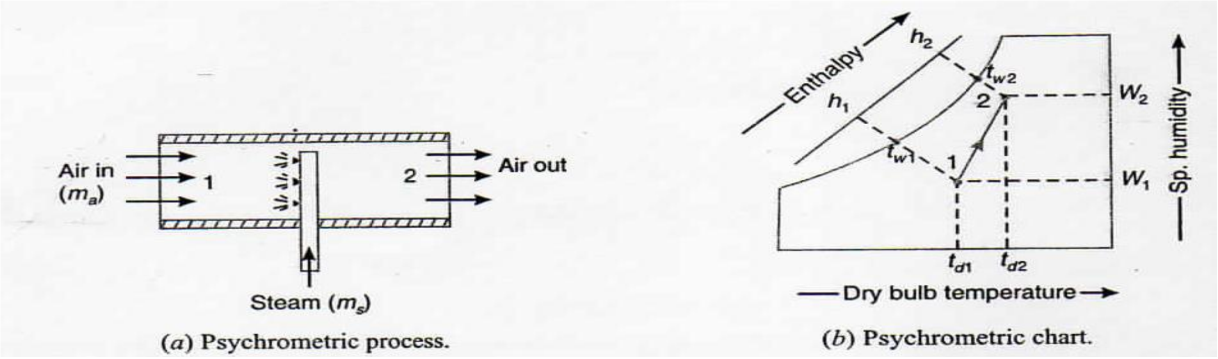
Note: The line 1-2 in Fig. 24 (b) is called sensible heat factor line.

### 3.11 Heating and Humidification by Steam Injection

The steam is normally injected into the air in order to increase its specific humidity as shown in Fig. 25 (a). This process is used for the air conditioning of textile mills where high humidity is to be maintained. The dry bulb temperature of air changes very little during this process, as shown on the psychrometric chart in Fig. 25 (b).

Let $m_s$ = Mass of steam supplied,

$m_a$ = Mass of dry air entering,



(a) Psychrometric process. (b) Psychrometric chart.

Fig.25 heating and humidification by steam injection

$W_1$= Specific humidity of air entering,

$W_2$ = Specific humidity of air leaving,

$h_1$ = Enthalpy of air entering,

$h_2$ = Enthalpy of air leaving, and

$h_s$ = Enthalpy of steam injected into the air.

Now for the mass balance,

$$W_2 = W_1 + \frac{m_s}{m_a} \qquad \text{….}(i)$$

and for the heat balance,

$$h_2 = h_1 + \frac{m_s}{m_a} \times h_s = h_1 + (W_2 - W_1)\, h_s \qquad \text{... [From equation } (i)\text{]}$$

**Example 3: Atmospheric air at a dry bulb temperature of 16° C and 25% relative humidity passes through a furnace and then through a humidifier, in such a way that the final dry bulb temperature is 30° C and 50% relative humidity. Find the heat and moisture added to the air. Also determine the sensible heat factor of the-process.**

**Solution**: Given: $t_{dt}$ = 16° C; $\varnothing_1$ = 25% ; $t_{d2}$ = 30° C; 42 = 50%

Heat added to the air

First of all, mark the initial condition of air i.e. at 16°C dry bulb temperature and 25% relative humidity on the psychrometric chart at point 1, as shown in Fig. 16.47. Then mark the final condition of air at 30° C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 2. Now locate the point A by drawing horizontal line through point 1 and vertical line through point 2. From the psychrometric chart, we find that enthalpy of air at point 1,

$h_1$ = 23 k1/kg of dry air

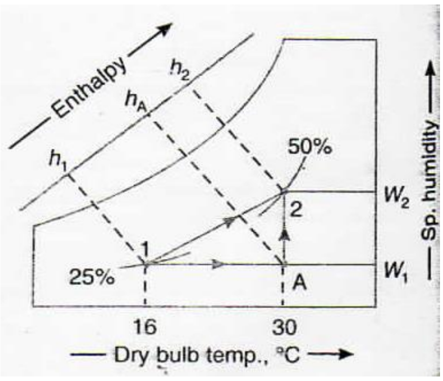


Fig.26

Enthalpy of air at point A,

$h_A$ = 38 kJ/kg of dry air

and enthalpy of air at point 2,

$h_2$ = 64 la/kg of dry air

∴ Heat added to the air

= $h_2$- $h_1$= 64 - 23 = 41 kJ/kg of dry air Ans.

***Moisture added to the air***

From the psychrometric chart, we find that the specific humidity in the air at point 1,

$W_1$ = 0.0026 kg/kg of dry air

and specific humidity in the air at point 2,

$W_2$ = 0.0132 kg /kg of dry air

∴ Moisture added to the air

= $W_2 – W_1$ = 0.0132 - 0.0026 = 0.0106 kg/kg of dry air Ans.

***Sensible heat factor of the process***

We know that sensible heat factor of the process,

$$SHF = \frac{h_A - h_1}{h_2 - h_1} = \frac{38 - 23}{64 - 23} = 0.366 \text{ Ans.}$$

**Example 4: Air at 10°C dry bulb temperature and 90% relative humidity is to be ated and humidified to 35°C dry bath temperature and 22.5°C wet bulb temperature. The air is pre-heated sensibly before passing to the air washer in which water is re-circulated. The relative humidity of the air coming out of the air washer is 90%. This air is again reheated sensibly to obtain the final desired condition. Find: 1. the temperature to which the air should be preheated. 2. the total heating required; 3. the makeup water required in the air washer ; and 4. the humidifying efficiency of the air washer.**

**Solution**: Given : $t_{dt}$ = 10°C; $\varnothing_1$ = 90%; $t_{d2}$,=35°C; $t_{w2}$=22.5°C

First of all, mark the initial condition of air i.e. at 10°C dry bulb temperature and 90% relative humidity, on the psychrometric chart at point 1, as shown in Fig. 16.48. Now mark the final condition of air i.e. at 35° C dry bulb temperature and 22.5° C wet bulb temperature at point 2.

From point I , draw a horizontal line to represent sensible heating and from point 2 draw horizontal line to intersect 90% relative humidity curve at point B. Now from point B, draw a constant wet bulb temperature line which intersects the horizontal line drawn through point 1 at point A. The line 1-A represents preheating of air, line AB represents humidification and line 8-2 represents reheating to final condition.

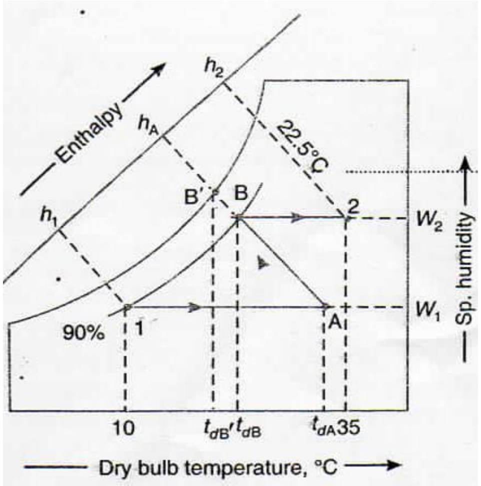


**Fig.27**

### *1.Temperature to which the air should be preheated*

From the psychrometric chart, the temperature to which the air should be preheated (corresponding to point A) is 4 $t_{dA}$ = 32.6°C Ans.

### *2. Total heating required*

From the psychrometric chart, we find that enthalpy of,air at point 1.

$$h_1 = 27.2 \text{ kJ /kg of dry air}$$

Enthalpy of air at point A,

$$h_A = 51 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 68 \text{ kJ/kg of dry air}$$

We know that heat required for preheating of air

$$= h_A - h_1 = 51 - 27.2 = 23.8 \text{ kJ/kg of dry air}$$

and heat required for reheating of air

$$= h_2 - h_B = 68 - 51 = 17 \text{ kJ/kg of dry air}$$

∴ Total heat required $= 23.8 + 17 = 40.8$ kJ/kg of dry air Ans.

### *3. Make up water required in the air washer*

From the psychrometric chart, we find that specific humidity of entering air,

$$W_1 = 0.0068 \text{ kg /kg of dry air}$$

and specific humidity of leaving air,

$$W_2 = 0.0122 \text{ kg /kg of dry air}$$

∴ Make up water required in the air washer

$$= W_B - W_A = W_2 - W_1$$

$$= 0.0122 - 0.0068 = 0.0054 \text{ kg/kg of dry air} \quad \text{Ans.}$$

***4. Humidifying efficiency of the air washer***

From the psychrometric chart, we find that

$$t_{dB} = 19.1°\text{C and } t_{dB} = 18°\text{C}$$

We know that humidifying efficiency of the air washer,

$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{t_{dA} - t_{dB}}{t_{dA} - t_{dB'}}$$

$$= \frac{32.6 - 19.1}{32.6 - 18} = \frac{13.5}{14.6} = 0.924 \text{ or } 92.4\% \quad \text{Ans.}$$

### 3.12 Heating and Dehumidification -Adiabatic Chemical Dehumidification

This process is mainly used in industrial air conditioning and can also be used for some comfort air conditioning installations requiring either a low relative humidity or low dew point temperature in the room.

In this process, the air is passed over chemicals which have an affinity for moisture. As the air comes in contact with these chemicals, the moisture gets condensed out of the air and gives up its latent heat. Due to the condensation, the specific humidity decreases and the heat of condensation supplies sensible heat for heating the air and thus increasing its dry bulb temperature.

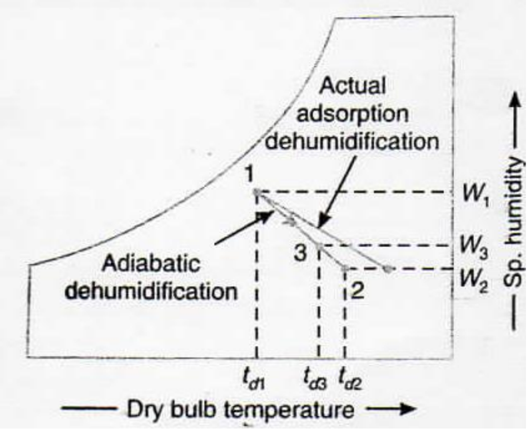


Fig.28

The process, which is the reverse of adiabatic saturation process, is shown by the line 1-2 on the psychrometric chart as shown in Fig. 28. The path followed during the process is along the constant wet bulb temperature line or-constant enthalpy line.

The effectiveness or efficiency of the dehumidifier is given as

$$\eta_H = \frac{\text{Actual increase in dry bulb temperature}}{\text{Ideal increase in dry bulb temperature}} = \frac{t_{d3} - t_{d1}}{t_{d2} - t_{d1}}$$

Notes: 1. In actual practice, the process is accompanied with a release of heat called heat of adsorption, which is very large. Thus the sensible heat gain of air exceeds the loss of latent heat and the process is shown above the constant wet bulb temperature line in Fig. 28.

2. Two types of chemicals used for dehumidification are absorbents and adsorbents. The absorbents are substances which can take up moisture from air and during this process change it chemically, physically or in both respects. These include water solutions or brines of calcium chloride, lithium chloride, lithium bromide and ethylene glycol. These are used as air dehydrators by spraying or otherwise exposing a large surface of the solution in the air stream.

The adsorbents are substances in the solid state which can take up moisture from the air and during this process do not change it chemically or physically. These include silca gel (which is a form of silicon dioxide prepared by mixing fused sodium silicate and sulphuric acid) and activated alumina (which is a porous amorphous form of aluminum oxide).

**Example 5: Saturated air at 21° C is passed through a drier so that its final relative humidity is 20%. The drier uses silica gel adsorbent. The air is then passed through a cooler until its final temperature is 21° C without a change in specific humidity. Determine : 1. the temperature of air at the end of the drying process; 2. the heat rejected during the cooling process ; 3. the relative humidity at the end of cooling process; 4. the dew point temperature at the end of the drying process ; and 5. the moisture removed during the drying process.**

**Solution:** Given: $t_{d1}$, = $t_{d3}$ = 21°C; $\phi_2$= 20%

***1.Temperature of air at the end of drying process***

First of all, mark the initial condition of air i.e. at 21°C dry bulb temperature upto the saturation curve (because the air is saturated) on the psychrometric chart at point 1, as shown in Fig. 29. Since the drying process is a chemical dehumidification process, therefore-. it follows a path along-the-constant wet bulb temperature or the constant enthalpy line as shown by the line 1- 2 in Fig. 29. Now mark the point 2 at relative humidity of 20%. From the psychrometric chart, the temperature at the end of drying process at point 2, $t_{d2}$ = 38.5°C Ans.

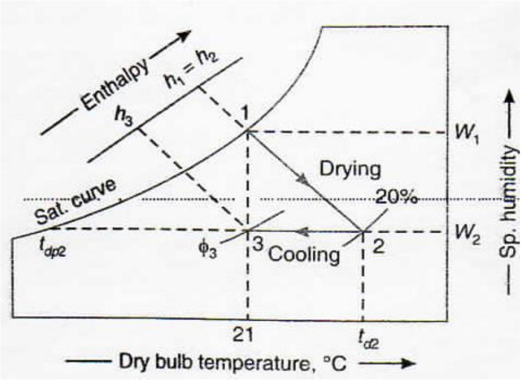


Fig.29

***2. Heat rejected during the cooling process***

The cooling process is shown by the line 2-3 on the psychrometric chart as shown in Fig.29. From the psychrometric chart. we find that enthalpy of air at point 2.

$h_2$ = 61 kJ /kg of dry air

and enthalpy of air at point 3,

$h_3$ = 43 kJ/kg of dry air

∴ Heat rejected during the cooling process

= $h_2$ - $h_3$ = 61— 43 = 18 kJ/kg of dry air A ns.

***3. Relative humidity at the end of cooling process***

From the psychrometric chart, we find that relative humidity at the end of cooling process (i.e. at point 3),

$\emptyset_3$ = 55% Ans.

***4. Dew point temperature at the end of drying process***

From the psychrometric chart, we find that the dew point temperature at the end of the drying process,

$t_{dp2}$ = 11.6° C Ans.

***5. Moisture removed during the drying process***

From the psychrometric chart, we find that moisture in air before the drying process at point 1,

$W_1$ = 0.0157 kg/kg of dry air

and moisture in air after the drying process at point 2,

$W_2$ = 0.0084 kg/kg of dry air

∴ Moisture removed during the drying process

= W, — W, = 0.0157 0.0084 = 0.0073 kg/kg of dry air Ans.

## AIR CONDITIONING SYSTEMS

### 3.13 Introduction

The air conditioning is that branch of engineering science which deals with the study of conditioning of air i.e., supplying and maintaining desirable internal atmosphere conditions for human comfort, irrespective of external conditions. This subject, in its broad sense, also deals with the conditioning of air for industrial purposes, food processing storage of food and other materials.

### 3.14 Factors affecting comfort Air Conditioning

The four important factors for comfort air conditioning are discussed as below:

***1. Temperature of air***: In air conditioning, the control of temperature means the maintenance of any desired temperature within an enclosed space even though the temperature of the outside air is above or below the desired room temperature. This is accomplished either by the addition or removal of heat from the enclosed space as and when demanded. It may be noted that a human being feels comfortable when the air is at 21°C with 56% relative humidity.

***2. Humidity of air:*** The control of humidity of air means the decreasing or increasing of moisture contents of air during summer or winter respectively in order to produce comfortable and healthy conditions. The control of humidity is not only necessary for human comfort but it also increases the efficiency of the workers. In general, for summer air conditioning, the relative humidity should not be less than 60% whereas for winter air conditioning it should not be more than 40%.

***3. Purity of air:*** It is an important factor for the comfort of a human body. It has been noticed that people do not feel comfortable when breathing contaminated air, even if it is within acceptable temperature and humidity ranges. It is thus obvious that proper filtration, cleaning and purification of air is essential to keep it free from dust and other impurities.

***4. Motion of air:*** The motion or circulation of air is another important factor which should be controlled, in order to keep constant temperature throughout the conditioned space. It is, therefore, necessary that there should be equi-distribution of air throughout the space to be air conditioned.

### 3.15 Air Conditioning System

We have already discussed in Art. the four important factors which affect the human comfort. The system which effectively controls these conditions to produce the desired effects upon the occupants of the space is known as an *air conditioning system*.

## 3.16 Equipments Used in an Air Conditioning System

Following are the main equipments or parts used in an air conditioning system:

1. Circulation fan. The main function of this fan is to move air to and from the room.

2. Air conditioning unit. It is a unit which consists of cooling and dehumidifying processes for summer air conditioning or heating and humidification processes for winter air Conditioning.

3. Supply duct. It directs the conditioned air from the circulating fan to the space to be air conditioned at proper point

4. Supply outlets. These are grills which distribute the conditioned air evenly in the room.

5. Return outlets. These are the openings in a room surface which allow the room ait to enter the return duct.

6. Filters. The main function of the filters is to remove dust, dirt and other harmful bacteria from the air.

## 3.17 Classification of Air Conditioning Systems

The air conditioning systems may be broadly classified as follows:

1. *According to the purpose*

    (a) Comfort air conditioning system, and

    (b) Industrial air conditioning system.

2. *According to season of the year*

    (a) Winter air conditioning system,

    (b) Summer air conditioning system, and

    (c) Year-round air conditioning system.

3. *According to the arrangement of equipment*

    (a) Unitary air conditioning system, and

    (b) Central air conditioning system.

In this chapter, we shall discuss all the above mentioned air conditioning systems one by one.

## 3.18 Comfort Air Conditioning System

In comfort air conditioning, the air is brought to the required dry bulb temperature and relative humidity for the human health, comfort and efficiency. If sufficient data of the required condition is not given, then it is assumed to be 21°C dry bulb temperature and 50% relative humidity. The sensible heat factor is, generally, kept as following:

For residence or private office = 0.9

For restaurant or busy office = 0.8

Auditorium or cinema hall = 0.7

Ball room dance hall etc. = 0.6

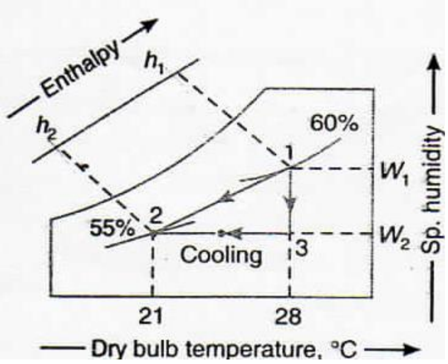
The comfort air conditioning may be adopted for homes, offices, shops, restaurants, theatres, hospitals, schools etc.

**Example 1:** ***An air conditioning plant is required to supply 60 m of air per minute at a DBT of 21°C and 55% RH. The outside air is at DBT of 28°C and 60% RH. Determine the mass of water drained and capacity of the cooling coil. Assume the air conditioning plant first to dehumidify and then to cool the air.***

**Solution:** Given $v_2 = 60\ m^3/min$; $t_{d2} = 21°C$; $\emptyset_2 = 55\%$; $t_{dt} = 28°C$; $\emptyset_1 = 60\%$

*Mass of water drained*

First of all, mark the initial condition of air at 28°C dry bulb temperature and 60% relative humidity on the 1 h psychrometric chart as point 1, as shown in Fig. 1. Now mark the final condition of air at 21°C dry bulb temperature % and 55% relative humidity as point 2. From the psychrometric chart, we find that



Specific humidity of air at point 1,

$W_1 = 0.0141$ kg/kg of dry air

Specific humidity of air at point 2,

$W_2 = 0.0084$ kg / kg of dry air

and specific volume of air at point 2,

$v_{s2} = 0.845$ m3/ kg of dry air

We know that mass of air circulated,

$$m_a = \frac{v_2}{v_{s2}} = \frac{60}{0.845} = 71\ \text{kg / min}$$

∴ Mass of water drained

$= m_a (W_1 - W_2) = 71(0.0142 — 0.0084) = 0.412$ kg / min

$= 0.412 \times 60 = 24.72$ kg / h Ans.

Capacity of the cooling coil

From the psychrometric chart, we find that

Enthalpy of air at point 1,

$h_1 = 64.8$ kJ / kg of dry air

and enthalpy of air at point 2,

$h_2 = 42.4$ kJ / kg of dry air

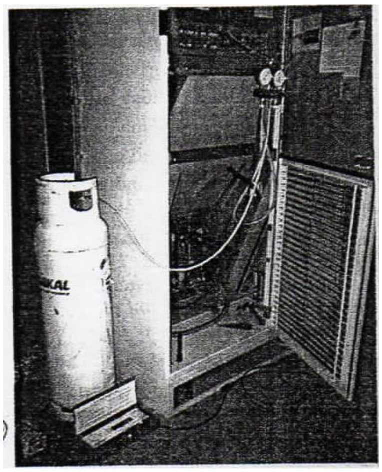
∴ Capacity of the cooling coil

$= m_a (h_1 - h_2) = 71(64.8 - 42.4) = 1590.4$ kJ / min

$= 1590.4 / 210 = 7.57$ TR Ans.

### 3.19 Industrial Air Conditioning System

It is an important system of air conditioning these days in which the inside dry bulb temperature and relative humidity of the air is kept constant for proper working of the machines and for the proper research and manufacturing processes. Some of the sophisticated electronic and other machines need a particular dry bulb temperature and relative humidity. Sometimes, these machines also require a particular method of psychrometric processes. This type of air conditioning system is used in textile mills, paper mills, machine-parts manufacturing plants, tool rooms, photo-processing plants etc.



Industrial air-conditioning system

**Example 2:** ***Following data refers to an air conditioning system to be designed for an industrial***

***process for hot and wet climate:***

***Outside conditions = 30° C DBT and 75% RH***

***Required inside conditions = 20° C DBT and 60% RH***

***The required condition is to be achieved first by cooling and dehumidifying and then by heating. If 20 nil of air is absorbed-by the plant every minute, find : 1. capacity of the cooling coil in tones of refrigeration; 2. capacity of the heating coil in kW; 3. amount of water removed per hour; and 4. By-pass factor of the heating coil, if its surface temperature is 35°C.***

Solution: Given $t_{dt}$ = 30°C; $\emptyset_1$= 75%; $t_{d3}$= 20°C; $\emptyset_3$ = 60%; $v_l$ = 20 $m^3$/ min; $t_{d4}$= 35°C

***1.Capacity of the cooling coil in tones of refrigeration***

First of all, mark the initial condition of air at 30°C dry bulb temperature and 75% relative humidity on the psychrometric chart as point 1, as shown in Fig. 2. Then mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity oil the chart as point 3.

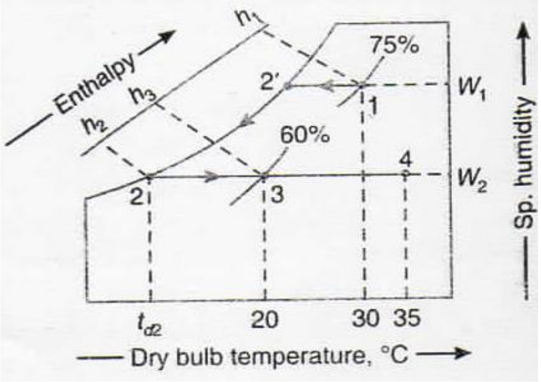


**Fig.2**

Now locate the points 2' and 2 on the saturation curve by drawing horizontal lines through points 1 and 3 as shown in Fig. 2. On the chart, the process 1-2' represents the sensible cooling. 2'-2 represents dehumidifying process and 2-3 represents the sensible heating process. From the psychrometric chart, we find that the specific volume of air at point 1.

$v_{x1}$ = 0.886 m3/ kg of dry air

Enthalpy of air at point 1,

$h_1$ = 81.8 kJ/kg of dry air

and enthalpy of air at point 2,

$h_2$ = 34.2 kJ / kg of dry air

We know that mass of air absorbed by the plant,

$$m_a = \frac{v_1}{v_{s1}} = \frac{20}{0.866} = 22.6 \text{ kg / min}$$

∴ Capacity of the cooling coil

= $m_a$ ($h_1 – h_2$) = 22.6 (81.8 - 34.2) = 1075.76 kJ / min

= 1075.76 / 210 = 5.1 TR Ans.

***2. Capacity of the heating coil in kW***

From the psychrometric chart, we find that enthalpy of air at point 3,

$h_3$ = 42.6 kJ / kg of dry air

∴ Capacity of the heating coil

= $m_a$ ($h_1 – h_2$) = 22.6 (42.6 - 34.2) = 189.84 Id / min

= 189.84/ 60 = 3.16 kW Ans.

***3. Amount of water removed per hour***

From the psychrometric chart, we find that specific humidity of air at point 1,

$W_1$ = 0.0202 kg / kg of dry air

and specific humidity of air at point 2,

$W_2$= 0.0088 kg / kg of dry air

∴ Amount of water removed per hour

= $m_a$ ($W_1$ - $W_2$) = 22.6 (0.0202 — 0.0088) = 0.258 kg / min

= 0.258 x 60 = 15.48 kg / h Ans.

***4. By-pass factor of the heating coil***

We know that by-pass factor,

$$BPF = \frac{t_{d4} - t_{d3}}{t_{d4} - t_{d2}} = \frac{35 - 20}{35 - 12.2} = 0.65\dot{8} \text{ Ans.}$$

... [From psychrometric chart, $t_{d2}$, = 12.2°C]

### 3.20 Winter Air Conditioning System

In winter air conditioning, the air is heated, which is generally -accompanied by humidification. The schematic arrangement of the system is Damper shown in Fig. 3.

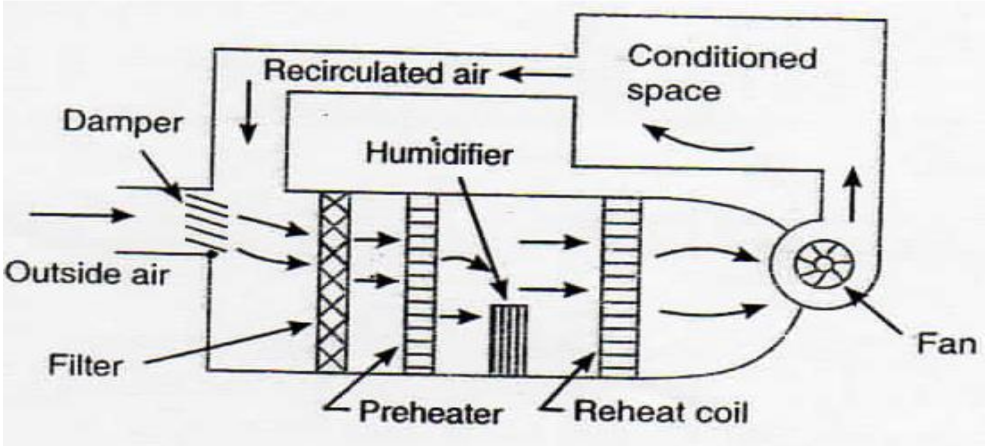


**Fig.3** Winter air conditioning system

The outside air flows through a damper and mixes up with the Outside air re-circulated air (which is obtained Fan from the conditioned space). The Filter mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a preheat coil in order to prevent the possible freezing of water and to control the evaporation of water in the humidifier. After that, 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 a 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 used air (known as re-circulated air) is again conditioned as shown in Fig.3.

The outside air is sucked and made to mix with re-circulated air, in order to make for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

### 3.21 Summer Air Conditioning System

It is the most important type of air conditioning, in which the air is cooled and generally dehumidified. The schematic arrangement of a typical summer air conditioning system is shown in Fig. 4.

The outside air flows through the damper, and mixes up with re-circulated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry bulb temperature of the air in the conditioned space. The cooled air passes through a perforated membrane and loses its moisture in the condensed form which is collected in a sump. After that, the air is made to pass through a heating coil which heats up the air slightly. This is done to bring the air to the designed dry bulb temperature and Damper relative humidity.

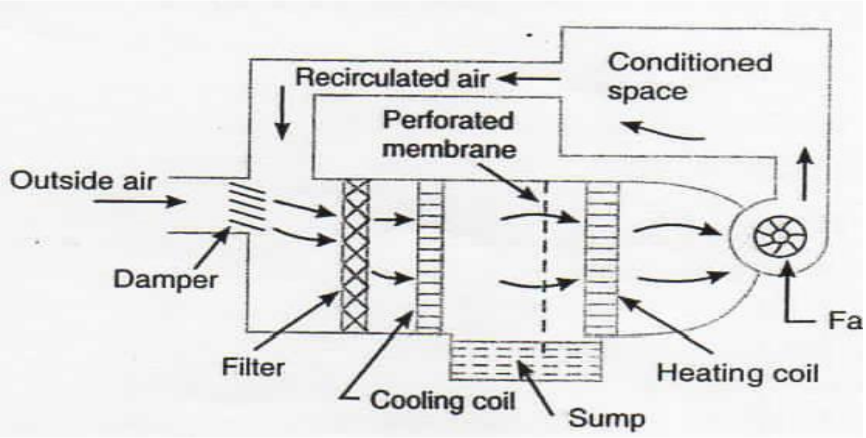


Fig 4 summer air conditioning system

Now the conditioned air is supplied to the conditioned space by a 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 used air (known as re-circulated air) is again conditioned -as shown in Fig. 4: The outside air is sucked and made-I6 mix with the re-circulated air in order to make up for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

**Example 3: The amount of air supplied to an air conditioned hall is 300m$^3$/min. The atmospheric conditions are 35°C DBT and 55% RH. The required conditions are 20°C DBT and 60% RH. Find out the sensible heat and latent heat removed from the air per minute. Also find sensible heat factor for the system.**

**Solution:** Given $v_1 = 300$ m$^3$/min; $t_{dt}$, = 35°C ; $\emptyset_1 = 55\%$; $t_{d2} = 20$°C; $\emptyset_2 = 60\%$

First of all, mark the initial condition of air at 35°C dry bulb temperature and 55% relative humidity on the psychrometric chart at point 1, as shown in Fig. 5. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Locate point 3 on the chart by drawing horizontal line through point 2 and vertical line through point 1. From the psychrometric chart, we find that specific volume of air at point 1,

$$v_{s1} = 0.9 \text{ m}^3/\text{kg of dry air}$$

∴ Mass of air supplied,

***Sensible heat removed from the air***

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 85.8 \text{ kJ/kg of dry air}$$

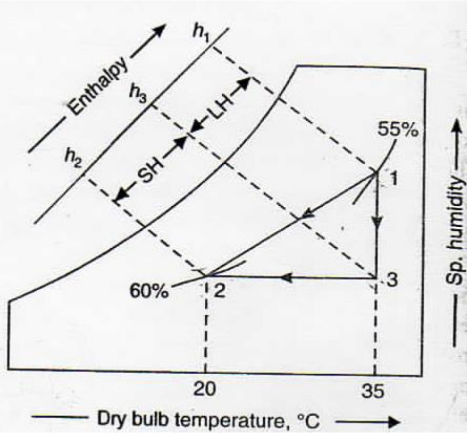


**Fig.5**

Enthalpy of air at point 2,

$$h_2 = 42.2 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 3,

$$h_3 = 57.4 \text{ kJ/kg of dry air}$$

We know that sensible heat removed from the air,

$$\text{SH} = m_a (h_3 - h_2)$$

$$= 333.3 (57.4 - 42.2) = 5066.2 \text{ kJ/min Ans.}$$

***Latent heat removed from the air***

We know that latent heat removed from the air,

$$\text{LH} = m_a (h_1 - h_3)$$

$$= 333.3 (85.8 - 57.4) = 9465.7 \text{ kJ/min Ans.}$$

***Sensible heat factor for the system***

We know that sensible heal factor for the system,

$$SHF = \frac{SH}{SH + LH} = \frac{5066.2}{5066.2 + 9465.7} = 0.348 \text{ Ans.}$$

**Example 4: An air handling unit in an air conditioning plant supplies a total of 4500 m$^3$/min of dry air which comprises by mass 20% of fresh air at 40°C DBT and 27°C WBT and 8,%, re-circulated air at 25°C DBT and 50% RH. The air leaves the cooling coil at 13°C saturated. Calculate the total cooling load and room heat gain. The following data can be used:**

| **Condition** | **DBT**<br>℃ | **WBT**<br>℃ | **RH**<br>% | **Sp. Humidity**<br>**% of water vapour / Kg of dry air** | **Enthalpy**<br>**kJ/kg of dry air** |
|---|---|---|---|---|---|
| **Outside** | **40** | **27** | **--** | **17.2** | **85** |
| **Inside** | **25** | **--** | **50** | **10.0** | **51** |
| **ADP** | **13** | **--** | **100** | **9.4** | **36.8** |

*Specific volume of air entering the cooling coil is 0.869 $m^3$/ kg of dry air.*

**Solution:** Given $v_3$= 4500 $m^3$/min; $t_{dt}$, = 40°C; = 27°C; $t_{d2}$, = 25°C; $\emptyset_2$ = 50%; $t_{d4}$= ADP = 13°C; $W^1$ = 17.2 g / kg of dry air = 0.0172 kg/kg of dry air; $W_2$ = 10g / kg of dry air =0.01 kg/kg of dry air; $W_4$=9.4g/kg of dry air = 0.0094 kg/kg of dry air; $h_1$=85 kJ/kg of dry air; $h_2$ = 51 kJ/kg of dry air; $h_4$ = 36.8 kJ/kg of dry air ; $v_{s3}$ = 0.869 ni3/kg of dry air.

First of all, mark the condition of fresh air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 6. Now mark the condition of re-circulated air at 25°C dry bulb temperature and 50% relative humidity as point 2. The condition of air entering the cooling coil *(point 3) is marked on the line 1-2, such that the specific volume of air at this point is 0.869 $m^3$/kg of dry air. The point 4 represents the condition of air leaving the cooling coil at 13°C on the saturation curve.

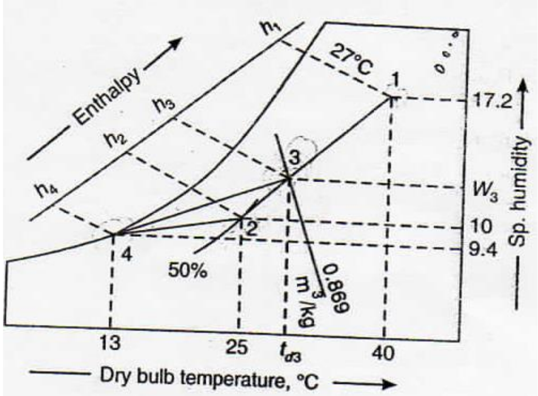


**Fig.6**

From the psychrometric chart, we find that enthalpy of air entering the cooling coil at point 3,

$$h_3 = 57.8 \text{ kJ / kg of dry air}$$

Specific humidity of air entering the cooling coil at point 3,

$$W_3 = 0.0116 \text{ kg / kg of dry air}$$

and dry bulb temperature of air entering the cooling coil at point 3,

$$t_{d3} = 28.3°C$$

***Total cooling load***

We know that mass of air entering the cooling coil,

$$m_{a3} = \frac{v_3}{v_{s3}} = \frac{4500}{0.869} = 5178 \text{ kg / min}$$

Total cooling load = $m_{a3}$ ($h_3$ - $h_4$) = 5178 (57.8 — 36:8) = 108 738 kJ / min

= 108 738 / 210 = 517.8 TR Ans.

***Room heat gain***

Since the total mass of air ($m_a$, = 5178 kg / min) comprises 20% of fresh air, therefore mass of fresh air supplied at point 1.

$$m_{a1} = 0.2 \text{ x } 5178 = 1035.6 \text{ kg / min}$$

and fresh air load = $m_{a1}$ ($h1 – h_2$) = 1035.6 (85 — 51) = 35 210 la / min

= 35 210 / 210 = 168 TR Ans.

∴ Room heat gain = Total cooling load Fresh air load

= 517.8 — 168 = 349.8 TR Ans.

**Example 5: A conference room of 60 seating capacity is to be air conditioned for comfort conditions of 22°C dry bulb temperature and 55% relative humidity. The outdoor conditions are 32°C dry bulb temperature and 22°C wet bulb temperature. The quantity of air supplied is 0.5m$^3$/min/person. The comfort conditions are achieved first by chemical dehumidification and by cooling coil. Determine I. Dry bulb temperature of air at exit of dehumidifier; 2. Capacity of dehumidifier; 3. Capacity and surface temperature of cooling coil, if the by-pass factor is 0.30.**

**Solution:** Given: Seating capacity = 60; $t_{d2}$= 22°C; $\emptyset_2$ = 55%; $t_{dt}$ = 32°C; $t_{wt}$ = 22°C; = 0.5 m$^3$/min / person = 0.5 x 60 = 30 m$^3$/min; BPF = 0.3.

First of all, mark the outdoor conditions of air re. at 32°C dry bulb temperature and 22°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.8. Now mark the required comfort conditions of air i.e. at 22°C dry bulb temperature and 55% relative humidity, as point 2. In order to find the condition of air leaving the dehumidifier, draw a constant wet bulb temperature line from point 1 and a constant specific humidity line-from point 2. Let these two lines intersect at point 3. The line 1-3 represents the chemical dehumidification and the line 3-2 represents sensible cooling.

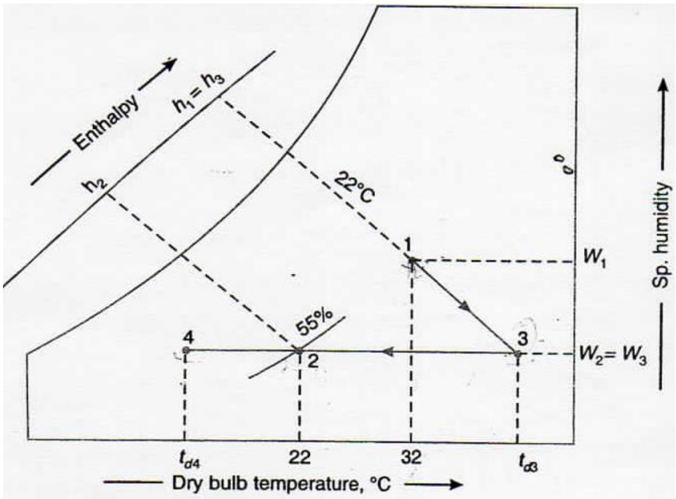


**Fig.7**

***1. Dry bulb temperature of air at exit of dehumidifier***

From the psychrometric chart, we find that dry bulb temperature of air at exit of dehumidifier i.e. at point3,

$$t_{d3} = 41°C \text{ Ans.}$$

***2. Capacity of dehumidifier***

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = = 64.5 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 45 \text{ kJ / kg of dry air}$$

Specific humidity of air at point 1,

$$W_1 = 0.0123 \text{ kg / kg of dry air}$$

Specific humidity of air at point 3,

$$W_3 = W_2 = 0.0084 \text{ kg/kg of dry air}$$

and specific volume of air at point 1,

$$v_{s1} = 0.881 \text{ .m3/ kg of dry air}$$

We know that mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{30}{0.881} = 34.05 \text{ kg / min}$$

∴ Capacity of the dehumidifier

$= m_a (W_1 - W_3)$

$= 34.05\ (0{:}0123 - 0.0084) = 0.1328$ kg / min

$= 0.1328 \times 60 = 7.968$ kg / h Ans.

***3. Capacity and surface temperature of cooling coil***

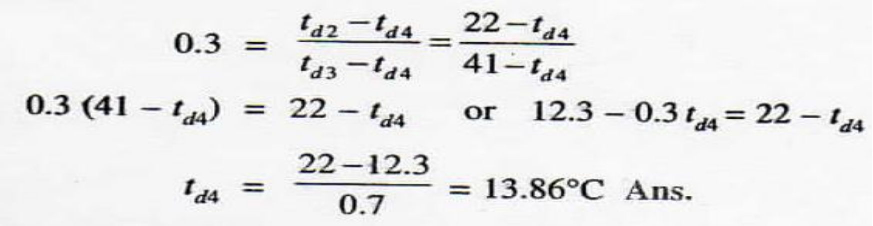
We know that capacity of the cooling coil

$= m_a (h_3 - h_2) = 34.05\ (64.5 — 45) = 664$ kJ/min

$= 664 / 210 = 3.16$TR Ans. .... ($\because$ 1TR = 210 kJ/min)

Let $t_{d4}$ = Surface temperature of the cooling coil.

We know that by-pass factor (BPF),



$$0.3 = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}} = \frac{22 - t_{d4}}{41 - t_{d4}}$$

$$0.3\ (41 - t_{d4}) = 22 - t_{d4} \quad \text{or} \quad 12.3 - 0.3\, t_{d4} = 22 - t_{d4}$$

$$t_{d4} = \frac{22 - 12.3}{0.7} = 13.86°\text{C Ans.}$$

**Example 6:** ***The following data refer to air conditioning of a public hall:***

***Outdoor conditions = 40°C DBT, 20°C WBT***
***Required comfort conditions = 20°C DBT, 50% RH***
***Seating capacity of hall = 1000***
***Amount of outdoor air supplied = 0.3 m³/min/person***

***If the required condition is achieved first by adiabatic humidifying and then cooling, find: 1. The capacity of the cooling coil and surface temperature of the coil if the by-pass factor is 0.25; and 2. The capacity of the humidifier and its efficiency.***

**Solution:** Given $t_{dt} = 40°C$; $t_{w1} = 20°C$; $t_{d2} = 20°C$; $\emptyset_2 = 50\%$; Seating capacity = 1000; $v_1 = 0.3\ m^3$/min/person = 0.3 x 1000 = 300 m3/min; BPF = 0.25

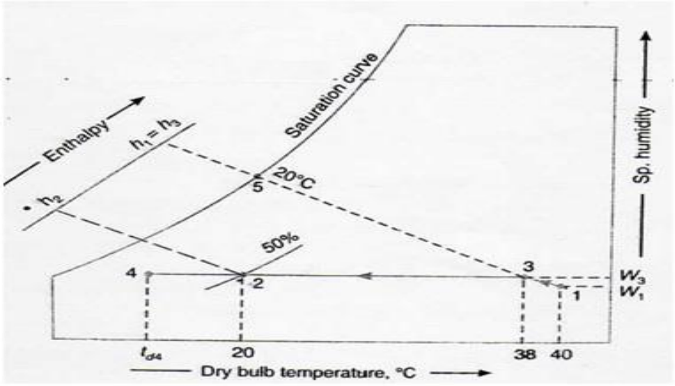


**Fig.8**

First of all, mark the outdoor conditions of air i.e., at 40°C dry bulb temperature and 20°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 8. Now mark the required comfort conditions of air i.e. at 20°C dry bulb temperature and 50% relative humidity, as point 2. From point 1, draw a constant wet bulb temperature line and from point 2 draw a constant specific humidity line. Let these two lines intersect at point 3. The line 1-3 represents adiabatic humidification and the line 3-2 represents sensible cooling.

From the psychrometric chart, we find that specific volume of air at point 1,

$$v_{s1}, = 0.896 \text{ m}^3/\text{ kg of dry air}$$

∴ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{300}{0.896} = 334.8 \text{ kg / min}$$

***1. Capacity of the cooling coil and surface temperature of the coil***

From the psychrometric chart, we find that enthalpy of air at point 3,

$$h_3 = 57.6 \text{ k)1 kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 39 \text{ / kg of dry air}$$

Dry bulb temperature of air after humidification i.e., at point 3.

$$t_{d3} = 38°\text{C}$$

We know that capacity of the cooling coil

$$= m_a (h_3 - h_2) = 334.8 (57.6 - 39) = 6227 \text{ kJ/min}$$

$$= 6227 / 210 = 29.6 \text{ TR Ans.}$$

Let $t_{d4}$ = Surface temperature of the coil

We know that by-pass factor (BPF),

$$0.25 = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}} = \frac{20 - t_{d4}}{38 - t_{d4}}$$

$$0.25 (38 - t_{d4}) = 20 - t_{d4} \quad \text{or} \quad 9.5 - 0.25\, t_{d4} = 20 - t_{d4}$$

$$t_{d4} = \frac{20 - 9.5}{0.75} = 14°\text{C Ans.}$$

***2. Capacity of the humidifier and its efficiency***

From the psychrometric chart, we find that specific humidity at point 1,

$$W_1 = 0.0064 \text{ kg / kg of dry air}$$

Specific humidity at point 3,

$$W_3 = 0:0074 \text{ kg / kg of dry air}$$

and dry bulb temperature at point 5,

$$t_{d5} = 20°\text{C}$$

We know that capacity of the humidifier

$$= m_a (W_3 - W_1) = 334.8 (0.0074 - 0.0064) = 0.3348 \text{ kg / min}$$

$$= 0.3148 \times 60 = 20.1 \text{ kg / h Ans.}$$

and efficiency of the humidifier,

$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{t_{d1} - t_{d3}}{t_{d1} - t_{d5}}$$

$$= \frac{40 - 38}{40 - 20} = 0.10 \text{ or } 10\% \text{ Ans.}$$

## 3.22 Year-Round Air Conditioning System

The year-round air conditioning system should have equipment for both the summer and winter air conditioning. The schematic arrangement of a modern summer year-round air conditioning system is shown in Fig.9.

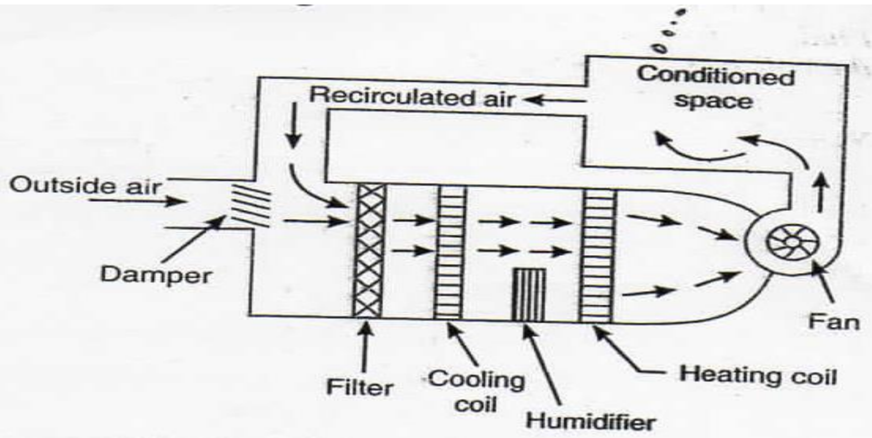


Fig.9 Year-round air conditioning system

The outside air flows through the damper and mixes up with the Damper re-circulated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. In summer air conditioning, the cooling coil operates to cool the air to the desired value. The dehumidification is obtained by operating the cooling coil at a temperature lower than the dew point temperature (apparatus dew point). In winter, the cooling coil is made inoperative and the heating coil operates to heat the air. The spray type humidifier is also made use of in the dry season to humidify the air.

**Example 7:** *An air conditioning plant is to be designed for a small office for winter conditions with the following data:*

*Outdoor conditions = 10°C DBT and 8°C WBT*

*Required indoor conditions = 20°C DBT and 60% RH*

*Amount of air circulation = 0.3* $m^3$*/min/person*

*Seating capacity of the office = 50 persons*

The required condition is achieved first by heating and then by adiabatic humidifying. Find: I. Heating capacity of the coil in kW and the surface temperature, if the by-pass factor of the coil is 0.32 and 2. capacity of the humidifier.

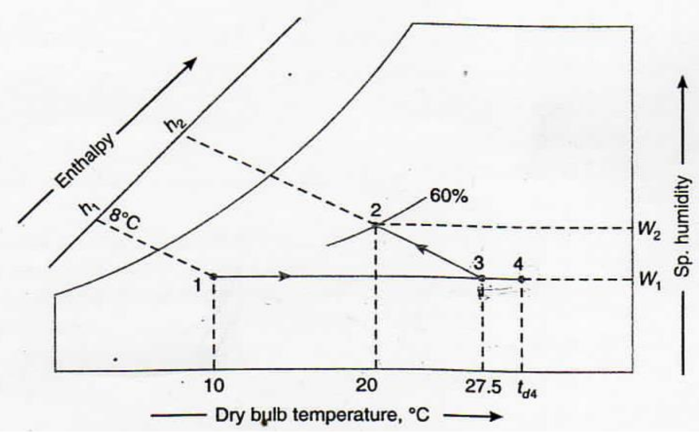


**Fig.10**

**Solution:** Given $t_{dt}$, = 10°C; $t_{wl}$ = 8°C; $t_{d2}$ = 20°C; $\varnothing_2$= 60%; seating capacity = 50 persons; $v_1$ = 0.3 $m^3$/min/person = 0.3 x 50 = 15 $m^3$/min; BPF = 0.32

First of all, mark the initial condition of air at 10°C dry bulb temperature and 8°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.11. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Now locate point 3 on the chart by drawing horizontal line through point I and constant enthalpy line through point 2, From the psychrometric chart, we find that the specific volume at point 1,

$$v_{sl} = 0.81 \text{ m}^3\text{/kg of dry air}$$

∴ Mass of air supplied per minute,

$$m_a = \frac{v_1}{v_{s1}} = \frac{15}{0.81} = 18.52 \text{ kg / min}$$

***1. Heating capacity of the coil in kW and the surface temperature***

From the psychrometric chart, we find that enthalpy at point 1,

$$h_1 = 24.8 \text{ kJ / kg of dry air}$$

and enthalpy at point 2, h2= 42.6 kJ / kg of dry air

We know that heating capacity of the coil

$$= m_a (h_2 - h_1) = 18.52 (42.6 - 24.8) = 329.66 \text{ kJ/min}$$

$$= 329.66 / 60 = 5.5 \text{ kW Ans.}$$

Let $t_{d4}$ = Surface temperature of the coil.

We know that by-pass factor (*BPF*),

$$0.32 = \frac{t_{d4} - t_{d3}}{t_{d4} - t_{d1}} = \frac{t_{d4} - 27.5}{t_{d4} - 10}$$

….[From psychrometric chart, $t_{d3}$, = 27.5°C]

or $0.32 (t_{d4} - 10) = t_{d4} - 27.5$ or $0.32\, t_{d4} - 3.2 = t_{d4} - 27.5$

$$t_{d4} = 24.3 / 0.68 = 35.7°\text{C Ans.}$$

***2. Capacity of the humidifier***

From the psychrometric chart, we find that specific humidity at point 1,

$$W_1 = 0.0058 \text{ kg / kg of dry air}$$

and specific humidity at point 2,

$$W_2 = 0.0088 \text{ kg / kg of dry air}$$

We know that capacity of the humidifier,

$$= m_a (W_2 - W_1) = 18.52 (0.0088 — 0.0058) = 0.055 \text{ kg / min}$$

$$= 0.055 \times 60 = 3.3 \text{ kg / h Ans.}$$

**Example 8**: *A small office hall of 25 person capacity is provided with summer air conditioning system with the following data:*

*Outside conditions = 34°C DBT and 28°C WBT*

*Inside conditions = 24°C DBT and 50% RH*

*Volume of air supplied = 0.4 m3 / min / person*

*Sensible heat load in room = 125 600 k / h*

*Latent heat load in the room = 42 000 kJ / h*

*Find the sensible heat factor of the plant.*

**Solution:** Given Seating capacity = 25 persons; $t_{dt}$ = 34°C; $t_{wl}$ = 28°C; $t_{d2}$, = 24°C ; $\phi_2$ = 50% ; $v_1$ = 0.4 $m^3$/min/person = 0.4 x 25 = 10 m3/min; S.H. load = 125 600 kJ / h; L.H. load = 42 000 kJ / h

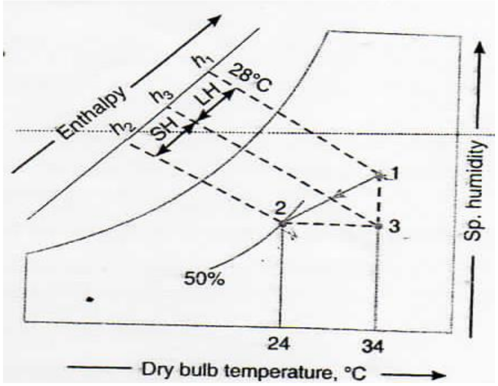


Fig.11

First of all, mark the initial condition of air at 34°C dry bulb temperature and 28°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.12. Now mark the final condition of air at 24°C dry bulb temperature and 50% relative humidity on the chart as point 2. Now locate point 3 on the chart by drawing horizontal line through point 2 and vertical line through point 1. From the psychrometric chart, we find that specific volume at point 1,

$$v_{s1} = 0.9 \text{ m3/ kg of dry air}$$

Enthalpy of air at point 1,

$$h_1 = 90 \text{ kJ / kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 48 \text{ ld / kg of dry air}$$

and enthalpy of air at point 3,

$$h_3 = 58 \text{ / kg of dry air}$$

We know that mass of air supplied per min,

$$m_a = \frac{v_1}{v_{s1}} = \frac{10}{0.9} = 11.1 \text{ kg / min}$$

and sensible heat removed from the air

$$= m_a (h_3 - h_2) = 11.1(58 - 48) = 111 \text{ kJ / min}$$

$$= 111 \times 60 = 6660 \text{ kJ/h}$$

Total sensible heat of the room,

$$SH = 6660 + 125\,600 = 132\,260 \text{ kJ / h}$$

We know that latent heat removed from the air

$$= m_a (h_1 - h_3) = 11.1(90 - 58) = 355 \text{ kJ / min}$$

$$= 355 \times 60 = 21\,300 \text{ kJ / h}$$

∴ Total latent heat of the room,

$$LH = 21\,300 + 42000 = 63300 \text{ kJ / h}$$

We know that sensible heat factor,

$$SHF = \frac{SH}{SH + LH} = \frac{132\,260}{132\,260 + 63\,300} = 0.676 \text{ Ans.}$$

**Example 9:** A restaurant with a capacity of 100 persons is to be air-conditioned with the following conditions:

| | |
|---|---|
| Outside conditions | : 30°C DBT and 70% RH |
| Desired inside conditions | : 23°C DBT and 55% RH |
| Quantity of air supplied | : 0.5 rn3 / min / person |

The desired conditions are achieved by cooling, dehumidifying and then heating. Determine: I. Capacity of cooling coil in tones of refrigeration; 2. Capacity of heating coil; 3. Amount of water removed by dehumidifier; and 4. By-pass factor of the heating coil if its surface temperature is 35°C.

**Solution:** Given: Number of persons = 100; $t_{dt}$ = 30°C; $\emptyset_1$ = 70%; $t_{d4}$ = 23°C; $\emptyset_4$ = 55% ; $v_l$ = 0.5 $m^3$/ min / person = 0.5 x 100 = 50 $m^3$/train

First of all, mark the outside conditions of air at 30°C dry bulb temperature and 70% relative humidity on the psychrometric chart as point 1, as shown in Fig. 18.13. Now mark the desired inside conditions of air at 23°C dry bulb temperature and 55% relative humidity on the chart as point 4. The process 1-2 represents the sensible cooling, process 2-3 represents dehumidification and the process 3-4 represents the sensible heating.

From the psychrometric chart, we find that the specific volume at point 1.

$$v_{s1} = 0.885 \text{ m}^3\text{/kg of dry air}$$

∴ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{50}{0.885} = 56.5 \text{ kg / min}$$

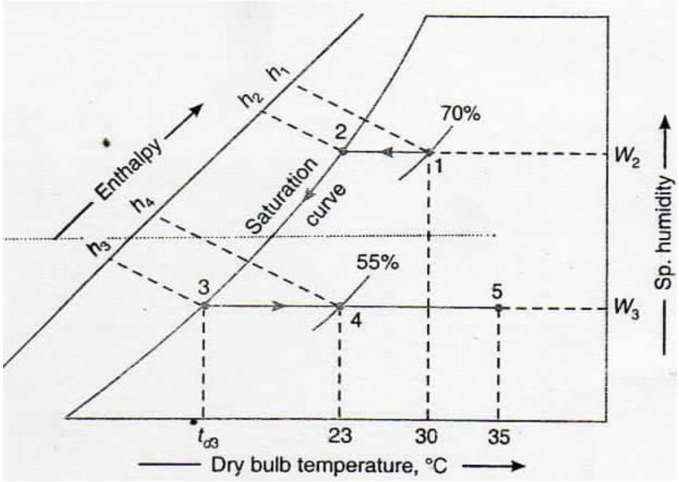


**Fig.12**

***1. Capacity of cooling coil in tones of refrigeration***

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 78.5 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 3,

$$h_3 = 37.8 \text{ kJ/kg of dry air}$$

∴ Capacity of the cooling coil

$$= m_a (h_1 - h_3) = 56.5 (78.5 - 37.8) = 2300 \text{ kJ/min}$$

$$= 2300 / 210 = 10.95 \text{ TR Ans.}$$

***2. Capacity of heating coil***

From the psychrometric chart, we find that enthalpy of air at point 4,

$$h_4 = 47.6 \text{ kJ / kg of dry air}$$

∴ Capacity of the heating coil

$$= m_a (h_4 - h_3) = 56.5 (47.6 - 37.8) = 554 \text{ kJ/min}$$

$$= 554 / 60 = 9.23 \text{ kW Ans.}$$

***3. Amount of water removed by dehumidifier***

From the psychrometric chart, we find that specific humidity at point 2,

$W_2$ = 0.0188 kg / kg of dry air

and specific humidity at point 3,

$W_3$ = 0.0095 kg / kg of dry air

∴ Amount of water removed by dehumidifier

= $m_a$ ($W_2$ - $W_3$) = 56.5 (0.0188 — 0.0095) = 0.525 kg I min

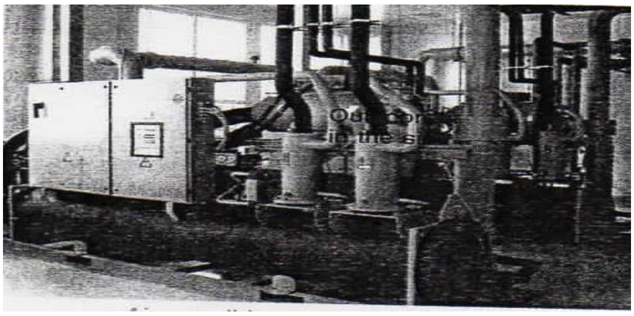
= 0.525 x 60 = 31.5 kg / h Ans. 4.

***4. By-pass factor of the heating coil***

Let $t_{d5}$ = Surface temperature of the heating coil = 35°C …..(Given)

From the psychrometric chart, we find that dry bulb temperature at point 3,

$t_{d3}$ = 13.5°C



Air conditioning system

We know that by-pass factor of the heating coil,

$$BPF = \frac{t_{d5} - t_{d4}}{t_{d5} - t_{d3}} = \frac{35 - 23}{35 - 13.5} = 0.558 \text{ Ans.}$$

**Room Sensible Heat Factor**

It is defined as the ratio of the room sensible heat to the room total heat. Mathematically, room sensible heat factor,

$$RSHF = \frac{RSH}{RTH} = \frac{RSH}{RSH + RLH}$$

where *RSH* = Room sensible heat,

*RLH* = Room latent heat, and

*RTH* = Room total heat.

The conditioned air supplied to the room must have the capacity to take up simultaneously both the room sensible heat and room latent heat loads. The point S on the psychrometric chart, as shown in Fig. 18.14, represents the supply air condition and the point R represents the required final condition in the room (i.e., room design condition). The line SR is called the room sensible heat factor line (RSHF line). The slope of this line gives the ratio of the room sensible heat (RSH) to the room latent heat (RLH). Thus the supply air having its conditions given by any point on this line will satisfy the requirements of the room with adequate supply of such air. In other words, the supply air having conditions marked by points $S_1$, $S_2$, $S_3$, $S_4$ etc., will satisfy the requirement but the quantity of air supplied will be different for different supply air points. The supply condition at S requires minimum air and at point 54, it is maximum of all the four points.

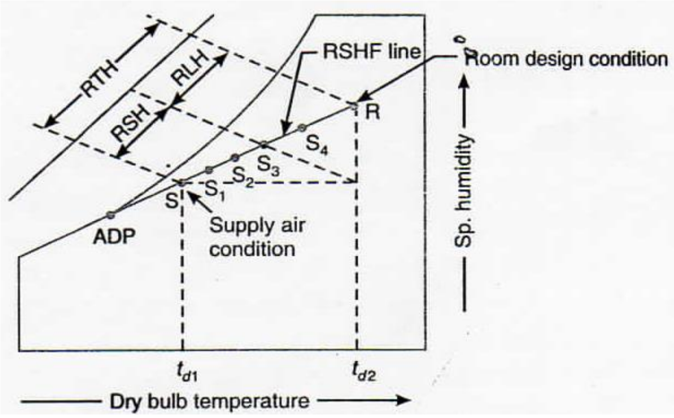


Fig. 13. Representation of supply air condition and room design condition.

When the supply air conditions are not known, which in fact is generally required to be found out, the room sensible heat factor line may be drawn from the calculated value of room sensible heat factor (RSHF), as discussed below:

1. Mark point a on the sensible heat factor scale given on the right hand corner of the psychrometric chart as shown-in Fig.14. The point a represents the calculated value of *RSHF*.

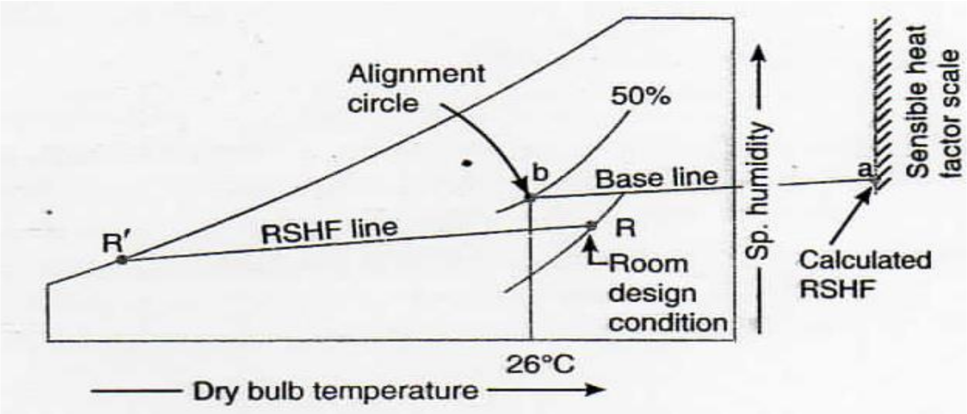


Fig. 14 Room sensible heat factor (RSHF) line.

2. Join point *a* with the *alignment circle or the reference point *b*. The line *ab* is called base line.

3. Mark point *R* on the psychrometric chart to represent the room design conditions.

4. Through point *R* draw a line *RR*' parallel to the base line *ab*. This line is the required room sensible

heat factor line.

Note: In a cooling and dehumidification process, the temperature at which the room sensible heat factor line intersects the saturation curve is called *room apparatus dew point* (*ADP*).

### 3.23 Grand Sensible Heat Factor

It is defined as the ratio of the total sensible heat to the grand total heat which the cooling coil or the conditioning apparatus is required to handle. Mathematically, grand sensible heat factor,

$$GSHF = \frac{TSH}{GTH} = \frac{TSH}{TSH + TLH} = \frac{RSH + OASH}{(RSH + OASH) + (RLH + OALH)}$$

where TSH = Total sensible heat = RSH + OASH

TLH = Total latent heat = RLH + OALH

GTH = Grand total heat = TSH + TLH = RSH + RLH + OATH

= RSH + RLH + (DASH + OALH)

Let $v_1$ = Volume of outside air or ventilation in $m^3$/min,

$t_{d1}$ = Dry bulb temperature of outside air in °C,

$W_1$ = Specific humidity of outside air in kg / kg of dry air,

$h_i$ = Enthalpy of outside air in kJ / kg of dry air,

$t_{d2}$ = Dry bulb temperature of room air in °C,

$W_2$ = Specific humidity of room air in kg / kg of dry air, and

$h_2$ = Enthalpy of room air in kJ / kg of dry air.

∴ Outside air sensible heat,

$$OASH = 0.020\,44\, v_1\, (t_{d1} - t_{d2}) \text{ kW}$$

Outside air latent heat,

$$OALH = 50\, v_1\, (W_1 - W_2) \text{ kW}$$

and outside air total heat,

$$OATH = OASH + OALH$$

The outside air total heat may also be calculated from the following relation:

$$\text{OATH} = 0.02\, v_1 (h_1 - h_2) \text{ kW}$$

Generally, the air supplied to the air conditioning plant is a mixture of fresh air (or outside air or ventilation) and the re-circulated air having the properties of room air. On the psychrometric chart, as shown in Fig. 15, the point 1 represents the outside condition of air, the point 2 represents the• room air condition and the point 3 represents the mixture condition of air entering the cooling coil. When the mixture condition enters the cooling coil or conditioning apparatus, it is cooled and dehumidified. The point 4 shows the supply air or leaving condition of air from the cooling coil or conditioning apparatus. When the point 3 is joined with the point 4, it gives a grand sensible heat factor line (*GSHF* line) as shown in Fig. 5. This line, when produced up to the saturation curve, gives apparatus dew point (*ADP*).

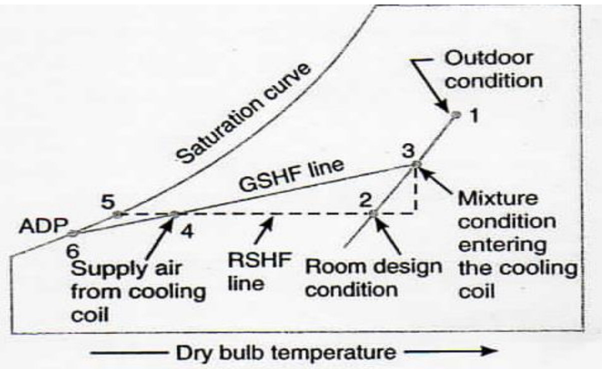


Fig.15 Grand sensible heat factor

If the mixture condition entering the cooling coil or conditioning apparatus and the grand sensible heat factor (*GSHF*) are known, then the *GSHF* line may be drawn on the psychrometric chart in the similar way as discussed for RSHF line. The point 4, as shown in Fig. 15, is the intersection of *GSHF* line and *RSHF* line. This point gives the ideal conditions for supply air to the room.

### 3.24 Effective Room Sensible Heat Factor

It is defined as the ratio of the effective room sensible heat to the effective room total heat. Mathematically, effective room sensible heat factor,

$$ERSHF = \frac{ERSH}{ERTH} = \frac{ERSH}{ERSH + ERLH}$$

Where $ERSH$ = Effective room sensible heat = $RSH + OASH \times BPF$

$= RSH + 0.020\ 44\ v_1\ (t_{dt} – t_{d2})\ BPF$

$ERLH$ = Effective room latent heat = $RLH + OALH \times BPF$

$= RLH + 50\ v_1\ (W_1 - W_2)\ BPF$

$ERTH$ = Effective room total heat = $ERSH + ERLH$

$BPF$ = By-pass factor

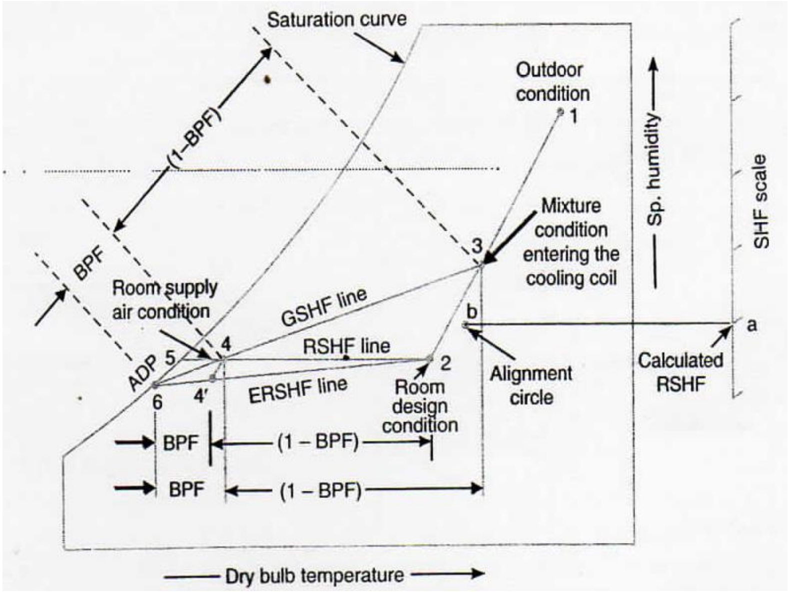


Fig. 16. Effective room sensible heat factor.

The line joining the point 2 and point 6 i.e. ADP, as shown in Fig. 18.17, gives the effective room sensible heat factor line (*ERSHF* line). From point 4, draw 4.4' parallel to 3-2. Therefore from similar triangles 6-4-4' and 6-3-2,

$$BPF = \frac{\text{Length 4-6}}{\text{Length 3-6}} = \frac{\text{Length 4}'\text{-6}}{\text{Length 2-6}}$$

The by-pass factor is also given by,

$$BPF = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4'} - ADP}{t_{d2} - ADP}$$

**Notes:1**. The effective room sensible heat (*ERSH*), effective room latent heat (*ERLH*) and effective room total heat (ERTH) may also be obtained from the following relations:

$ERSH = 0.020\ 44\ v_d(t_{d2} - ADP)\ (1 - BPF)$ kW

$ERLH = 50\ v_d\ (W_2 - W_{ADP})\ (1 - BPF)$ kW

and $ERTH = 0.02\ v_d(h_2 – h_{ADP})\ (1 - BPF)$ kW

where $v_d$ = Volume of dehumidified air to room or space in m$^3$/min,

$ADP$ = Apparatus dew point in °C,

$W_{ADP}$ = Specific humidity at apparatus dew point in kJ / kg of dry air, and

$h_{ADP}$ = Enthalpy at apparatus dew point in kJ / kg of dry air.

2. The mass of dehumidified air is given by

$$m_d = \frac{\text{Room total heat}}{h_2 - h_4}$$

where $h_2$ = Enthalpy of air at room condition, and

$h_4$ = Enthalpy of supply air to room from the cooling coil.

**Example 10:** *In an air conditioning system, the inside and outside conditions are dry bulb temperature 25°C, relative humidity 50% and dry bulb temperature 40°C, wet bulb temperature 27°C respectively. The room sensible heat factor is 0.8. 50% of the room air is rejected to atmosphere and an equal quantity of fresh air added before air enters the air conditioning apparatus. If the fresh air added is 100* $m^3$*/min, determine:*

*1. Room sensible and latent heat load;*

*2. Sensible and latent heat load due to fresh air;*

*3. Apparatus dew point;*

*4. Humidity ratio and dry bulb temperature of air entering air conditioning apparatus.*

*Assume by-pass factor as zero, density of air as 1.2 kg /* $m^3$ *at a total pressure of 1.01325 bar.* **Solution:** Given $t_{dt}$ = 40°C; $t_{w1}$, = 27°C; $t_{d2}$= 25°C; $\varnothing_2$ = 50% ; RSHF= 0.8; $v_1$ = 100 $m^3$/min; $\rho_a$ =1.2 kg/$m^3$

The flow diagram for the air conditioning system is shown in Fig. 17, and it is represented on the psychrometric chart as discussed below:

First of all, mark the outside condition of air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18. Now mark the inside condition of air at 25°C dry bulb temperature and 50% relative humidity as point 2. Since 50% of the room air and 50% of fresh air is added before entering the air conditioning apparatus, therefore mark point 3 on the line 1-2 such that

$$\text{Length 2-3} = \frac{\text{Length 1-2}}{2}$$

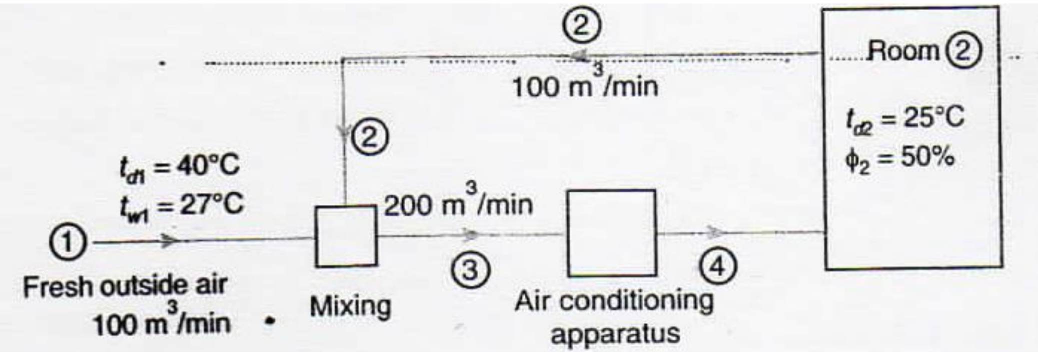


**Fig.17**

Now mark the given value of RSHF (i.e. 0.8) on the room sensible heat factor scale and join this with the alignment circle (i.e. 26°C DBT and 50% RH). From point 2, draw a line 2-4 parallel to this line. This line is called RSHF line. The point 4 represents the apparatus dew point (*ADP*). From the psychrometric chart, we find the enthalpy of air at point 1,

$$h_1 = 85.2 \text{ kJ / kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 50 \text{ kJ / kg of dry air}$$

and enthalpy of air at point 4,

$$h_4 = 33 \text{ kJ / kg of dry air}$$

***1. Room sensible and latent heat load***

We know that mass of air supplied to the room,

$m_a = v_3 \times \rho_a = (100 + 100)1.2 = 240$ kg/min

∴ Room sensible heat load,

$$\text{RSH} = m_a C_{pm} (t_{d2} - t_{d4})$$

$$= 240 \times 1.022 (25 - 11.8) = 3238 \text{ kJ / min}$$

= 3238/60 = 53.96.kJ/s or kW … [∵ From psychrometric chart, $t_{d4} = 11.8$°C]

and room total heat load,

$$RTH = m_a (h_2 - h_4) = 240 (50 - 33) = 4080 \text{ kJ/min}$$

$$= 4080/60 = 68 \text{ kJ/s or kW}$$

∴ Room latent heat load,

$RLH$ = RTH - RSH

= 68 — 53.96 = 14.04 kW

2. Sensible and latent heat load due to fresh air

We know that mass of fresh air supplied,

$m_F = v_1 \times \rho_a = 100 \times 1.2 = 120$ kg / min

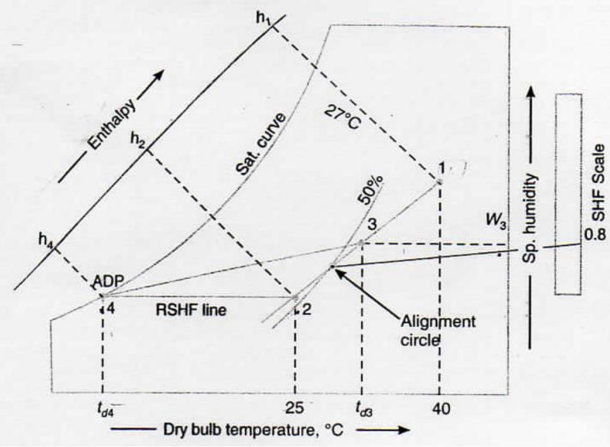


**Fig.18**

∴ Sensible heat load due to fresh air

$= m_F C_{pm} (t_{d1} - t_{d2})$

= 120 x 1.022 (40 - 25) = 1840 kJ/min

= 1840 / 60 = 30.67 kJ/s or kW Ans.

and total heat load due to fresh air

$= m_F (h_1 - h_2) = 120 (85.2 - 50) = 4224$ kJ/min

= 4224 / 60 = 70.4 kJ/s or kW

∴Latent heat load due to fresh air

= Total heat load — Sensible heat load

= 70.4 — 30.67 = 39.73 kW Ans.

**3. Apparatus dew point**

From the psychrometric chart, we find that apparatus dew point ($ADP$) corresponding to point 4 is

$t_{d4}$ = 11.8°C Ans.

***4. Humidity ratio and dry bulb temperature of air entering air conditioning apparatus***

The air entering the air conditioning apparatus is represented by point 3 on the psychrometric chart as shown in Fig. 18. From the psychrometric chart, we find that humidity ratio corresponding to point 3,

$$W_3 = 0.0138 \text{ kg / kg of dry air Ans.}$$

and dry bulb temperature corresponding to point 3.

$$t_{d4} = 32.5°\text{C Ans.}$$

**Example 11:** *An air conditioned space is maintained at 27°C dry bulb temperature and 40% relative humidity. The ambient conditions are 40°C dry bulb temperature and 27°C wet bulb temperature. The space has a sensible heat gain of 14kW. The air is supplied to the space at 7°C saturated. Calculate:*

1. Mass of moist air supplied to the space in kg / h; 2. Latent heat gain of space in kW; and 3. Cooling load of air-washer in kW if 30 per cent of air supplied to the space is fresh, the remainder being re-circulated.

**Solution:** Given $t_{d2}$=.27°C; $\emptyset_2$ = 50%; $t_{dt}$ = 40°C; $t_{w1}$ = 27°C; $Q$s = 14 kW = 14 kJ/s = 14x3600 kJ/ h; $t_{d4}$ = 7°C

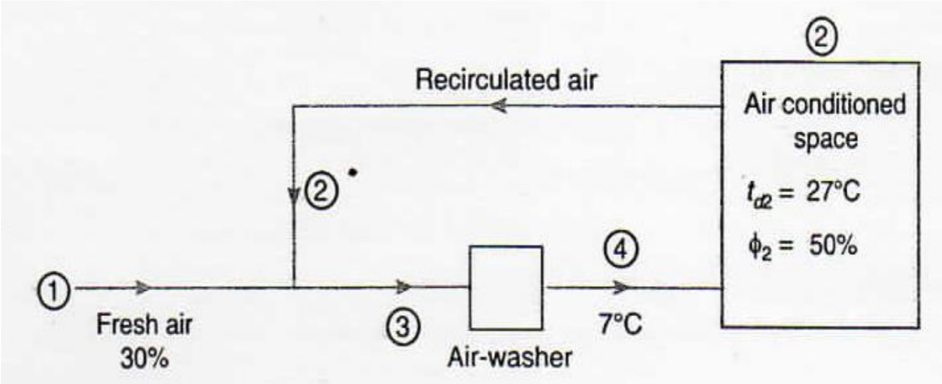


**Fig. 19**

The line diagram for the air conditioned space is shown in Fig. 19 and it is represented on the psychrometric chart as discussed below:

First of all, mark the ambient (outside) conditions of air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 20. Now mark the inside conditions of the space at 27°C dry bulb temperature and 50% relative humidity. Since the air is supplied to the space at 7°C saturated, therefore mark point 4 on the saturation curve at 7°C. Also 30 per cent of air supplied to the space (i.e. at point 2) is fresh, therefore mark point 3 on the line 2-1, such that

Length 2-3 = 0.3 x Length 2- I

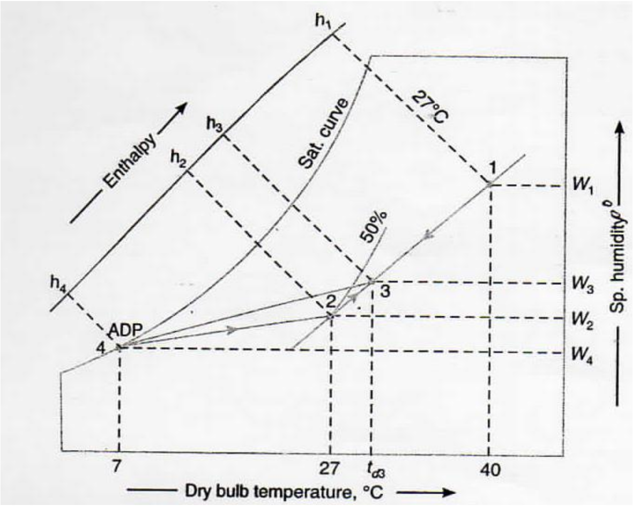


**Fig.20**

Now from the psychrometric chart, we find that enthalpy of air at point 1,

$h_1$= 85 kJ / kg of dry air

Specific humidity of air at point 1,

$W_1$= 0.0172 kg / kg of dry air

Enthalpy of air at point 2,

$h_2$ = 56 kJ / kg of dry air

Specific humidity of air at point 2,

$W_2$ = 0.0112 kg / kg of dry air

Enthalpy of air at point 4,

$h_4$ =23 kJ / kg of dry air

Specific humidity of air at point 4,

$W_4$ = 0.0062 kg / kg of dry air

***1. Mass of moist air supplied to the space in kg / h***

We know that mass of dry air supplied to the space,

$$m_a = \frac{Q_S}{c_{pm}(t_{d2} - t_{d4})} = \frac{14 \times 3600}{1.022\,(27-7)} = 2465.75 \text{ kg / h}$$

…. [∵ $C_{pm}$ = Humid specific heat = 1.022 kJ / kg K]

∴ Mass of moist air supplied to the space ,

$$= m_a (1 + W_4) = 2465.75 (1 + 0.0062)$$

$$= 2481 \text{ kg / h} \quad \text{Ans.}$$

***2. Latent heat gain of space in kW***

We know that latent heat gain of space,

$$Q_L = m_a (W_2 - W_4) h_{fg}$$

$$= 2465.75 (0.0112 — 0.0062) 2500 = 308\ 22 \text{ kJ/h}$$

$$= 308\ 22 / 3600 = 8.56 \text{ kJ/s or kW Ans.}$$

***3. Cooling load of air-washer in kW***

From the psychometric chart, we find that dry bulb temperature of air at point 3,

$$*t_{d3} = 31°C$$

and enthalpy of air at point 3,

$$h_3 = 64.6 \text{ kJ / kg of dry air}$$

We know that cooling load of air-washer

$$= m_a (h_3 - h_4) = 2465.75 (64.6 - 23) = 102\ 575 \text{ kJ/h}$$

$$= 102\ 575 / 3600 = 28.5 \text{ k-J/s or kW Ans}$$

**Example 12:** Air flowing at the rate of 100m$^3$/min at 40°C dry bulb temperature and 50% relative humidity is mixed with another stream flowing at the rate of 20m$^3$/min at 26°C dry: bulb temperature and 50% relative humidity. The mixture flows over a cooling coil whose apparatus dew point temperature is 10°C and by-pass factor is 0.2. Find dry bulb temperature and relative humidity of air leaving the coil. If this air is supplied to an air-conditioned room where dry bulb temperature of 26°C and relative humidity of 50% are maintained, estimate 1. Room sensible heat factor; and 2. Cooling load capacity of the coil in tones of refrigeration

Solution: Given $v_1$ =100 m$^3$/min; $t_{dt}$ = 40°C; $\emptyset_1$ = 50%; $v_2$ = 20 m$^3$/min ; $t_{d2}$ = 26°C ; $\emptyset_2$ = 50% ; *ADP* =10°C ; *BPF* = 0.2

The flow diagram for an air-conditioned room is shown in Fig 21 and it is represented on the psychrometric chart as discussed below:

First of all, mark the initial condition of air at 40°C dry bulb temperature and 50% relative humidity on the psychrometric chart as point 1, as shown in Fig. 22. Now mark the room condition of air at 26°C dry bulb temperature and 50% relative humidity as point 2. From the psychrometric chart, we find that enthalpy of air at point 1,

$h_1$ = 99.8 kJ / kg of dry air

Enthalpy of air at point 2,

$h_2$ = 53.5 Id / kg of dry air

Specific volume of air at point 1,

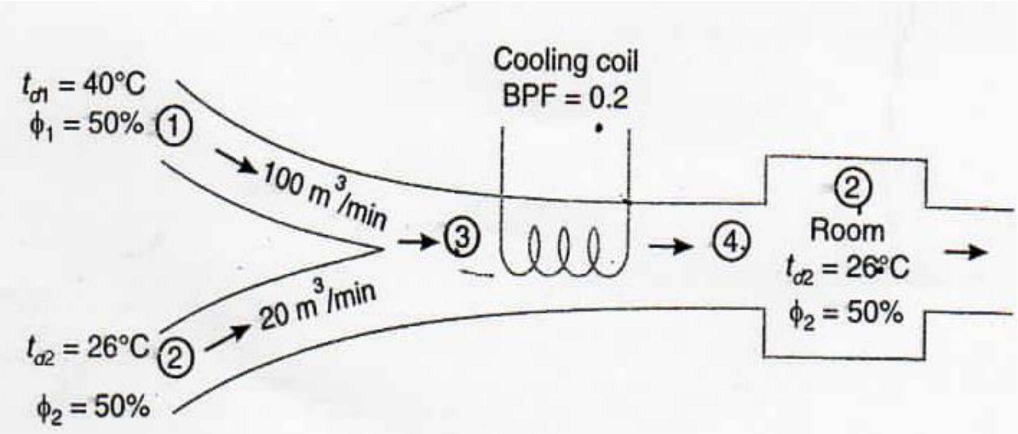
$v_{s1}$= 0.92 m3/ kg of dry air

and specific volume of air at point 2,

$v_{s2}$ = 0.862 $m^3$/ kg of dry air

We know that mass of air supplied at point 1,

$$m_{a1} = \frac{v_1}{v_{s1}} = \frac{100}{0.92} = 108.7 \text{ kg / min}$$



Fig,21

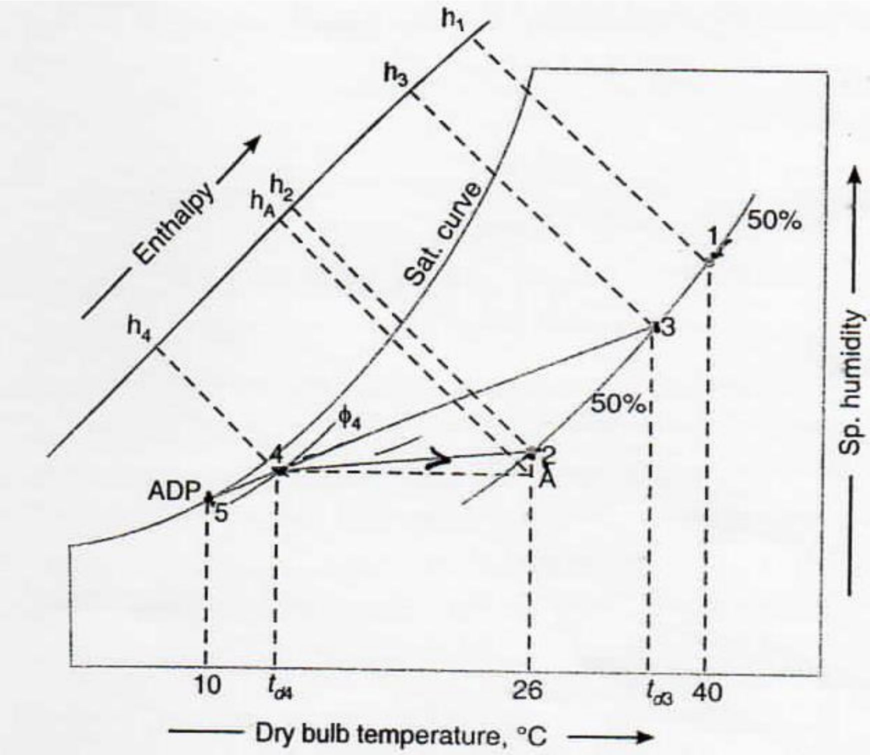


Fig.22

and mass of air supplied at point 2,

$$m_{a2} = \frac{v_2}{v_{s2}} = \frac{20}{0.862} = 23.2 \text{ kg / min}$$

∴ Mass of air flowing through the cooling coil at point 3.

$$m_{a3} = m_{al} + m_{a2} = 1083 + 212 = 131.9 \text{ kg / min}$$

For the energy balance,

$$m_{a1}h_1 + m_{a2}\,h_2 = m_{a3}\,h_3$$

$$\therefore \qquad h_3 = \frac{m_{a1}\,h_1 + m_{a2}\,h_2}{m_{a3}} = \frac{108.7\times 99.8 + 23.2\times 53.5}{131.9}$$

$$= 91.65 \text{ kJ / kg of dry air}$$

Now draw a constant enthalpy line corresponding to h3 = 91.65 k7 / kg of dry air which intersects the line 1-2 at point 3. From the psychrometric chart, we find that dry bulb temperature of air entering the cooling coil at point 3 is

$$t_{d3} = 37.6°\text{C}$$

Mark point 5 on the saturation curve such that ADP = 10°C, and draw a line 3-5. The point 4 lies on this line.

**Dry bulb temperature and relative humidity of air leaving the coil**

Let $t_{d4}$ = Dry bulb temperature of air leaving the coil.

We know that by-pass factor (BPF),

$$0.2 = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4} - 10}{37.6 - 10}$$

$\therefore t_{d4} = 15.52°\text{C}$ Ans.

From the psychrometric chart, we find that relative humidity of air leaving the coil at point 4 is

$\emptyset_4 = 92\%$ Ans.

***1. Room sensible heat factor***

From the psychrometric chart, we find that enthalpy of air at point 4, h

$h_4$ = 42 kJ / kg of dry air.

and enthalpy of air at point A (which is the intersection of horizontal line from point 4 and vertical line from point 2),

$h_A$ = 52.5 kJ / kg of dry air

We know that room sensible heat factor,

$$RSHF = \frac{h_A - h_4}{h_2 - h_4} = \frac{52.5 - 42}{53.5 - 42} = 0.913$$ Ans

***2. Cooling load capacity of the coil***

We know that cooling load capacity of the coil

$= m_{a3} (h_3 - h_4) = 131.9 (91.65 - 42) = 6548.8$ kJ/min

$= 6548.8 / 210 = 31.185$ TR Ans.

… (∵ 1 TR= 210 1J/min)

**Example 13**: *An air conditioned auditorium is to be maintained at 27°C dry bulb temperature and 60 % relative humidity. The ambient condition is 40°C dry bulb temperature and 30°C wet bulb temperature. The total sensible heat load is 100 000 kJ/h and the total latent heat load is 40 000 kJ/h. 60% of the return air is re-circulated and mixed with 40% of make-up air after the cooling coil. The condition of air leaving the cooling coil is at 18°C.*

*Determine: 1. Room sensible heat factor; 2. The condition of air entering the auditorium; 3. The amount of make-up air; 4. Apparatus dew point; and 5. By-pass factor of the cooling coil. Show the processes on the psychrometric chart.*

**Solution:** Given $t_{d4}$ =27°C; $\phi_4$ = 60%; $t_{dt}$, = 40°C; $t_{w1}$= 30°C; RSH= 100 000kJ / h; RLH = 40 000 kJ / h; $t_{d2}$ = 18°C

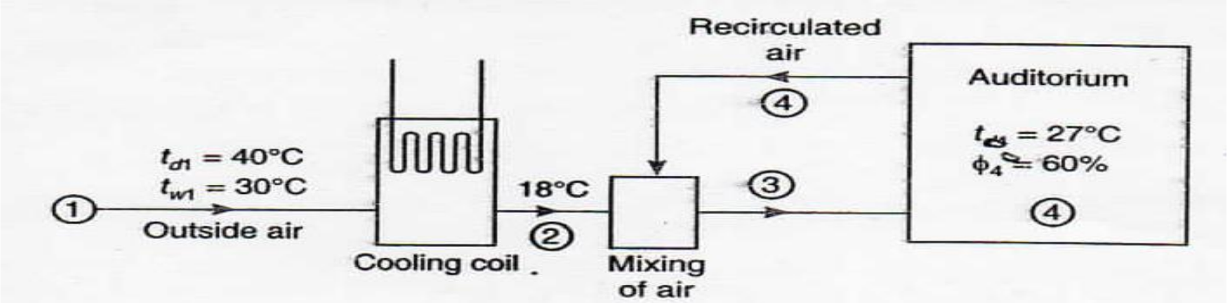
1. Room sensible heat factor

We know that room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{100\,000}{100\,000 + 40\,000} = 0.714 \text{ Ans.}$$

***2. Condition of air entering the auditorium***

The line diagram for processes involved in the air conditioning of an auditorium is shown in Fig. 23. These processes are shown on the psychrometric chart as discussed below



:

**Fig.23**

First of all, mark the ambient condition of air (outside air) i.e. at 40°C dry bulb temperature and 30°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.26. Now mark the condition of air in the auditorium, i.e. at 27°C dry bulb temperature and 60% relative humidity, as point 4.

Mark the calculated value of *RSHF* = 0.714 on the sensible heat factor scale as point *a* and join with point b which is the alignment circle (i.e. 26°C *DBT* and 50% *RH*) as shown in Fig. 24. Now from point 4, draw a line 4-5 (known as *RSHF* line) parallel to the line *ab*. Since the condition of air leaving the cooling coil is at 18°C, therefore, mark point 2 such that $t_{d2}$= 18°C. Join points 1 and 2 and produce up to point 6 on the saturation curve. The line 1-2-6 is the *GSHF* line. It is given that 60% of the air from the auditorium is re-circulated and mixed with 40% of the make-up air after the cooling coil. The mixing condition of air is shown at point such that

$$\frac{\text{Length } 2-3}{\text{Length } 2-4} = 0.6$$

The condition of air entering the auditorium is given by point 3. From the psychrometric chart, we find that at point 3,

Dry bulb temperature, $t_{d3}$ = 23°C Ans.

Wet bulb temperature, $t_{w3}$= 19.5° C Ans.

and relative humidity, $\emptyset_3 = 72\%$ Ans.

3. Amount of make-up air

From the psychrometric chart, we find that enthalpy of air at point 4,

$$h_4 = 61 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 3, $h_3$ = 56 kJ/kg of dry air

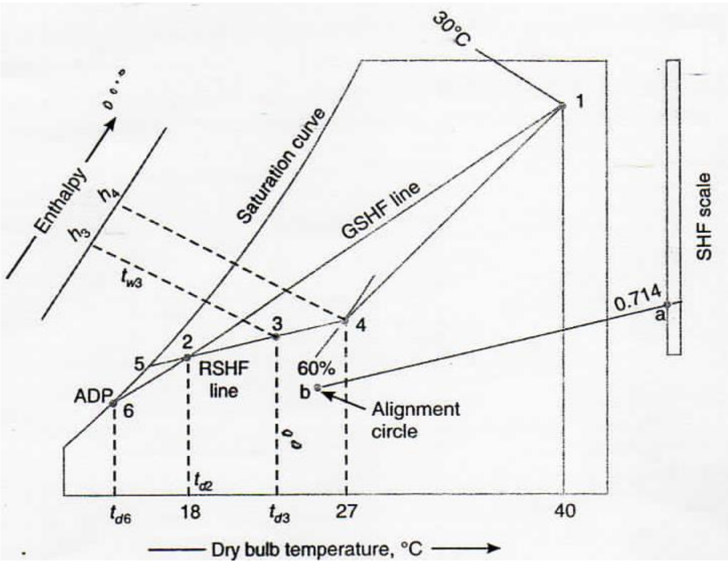


**Fig.24**

We know that mass of supply air to the auditorium,

$$m_S = \frac{\text{Room total heat}}{h_4 - h_3} = \frac{RSH + RLH}{h_4 - h_3}$$

$$= \frac{100\,000 + 40\,000}{61 - 56} = 28\,000 \text{ kg / h}$$

Since the make-up air is 40% of supply air, therefore mass of make-up air

= 0.4 x 28 000= 11 200 kg / h Ans.

**4. Apparatus dew point**

From the psychromettic chart, we find that the apparatus dew point of the cooling coil at point 6 is

ADP = $t_{d6}$= 13° C Ans. 3

***5. By-pass factor .of the coaling coil***

We know that by-pass factor of the cooling coil,

$$BPF = \frac{t_{d2} - ADP}{t_{d1} - ADP} = \frac{18 - 13}{40 - 13} = \frac{5}{27} = 0.185 \text{ Ans.}$$

**Example 14:** *An air conditioned hall is to be maintained at 27°C dry bulb temperature and 21°C wet bulb temperature. It has a sensible heat load of 46.5 kW and latent heat load of 17.5 kW. The air supplied from outside atmosphere at 38°C dry bulb temperature and 27°C wet bulb temperature is 25m$^3$/min, directly into the room through ventilation and infiltration. Outside air to be conditioned is passed through the cooling coil whose apparatus dew point is 15°C. The quantity of re-circulated air from the hall is 60%. This quantity is mixed with the conditioned air after the cooling coil. Determine : 1. condition of air after the coil and before the re-circulated air mixes with it; 2. condition of air entering the hall, i.e. after mixing with re-circulated air; 3. mass of fresh air entering the cooler; 4. by-pass factor of the cooling coil: and 5. refrigerating load on the cooling coil.*

**Solution:** Given $t_{d4}$ = 27°C; $t_{w4}$ = 21°C; $Q_{s4}$ = 46.5 kW; $Q_{L4}$ = 17.5 kW; $t_{dt}$, = 38°C; $t_{w1}$= 27°C ; $v_1$ = 25m$^3$/min; ADP = 15°C

The line diagram for the processes involved in the air conditioning of a hall is shown in Fig. 18.27. These processes are shown on the psychrometric chart as discussed below:

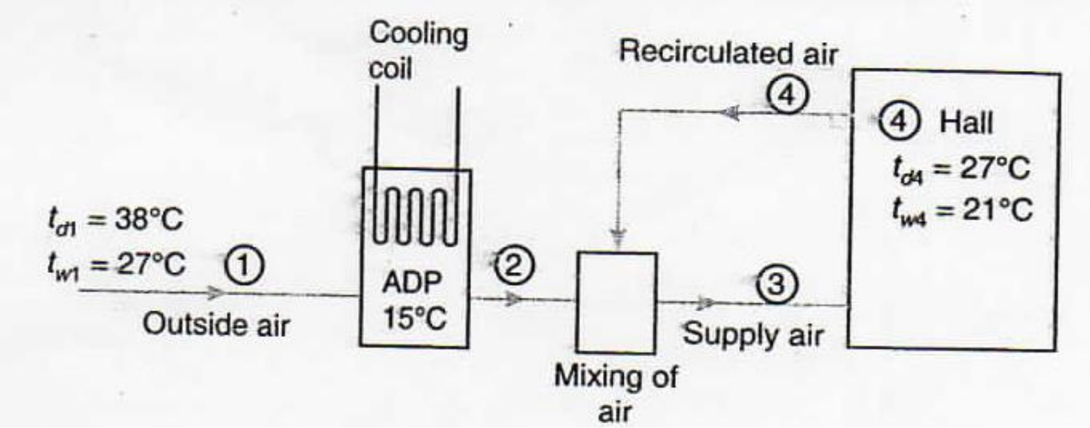


**Fig. 25**

First of all, mark the condition of outside air i.e. at 38°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.28. Now mark the condition of air in the hall, i.e. at 27°C dry bulb temperature and 21°C wet bulb temperature, at point 4. Mark point A by drawing vertical and horizontal lines from points 1 and 4 respectively. Since 25 m3/min of outside air at Id, = 38°C and $t_{w1}$ = 27°C is supplied directly into the room through ventilation and infiltration, therefore the sensible heat and latent heat of 25 m$^3$/min infiltrated air are added to the hall in addition to the sensible heat load of 46.5 kW and latent heat load of 17.5 kW.

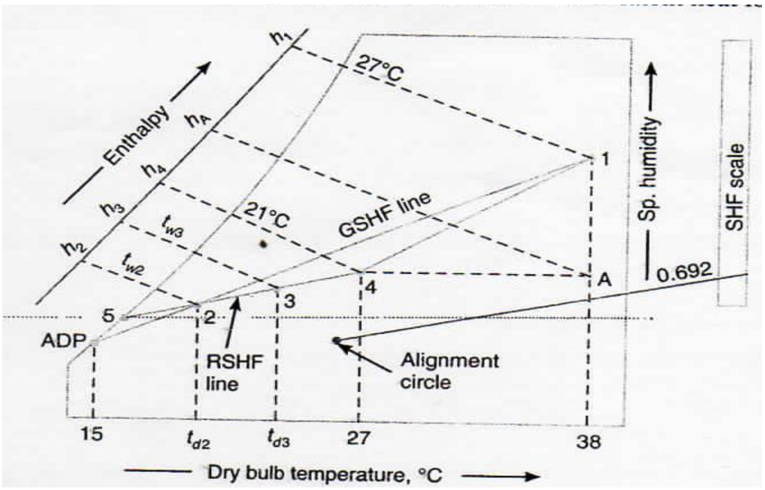


**Fig.26**

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 85 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 4

$$H_4 = 61 \text{ kJ/kg of dry air}$$

and enthalpy of air at point A ,

$$h_A = 72.8 \text{ kJ / kg of dry air}$$

Also specific volume of air at point 1,

$$v_{s1}, = 0.907 \text{ m}^3\text{/ kg of dry air}$$

∴ Mass of air infiltrated into the hall,

$$m_a = \frac{v_1}{v_{s1}} = \frac{25}{0.907} = 27.56 \text{ kg / min}$$

Sensible heat load due to the infiltrated air,

$$Q_{s1} = m_a (h_A - h_4) = 27.56 (72.8 - 61) = 325.21 \text{ kJ/min}$$

$$= 325.21/60 = 5.42 \text{ kW}$$

and latent heat load due to the infiltrated air,

$$Q_{L1} = m_a (h_1 - h_A) = 27.56 (85 — 72.8) = 336.23 \text{ kJ/min}$$

$= 336.23/60 = 5.6$ kW

∴ Total room sensible heat load,

$$RSH = Q_{s4} + Q_{s1} = 46.5 + 5.42 = 51.92 \text{ kW}$$

and total room latent heat load

$$RLH = Q_{L4} + Q_{L1} = 17.5 + 5.6 = 23.1 \text{ kW}$$

We know that room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{51.92}{51.92 + 23.1} = 0.692$$

Now mark this calculated value of *RSHF* on the sensible heat factor scale and join with the alignment circle (i.e. 26°C *DBT* and 50% *RH*) as shown in Fig. 26. From point 4, draw a line 4-5 (known as RSHF line) parallel to this line. Since the outside air marked at point 1 is passed through the cooling coil whose *ADP* = 15°C, therefore join point 1 with *ADP* = 15°C on the saturation curve. This line is the *GSHF* line and intersects the RSHF line at point 2, which represents the condition of air leaving the cooling coil. Also 60% of the air from the hall is re-circulated and mixed with the conditioned air after the cooling coil. The mixing condition of air is shown at point 3 such that

$$\frac{\text{Length 2-3}}{\text{Length 2-4}} = 0.6$$

### *1. Condition of air after the coil and before the re-circulated air mixes with it*

The condition of air after the coil and before the re-circulated air mixes with it is shown by point 2 on the psychrometric chart, as shown in Fig. 26. At point 2, we find that

Dry bulb temperature, $t_{d2} = 19$°C Ans.

Wet bulb temperature, $t_{w2} = 17.5$°C Ans.

### *2. Condition of air entering the hall, i.e. after mixing with re-circulated air*

The condition of air entering the hall, i.e. after mixing with re-circulated air, is shown by point 3 on the psychrometric chart, as shown in Fig. 26. At point 3, we find that

Dry bulb temperature, $t_{d3} = 24$°C Ans.

Wet bulb temperature, $t_{w3} = 19.8$°C Ans.

### *3. Mass of fresh air entering the cooler*

The mass of fresh air passing through the cooling coil to take up the sensible and latent heat of the hall is given by

$$m_F = \frac{\text{Total heat removed}}{h_4 - h_2} = \frac{RSH + RLH}{h_4 - h_2}$$

$$= \frac{51.92 + 23.1}{61 - 49} = 6.25 \text{ kg/s} = 6.25 \times 60 = 375 \text{ kg / min Ans.}$$

... (From psychrometric chart, $h_2$ = 49 kJ / kg of dry air)

***4. By-pass factor of the cooling coil***

We know that by-pass factor of the cooling coil,

$$BPF = \frac{t_{d2} - ADP}{t_{d1} - ADP} = \frac{19 - 15}{38 - 15} = 0.174 \text{ Ans.}$$

***5. Refrigerating load on the cooling coil***

We know that the refrigerating load on the cooling coil

$= m_f (h_1 - h_2) = 375(85 — 49) = 13\ 500$ kJ/min

$= 13\ 500/210 = 64.3$ TR Ans.

**Example 15:** *The room sensible and latent heat loads for an air conditioned space are kW and 5 kW respectively. The room condition is 25°C dry bulb temperature and 50% relative humidity. The outdoor condition is 40°C dry bulb temperature and 50% relative humidity. The ventilation requirement is such that on mass flow rate basis 20% of fresh air is introduced and 80% of supply air is re-circulated. The by-pass factor of the cooling coil is 0.15.*

*Determine: I. supply airflow rate; 2. outside air sensible heat; 3. outside air latent heat; 4. grand total heat; and 5. effective room sensible heat factor.*

**Solution:** Given $RSH$ = 25kW; $RLH$ = 5kW; $t_{d2}$= 25°C; $\emptyset_2$ = 50%; $t_{dt}$ = 40°C; $\emptyset_1$ = 50%; $BPF$ = 0.15

The flow diagram for the air conditioned space is shown in Fig. 18.29 and it is represented on the psychrometric chart as discussed below:

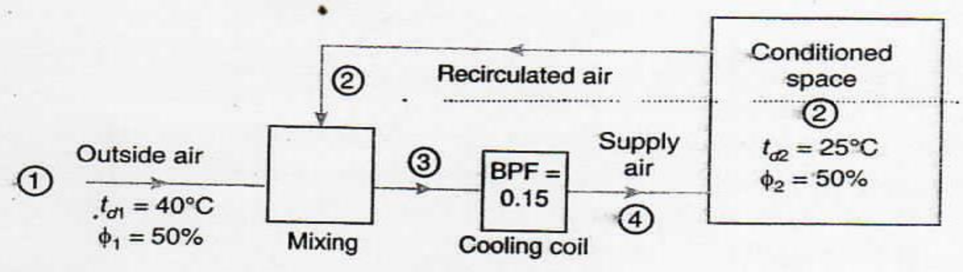


**Fig.27**

First of all, mark the initial condition of air at 40°C dry bulb temperature and 50% relative humidity on the psychrometric chart as point 1, as shown in Fig. 28. Now mark the room, condition of air at 25°C dry bulb temperature and 50% relative humidity as point 2. We know that i room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{25}{25 + 5} = 0.833$$

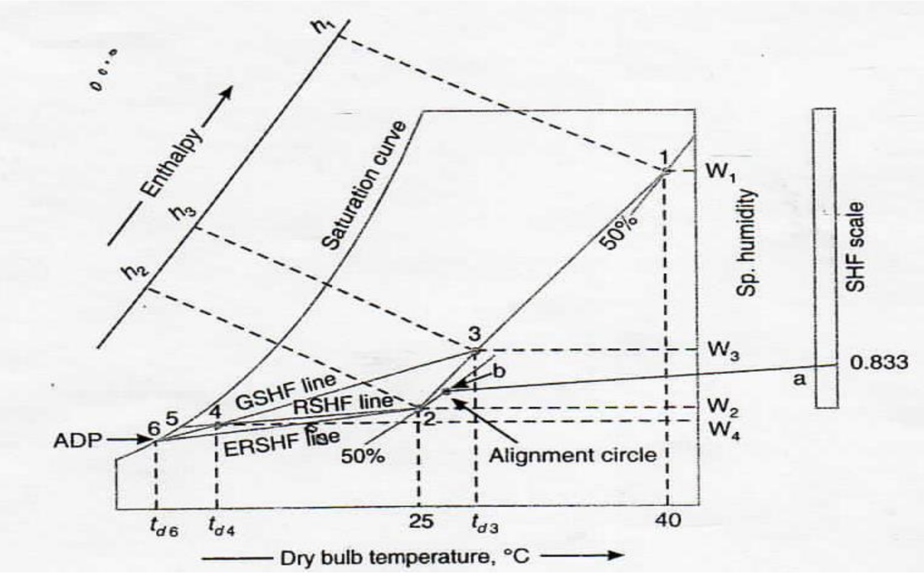


**Fig. 28**

Now mark this calculated value of *RSHF*=0.833 on the sensible heat factor scale as point *a* and join with point *b* which is the alignment circle (i.e. 26°C dry bulb temperature and 50% relative humidity). From point 2, draw a line 2-5 parallel to the line *ab*. The line 2-5 is called *RSHF* line. Since 20% of fresh or outside air is mixed with 80% of supply air, therefore the condition of air entering the cooling coil after mixing process is marked on the line 1-2 by point 3, such that

Length 2-3 = Length 1-2 x 0.2

Through point 3,-draw a line 3-6 (known as *GSHF* line) intersecting the *RSHF* line at point 4 and the saturation curve at point 6, such that

$$\frac{\text{Length 4-6}}{\text{Length 3-6}} = BPF = 0.15$$

***1. Supply air flow rate***

Let v = Supply air flow rate in m3/min,

$t_{d4}$ =Dry bulb temperature of air leaving the cooling coil, and

$t_{d6}$ = Apparatus dew point (ADP).

From the psychrometric chart, we find that dry bulb temperature of air entering the cooling coil at point 3,

$$t_{d3} = 28°\text{C}$$

We know that by-pass factor (BPF)

$$0.15 = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4} - t_{d6}}{28 - t_{d6}}$$

By trial and error, we find that

$$t_{d4} = 13.72°\text{ C and } t_{d6} = 11.2°\text{C}$$

We know that room sensible heat load,

$$RSH = 0.020\ 44\ v\ (t_{d2} - t_{d4})$$

$$25 = 0.020\ 44\ v\ (25 - 13.72) = 0.23\ v$$

$$v = 25 / 0.23 = 108.7 \text{ m3/min Ans.}$$

***2. Outside air sensible heat***

Since the outside air is 20% of the supply air, therefore outside air flow rate,

$$v_0 = 0.2\ v = 0.2 \times 108.7 = 21.74 \text{ m}^3\text{/min}$$

We know that outside air sensible heat,

$$\text{OASH} = 0.020\ 44\ v_0\ (t_{dt} - t_{d2})$$

$$= 0.020\ 44 \times 21.74\ (40 - 25) = 6.66 \text{ kW Ans.}$$

***3. Outside air latent heat***

From the psychrometric chart, we find that specific humidity of outside air at point 1,

$$W_1 = 0.0236 \text{ kg / kg of dry air}$$

and specific humidity of room air at point 2,

$$W_2 = 0.0098 \text{ kg / kg of dry air}$$

We know that outside air latent heat,

$$\text{OALH} = 50\ v_0\ (W_1 - W_2)$$

$$= 50 \times 21.74\ (0.0236 - 0.0098) = 15 \text{ kW Ans.}$$

4. Grand total heat

We know that total sensible heat,

$$TSH = RSH + OASH = 25 + 6.66 = 31.66 \text{ kW}$$

and total latent heat, $TLH = RLH + OALH = 5 + 15 = 20$ kW

$\therefore$ Grand total heat,

$$GTH = TSH + TLH = 31.66 + 20 = 51.66 \text{ kW Ans.}$$

Note: The total sensible heat (*TSH*) and total latent heat (*TLH*) may also be calculated as follows:

From psychrometric chart, we find that specific humidity at point 3,

$$W_3 = 0.0127 \text{ kg / kg of dry air}$$

and specific humidity at point 4.

$$W_4 = 0.009 \text{ kg / kg of dry air}$$

We know that total sensible heat,

$$TSH = 0.020\,44\, v\, (t_{d3} - t_{d4})$$

$$= 0.020\,44 \times 108.7\,(28 - 13.72) = 31.7 \text{ kW}$$

and total latent heat, $TLH = 50\, v\, (W_3 - W_4)$

$$= 50 \times 108.7\,(0.0127 - 0.009) = 20.1 \text{ kW}$$

**5. Effective room sensible heat factor**

We know that effective room sensible heat,

$$ERSH = RSH + OASH \times BPF = 25 + 6.66 \times 0.15 = 26 \text{ kW}$$

## 3.25 COMFORT CONDITIONS

### 3.26 Introduction

Strictly speaking, the human comfort depends upon physiological and psychological condition. Thus it is difficult to defang the term ‘human comfort’. There are many definitions given for this term by different bodies. But the most accepts definition, from the subject point of view, is given by the American Society of Heating, Refrigeration and air Conditioning Engineers (ASHRAE) which states : human comfort is that conditions of mind, which expressed satisfaction with the thermal environment

### 3.27 Thermal Exchanges of Body with Environment

The human body works best at a certain temperature, like any other machine, but it cannot tolerate wide range of variations in their environment temperatures like machines. The human body maintains its thermal equilibrium with the environment by means of three modes of hat transfer i.e evaporation, radiation and convection. The way in which the individual’s body maintains itself in comfortable equilibrium will be by its automatic use of one or more of the three models of heat transfer. A human body feels comfortable when the heat produced by metabolism of human body is equal to the of the heat dissipated to the surroundings and the heat stared inhuman body by rising the temperature of body tissues. This phenomenon may be represented by the following equation.

It may be noted that

1. The metabolic heat produced ($Q_{M)}$ depends upon the rate of food energy consumption in the body. A fasting, weak or sick man, will have less metabolic heat production.
2. The heat loss by evaporation is always positive. It depends upon the vapour pressure difference between the skin surface and the surrounding air. The heat loss of evaporation ($Q_{E)}$ is given by

$$Q_E = C_d\, A\, (Ps - Pv)\, h_{fg}\, C_C$$

$C_d$ = Diffusion coefficient in kg of water evaporated per Unit surface area and pressure difference per hour.

A = Skin surface area = 1.08 m2 for normal man,

Ps = Saturation vapour pressure corresponding to skin temperature

$P_v$ = Vapour pressure of surrounding air,

$H_{f8}$ = Latent heat of vaporization = 2450 kJ/kg,

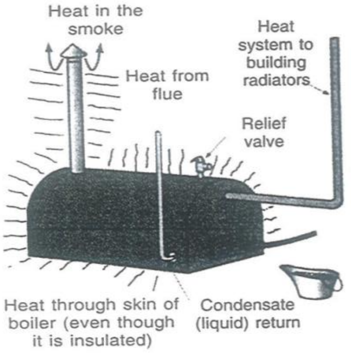
$C_c$ = Factor which accounting for clothing worn.

The value of $Q_E$ becomes zero when Ps = Pc, i.e when the surrounding air temperature is equal to the skin temperature and air is saturated or when it is higher than the erature and the air is nearly saturated.

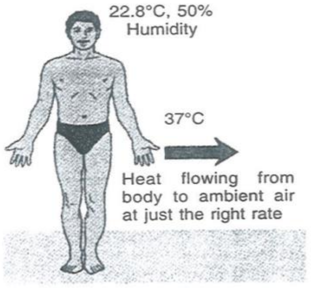
- The skin tempplus sing is used when heat is lost to the surrounding and negative sing is used when heat is gained from surroundings.
- The plus sign is used when the temperature of the body rises and negative sign is used when the temperature of the body falls.

The value of $Q_e$ is never negative as when $p_s$ is less than $p_v$, the skin will not absorb moisture from the surrounding air as it is in sutures state. The only way for equalizing the pressure difference is by increasing $p_s$ to p v by rise of skin temperature from the sensible heat flow from air skin.

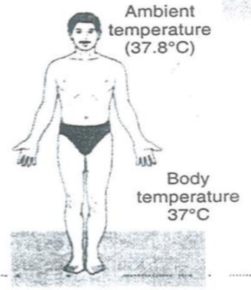
3. The heat loss or gain by radiation ($Q_R$) from the body to the surroundings depends upon the mean radiant temperature. It is the average surface temperature of the surrounding object when properly weighted, and varies from place to place inside the room. When the mean radiant temperature is lower the day bulb temperature of air in the room, $Q_R$ is positive i.e the body will undergo a radiant heat loss. On the other hand, if the mean radiant temperature is higher than will undergo radiant heat loss. On the other hand, if the mean radiant temperature is higher than the dry bulb temperature of air in the room, $Q_R$ is negative i.e the body will undergo a radiant heat gain.



The boiler can be compared to the human body in that it uses fuel for energy.

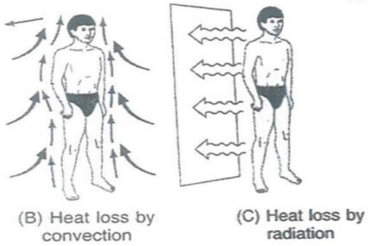


The body's normal temperature is 37°C



The human body must give off some of its generated heat to the surroundings.

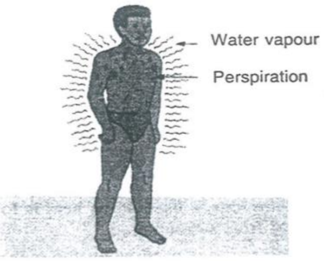
(A) Heat loss by conduction



(B) Heat loss by convection

(C) Heat loss by radiation

The three direct ways the human body gives off heat (A) conduction (B) convection (C) radiation.



The evaporation of respiration.

4. The heat loss by convection (QC) from the body to the surroundings is given by where

$Q_C = UA\ (t_B - t_s)$

U = Body film coefficient of heat transfer,
A = Body surface area = 1.8 m2 for normal man,
$T_b$ = Temperature of the body, and
Ts = Temperature of the surroundings.

When the temperature of the surroundings ($t_s$) is higher than the temperature of the body ($t_B$), then QC will be negative, i.e the heat will be gained by the body. On the other hand if the temperature of the surroundings ($t_S$) is lower than the temperature of the body (TB), then QC will be the positive, i,e the heat will be lost by the body. Since the body film coefficient of heat transfer increases with the increase in air velocity, therefore higher air velocities will produce uncomforted t when Ts is higher than $t_B$. The higher air velocities are recommended when $t_S$ is lower than $t_B$.

5. When $Q_E$, $Q_R$ and $Q_C$ are high and positive and ($Q_E + Q_R + Q_C$) is greater than ($Q_M$-W), the heat stored in the body (Qs) will be negative i.e the body temperature falls sown. Thus the sick, weak, old or a fasting man feels colder. On the other a man gets fever when high internal body actives increases $Q_M$ to such extent so that $Q_S$ becomes positive for the given $Q_E$, $Q_R$ and $Q_C$.

   The heat stored in the body has maximum and minimum limits which when exceed bring death. The usual body temperature, for a normal man (when Qs =0) is 37°C (98.6°F). the temperature of the body temperature of the body when falls below 36.5°C (98°F) and exceeds 40.5°C (105°F) is dangerous. There is some kind of thematic control called vasomotor control mechanism in the human body which maintains the temperature of body at the normal level of 37°C, by regulating the blood supply to the skin. When the temperature of the body falls (i.e the heal stored QS in the body is negative), then the vasomotor control decreases the circulation of blood which decreases conductivity of never cells and other tissues between the skin and the inner body cells. This allows temperature to fall but allows higher inner temperature of body cells beneath. When the temperature of the body rises (i.e the heat stared Qs in the body is positive), then the vasomotor control increases blood circulation which increases conductivity of tissues and hence allows less temperature drop between the skin and inner body cell.

   The human body feels comfortable when there is no changes in the body temperature, i.e when the heat stored in the body Qs is zero. Any variation in the body temperature acts as a stress to the brain which ultimately results in either perspiration or shivering.

### 3.28 Physiological Hazards Resulting from Heat

In summer the temperature of the surrounding is always higher than temperature of the body. Thus the body will gin heat from the surrounding by means of radiation and convection processes. The body can dissipate eat only through evaporation of sweat. When the health loss by evaporation in unable to cope with the heat gain, there will bestrgae of heat in the body and the temperature of body rises. Several physiological hazards exist, the severity of which depends upon the extent and time duration of body temperature rise. Following are some of the physiological hazards which may result due to the rise in body temperature.

1. Heat exhaustion. It is due to the failure of normal blood circulation. The symptoms of heat exhaustion ileuses fatigue, headache, dizziness, vomiting and abnormal mental reaction such as irritability, severe heat exhaustion may close fainting. It does not cause permanent injury to the body and recovery is usually rapid when the person is removed to a cool places.
2. Heat cramp. It results from loss of salt due to an excessive rate of body perspiration. It cases sever pain in the calf and thing muscles. The heat cramp may be largely avoided by using salt tablets.
3. Heat stroke. It is most serious hazarded, when man is exposed to excessive heat and work, the body temperature may rise rapidly to 40.5°C (105°F) or higher. At such elevated temperatures, sweating cease and the man may enter a soma, with death imminent. A person experiencing a hat stroke may have permanent damage to the brain. The heat stroke may be avoided by taking sufficient water at frequent intervals. It has been found that man doing hard work in the sun requires one liter of water per hour.

### 3.29 Factors Affecting Human Comfort

In designing winter or summer air conditioning system, the designer should be well conversant with a number of factors which physiologically affect human comfort. The important farces are as follows:

1. Effective temperature, 2. Heat production and regulation in human body 3. Heat and moister losses from the human body, 4. Moisture content of air, 5. Quality and quantity of air. 6. Air motion, 7. Hot and cold surfaces and 8. Air stratification

   These factors are discussed, in detail, in the following articles:

**Effective Temperature** The degree of warmth or cold felt by a human body depends mainly on the following three factors:

1. Dry bulb temperature, 2. Relative humidity and 3. Air velocity.
   In order to evaluate the combined effect of these factors, the effective temperature is employed. It is defined as that index which collates the com binned effects of air temperature, relative humidity and air velocity on the human body. The numerical value of effective temperate is made equal to the temperature of stills (i.e 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or clones as produced under the given conditions.

The practical application of the concept of effective temperature is presented by the comfort chart, as shown in Fig 171. This chart is the result of research made on different kinds of people subjected to wide range of environmental temperature, relative humidity and air movement by the American Society of Heating, Refrigeration and Air conditioning Engineers (ASHRAE). It is appreciable to reasonably still air (5 to 8 m/min air velocity) to situations where the occupants are seated at rest or doing light work and to space whose enclosing surfaces are at a mean temperature equal to the air dry bulb temperature.

In the comfort chart, as shown in Fig.17.1, the dry bulb temperature is taken as abscissa and the wet bulb temperature of ordinates. The relative humidity lines are repotted from the psychometric chart. The statistically prepared graphs corresponding to summer and winter season are also superimposed. These graphs have effective temperature scale as abscissa and % of people feeling comfortable as ordinate.

A close study of the chart reveals that the several combination of wet and dry bulb temperatures with different relative humidity will produces the same effective temperature.

However, all points located on a given effective temperature line do not indicate conditions of equal comfort or discomfort. The extremely high or low relative humidifies may process conditions of discomfort repulses or existent effective temperate. The moist deniable relive humidly rages lies between 30 and 70 per cent. When the relative humidity is much below 30 per cent, the mucous membranes and the skin surface become too dry for comfort and health. On the other hand, if the relative humidity is above 70 per cent, there is a tendency for a clammy or sticky sensation to develop. The curves at the top and bottom, as shown in Fig. 17.1, indicate the parentages of person participating in tests, who found various effective temperatures satisfactory for comfort.

The comfort chart shows the range for both summer and winter condition within which a condition of comfort exists for most people. For summer conditions, the chart indicates that a maximum of 98 percent people felt comfortable for an effective temperature of 21.6°C. For winter conditions, chart indicates that an effective temperature of 20°C was desired by 97.7 percent people. It has been found that comfort; women require 0.5°C higher effect give temperature than men. All men and women above 40 years of age preset 0.5°Chiger effective temperature than the person below 40 years of age.

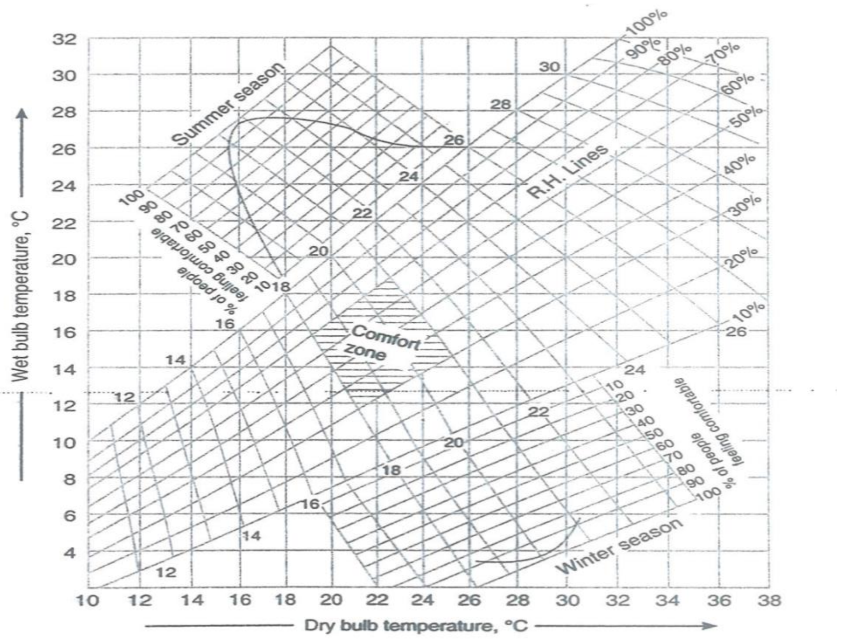


Fig. 17.1. Comfort chart for still air (air velocities from 5 to 8 m/min)

It may be noted that the comfort chart, as shown in \fig.17.1, does not take into account the carnations in comfort conditions when there are wide variations in the mean radiant temperature (MRT). In the range of 26.5°C, a rise of 0.5°C in mean radiant temperature above the room dry bulb temperature raises the effete temperature by 0.5°C. The effect of mean radiant temperature on comfort is less pronounced at high temperatures than at low temperatures.

The comfort conditions for persons at work vary with the rate of work and the amount of clothing worn. In general, the greater the dredge activity, the lower the effective temperature necessary for comfort.

Fig.17.2 shows the variation in effective temperature with different air velocities. We see that for the atmospheric conditions of 24°C dry bulb temperature and16°C wet bulb temperature correspond to about 21°C with nominally still air ( velocity 6m/min) and it is about 17°C at an air velocity of 210 m/min. the same effective temperature is observed at higher dry bulb and wet bulb temperatures it higher velocities. The case is reversed after 37.8°C as in that case higher velocities will increases sensible heat flow from air to body and will decrease comfort. The same effective temlperarue means same feeling of warmth, but it does not means same comfort.

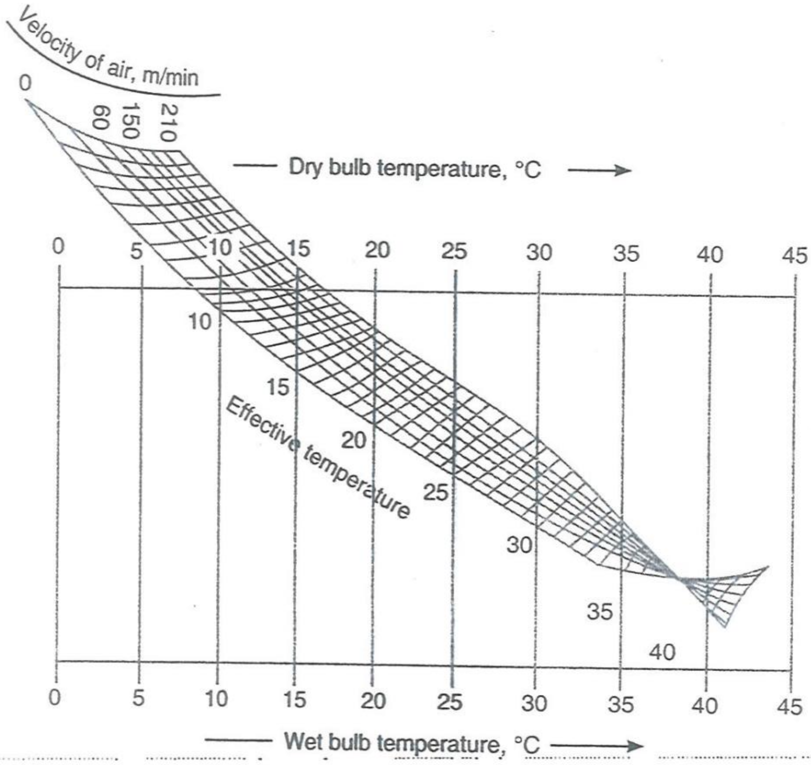


Fig. 17.2. Variation of effective temperature with air velocity.

## 3.30 Modified Comfort Chart

The comfort chart, as shown in Fig.17.1 has become obsolete now-a-days due to its short comings of over exaggeration of humidity at lower temlperarue and under estimation of humidity at heat to trance level. The modified comfort char according to ASHRAE is shown in Fig.17.3 and it is commonly use these days. This chart was developed on the basis or research done in 1963 by then stirred fro environmental research at Kansas State University. The mean radiant temperature was kept equal to dry bulb temperature and air velocity was less than 0.17 m/s.

## 17.7 Heat Production and Regulation in Human Body

The human body acts like a heat entice which gets its energy from the combustion of food within the body. The process of combustion (called metabolism) produces het and energy due to the oxidation of redacts in the body by oxygen obtained from in held air. The rate of heat production depends upon the individual's health, his physical activity and his environment. The rate at which the body produces heat is metabolic rate. The heat production from a normal healthy person when a sleep (called based metabolic rate) is about 60 wtts and it is about ten times more for a person carrying out sustained very hard work.

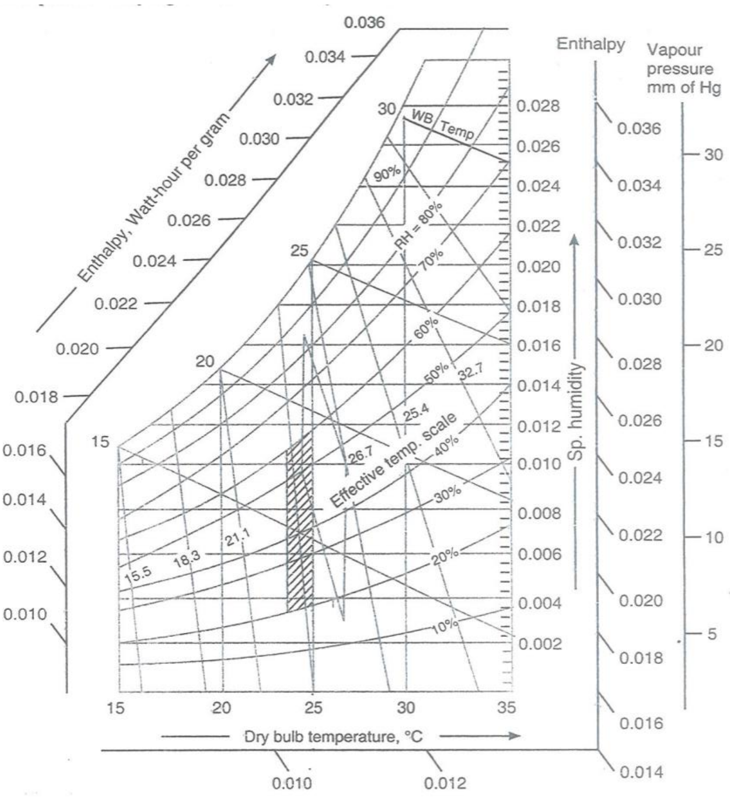


Fig. 17.3. Modified comfort chart.

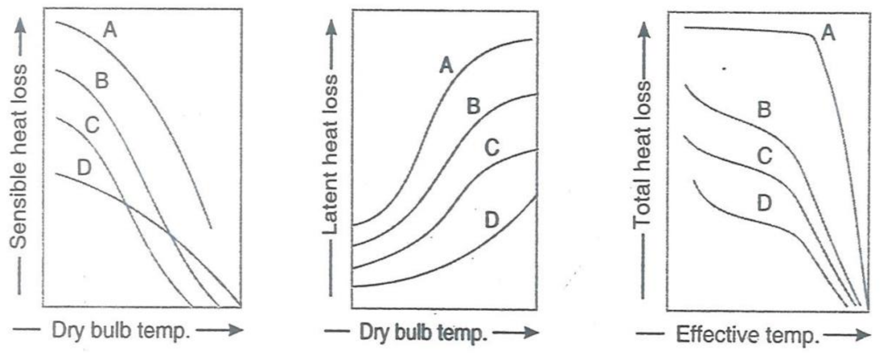
Since the body has a thermal efficiency of 20 per cent, therefore the remaining 80 per cent of the heat must be rejected to the surrounding environment, otherwise accumulation of heat result which causes discomfort. The rate and the manner of rejection of heat is controlled by the automatic regulation system of a human body.

In order to effect the loss of heat from the body to race cold, the body may react to bring more blood to the capillaries in the skin. The heat looses from the skin, now, may take place by radiation, convection and by evaporation. When the process of radiation or convection or both fails process necessary loss of heat, the sweat glands become more active and ore moister is debited on the kin, carrying heat always as it evaporates. It may be noted that when the temperature of surrounding air and objects is below the blood temperature, the heat is removed by rendition and convection. On the other hand, when the temperature of surrounding air is above the blood temperature, the heat is removed by evaporation only. In case the body fails to throw off the requisite amount of heat, the blood temperature rises. This results in the accumulation of heat which will cause discomfort.

The human body attempts to maintain its temperatures when exposed to cold by the with drawl of blood from the outer portions of the skin, by decreased blood circulation and by an increased rate of metabolism.

### 3.31 Heat and Moisture Losses from the Human Body

This heat is given off from the human body as either sensible or latent heat or both. In order to design any air-conditioning system for spaces which human bodies are to occupy, it is necessary to know the rates at which these two forms of heat are given off under different conditions of air temperature and bodily activity.



(*a*) Relation between sensible heat loss from the human body and dry bulb temperature for still air.

(*b*) Relation between latent heat loss from the human body and dry bulb temperature for still air.

(*c*) Relation between total heat loss from the human body and effective temperature for still air.

**Fig. 17.4**

Fig.17.4(a) shows the graph between sensible heat loss by radiation and convection for an average man and the dry bulb temperate for different types of acidity. Fig.17.4 (b) shows the graph between the latent heat loss by evaporation for an average man and dry bulb temperature for different types activity.

The total heat loss from the human body under varying effective temperatures is show in Fig.17.4(c). From curve D, which apples to men at rest, we see that from about 19°C to 30°C effective temperature, the heat loss inconstant. At the lower effective temperature, the heat dissipation increases which results in a feeling of coolness. At higher effective temperature, the ability to lose heat rapidly decreases resulting in sever discomfort. The curves A, B, C and D show in Fig.17.4 represents as follows:

Curve A – Men working at the rate of 90 kN-m/h

Curve B – Men working at the rate of 45 kN-m/h

Curve C – Men working at the rate of 22.5 kN-m/h

Curve D -- Men at rest.

### 3.32 Moisture Content of Air

The dry bulb temperature, relative humidity and air motion are inter-related. The moisture content of outside air during winter is generally low and it is above the average during simmer, because the capacity of the air to carry moisture is dependent upon its dry bulb temperature. This means that in winter, if the cold outside air having a low mist rue content leaks into the conditioned space, it will cause a low relative humidity unless minster is assed to the air by the processes of humidification. In summer, the reveres will take place unless moisture is removed from the inside air by the dehumidification process. Thus, while designing an air-conditioning system, the proper dry bulb temperature for either summer or winter must be selected in accordance with the practical consideration of relative humidity's which are feasible. In general, for winter conditions in the average residence, relative humidity above 35 to 40 per cent are not practical. In summer comfort cooling, the air of he occupied space should not have a relative humidity above 0 per cent. With theses limitations the necessary dry bulb temperature for the air may be depend from the comfort chart.

### 3.33 Quality and Quantity of Air

The air in an occupied space should, at all times, be free room toxic, unhealthful or disagreeable fumes such as carbon dioxide. It should also be free from dust and odour. In order to obtain these conditions, enough clean outside in must allows be supplied to an occupied space to counteract or adequately dilute the sources of contamination.

The concentration of odour in a room depends upon many factors such as directory and hygienic habits of occupant, type and amount f outdoor air supplied, room volume per occupant and types of odour sources. In general. When here is no smoking in a room, 1m3/min per person of outside air will take care of all the conditions. But when smoking takes place in a room,1.5 m3/min per person of outside air is necessary. In most air-conditioning systems, alarge amount of air is reticulated over and above the required amount of outside air to satisfy the minimum ventilation conditions in regard to odour and purity. For general application, a minimum of 0.3 m3/min of outside air pr person, mixed with 0.6m3/min of recalculated air is good. The recommended and minimum values for the outside air require per person are given in Chapter 19 on cooling load estimation

### 3.34 Air Motion

The air motion which included the distribution of air is very important to maintain uniform temperate in the conditioned space. No air conditioning system is satisfactory unless the air handled is properly circulated and distributed. Ordinarily, the air velocity in the occupied zone should not exceed 8 to 12m/min. The air velocities in the space above the occupied zone should be very high in order to produce goof distribution of air in the occupied zone, provided that the air in motion does not produce any objectionable noise. The flow of air should be preferably towards the faces of the indicial rather than form the rear in the occupied zone. Also for the proper and perfect distribution of air the air-conditioned space, down flow should be preferred instead of up flow.

The air motion without proper air distribution produces local cooling sensation known as draft.

### 3.35 Cold and Hot Surfaces

The cold or hot objects in a conditioned space may cause discomfort to the occupants. A single glass of large area when exposed to the outdoor air during winter will produce Discomfort

The atmospheric air contains 0.03% to 0.04% by volume of carbon dioxide and it should not increase 0.6% which is necessary for proper functioning of respiratory system. The carbon dioxide, in excess of 2% dilutes oxygen contents and makes breathing difficult. When the carbon dioxide exceeds 6%, breathing is very difficult and 10% carbon dioxide causes loss of consciousness. A normal man at rest in breathing exhales about 0.015 to 0.018 m3/h of carbon dioxide to the occupants of a room by absorbing heat from them by radiation. On the other hand, a ceiling that is warmer than the room air during summer cusses discomfort. Thus, in the designing of an air conditioning system, the temperature of the surfaces to which the body may be exposed must be given considerable

### 3.36 Air Stratification

When air is heated, its density decreases and thus it rises to the upper part of the confined space. This results in a considerable variation in the temperatures between the floor and ceiling levels. The movement of the air to produce the temperature gradient from floor to ceiling is termed as air stratification. In order to achieve comfortable conditions in the occupied space, the air conditioning system must be designed to reduce the air stratification to a minimum.

### 3.37 Factors Affecting Optimum Effective Temperature

The important factors which affect the optimum effective temperature are as follows:

1. Climatic and seasonal differences. It is a known fact that the people living in colder climates feel comfortable at lower effective temperatures than those living in warmer religions. There is a relationship between the optimum indoor effective temperature

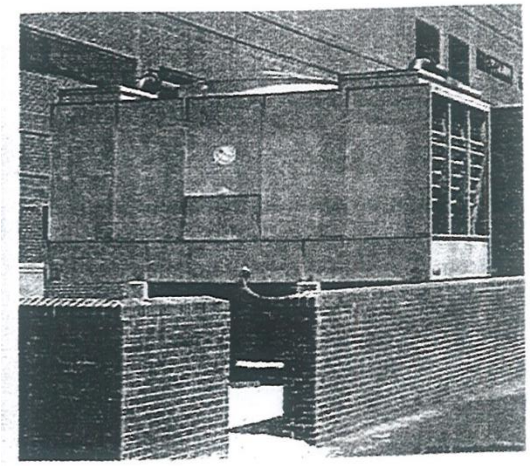
and the optimum outdoor temperature, which changes with seasons. We see from the comfort chart (Fig.171.1) that in winter the optimum effective temperature is 19°C whereas in summer this temperature is 22°C.

2. **Clothing:** It is another important factor which affects the optimum effective temperature. It may be noted that the person with light clothing need less optimum temperature than a person with heavy clothing.
3. **Age and sex**. We have already discussed that the women of all ages require high reflective temperature (about 0.5°C) than men, similar is the case with young and old people. The children also need higher effective temperature than adults. Thus, the maternity halls are always kept at an effective temperature of 2 to 3°C higher than the effective temperature used for adults.
4. **Duration of stay.** It has been established that if the stay in a room is shorter (as in the case of persons going to banks), then higher effective temperature is required than that needed for long stay (as in the case of persons working in an office).
5. **Kind of activity.** When the activity of the person is heavy such as people working in a factory, dancing hall, then low effective temperature is need than the people sitting in cinema hall or auditorium**.**
6. **Density of Occupants.** The effect of body radiant heat from person to person particularly in a densely occupied space like auditorium is large enough which rewire alight lower effective temperature.

# UNIT IV

# COOLING LOAD AND DESGN OF AR-CONDITIONING SYSTEMS

## Introduction



The total heat required to be removed from the space in order to bring it at the desired temperature by the air conditioning and refrigeration equipment is known as cooling load.

The purpose of a load estimation is to determine the size of the air conditioning and refrigeration equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The design load is based on inside and outside design conditions and it is air conditioning and refrigeration equipment capacity to produce and maintain satisfactory inside conditions.

19.2 Components of a Cooling Load

The two main components of a cooling load imposed on an air conditioning plant operating during hot weather are as follows:

1. Sensible heat gain. When there is a direct addition of heat to the enclosed space, a gain in the sensible heat is said to occur. This sensible heat is to be removed during the process of summer air conditioning. The sensible heat gain may occur due to any one or all of the following sources of heat transfer:

   (a) The heat flowing into the building by conduction through exterior walls, floors, ceilings, doors and windows due to the temperature difference on their two sides.

   (b) The heat received from solar radiation. It consists of

   (i) The heat transmitted directly through glass of windows, ventilators or doors, and

(ii) The heat absorbed by walls and roofs exposed to solar radiation and later on transferred to the room by conduction.

(c) The heat conducted through interior partition from rooms in the same building which are not conditioned.

(d) The heat given off by lights, motors, machinery, cooking operations, industrial processes etc.

(e) The heat liberated by the occupants.

(f) The heat carried by the outside air which leaks in (infiltrating air) through the cracks in doors, windows, and through their frequent openings.

(g) The heat gain through the walls of ducts carrying conditioned air through unconditioned space in the building

(h) The heat gain from the fan work.

2. Latent heat gain. When there is an addition of water vapour to the air of enclosed space, a gain in latent heat is said to occur. This latent heat is to be removed during the process of summer air-conditioning. The latent heat gain may occur due to any one or all of the following sources:

(a) The heat gain due to moisture in the outside air entering by infiltration.
(b) The heat gain due to condensation of moisture from occupants.
(c) The heat gain due to condensation of moisture from any process such as cooking foods which takes place within the conditioned space.
(d) The heat gain due to moisture passing directly into the conditioned space through permeable walls or partitions from the outside or from adjoining regions where the water vapour pressure is higher.

The total heat load to be removed by the air-conditioning and refrigeration equipment is the sum of sensible and latent heat loads as discussed above.

Note: When the outside air is introduced for ventilation purposes, there is a sensible heat gain as well as latent heat gain. The sensible heat gain is due to the temperature difference between the fresh air and the air in space whereas the latent heat gain is due to the difference of humidity.

**Sensible Heat Gain through Building Structure by conduction**

The heat gain through a building structure such as walls, floors, ceilings, doors and windows constitutes the major portion of sensible heat load.

Consider a building wall composed of a single homogeneous material as shown in Fig. 19.1. A little consideration will show that the heat passing through a wall is first received at the wall surface exposed to the region of higher air temperature by radiation, convection and conduction. It then flows through the material of the wall to the surface exposed at the region of lower air temperature. Now the heat is dispersedthrough the processes of radiation. Convenction and conduction. Thus, the heat transferred or gained through a wall under steady state condition is

Where $Q=f_0(t_0-t_1)A+\frac{k}{x}(t_1-t_2)A+f_1(t_2-t_1)|A$

$=UA(t_0-t_i)$

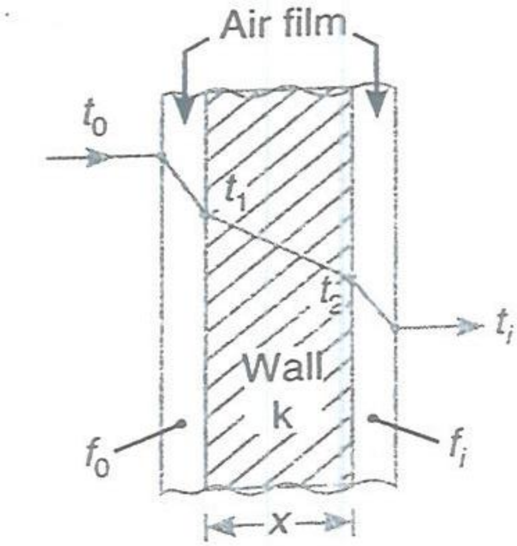


Fig. 19.1. Heat transfer through a single wall.

Where

$f_0$ =Outside film or surface conductance,

f1= Inside film or surface conductance,

A= Outside area of wall,

$t_o$= Outside air temperature,

$t_i$ =Inside air temperature,

x=Thickness of wall,

k=Thermal conductivity for the material of the wall, and

U=Overall coefficient of heat transmission of the wall

$$=\frac{1}{\frac{1}{f_o}+\frac{x}{k}+\frac{1}{f_i}}$$

When a wall is made up of layers of different materials as shown in Fig. 19.2, then the overall coefficient of heat transmission is given by

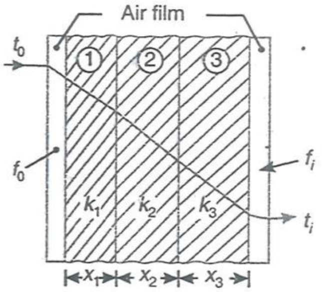


Fig. 19.2. Heat transfer through a composite wall.

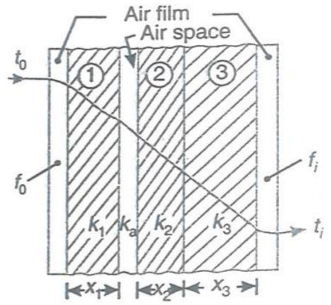


Fig. 19.3. Heat transfer through a composite wall with air space.

These types of walls are made for a cold storage with a maximum low temperature insulation of 10 cm.

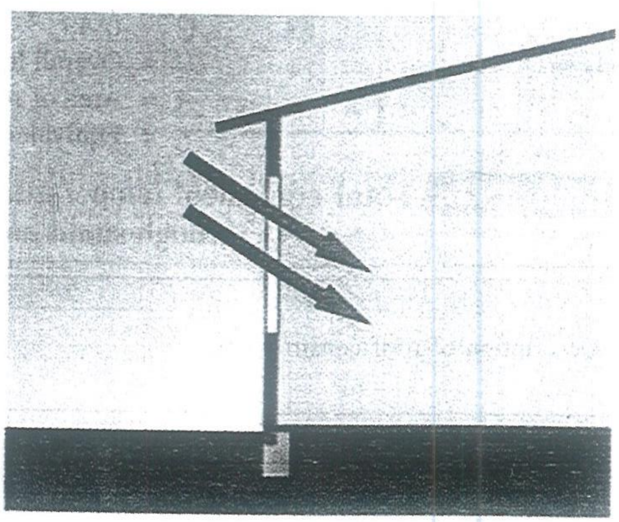
When air space is provided between the materials as shown in fig. 19.3., then the overall coefficient of heat transmission is given by

In case of interior walls or partitions, floors and ceilings, the temperature in the adjoining unconditioned space is usually 5 to 10 degree below the outdoor design temperatures. But for kitchens, boiler rooms, attics and show windows, it is often higher than the outdoor temperature.

The values of overall coefficient of heat transmission for structures, thermal conductivity of building materials and insulating materials, conductance of building materials and construction and film or surface conductance for air film and air spaces, are given in the following tables.

## 19.4 heat Gain from solar Radiation

The solar radiation striking the outside surfaces of a building may contribute appreciable to the peak load on the air conditioning and refrigeration equipment and must, therefore, be considered. The amount of heat that flows towards the interior of a building due to solar radiation depends upon the following factors:



Solar heat gain coefficient is the fraction of incident solar radiation admitted through a window.

1. Altitude angle of the sun,
2. Clearness of the sky,
3. Position of the surface with respect to the direction of the sun's rays,
4. Absorptivity of the surface,
5. Ratio of the overall coefficient of heat transfer of the wall to the coefficient of heat transfer of the outside air film and
6. Temperature of the ground and surrounding objects with which the heated surface may interchange radiant heat.

The altitude angle of the sun, in turn depends upon the altitude of the locality, season of year and hour of day.

The heat from solar radiation is received by building surfaces in two forms i.e., direct radiation and sky or diffuse radiation. The direct radiation is the impingement of the sun's rays upon the surface. The sky or diffuse radiation is received from moisture and dust particles in atmosphere which absorbs part of the energy of the sun's rays thereby becoming heated to a temperature above that of the air. The sky radiation is received by surfaces which do not face the sun.

### 19.5 Solar Heat gain (sensible through outside walls and Roofs)

The transmission of heat through the walls exposed to the outdoors and roof is not steady (i.e. the flow of heat is periodic) due to variation in the outside air temperature and the solar radiation intensity over a period of 24 hours. A little consideration will show that the temperature of wall rises with the rise in outside air temperature and the heat is stored in the wall which has a considerable storage capacity. Thus the heat transferred to the room is reduced. The stored heat in The wall is given off to the room when the outside air temperature falls. Since the outside air temperature changes continuously over a cycle of 24 hours, therefore instantaneous heat gain from outside is not equal to the instantaneous heat gain inside the room, the difference being stored or rejected by the wall. The heat stored by the wall is given off later in the evening. Thus peak of incoming heat rate is delayed by the storage effect of the walls and it is also reduced. Fig. 19.4 shows the curves of instantaneous load coming from outside and the actual load felt inside. The area under the two curves is equal. The shaded area above the actual load shows the heat stored and below the actual load shows the heat released by the walls and other structures.

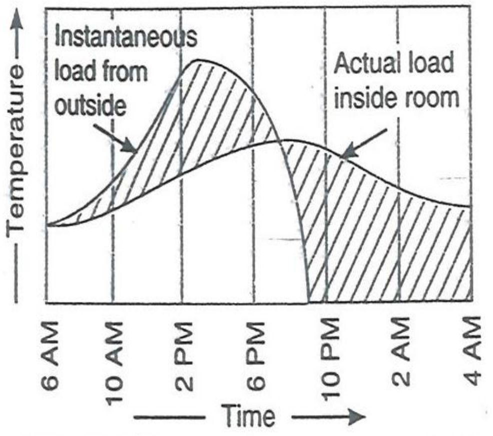


Fig. 19.4. Curves of instantaneous load.

A convenient method of taking into account this lagging effect of storage and the solar radiation is to use an equivalent temperature differential. The Equivalen temperature differentials for roofs are given in Table 19.6, and those applying to walls is given in Table 19.7. the heat gain through outside walls and roofs is given by

$Q = UAt_e$

U=Overall heat transmission coefficient of roof or wall,

A=Area of roof or wall, and

Where $t_e$ =Equivalent temperature differential.

### 19.6 Sol Air Temperature

It is a hypothetical temperature used to calculate the heat received by the outside surface of a building wall by the combined effect of convection and radiation. The heat received by the outside surface of the wall by convection is given by

Where $q_c=f_o(t_o-t_{os})$

fo = Outside film coefficient.

$t_o$ = Temperature of outside air, and

$t_{os}$= Temperature of the outside surface of the wall.

The heat received by the outside surface of the wall by radiation is given by

1. $q_r = I\alpha$

Where,

I=Total radiation intensity, and

α= Absorptivity of the surface.

· Total heat

Received by the outside surface of the wall, Where

This temperature $t_e$ is known as sol air temperature.

## 19.7 Solar Heat Gain Through Glass Areas

The heat gain through the glass areas constitutes a major portion of the load on the cooling apparatus. When a sheet of glass is subjected to solar radiation (direct and diffuse), a part of it is absorbed, a part is reflected and the remainder is transmitted directly to the interior of the building.

In the absorption process, the temperature of the glass increases until it is in a position to lose heat at the same rate in an interchange of energy with surfaces inside and outside of the building. A simultaneous interchange of radiant energy takes place between the exterior surface of the glass and the heated particles in the atmosphere and between the interior surface and the various objects in the room. In addition to the radiant effects, the net heat gain into the interior of a building through a sheet of glass is affected by convection air currents on both sides. If the temperature of the glass is higher than that of the out-door air, there will be no gain by conduction in spite of the fact that the outdoor air is warmer than that of inside. In that case, a portion of the radiation absorbed by the glass will be removed by inside air currents, another portion by outside air currents and the remainder by the net loss in interchanges of radient energy between the glass and all of the surfaces it can see. The complete heat balance can be written as follows:

Net heat gain = Transmitted solar radiation + Heat flow by convection and radiation heat exchanges between glass and indoor surfaces.

Neglecting the heat which may be stored in the glass or which may be given up by the glass under changing conditions, the second term of the above expression may be written as follows:

Heat flow by convection and radiation heat exchanges between glass and indoor surfaces

= Absorbed solar radiation ± Radiation and convection heat exchanges, between glass and outdoor surfaces

Notes: 1. The solar heat gain through glass areas varies from hour to hour, from day to day and from latitude to latitude. For complete details of the solar radiation with respect to time of day and situation of glass areas, the guide published by American Society of Heating and ventilating Engineers (ASHVE) may be referred.

2. The values of solar heat gain are usually given in examination problems.

## 19.8 Heat Gaub due to Infiltration

The infiltration air is the air that enters a conditioned space through window cracks and opening of doors. This is caused by pressure difference on the two sides of the windows and doors and it depends upon the wind velocity and its direction and difference in densities due to the temperature difference between the inside and outside air.

There are two methods of estimation the infiltrated air:

1. Crack length method, and 2. Air change method.

The crack length method is usually used where greater accuracy is required. In most cases, the air change method is used for calculating the quantity of infiltrated air. According to this method, the amount of infiltrated air through windows and walls is

The total room infiltration air for an entire building is taken building is taken one-half of the above calculated value because infiltration takes place on the windward side of a building.

The following table shows the number of air changes per hour for a variety of room types and exposures.

For each person passing through a door leading to the outside or to an unconditioned space, the values given in the following table for the door infiltration should be added to the infiltration air through windows and walls, in order to find the total building infiltration.

Notes: 1. For rooms with weather stripped windows or storm cash, use ½ of the values given in Table 19.8 but it should never be less than 0.5.

2. The figures in Table 19.9 are based on the assumption that there is no wind pressure and the swinging doors are in use in one wall only. Any swinging door in other walls should be kept closed to ensure air conditioning in accordance with the recommended standards.
3. The door infiltration for a 0.9m swinging door may be taken as 3 $m^3$/min.

## 19.9 Heat Gain due to ventilation

The ventilation (i.e., supply of outside air) is provided to the conditioned space in order to minimize odour, concentration of smoke, carbon dioxide and other undesirable gases so that freshness of air could be maintained. The quantity of outside air used for ventilation should provide at least one-half air change per hour in buildings with normal ceiling heights. Also, if the infiltration air quantity is larger than the ventilation quantity, then the latter should be increased to at least equal to the infiltration air. The outside air adds sensible as well as latent heat.

The following table shows the recommended and minimum values for the outside air required per person in the building.

## 19.10 Heat Gain from Occupants

The human body in a cooled space constitutes cooling load of sensible heat and latent heat.

**Notes:**

1. The heat gain from occupants is based on the average number of people that are expected to be present in the conditioned space. The heat load produced by each person depends upon the activity of the person. Table 19.11 Shows a wide range of activities from the rest position to one of heavy work.
2. The adjusted total heat gain is based on normal percentage of men, women and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male and that that gain from a child is 75 percent of that for an adult male.
3. The adjusted total heat value for sedentary work, restaurant includes 17.5 W for food per individual (8.75 W sensible and 8.75 W latent).
4. For bowling, figure one person per alley actually bowling and all other as sitting (117 W) and standing (161 W).

## 19.11 Heat Gain from Appliances

The appliances frequently used in air conditioned spaces may be electrical, gas-fired or steam heated. Following table gives most of the commonly used appliances together with approximate values of sensible heat and latent heat.

## 19.12 Heat Gain from Products

The heat emitted from the products to be stored is very important in case of cold storages.

The loads to be considered in the cold storages .

1. Chilling load above freezing. The product chilling load above freezing depends upon the mass of product (m), mean specific heat of the product ($C_{pm}$), entering product temperature ($T_1$), final product temperature desired ($T_2$) and the chilling time ($t_{ch}$). Mathematically, chilling load above freezing,

$Q_{ch}$ =

2. Freezing load. The freezing load depends upon the mass of the product (m), its latent heat of freezing ($h_{fs}$) and the freezing time ($t_F$). Mathematically, freezing load,

3. Cooling load below freezing. The cooling load below freezing depends upon the mass of the product(m), mean specific heat of the freezing product ($c'_{pm}$), actual storage temperature of the product ($T_1'$), desired freezing temperature of the product ($T_2'$), and the cooling time ($t_c$), Mathematically, cooling load below freezing,

4. Product reaction or respiration heat. During the maturing of some food products, even in cold storage, reaction or respiration heat is evolved. This heat gain is given by

The rate of heat production for different types of food is given in the following table.

19.13 Heat Gain from Lighting Equipment's

The heat gained from electric lights depends upon the rating of lights in watts, use factor and allowance factor. Mathematically, the heat gained from electric lights is given by

The use factor is the ratio of actual wattage in use to the installed wattage. Its value depends upon the type of use to which the room is put. In case of residences, commercial stores and shops, its value is usually taken as unity, whereas for industrial workshops it is taken below 0.5.

The allowance factor is generally used in case of fluorescent tubes to allow for the power used by the ballase. Its value is usually taken as 1.25.

## 19.14 Heat Gain from Power Equipments

The power equipments such as fan, motor or any other equipment of this type also adds heat in the air-conditioned space. The power consumed by the air-conditioning fan is converted into heat energy and imparted to the air. If the fan is located before the air conditioner, the heat energy must be added to the total load. If the fan is located after the air-conditioner, the heat energy is added to the room sensible heat load.

The electric motors used to operated the conditioning equipment within the conditioned space adds at least a part of the heat equivalent of the power consumed to the cooling load. The heat gain from the electric motor is given by

The load factor is the fraction of the total load at which the motor is working.

## 19.15 Heat Gain through Ducts

The heat gain due to supply-duct depends upon the temperature of air in the duct and the temperature of the space surrounding the duct. The heat gain through the duct ($Q_D$) is calculated by using the relation.

Where $Q_D=UA_D(t_a-t_s)$

U=Overall heat-transfer coefficient,

$A_D$ = Surface area of the duct,

$t_a$ =Temperature of ambient air, and

$t_s$=Temperature of supply air.

When all of the duct is in the air conditioned space, then the heat gain is zero. If the duct is located in an unconditioned space, there is a gain in heat and condensation takes place. In order to prevent condensation and to reduce duct heat gain. The duct must be insulated. The heat gain through supply duct is roughly taken as 5% of the room sensible heat.

The loss due to supply air leakage is not easy to estimate, because it depends upon the workmanship of duct construction and the length of run. It has been found that the duct leakages are of the order of 5 to 30 percent. The air leakage from supply ducts results in the loss of cooling capacity unless the leakages take place withing the conditioned space. The following are the minimum recommended values of air leakages through ducts:

Long runs ------------10%

Medium runs ------5%

Short runs ------Neglect

Note: the return-duct heat gain is calculated in the similar way as discussed above. But the air-leakage in this case is not appreciable for short lengths of ducts. If the ducts are long and runs through the unconditioned space, a leakage upto 3 percent may be assumed.

Example 19.1 A conference room for seating 100 persons is to be maintained at 22ºC dry bulb temperature and 60% relativehumidity. The outdoor conditions are 40ºC dry bulb temperature and 27ºC wet bulb temperature. The various loads in the auditorium are as follows:

Sensible and latent heat loads per person. 80W and 50 W respectively: lights and fans 15000W: sensible heat gain through glass, walls, ceiling etc., 15000W. the air infiltration is 20m$^3$/min and fresh air supply is 100m$^3$/min. Two-third of recirculated room air and one-third of fresh air are mixed before entering the cooling coil. The by-pass factor of the coil is 0.1.

Determine apparatus dew point, the grand total heat load and effective room sensible heat factor.

Solution. Given: No. of persons=100: $t_{d2}$=22ºC; $\phi_2$=60%; $t_{d1}$=40ºC: $t_{w1}$=27ºC:

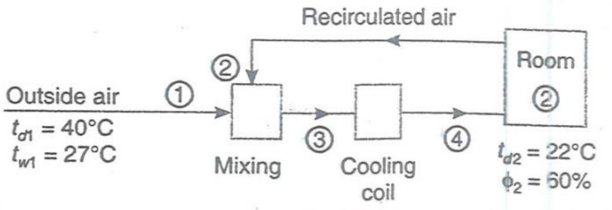


Fig. 19.6

$Q_s$ per person = 80W; $Q_l$ per person = 50 W; $Q_{SL}$ = 15000 W = 15 Kw; $Q_{SG}$=15000 W = 15KW; $V_1$=20m$^3$/min; $V_{F\,=}$ 100m$_3$/min; BPF = 0.1

The flow diagram for the given air conditioning system is shown in Fig. 19.6. the various points on the psychrometric chart, as shown in Fig. 19.6. The various points on the psychrometric chart, as shown in Fig. 19.7 , are marked as discussed below:

First of all, mark the outside condition of air i.e., 40ºC dry bulb temperature and 27ºC wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 19.7., Now mark the inside condition of air in the room i.e., 22ºC dry bulb temperature and 60% relative humidity as point 2. Locate point A by drawing vertical and horizontal lines through points 1 and 1 respectively.

From the psychrometric chart, we find that specific volume of air at point 1.

$V_{s1}$=0.911 m$^3$/kg of dry air

Enthalpy of air at point 1,

$h_1$=85 kJ/kg of dry air

Enthalpy of air at point 1,

$h_2$ = 47.5 kJ/kg of dry air

and Enthapy of air at point A, $h_A$=66kJ/kg of dry air

we know that mass of infiltrated air at point 1,

$$m_1=\frac{v_1}{v_{s1}}=\frac{20}{0.911}=21.95 \text{ kg/min}$$

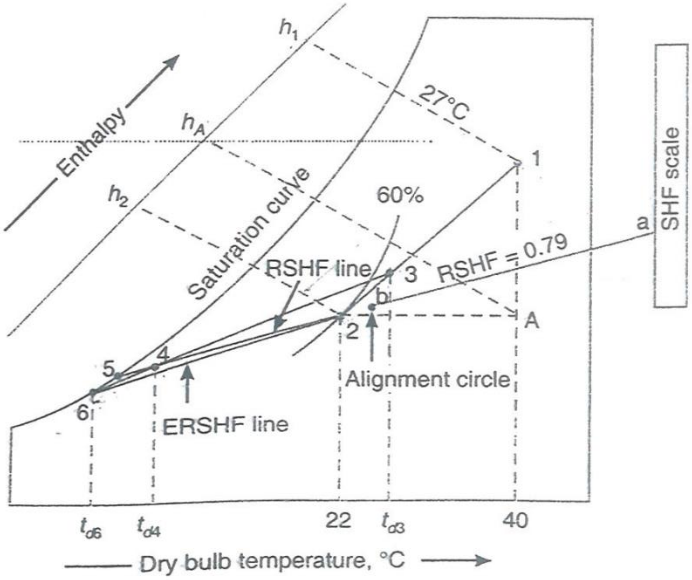


Fig. 19.7

∴ Sensible heat gain due to infiltration air

= $m_1$ ($h_A$ – h2) = 21.95 (66-47.5) = 406 kJ/min

=406/60 = 6.77 kW

and latent heat gain due to infiltration air

=$Q_s$ per person x No. of persons

=80 X 100 = 8000W = 8 kW

Total latent heat gain from perso

ns

=$Q_L$ Per person x No. of persons

=50x100 = 5000W = 5kW

$\therefore$ Total sensible heat gain in the room

RSH = Sensible heat gain due to infiltration air + Sensible heat gain from persons + Sensible heat gain due to lights and fans ($Q_{SL}$) + Sensible heat gain through glass, walls and ceiling etc. ($Q_{SG}$)

=6.77+8+15+15 = 44.77kW

and total latent heat gain in the room,

RLH = Latent heat gain due to infiltration air + Latent heat gain from persons

=6.95 + 5 =11.95kW

We know that room sensible heat factor,

$$\text{RSHF} = \frac{RSH}{RSH + RLH} = \frac{44.77}{44.77 + 11.95} = 0.79$$

Now mark this calculated value of RSHF = 0.79 on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e 26°C DBT and 50% RH), as shown in Fig 19.7., From point 2, draw a line 2-5 parallel to the line ab. This line 2-5 is known as RSHF line.

Since two-third of the recirculated air and one-third of fresh air are mixed before entering the cooling coil, therefore mark the mixing point 3 on the line 1-2, such that

$$\text{Length 2-3} = \text{Length 2-1} \times \frac{1}{3}$$

From the psychrometric chart, we find that dry bulb temperature at point 3,

$t_{d3} = 28^\circ C$

Apparatus dew point

Let $t_{d6}$= Apparatus dew point i.e. dew point temperature of the coil

We know that by-pass factor (BPF),

$$0.1 = \frac{t_{d4} - t_{d6}}{t_{d3} - t_{d6}} = \frac{t_{d4} - t_{d6}}{28 - t_{d6}}$$

By trial and error method, we find that

$t_{d4} = 13.15^\circ C$ and $t_{d6} = 11.5^\circ C$ Ans.

We know that outside air total heat,

$OATH = 0.02\ V_1\ (h_1 - h_2)$

$= 0.02 \times 20\ (85 - 47.5 = 15$ kW

$\therefore$ Grand total heat load,

GTH = RSH + RLH + OATH

=44.77 +11.95 +15 = 71.72 kW Ans.

Effective room sensible heat factor

Join point 6 with point 2 and produce upto the sensible heat factor scale on the psychrometric chart. The line 6-2 is known as effective room sensible heat factor (ERSHF) line and its value is given as

ERSHF = 0.77 aNS.

Example 19.2. The following data relates to the office air conditioning plant having maximum seating capacity of 25 occupants:

Outside design conditions = 34 ° C DBT, 28 ° C WBT

Inside design conditions = 24 ° C DBT, 50% RH

Solar heat gain = 9120 W

Latent heat gain per occupant = 105 W

Sensible heat gain per occupant = 90W

Lightening load = 2300W

Sensible heat load from other sources = 11630 W

Infiltration load = 14 $m^3$/min

Assuming 40% fresh air and 60% of recirculated air passing through the evaporator coil and the by-pass factor of 0.15, find the dew point temperature of the coil and capacity of the plant.

Solution. Given: No. of occupants = 2; $t_{d1}$=34 ° C; $t_{w1}$=28 ° C, $t_{d2}$=24 ° C; $\phi$=50%;

$Q_{ss}$=9120 W: $Q_L$ per occupant = 105 W ; $Q_s$ per occupant = 90 W: $Q_{SL}$ =2300 W: $Q_s$ from other sources = 11630 W; $V_1$ =14$m^3$/min; BPF = 0.15

The flow diagram for the given air conditioning system is shown in Fig. 19.8. The various points on the psychrometric chart, as shown in Fig. 19.9 are marked as discussed below.

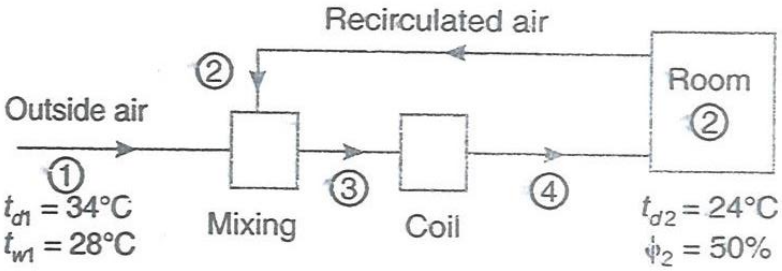


Fig. 19.8

First of all, mark the outside condition of air i.e., $34^\circ$Cdry bulb temperature and $28^\circ$C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 19.9. Now mark the inside condition of air i.e., $24^\circ$C dry bulb temperature and 50% relative humidity as point 2. Locate point A by drawing vertical and horizontal lines through points 1 and 2 respectively.

From the psychrometric chart, we find that specific volume of air at point 1.

$V_{s1}$=0.9 $m^3$/kg of dry air

Enthalpy of air at point 1

$h_1$=90kJ/kg of dry air

Enthalpy of air at point 2,

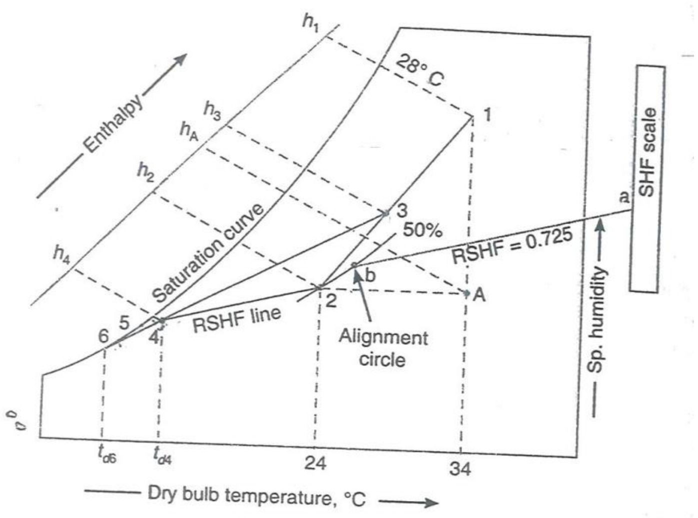


Fig. 19.9

$h_2$=48kJ/kg of dry air

we know that mass of infiltrated air at point 1,

$$m_1 = \frac{v_1}{v_{s1}} = \frac{14}{0.9} = 15.56 \text{ kg/min}$$

$\therefore$ Sensible heat gain due to infiltration air

$=m_1(h_A-h_2) = 15.56\ (59-48) = 171.16$ kJ/min

$=171.16/60 = 2.853$kW$=2853$W

and latent heat gain due to infiltration air

$= m_1(h_1-h_A) = 15.56\ (90-59) = 482.36$ kJ/min

$=482.36/60 = 8.039$kW $= 8039$W

Total sensible heat gain from occupants

$=Q_s$ per occupant x No. of occupants

$=90 \times 25 = 2250$ W

Total latent heat gain from occupants

$=Q_L$ per occupant x No. of occupants

$=105 \times 25 = 2625$ W

$\therefore$ Total sensible heat gain in the room,

RSH = Solar heat gain + Sensible heat gain due to infiltration air + Sensible heat gain from occupants + Sensible heat gain due to lightening + Sensible heat gain from other sources.

$=9120 + 2853 + 2250 + 2300+ 11630 = 28153$ W

$= 28.153$ kW

and total latent heat gain in the room,

RLH = Latent heat gain due to infiltration air + Latent heat gain from occupants

$= 8039+ 2625=10664$ W$= 10.664$ kW

We know that room sensible heat factor,

RSHF = $\frac{RSH}{RSH+RLH}=\frac{28.153}{28.153+10.664}$ =0.725

Now mark this calculated value of RSHF = 0.725 on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e 26 °C DBT and 50% RH), as shownin Fig.19.9. From point 2, draw a line 2-5 parallel to the line ab. the line 2-5 is known as RSHF line. Since 40% of fresh air (or outside air) and 60% of recirculated air is passed through the evaporator coil, therefore mark the mixing point 3 on the line 1-2, such that

Length 2-3 = Length 2-1 0.4

Let $t_{d6}$ = Dew point temperature of the coil

we know that by-pass factor of the coil (BPF),

0.15 = = $\frac{t_{d4}-t_{d6}}{t_{d3}-t_{d6}}=\frac{t_{d4}-t_{d6}}{28-t_{d6}}$

By trial and error method, we find that

$t_{d6}$ =10.4 ºC and $t_{d6}$=7.3 ºC Ans.

Capacity of the plant

From the psychrometric chart, we find that enthalpy of air at point 4,

$h_4$=29kJ/kg of dry air

and enthalpy of air at point 3,

$h_3$=64.8kJ/kg of dry air

we know that mass of air entering the room,

$m_a$= $\frac{Totalroomheat}{Totalheatremoved}=\frac{RSH+RLH}{h_2-h_4}$

$= \dfrac{28.153+10.664}{48-29}$ =2.043 kg/s = 122.58 kg/min

$\therefore$ Capacity of the plant

= $m_a$ ($h_3$-$h_4$) = 122.58 (64.8 -29) = 4388.4 kJ/min

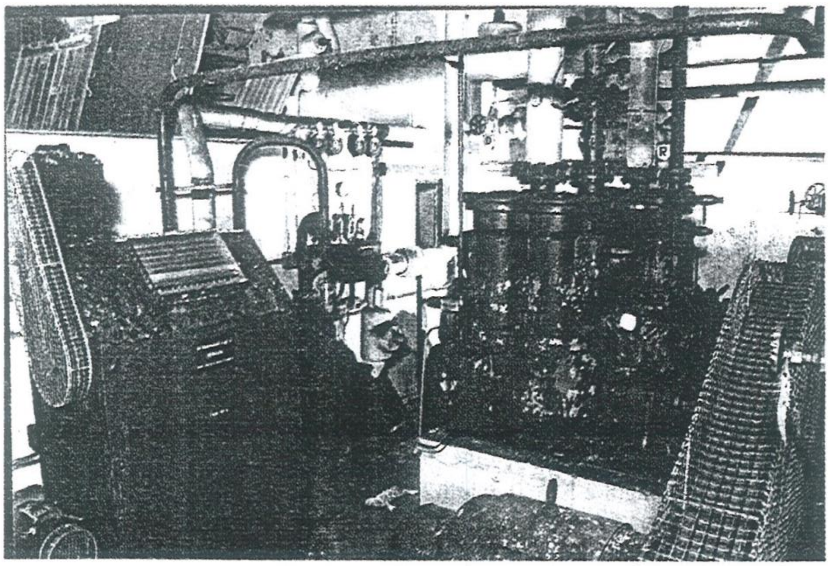
=4388.4/210 = 20.9 TR Ans.

Example 19.3. An air conditioning system is designed for a restaurant when the following data is available:

Total heat flow through the walls, roofs and floor = 6.2 kW

Solar heat gain through glass = 2 kW

Equipment sensible heat gain = 2.9 kw



*Main air conditioning plant room with all original fittings.*

Equipment latent heat gain = 0.7 kW

Total infiltration air = 400 $m^3$/h

Outdoor conditions = 35 ºC DBT; 26 ºc WBT

Inside designed conditions = 27 ºC DBT; 55% RH

Minimum temperature of air supplied to room = 17 ºC DBT

Total amount of fresh air supplied = 1600 $m^3$/h

Seating chairs for dining = 50

Employees serving the meals = 5

Sensible heat gain per person = 58W

Latent heat gain per sitting person = 44W

Latent heat gain per employee = 76 W

Sensible heat added from meals = 0.17 kW

Latent heat added from meals = 0.3 kW

Motor power connected to fan = 7.6 kW

If the fan is situated before the conditioner, then find the following:

(a) Amount of air delivered to the room in $m^3_{/h;}$

(b) Percentage of recirculated air;

(c) Refrigeration load on the coil in tonnes of refrigeration; and

(d) Dew point temperature of the cooling coil and by-pass factor.

**Solution.** Given: $Q_{sw}$=6.2 kW; $Q_{SG}$=2 kW; $Q_{SE}$ = 2.9 kW; $Q_{LE}$=0.7 kW; $V_1$=400 $m^3$/h; $t_{d1}$=35 °C;

$t_{w1}$=26ºC; $t_{d2}$=27 ºC; $\phi_2$=55%; $t_{d4}$=17 ºC; $V_f$=1600$m^3$/h; Dinning chairs = 50; Employees = 5; $Q_s$ per person= 58W; $Q_L$ per person = 44W; $Q_L$ per employee = 76 W; $Q_{sm}$=0.17kW; $Q_{LM}$=0.3kW; $Q_M$=7.6 kW.

The flow diagram for the given air conditioning system is shown in Fig. 19.10.

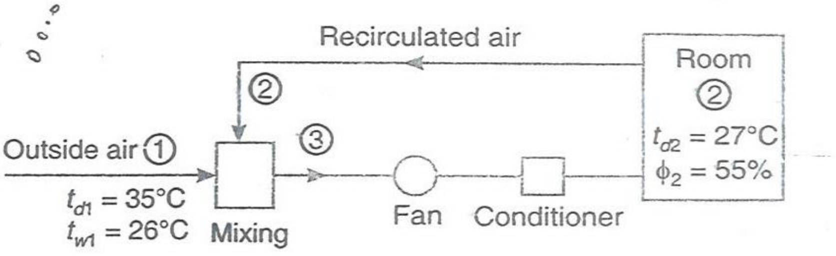


Fig. 19.10

First of all, mark the outdoor conditions of air i.e., 35 ºC DBT and 26 ºC WBT on the psychrometric chart as point1, as shown in Fig. 19.11. Now mark the inside designed conditions of air i.e. 27 ºCDBT and 55% RH as point 2. Locate point A by drawing vertical and horizontal lines from points 1 and 2 respectively.

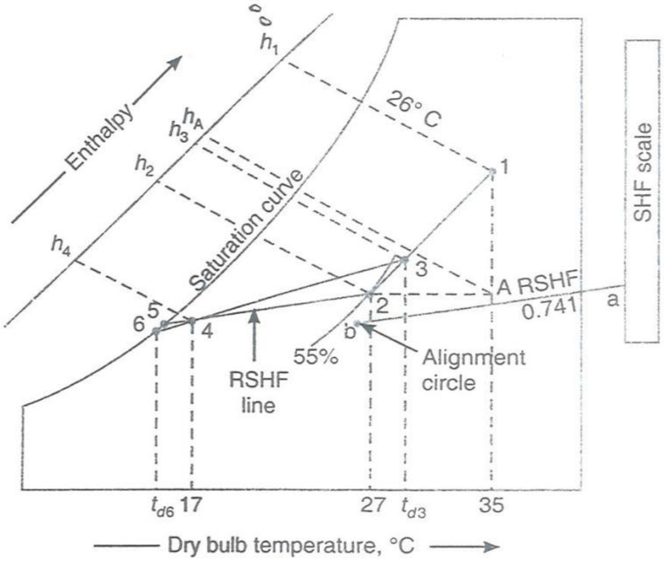


Fig. 19.11

From the psychrometric chart, we find that specific volume of air at point 1,

$V_{S1}$=0.897 $m^3$/kg of dry air

Enthalpy of air at point 1,

$h_1$=80.6 kJ/kg of dry air,

Enthalpy of air at point 2,

$h_2$=58.2 kJ/kg of dry air

and enthalpy of air at point A,

$h_A$=66kJ/kg of dry air

We know that mass of infiltration air,

$$m_1=\frac{v_1}{v_{s1}}=\frac{400}{0.897}=446\text{kg/h}$$

$\therefore$ Sensible heat gain due to infiltration air

$=m_1(h_A-h_2) = 446\ (66-58.2)= 3480$kJ/h

=3480/3600 = 0.97kW …….($\because$ 1 kW=1kJ/s=3600kJ/h)

and latent heat gain due to infiltration air

$=m_1\ (h_1-h_A)=446\ (80.6-66)=6512$ kJ/h

=6512/3600=1.8 kW

Sensible heat gain from persons

$=Q_s$ per person x No. of persons

58 x 50 = 2900 W= 2.9 kW

Latent heat gain from persons

$=Q_L$ per person x No. of persons

= 44 x 50 = 2200 W = 2.2kW

Sensible heat gain from employees serving the meals

= $Q_S$ per employee x No. of employees

=58 x 5 = 290 W = 0.29 kW……(Taking $Q_S$ per employee = 58 W

Latent heat gain from employees serving the meals

=$Q_L$ per employee x No. of employees

=76 x 5 = 380 W = 0.38 kW

Total sensible heat gain in the room,

RSH = Heat gain from walls, roof and floor ($Q_{sw}$) +Solar heat gain through glass ($Q_{SG}$)+Sensible heat gain due to infiltration air + Sensible heat gain from persons taking meals + Sensible heat gain from employees + Sensible heat gain from meals ($Q_{SM}$)

=6.2+2+2.9+0.97+2.9+0.29+0.17 = 15.43kW

Total latent heat gain in the room,

RLH=Latent heat gain from equipment ($Q_{LE}$)+Latent heat gain due to infiltration air + Latent heat gain from employees + Latent heat gain from meals ($Q_{LM}$)

=0.7+1.8+2.2+0.38+0.3 = 5.38 kW

∴ Room sensible heat factor,

$$\text{RSHF}=\frac{RSH}{RSH+RLH}=\frac{15.43}{15.43+5.38}=0.741$$

Now mark this calculated value of RSHF = 0.741 on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e 26°CDBT and 50% RH), as shown in Fig. 19.11. From point 2, draw a line 2-5 parallet to line ab. This line 2-5 is known as RSHF line. Now draw a vertical line through 17 °C DBT (minimum temperature of air supplied to the room which cuts RSHF line at point 4.

From the psychrometric chart, we find the specific volume of air supplied to the room at point 4,

$V_{S4}$=0.836 $m^3$/kg of dry air

andenthalpy of air at point 4,

$h_4$= 45 kJ/kg of dry air

(A) Amount of air delivered to the room in $m^3$/h

We know that the amount of air delivered to the room.

$$m_a = \frac{Totalroomheat}{Totalheatremoved} = \frac{RSH + RLH}{h_2 - h_4}$$

$$= \frac{15.43 + 5.38}{58.2 - 45} = 1.576 \text{ kg/s} = 5675 \text{ kg/h}$$

∴ Amount of air delivered to the room in $m^3$/h

(b) percentage of recirculated air

we know that mass of fresh air supplied,

$$m_F = \frac{V_F}{V_{S1}} = \frac{1600}{0.897} = 1784 \text{ kg/h}$$

∴ Mass of recirculated air = $m_a$-$m_F$= 5675-1784 = 3891 kg/h

and percentage of recirculated air

$$= \frac{3891}{5675} = 0.686 = 68.6\% \text{ Ans.}$$

(b) Refrigeration load on the coil

Since the recirculated air at point 2 is 68.6%, therefore the fresh air supplied to the room at point 1 is 31.4%. The mixing of recirculated air and fresh air before entering to the cooling coil is marked at point 3 on the psychrometric chart, such that

Length 2-3 = Length 1-2 x 0.314

From the psychrometic chart, we find that enthalpy of air at point 3,

$h_3 = 64.6$ kJ/kg of dry air

$\therefore$ Refrigeration load on the coil

$= m_a (h_3 - h_4) +$ heat added by fan motor

$= 1.576\,(64.6 - 45) + 7.6 = 38.5$ kW

$= 38.5/3.5 = 11$TR Ans. ….($\because$ 1 TR = 3.5 kW)

$= 38.5/3.5 = 11$ TR Ans

d Dew point temperature of the cooling coil and by-pass factor

Join point 3 with point 4 and produce the line to intersect the saturation curve at point 6. From the psychrometric chart, we find that dew point temperature of the cooling coil.

$t_{dp} = t_{d6} = 14.6\,^\circ$C Ans.

We know that by -pass factor,

$$BPF = \frac{t_{d4} - t_{d6}}{t_{d3} - t_{d6}} = \frac{17 - 14.6}{29.5 - 14.6} = 0.161 \text{ Ans.}$$

Example 19.4. A hall is to be maintained at 24°Cdry bulb temperature and 60% relative humidity under the following conditions:

Outdoor conditions = 38°C DBT and 28°C WBT

Sensible heat load in the room = 46.4 kW

Latent heat load in the room = 11.6 kW

Total infiltration air = 1200 $m^3/h$

Apparatus dew point temperature = 10 °C

Quantity of recirculated air from the hall = 60%

If the quantity of recirculated air is mixed with the conditioned air after the cooling coil,

find the following:

(a) The condition of air leaving the conditioner coil and before mixing with the recirculated air,

(b) The condition of air before entering the hall

(c) The mass of air entering the cooler;

(d) The mass of total air passing through the hall;

(e) The by-pass factor of the cooling coil; and

(f) the refrigeration load on the cooling coil in tonnes of refrigeration.

Solution. Given: $t_{d2}$=24 °C; $\phi_2$=60%; $t_{d1}$ =38 °C; $t_{w1}$=28 °C; $Q_S$=46.4 kW,

$Q_L$=11.6kW; $V_1$=1200 $m^3/h$; ADP = 10 °C

The flow diagram for the given air conditioning system is shown in Fig. 19.12.

(a) Condition of air leaving the conditioner coil and before mixing with the recirculated air

First of all, mark the outdoor conditions of air i.e., 38 °CDBT and 28 °C WBT on the psychrometric chart as point 1, as shown in Fig. 19.13. Mark the inside conditions of air i.e 24 °CDBT and 60% RH as point 2. Locate point A by drawing vertical and horizontal lines from points 1 and 2 respectively. From the psychrometric chart, we find that specific volume of infiltrated air at point 1,

Enthalpy of air at point 1,

$h1$ = 89.6 kJ/kg of dry air

Enthalpy of air at point 2,

$h_2$=53 kJ/kg of dry air

and enthalpy of air at point A,

$h_A$=68 kJ/kg of dry air

We know that mass of infiltrated air

$$m_1=\frac{V_1}{V_{S1}}=\frac{1200}{0.907}=1323 \text{ kg/h}$$

$\therefore$ Sensible heat gain due to infiltrated air

$=m_1(h_A-h_2)$= 1323 (68-53)=19845 kJ/h

=19845/3600 = 7.94 kW

$\therefore$ Total sensible heat gain in the room,

RSH = 46.4 + 5.5 = 51.9 kW

and total latent heat gain in the room,

RLH = 11.6+7.94 = 19.54kW

We know that room sensible heat factor,

$$RSHF=\frac{RSH}{RSH+RLH}=\frac{51.9}{51.9+19.54}=0.726$$

Now mark this calculated value of RSHF = 0.726 on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e 26 $^\circ$C DBT and 50% RH), as shown in Fig. 19.13. From point 2, draw a line 2-3 parallel to the line ab. This line 2-3 is known as RSHF line. Draw a line joining the points 1 and ADP=10 $^\circ$C which intersects the line 2-3 at point 4. This point 4 represents the condition of air leaving the conditioner coil (or coiling coil) and before mixing with the recirculated air.

From the psychrometric chart, we find that dry bulb temperature at point 4,

$t_{d4}$=16°C Ans.

and relative humidity at point 4,

$\phi_4$=90% Ans.

(b) Condition of air before entering the hall.

Since 60% of the air from the room at point 2 is recirculated and mixed with 40% of air leaving the cooling or conditioner coil at point 4, therefore mark the mixing point 5, such that

Length 2-5 = Length 2-4 x 0.4

The point 5 represents the condition of air before entering the hall.

From the psychrometric chart, we find that dry bulb temperature at point 5.

$t_{d5}$=20.8°C Ans.

and relative humidity at point 5,

$\phi_5$ =70% Ans.

(c Mass of air entering the cooler

We know that mass of air entering the cooler,

$$m_c = \frac{Totalroomheat}{Totalheatremoved} = \frac{RSH + RLH}{h_2 - h_4}$$

$$= \frac{51.9 + 19.54}{53 - 42}$$ =6.5 kg/s=6.5 x 60 = 390 kg/min Ans.

(dMass of total air passing through the hall

Let $m_H$=Mass of total air passing through the hall.

Since the total air ($m_H$) passing through the hall consists of air passing through the cooler and recirculated air from the hall (i.e. 60% $m_H$), therefore,

$390+0.6\ m_H = m_H$

$m_H = 390/0.4 = 975$ kg/min Ans.

(e By-pass factor of the cooling coil

We know that by-pass factor of the cooling coil,

$$\text{BPF}=\frac{t_{d4}-ADP}{t_{d1}-ADP}=\frac{16-10}{38-10}=0.214 \text{ Ans.}$$

(f Refrigeration load on the cooling coil

We know that refrigeration load on the cooling coil

$=m_c(h_1-h_4)=390\ (89.6-42)=18564$ kJ/min

$=18564/210 = 88$TR Ans.

Example 19.5. An air conditioning system is to be designed for a restaurant with the following data.

Outside design conditions = 40 °C DBT, 28 °C WBT

Inside design conditions= 25 °C DBT, 50% RH

Solar heat gain through walls, roof and floor = 5.87 kW

Solar heat gain through glass = 5.52 kw

Occupants = 25

Sensible heat gain per person = 58W

Latent heat gain per person = 58W

Internal lighting load = 15 lamps of 100W

10 fluorescent tubes of 80W

Sensible heat gain from other sources = 11.63 kW

Infiltrated air = 15 $m^3$/min

If 25% fresh air and 75% recirculated air is mixed and passed through the conditioner coil, find:

(a) The amount of total air required in $m^3$/h:

(b) the dew point temperature of the coil;

(c) The condition of supply air to the room; and

(d) The capacity of the conditioning plant.

Assume the by-pass factor equal to 0.2.

Draw the schematic diagram of the system and show the system and show the system on skeleton psychrometric chart and insert the temperature and enthalpy values at salient points.Solution. Given: $t_{d1}$=40°C; $t_{w1}$=28°C; $t_{d2}$=25°C; $\phi_2$=50%; $Q_{SW}$=5.87kW; $Q_{SL}$=15 x 100 +10 x 80 = 2300W = 2.3kW; $Q_s$ from other sources = 11.63kW; $V_1$ =15 m$^3$/min; BPF = 0.2.

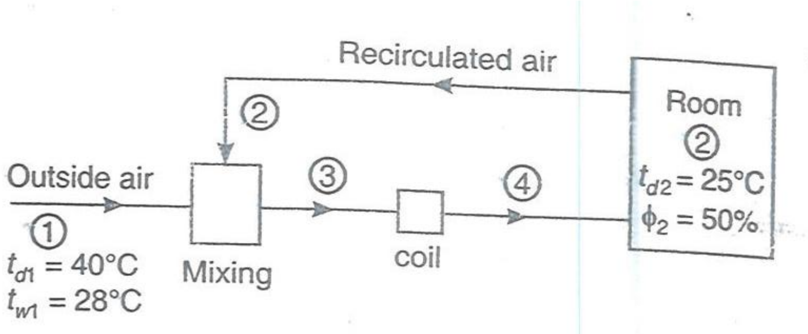


Fig. 19.14

The schematic diagram of the given air conditioning system is shown in Fig. 19.14. The various points on the psychrometric chart, as shown in Fig. 19.15, are marked as discussed below:

First of all, mark point 1 on the psychrometric chart representing the outside conditions of air i.e., 40°C DBT and 28°C WBT. Now mark point 2 representing the inside conditions of air i.e., 25°CDBT and 50% RH. Locate point A by drawing vertical and horizontal lines from points 1 and 2 respectively.

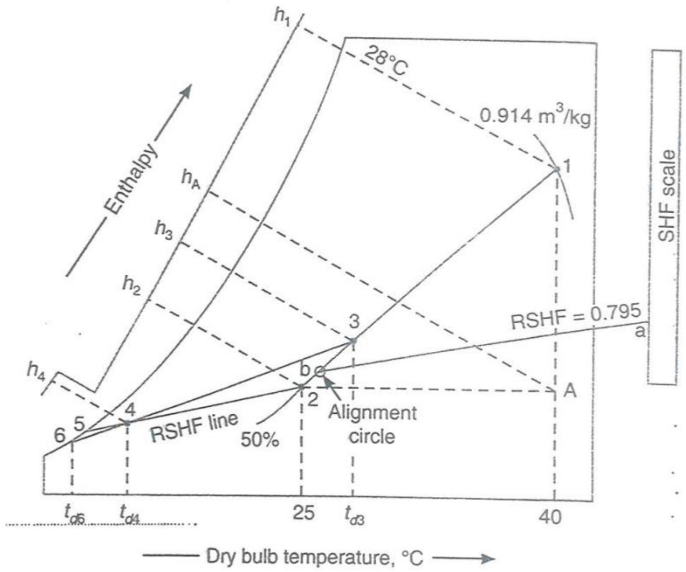


Fig. 19.15

From the psychrometric chart, we find that the specific volume of air at point 1,

$V_{S1}$ =0.914 $m^3$/kg of dry air

Enthalpy of air at point 1,

$h_1$ = 90 kJ/kg of dry air

Enthalpy of air at point 2,

$h_2$ = 50 kJ/kg of dry air

and enthalpy of air at point A,

$h_A$=66kJ/kg of dry air

We know that mass of infiltrated air at point 1,

$m_1=\frac{V_1}{V_{S1}}=\frac{15}{0.914}$=16.41 kg/min

∴ Sensible heat gain due to infiltrated air

$=m_1(h_A-h_2)$=16.41 (66-50)=262.56kJ/min

=262.56/60 = 4.376kW

and latent heat gain due to infiltrated air

$=m_1(h_1-h_2)$=16.41 (90-66)=393.84kJ/min

=393.84/60 = 6.564kW

Total sensible heat gain from occupants

$=Q_S$ per person x No. of occupants

=58 x 25 = 1450W = 1.45kW

Total latent heat gain from occupants,

$=Q_L$ per person x No. of occupants

=58 x 25 = 1450 W= 1.45 kW

Sensible heat gain due to internal lighting load,

=15 x 100 +10 x80 =2300W= 2.3kW

We know that sensible heat gain in the room,

RSH = Solar heat gain through walls, roof and floor + Solar heat gain through glass + Sensible heat gain due to infiltrated air + Sensible heat gain from occupants + Sensible heat gain due to internal lighting + Sensible heat gain from other sources.

=5.87 + 5.52 + 4.376 +1.45 +2.3+11.63 = 31.146kW

and total latent heat gain in the room,

RLH=Latent heat gain due to infiltrated air + Latent heat gain from occupants

=6.564 +1.45 = 8.014 kW

$\therefore$ Room sensible heat factor,

$$\text{RSHF} = \frac{RSH}{RSH + RLH} = \frac{31.146}{31.146 + 8.014} = 0.795$$

Now mark this calculated value of RSHF = 0.795 on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e. 26 $^\circ$C DBT and 50% RH), as shown in Fig. 19.15 . From point 2, draw a line 2-5 parallel to the line ab, This line 2-5 is known as RSHF line. Simnce 25% of fresh air and 75% of recirculated air is mixed and passed through the conditioner coil, therefore mark the mixing point 3 on the line 1-2, such that

Length 2-3 = Length 2-1 x 0.25

From the psychrometric chart, we find that dry bulb temperature of air at point 3,

$t_{d3}$=28.8 °C

and enthalpy of air at point 3,

$h_3$ =60kJ/kg of dry air

The point 4 on the line 2-5 and point 6 on the saturation curve (line 3-6) is marked as discussed below:

We know that by-pass factor of the coil (BPF),

$$0.2 = \frac{t_{d4} - t_{d6}}{t_{d3} - t_{d6}} = \frac{t_{d4} - t_{d6}}{28.8 - t_{d6}}$$

By trial and error method, we find that

$t_{d4}$=14.4 °C and $t_{d4}$=10.8 °C

The point 4 represents the condition of air leaving the coil and entering the room. The point 6 represents the dew point temperature of the coil. From the psychrometric chart, we find that enthalpy of air at point 4,

$h_4$=36.5 kJ/kg of dry air

and specific volume of air at point 4,

$V_{s4}$=0.826 $m^3$/kg of dry air

(a) Amount of total air required in $m^3$/h

Let

We know that the amount of total air required,

$$m_a = \frac{Totalroomheat}{Totalheatremoved} = \frac{RSH + RLH}{h_2 - h_4}$$

$=\frac{31.146+8.014}{50-36.5}$ =2.9kg/s=2.9 x 3600 = 10.440 kg/h

$V_a = m_a \times V_{s4}$

=10440 x 0.826 = 8623.4 $m^3$/h Ans.

b) Dew point temperature of the coil

We have obtained above that the dew point temperature of the coil.

$t_{dp} = t_{d6} = 10.8^\circ C$ Ans.

C) Condition of supply air to the room

The point 4, as shown in Fig. 19.15, represents the condition of supply air to the room. at this point 4, dry bulb temperature,

$t_{d4} = 14.4^\circ C$ Ans.

and relative humidity at point 4,

$\phi_4 = 88\%$ Ans.

d. Capacity of the conditioning plant

We kbnow that capacity of the conditioning plant

$= m_a (h_3 - h_4) = 2.9 (60-36.5) = 39.15$ kW

=39.15/3.5 =11.2 TR Ans. ------(∵ 1 TR = 3.5 kW)

# FANS

A fan is a kind of pump which is used for pumping or circulating the air through the entire duct system and duct system and the conditioned space. It is usually located at the inlet of the air conditioner. A fan, essentially, consists of a rotating wheel (called impeller) which is surrounded by a stationary member known as housing. The energy is transmitted to the air by the power driven wheel and a pressure difference is created to provide flow of air. The air may be moved by either creating an above atmospheric pressure (i.e. positive pressure) or a below- atmospheric pressure (i.e negative pressure). All fans produce both the conditions. This air at inlet to the fan is below atmospheric pressure(i.e negative pressure). All fans produce both the conditions. The air at inlet to the fan in blew atmospheric pressure while at the exhort or outlet of the fan is above atsmpsheric pressure. The air feed into a fan is called induced draft while the air exhaust from a fan is called forced draft.

The fans, irrespective of their type of construction, may function as either blowers or exhausters. The blower discharge air against a pressure at their outlet whereas exhausters remove gases from a space by suction.

## Types of Fans

The following two type of and may be used for the transmission of air:

1. Centrifugal or radial flow fans, and 2. Axial flow fans.

   When the air enters the impeller axially and is discharged radically from the impeller, it is called a centrifugal or radial flow fan.

   When the air flows parallel to the axis of impeller, it is called an axial flow fan.

## Centrifugal Fans

The centrifugal fans are widely use for duct air conditioning system, because

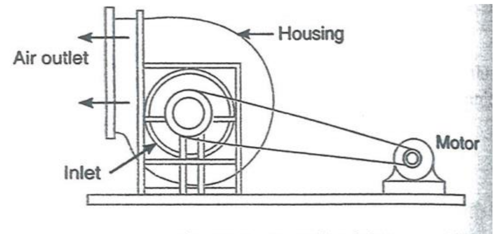


Fig. 21.1. Centrifugal fan.

they can conditioning system, because they can efficiently move large or small quantities of air over a fraters ranges of operating pressures. All centrifugal fans have an impeller or wheel mounted in a scroll type of housing, as shown in Fig.21.1. The impeller is turned either by the direct drive or more frequent by an electric motor employing pulleys and belt. The centrifugal force created by the rotating impeller moves the air outward along the blade channels. The outward moving air streams are combined by scroll into a single large air stream. This air stream leaves the fan through the discharge outlet.

The fan impeller may have the following three type of bases:

1. Radial or straight blades, 2. Forward curved blades, and 3. Backward curved blades.

The centrifugal fans with radial blades, as shown in Fig.21.2 (a), have simple impeller construction. The blades run straight out from a central hub. Some fans of this type have heavy steel bides with high structural strength. These fans parade very high pressure at high speeds.

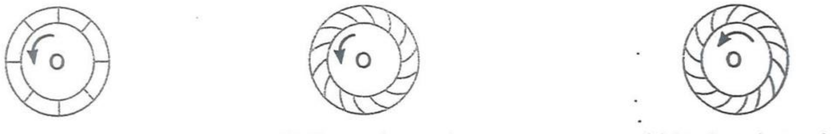
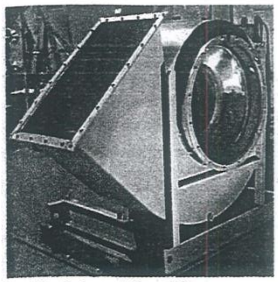

(a) Radial. (b) Forward curved. (c) Backward curved.

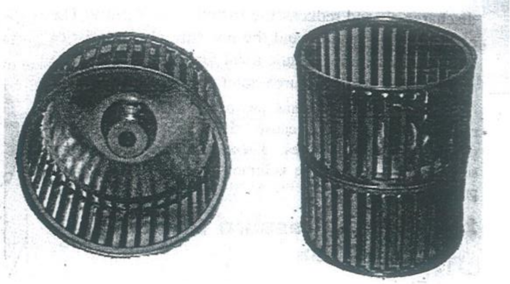
Fig. 21.2. Three types of fan impeller blades.

A large s have impeller blades are very pening more nearly approaches to that of the impeller. The ample ingle opening together with stream-lined hub of the wheel, promotes a smooth flow of air into the rotating blades. This increases the efficiency of the fan and recesses its noise when their operation is at low speeds.

The centrifugal fan impeller may have backward caved blades, as how in Fig.21.2 ©. The backward curved blades must be operated at much higher speed of rotation than the forward caved blades, if the same static pressures to be produced in each case. In some cases, the higher speed may be an advantage because of a possible direct connection to the deriving motor. The fan impellers having backward caved blades operate at high efficiently and have no overloading power characteristic. They also offer the advantage of wide ranges of capacity at constant speed with small changes in the power requirements.



Radial centrifugal fans.

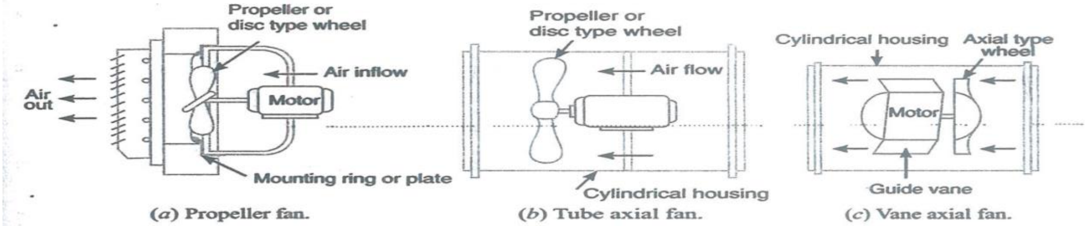


Centrifugal blowers.

## Axial Flow Fans

The axial flow fans are divided into the following three groups :

1. Propeller fan. An propeller type of axial flow fan consists of a propeller or disc type wheel which operated within a mounting as shown in Fig. 21.3 (a). The design of the ring surrounding the wheel is important because it prevents the air discharged from being drawn surrounding the wheel important because it around its periphery. The propeller fans are used only when the resistance to air movement is small. They are useful for the ventilation of attic spaces, lavatories and bathrooms, removal of cooking odious from kitchens and many other applications where little or no duct work is involved.



(*a*) Propeller fan. (*b*) Tube axial fan. (*c*) Vane axial fan.

**Fig. 21.3.** Types of axial flow fans.

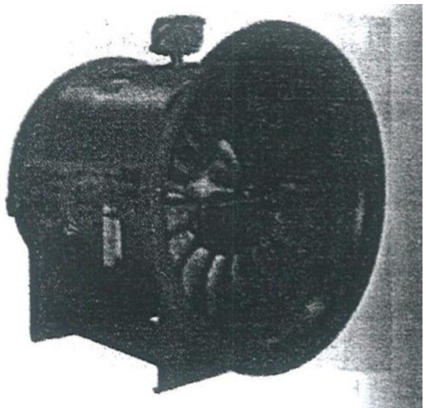
2. Tube axial fan. A tube axial fan consists of a propeller wheel housed in a same cylinder as shown in Fig. 21.3(b). The wheel may be driven either from an electric motor within the cylinder directly connected to its shaft or may be driven through a belt arrangement from a motor efficient than propeller fans. The air discharge from tube axial fan flows a spiral path as it leaves the cylindrical housing

3. Vane axial fan. A vane axial fan combines a tube axial fan wheel mounted in a cylinder with a set of air guide Vance, as shown in Fig.21.3(c). This fan eliminates spiral flow of the air and reduces the turbulence of flow. The efficiency of operation and the pressure characteristics are better than those of tube axial fan. The straight line flow fan assures quiet operation.

   Note: the axial flow fans are never used for duct air conditioning system because they are incapable of developing high pressure. These fans are particularly suitable for handling large volumes of air at relatively low pressures.

## Total pressure Developed by a Fan

We have already discarded the static pressure, velocity pressure and total pressure of air in ducts. In case of a fan, the fan static pressure (PSF) is the pressure increase produced by a fan. The fan velocity pressure (PVF) is the velocity pressure corresponding to the mean velocity of air at the fan outlet based on the total outlet area without any deductions formoitore, fairings, or other bodies. The total pressure created by a fan or the fan total pressure at the fan inlet. Mathematically, fan total pressure,



Axial flow fans.

Where $P_{TF} = P_{T2} - P_{T1}$

$P_{T2}$ = Total pressure at fan outlet

= Static pressure at fan outlet + velocity pressure at fan outlet

= $P_{S2} + P_{v2}$ and

$P_{T1}$ = Total pressure at fan inlet

= Static pressure at fan inlet + Velocity pressure at fan inlet

= $P_{s1} + P_{V1}$

We know that the total pressure eat a point is the sum of static pressure and velocity pressure at the point. Thus, for a fan,

Fan total pressure = Fan static pressure + Fan velocity pressure

$P_{TF} = P_{SF} + P_{VF}$

Since the fan velocity pressure ($P_{VF}$) is the velocity pressure at the fan outlet (pV2), therefore

PTF = PSF + PV2

Notes: 1. If fan has no suction duct, the entry losses to the fan housing are considered as part of the losses and are reflected in the mechanical efficiency of the fan.

In an actual system, the fan has a suction duct and apparatus such as filters and coiled. In such a sachem teetotal pressure at the fan inlet is always equal to in that part of the system. Also, the total pressure at the fan inlet in such a system is always negative and it is numerically less than the static pressure at the fan inlet.

2. If the fan has no discharge duct (i.e. the fan delivers air directly into a free open space), the discharge static pressure is zero (i.e Ps2=0). Thus the total pressure at the fan outlet is quall to the velocity pressure (i.e. PT2 = PV2). In an actual system, the fan has a discharge duct. In such a system, the total

pressure at the fan outlet is quall to the velocity pressure at the point of discharge plus all pressure losses in the path taken by air to reach that point.

## Fan Air Power

The power output of fan is expressed in terms of air power and represents the work done by the fan. Mathematically, total fan air power (based on fan total pressure, p $_{TF}$)

$$P_{at} = \frac{9.81Q \times p_{TF} \times K_P}{60} \text{ (in watts)}$$

where

Q = Total quantity of air flowing at the fan inlet in $m^3$/min,

$P_{TF}$ = Fan total pressure in mm of water, and

$K_p$ = Compressibility coefficient

Similarly, static fn air power based on the fan static pressure ($P_{SF}$),

$$P_{at} = \frac{9.81Q \times p_{SF} \times K_P}{60} \text{ (in watts)}$$

Note: If Q is expressed in $m^3$/s and $P_{TF}$ and PSF are in N/m2, then total fan air pwer (in water).

$P_{at} = Q \times P_{TF} \times K_P$

And static fan air power, $P_{as} = Q \times P_{SF} \times K_P$

## FAN EFFICIECIES

The ratio of the total fan air power to the driving power (or brake power) required at the fan shaft is known as total fan efficiency. It is also called mechanical efficiency of the fan. Mathematically, total fan efficiency

$$\eta_{TF} = \frac{Totalfanairpower(P_{at})}{Inputorbrakepower(B.P.)}$$

Similarly, static fan efficiency,

$$\eta_{SF} = \frac{StaticFanairpower(P_{as})}{Inputorbrakepower(B.P.)}$$

**Chillers**

Chiller is the name given to a special refrigeration device that cools water or any other fluid that is circulated to a remote location for cooling. Chillers are, similar to evaporators except that instead of cooling a room or an application, water is cooled. This water is then used for cooling by circulating it through cooling coils in the application area.

Thus, chillers, unlike evaporators, do not directly cool the application. Chillers may work on vapour compression or vapour absorption cycle. The capacity of a chiller ranges from 3 tons to 1000 tons of refrigeration. For a chiller working on the vapour compression cycle, the entire chiller package consists of a compressor, an air-cooled condenser, an expansion valve and a water cooler (similar to an evaporator). The heat extracted from the water which is cooled is rejected to the condenser. The condenser is sometimes water cooled. In that case, the heat absorbed by the cooling water in the condenser is rejected to a cooling tower.

If centrifugal compressors are used, then the chiller is called centrifugal chiller. For a chiller working on the vapour absorption cycle, the compressor is replaced by absorber-generator combination. The heating can be done by electrical heaters or steam or hot water. Absorption chillers can come in size as large as 1500 tons. In air-conditioned buildings, two or three chillers are use4 with one stand by unit. They are located in the basement or on the roof. Companies like York, Trane manufacture chillers.

**Pumps and Piping**

Pumps used in air-conditioning applications are single-inlet centrifugal pumps.

Pumps for larger systems are horizontal spilt case type with double suction impeller. Sometimes, end suction pumps are also used. Pumps are required for chilled water circulation, hot water and condense water circulation, steam condensate return, boiler feed circulation and for fuel oil supply.

Air-conditioning piping is of various types according to the fluid flowing through pipes. The following types of piping are required:

- Refrigerant piping

- Fuel piping

- Steam piping

- Water piping (hot and cold)

The steam and hot water piping is insulated using fibre glass wool. Copper tubing is used for refrigerant piping. Stainless steel or CI pipes are used for water piping. Steam traps are used in steam piping to prevent steam loss. All the piping is run in pipe chases or enclosures in the buildings.

**Diffusers and Grilles**

A grille is a decorative covering for the return air duct and diffuser is the supply device for the supply air duct. If the grille has a motorized damper controlling its operation, it is known as a register.

Grilles and diffusers improve the aesthetics of the air-conditioning system. They usually are at the same level as false ceiling, which is provided to conceal ducts. Figure .1 shows the supply diffuser and return

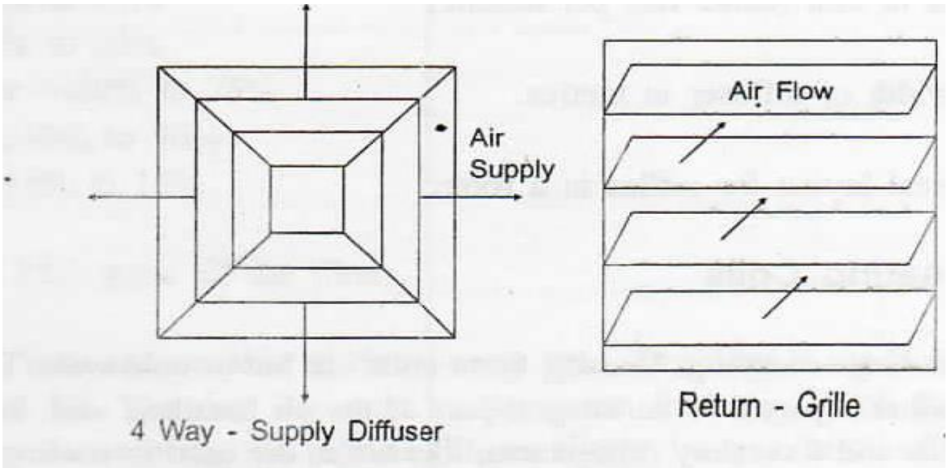


Fig. 1: Diffuser and Grille

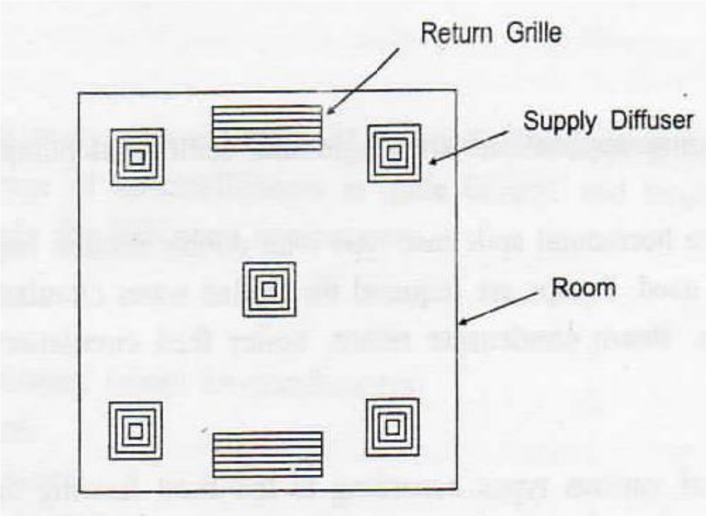


Fig. 2: Room Layout-Typical Pattern of Grilles and Diffusers

Diffusers are responsible for proper supply of conditioned air to the room. Usually, they are made up of a series of rectangular plates attached one above the other. The air is supplied in four directions through gaps between the plates. Figure 17.1 shows a four-way air diffuser. Three-way or two-way diffusers are also available, where air supply is in three or two directions. Round diffusers are also available. Diffusers should not be looted near walls and a typical pattern is shown in figure .2. A rectangular pattern for diffuser layout is usually followed for proper air distribution in the room.

Grilles can be located in the floor or in walls or in the ceiling. Grilles usually contain a series of slotted vanes fitted in a frame. The vanes can be operated by a damper driven by a motor. Usually, ceiling-to-floor air distribution is used to eliminate drafts and provide proper air circulation in the room. Usually, one or two grilles are provided for each room. It is necessary to consider throw and velocity of air through the diffuser to determine size of a diffuser. Same is the case for a grille. The throw can be calcglated as follows:

$$L = \frac{KQ}{\sqrt{ab}}$$

Where L = throw in metres

Q = quantity of air in cfm (cubic feet per minute)

K = constant

a, b = length and width of diffuser in metres.

Figure .2 Shows the typical layout for grilles in a room.

**Cooling Coils and Heating Coils**

These are used for either cooling or heating. Usually, these coils use hot or cold water for heating or cooling. The coils are either separate or an integral part of the air handling unit. Sometimes, steam is used in heating coils and secondary refrigerants, like brine, are used in cooling coil. The coils are bundled together usually in a rectangular fame, which can be fitted easily in the air handling unit. The flow rate of hot or cold water can be varied in the coil to control the amount of heating or cooling of the room. Coils are usually made of copper or stainless steel. Maximum surface area is exposed to the air to allow maximum heat transfer.

**Air-Conditioning Filters**

Filters are used in air-conditioning to remove dust and other contaminant particles from air in order to purify it. Filters are of various types and five groups of classification are as follows: 1] Dry filters 2] Electric filters 3] Viscous filters 4] Wet filters 5] Centrifugal filters.

It is assumed that there is no pressure drop in the filter. Let us study each type of filter. Filters are selected on the basis of the following factors:

1] Degree of air cleanliness required

2] Volume of air handled

3] Method of disposal of dust

4] Type and amount of particulate matter in air to be removed

Filters improve the performance of an air-conditioning system by maintaining air purity, improving life of the system and reducing maintenance of the system.

Filter efficiency is defined as:

$$\eta_f = \frac{W_1 - W_2}{W_1} \times 100\%$$

Where $W_1$ = weight of dust particles in unit volume of air entering air-filter

$W_2$ = weight of dust particles in unit volume of air leaving air-filter

The filter efficiencies of different filters at as follows:

Dry filter → 60% to 95%

Wet filter → 475% to 80%

Centrifugal filter →50% to 75%

Electric filter →70% to 90%

Viscous filter →35% to 19/0'

Figures 3 to 7 show all the filters.

**1] Dry filters**: These are reusable or throw-away type. Dry filter is made of cloth, coarse paper, wool or cellulose felt. The dust is caught by the screen from air, which flows over the filter in a steady stream at velocity 10 m/min. These filters have a high efficiency up to 65%.

Raping action can be used to clean these filters. The baghouse filter is a type of dry filter. These filters can remove small particles, but are not useful for high-dust concentration air streams.

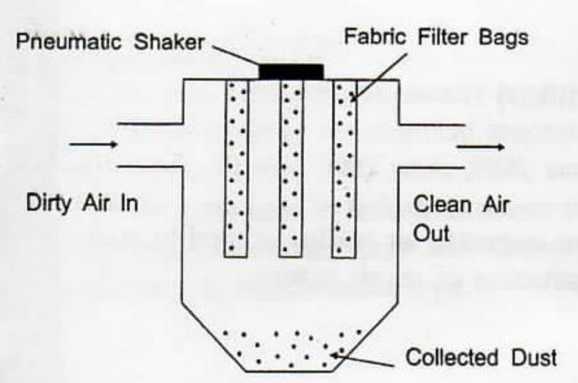
**2] Viscous filter**: These filters are in the form of pads or mats made of steel wool, copper mesh or plastic and are impregnated with oils and greases. This viscous layer traps dust and other particles from air. These are throw-away type only and are discarded periodically. Care should be taken to maintain viscosity of grease upon exposure to hot air. Also, growth of bacteria in the moist grease should be prevented by additives. The efficiency of such filters is low (only 5% to 15 %)

**3] Wet filters**: Air washer is a wet type filter. In this filter, dust particles are wetted by a water spray and heavy particles then settle down. The effectiveness of a wet filter depends on the ability of dust particles to get wet. These filters are ineffective for greasy particles like pollens in air. Their efficiency is about 75% to 80%. These filters are used for industrial air-conditioning for absorption of gases as well. They can be used in combination with dry and viscous filters.

**4] Electric filters**: These filters use ionization to capture dust particles in air. Air is passed between two charged plate conductors. The two conductors have opposite charges. The particles in air get ionized by passing through the electric field of the two plates. This ionized air is passed through a collection chamber. In this chamber, there are a series of vertical metal plates. These plates are alternately charged positive and negative. The positive negatively charged plates and vice versa. Thus, dust particles collection, the plates are washed by water sprays to remove advantage of this filter is its ease of operation, compactness, low and its effectiveness for high-dust concentration air streams. Efficiency ranges around 70%-90% for such filters. Drawbacks include danger of sparking and tendency to form ozone, a pollutant.

**5] Centrifugal filters:** These are dust collectors commonly used in industrial applications. It collects dust effectively. For powder processing or asbestos industries, dust collected can be reused saving costly raw material.

In this filter, a high-velocity air stream is used to produce a whirl in a chamber. The centrifugal force separates light and heavy dust particles. Heavy particles move towards the side and fall down. They are collected and removed. Lighter particles remain at the centre, whereas clean air is removed from the top of the chamber. Efficiency of this filter is 50% to 75%. Advantage is its rugged construction. Disadvantage is its high-power consumption.



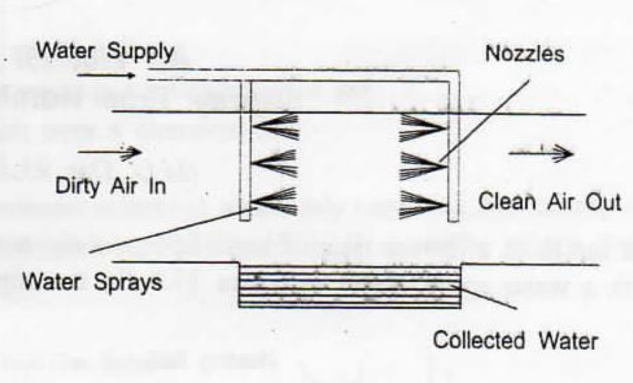
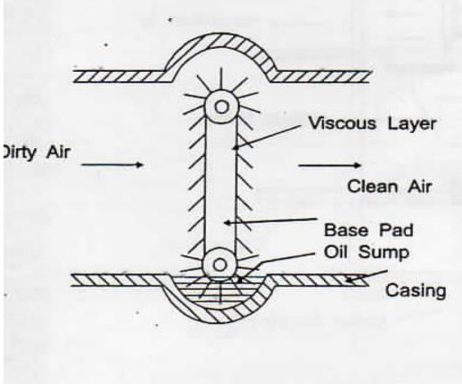


**Fig. 3: Dry Filter**

**Fig. 4: Wet Filter**



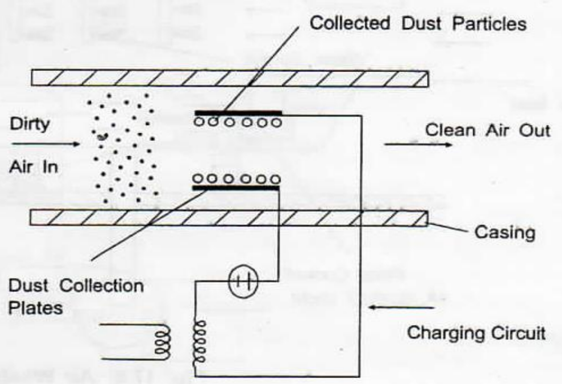


**Fig.5: Viscous Filter**

**Fig.8: Electric Filter**

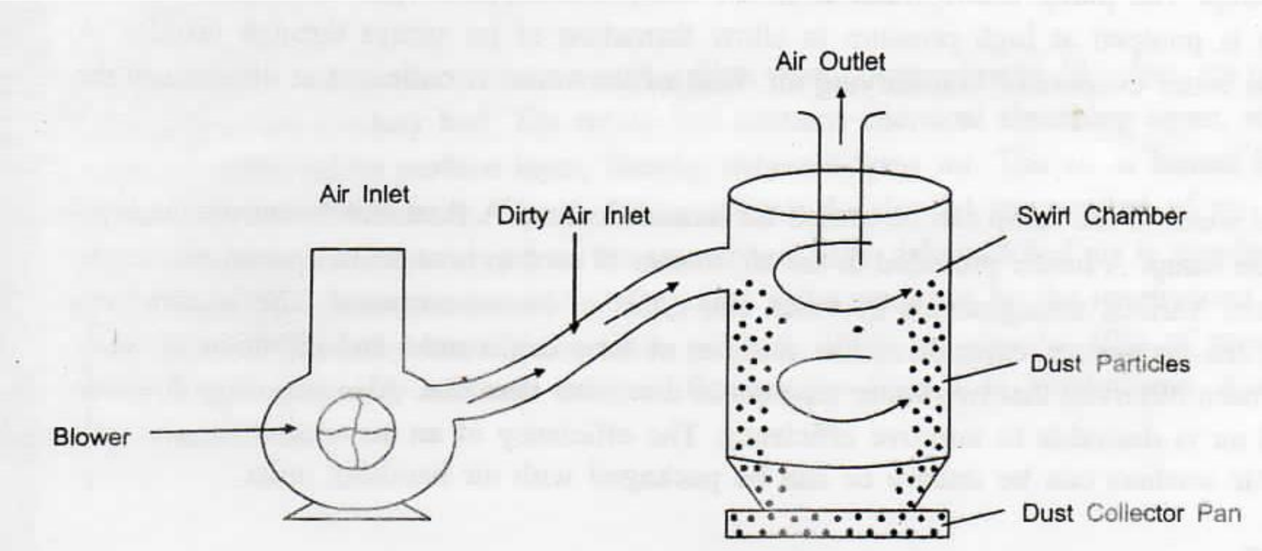


**Fig.7: Centrifugal Filter**

## Air Washer

## (Spray Type Humidifier)

*(M.U. Dec 95,June 1998, June 2000, Jan 01, June 01)*

This is the most effective type of humidifier. As the name suggests, air washer is used to wash air with a water spray. Refer to figure 17.8 for the construction of an air washer.

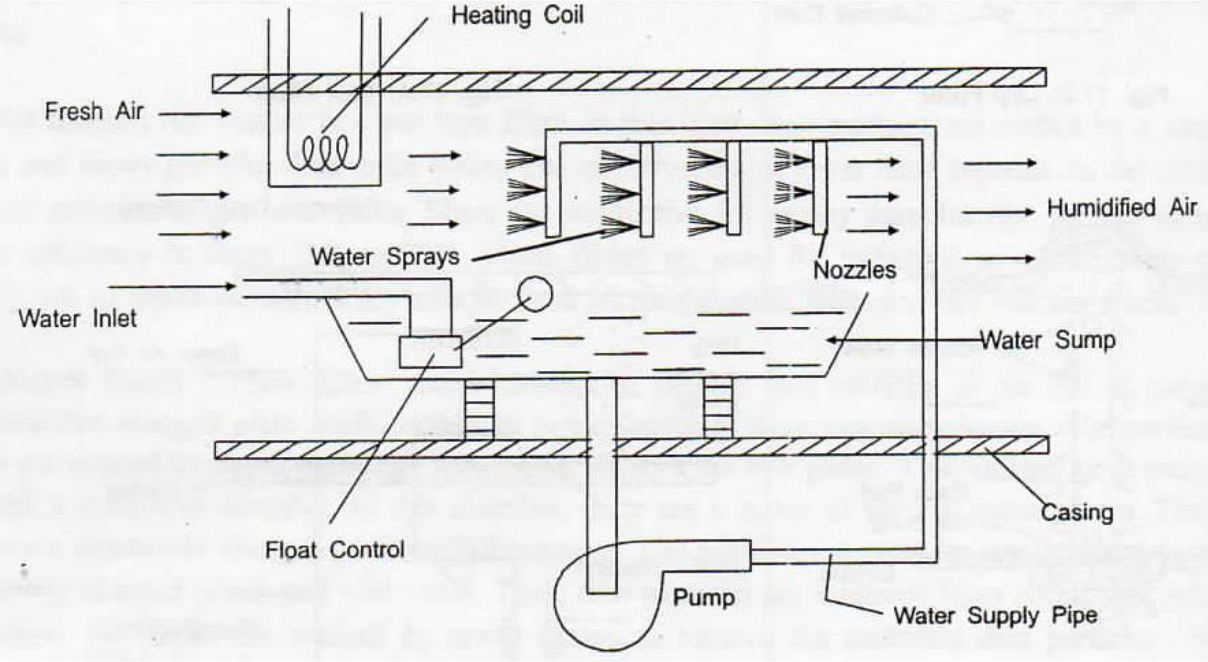


**Fig. 8: Air Washer**

Water is sprayed through nozzles fitted to a bank of tubes. Water is supplied through a water main to a water sump. The pump draws' water from the sump and supplies it to the tube bank. The supply water is pumped at high pressure to allow formation of jet sprays through nozzles. A portion of the water evaporates humidifying air. Rest of the water is collected at the base in the sump.

The collected water in the sump can be reused for humidification. A float valve controls the level of water in the sump. A heater provided in the air washer is used to heat air to desired conditions of temperature. Various arrangements of tubes and types of heater are used. The humidifying efficiency of the air washer depends on the number of tube banks used and direction of water spray. It has been observed that two water tube banks are better than one. Also opposing direction of water and air is desirable to improve efficiency. The efficiency of an air washer ranges from 60%-95%. Air washers can be unitary or can be packaged with air handling units.

## Dehumidifiers

Dehumidifiers are used for removal of moisture from air. The common techniques for dehumidification are:

1] To cool air below DPT

2] To absorb the moisture physically by a hygroscopic substance like salt

3] To remove moisture by chemical absorption over a chemical bed.

Dehumidifiers are used in industries where moisture control is absolutely essential. For example: Textile, pharmaceutical, food and electronics industries. Figure 17.9 shows a rotary bed dehumidifiers of the chemical type.

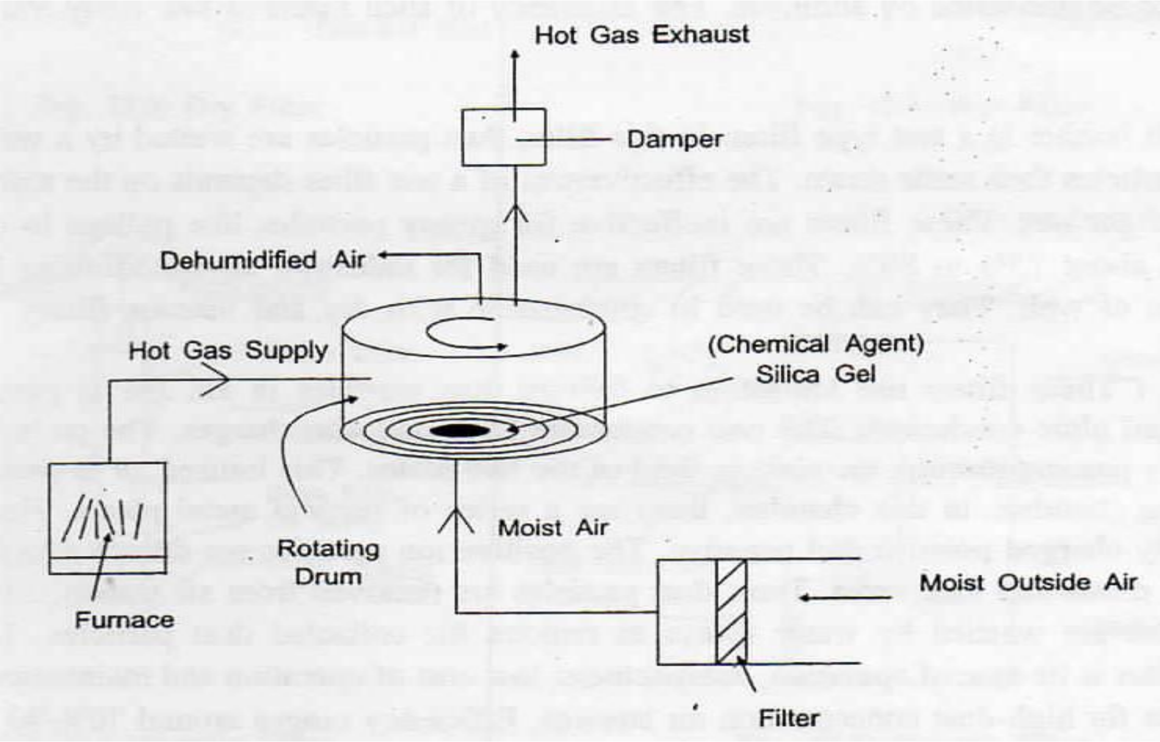


**Fig. 9: Dehumidifier - Rotary Chemical Type**

In this dehumidifier, outside air passes through a filter where contaminants, like dust, are removed. Moist air then enters a rotary bed. The rotary bed contains chemical absorbing agent, silica gel. It absorbs moisture on the surface layer, thereby dehumidifying air. The air is heated by using hot gas coming out of a furnace. This helps to achieve the desired temperature of air. The hot flue gases are exhausted by a damper to the atmosphere. The dehumidified air is supplied to the application area. The moisture absorbed in the silica gel is removed by the reactivation process. The reactivation is done by heating the silica gel bed. Silica gel absorbs 40% of moisture by weight of the gel. Capacity of these dehumidifiers ranges from 1 to 3000m3Imin. It can give continuous operation.

## Air Handling Unit (AHU)

*(M.0 June 2001)*

Air handling unit is a device which is a combination of several entities. It consists of supply fan, filter, a cooling coil, a heating coil, a sprayer (humidifier), a dehumidifier and dampers for regulation. Sometimes, preheat coils are also used along with return fans. The heating coil has a supply of steam or hot water for heating. The cooling coil has a supply of chilled water. The air enters through the inlet damper and gets purified by passing over the filter. For summer air-conditioning, cooling and dehumidification is required. The air is blown over cooling coil and dehumidifier by supply fan. The conditioned air is supplied to the room by the outlet damper. The dampers are motor controlled as shown in figure.10. For winter air-conditioning, air is heated and humidified. The air is blown by the fan over the heating coil and the spray humidifier sprays water in the air stream for humidification. The hot, humid air is supplied to the conditioned space by the outlet damper. Thus air handling unit can be used for year-round air-conditioning. Most components are assembled on site. York, Trane and Carrier are some big corporations, which manufacture air handling units. Air handling units are located on vibration-proof pads on the roof or in the basement of the building and connected to the duct system. A single air handling unit can supply air to several zones in the building. The fans used in the A1-111 are centrifugal type. The coils are usually finned and referred as 'decks'.

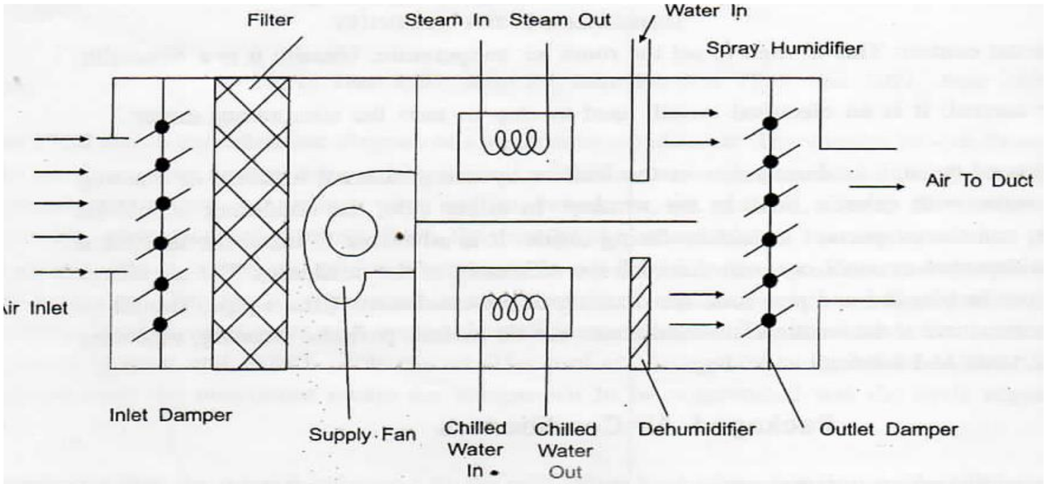


Fig. 10: Air Handling Unit

Evaporative cooling is a process of adiabatic saturation of air, in which a spray of water is made to evaporate without transfer of heat to or from the surrounding. That is, sensible heat removed from air is equal to the latent heat of vaporization of water. Evaporative cooling is the simplest air-conditioning technique with lowest cost. The evaporative cooling is limited by WBT of ambient air. Since it humidifies air, the technique does not work well in humid regions, like Mumbai. It works well for arid and dry regions, like Nagpur or Pune. Figure .11 shows the inner details of a desert cooler and figure .12 shows its outer details.

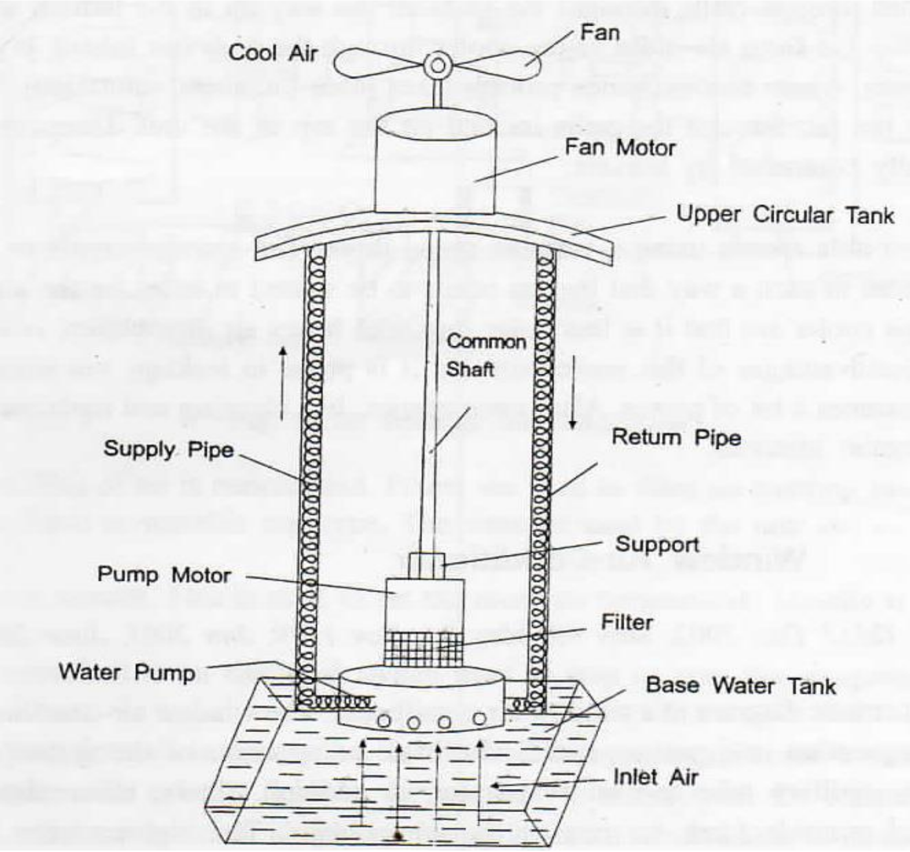


**Figure 11: Inner Details of a Desert Cooler**

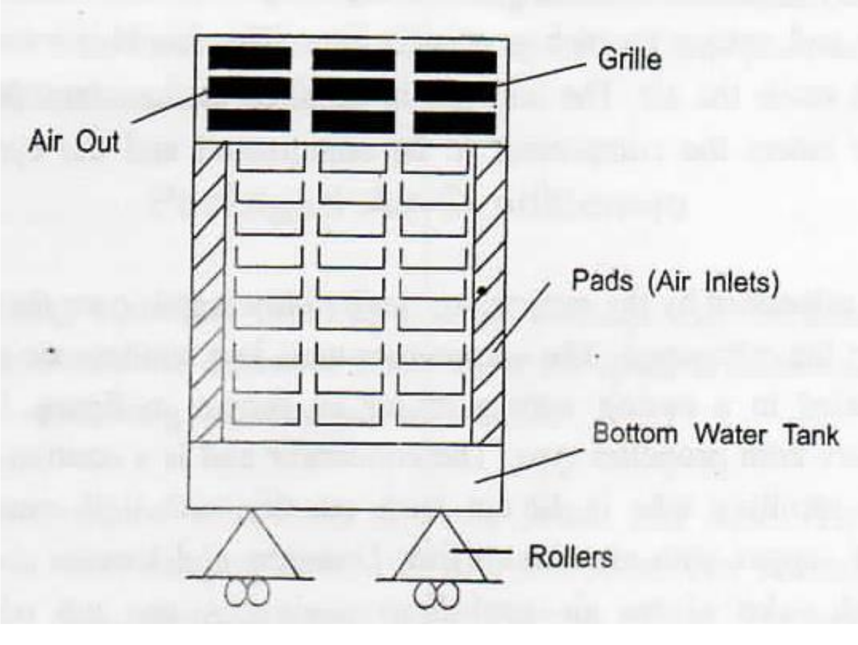


**Fig.12: Outer Details of a Desert Cooler**

The basic components of a desert cooler are fan, water pump and pads (air inlets). The air and water pump move in cross flow motion. Air is cooled by the evaporating water. The air cooling depends on relative humidity of air, air velocity and time of contact between air and water. The fan can be in vertical or horizontal plane. Figure 17.11 shows a vertical fan cooler. The pump lifts water from the bottom water tank and supplies it to the upper circular tank. The water from the upper tank then trickles through (falls through) the pads all the way up to the bottom tank. The air is sucked by the top fan from all sides of the cooler through the pads (air inlets). It gets cooled by contact with water, where heat exchange process takes place (adiabatic saturation). The cool air is discharged by the fan through the grills located on the top of the unit. Direction of air supply can be manually controlled by louvers.

Fan can be operated at variable speeds using a variable speed drive. The electric supply to fan and pump are interconnected in such a way that the fan needs to be started in order for the pump to start. Advantages of this cooler are that it is less noisy, provides better air distribution, is easy to operate and control. Disadvantages of this cooler are that it is prone to leakage, not suitable for humid regions and consumes a lot of power. Also, maintenance, like cleaning and replacement of pads, is required at regular intervals.

### Window Air-Conditioner

Figure .12 shows the schematic diagram of a window air-conditioner. The window air-conditioner works on the vapour compression refrigeration cycle. The basic components of the system are compressor, condenser, a capillary tube and an evaporator, in addition, filters, accumulators, motors, fan and electrical controls form the rest of the components. The high-pressure gas (refrigerant), which comes out of the compressor, is condensed in the condenser using ambient air as a coolant. The liquid refrigerant is expanded in a capillary tube and then enters the evaporator. Here, it evaporates and causes the refrigerating effect. The fan blows the fresh air over the evaporator coil, which cools the air. The cool air is supplied to the room. Meanwhile, refrigerant from the evaporator enters the compressor to be compressed and the cycle repeats itself.

The return air from the room is exhausted by the evaporator fan by blowing air over the condenser coil, where it picks up heat from the refrigerant. The compressor used in a window air-conditioner is hermetic type, which is sealed in a casing with a motor as shown in figure .12. The condenser and evaporator fans are both propeller type. The condenser coil is a continuous copper coil with aluminum fins. The capillary tube is 40 cm long, usually with 0.75 mm diameter. Evaporator coil is also made of copper with aluminum fins. Dampers and louvers are provided for intake and exhaust on both sides of the air-conditioner casing. A one ton window air-conditioner circulates $4m^3$/min of air in the room. Some fraction of air (25%) is-fresh, whereas some fraction (75%) of air is re-circulated. Filter are used to filter air entering the A/C unit. The filter is plastic foam or metallic mat type. The controls used by the unit are —

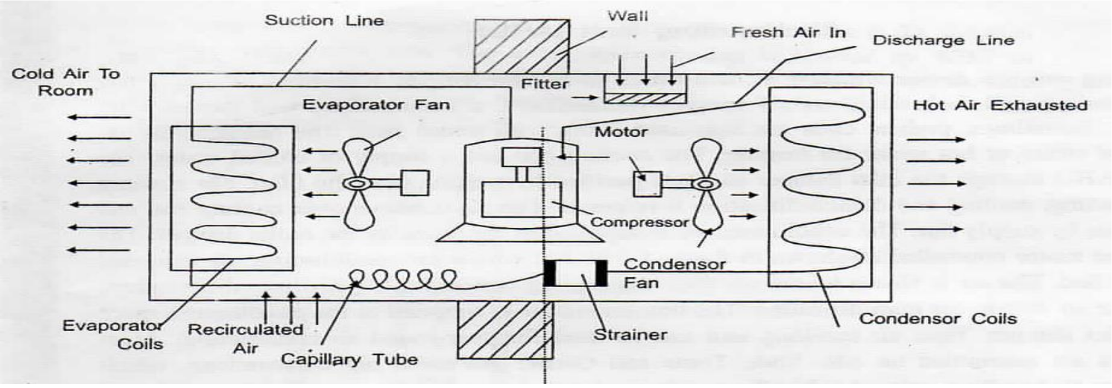


**Fig.12: Window Air-conditioner**

1] Thermostat control: This is used to set the room air temperature. Usually it is a bimetallic strip.

2] Master control: It is an electrical switch used to stop or start the compressor motor.

The installation of the unit is done either in the wall or by using external brackets or locating it inside the room with exhaust flush to the window. In either case, the condenser should be facing outside and the evaporator should be facing inside. It is advisable to install the unit for a minimum sun exposure as sun's rays can diminish the efficiency of the condenser. The air louver inlets should not be blocked and provision for draining of the condensate (like a pipe) should be provided. Maintenance of the window air-conditioner should include periodic cleaning, replacing filters and air seals and lubrication of fans.

## Packaged Air-Conditioners

These are air-conditioning units that come as a Single package. All the components are located in a single casing. These units are desirable where there are space restrictions and ducted systems with one central air-conditioning unit are not affordable.

The packaged unit is insulated with foam to avoid losses and leaks. The units are factory-assembled and come in sizes ranging from 100 to 1,000 tons. The heating and cooling coils, refrigeration equipment, fans, controls are all housed in one casing. The casing is usually made 1 decorative from aesthetic considerations.

The advantages of a packaged system are:

1] Installation and assembly labour charges are less.

2] Zoning or duct work is eliminated.

3] Units can be started or stopped easily.

4] Load factor of plant is low.

5] Maintenance is easy.

6] Individual room control is possible.

7] Failure of unit only affects one area unlike the failure of central system which affects entire building.

Disadvantages:

1] More noisy operation

2] Leakages are a problem.

3] Vibrations are more as compared to other units.

**Spilt Air-Conditioning Units**

These are preferred for offices, schools, auditoriums, etc. The idea of spilt units came from the fact that air-conditioning system operation was noisy. In order to eliminate the noise, compressor and condenser were located at a remote location and only evaporator coil and a fan were locate' near the application. This division or splitting of the air-conditioning unit gave the equipment it name. Figure 13 shows the schematic

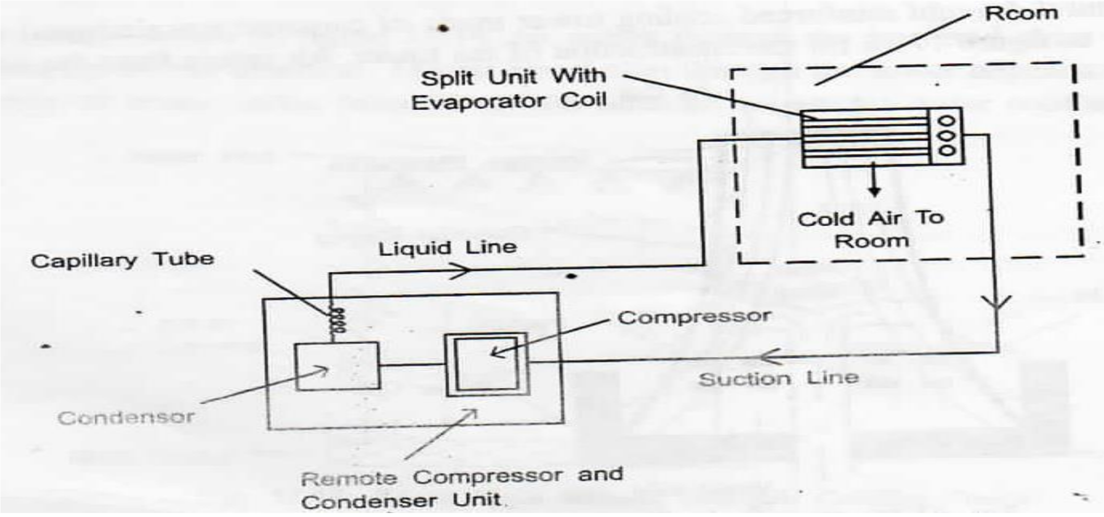


Fig. 13: Spilt Air-Conditioning Unit

Usually, spilt units are wall-mounted. They are preferred due to their noiseless operation. The refrigerant liquid lines and suction lines are long as they connect the compressor at the remote location to the spilt unit evaporator. This increases cost of the system. The operation is similar to the window air-conditioner as the basic cycle of operation is vapour compression refrigeration. The refrigerant is compressed, condensed, expanded at the remote site and evaporated only in the room evaporator. The air is blown over the evaporator coil and that causes cooling. Advantages of the unit are:

1] Noiseless operation

2] Easy maintenance

3] Less vibrations

4] Compact unit

5] Aesthetic quality of unit is high

Disadvantages of the unit are:

1] High cost of unit

2] Longer refrigerant lines and thus more leakage

3] COP of unit is poor.

4] Control over operation is difficult.

Despite drawbacks, split units are preferred more and more nowadays.

## Cooling Towers

*(MU. Dec. 2002, May 99, Nov 99, June 2002)*

A cooling tower is used to cool hot water coming out of a condenser. A 1,000 MW plant requires 100 thousand tons of cooling water. Thus cooling towers are expected to cool a large amount of hot water leaving the condenser. The cooled water can be reused or dumped to the source which may be a lake or sea.

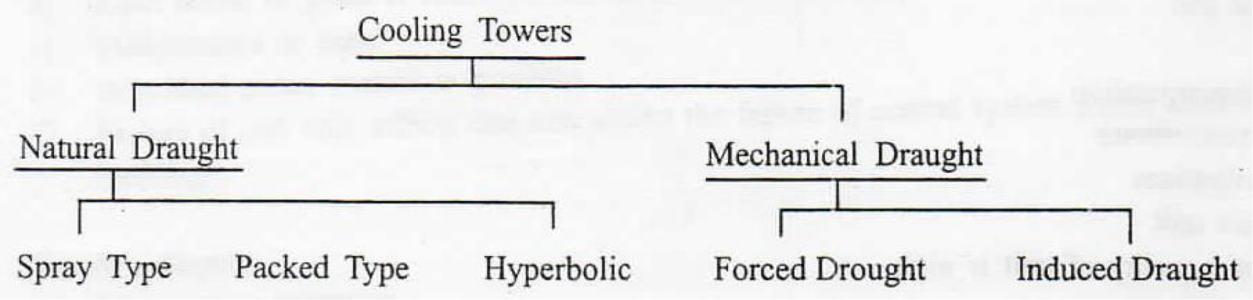
The cooling tower is a semi-enclosed device for evaporative cooling of water by contact with air. It is a wooden/steel/ concrete structure with baffles or trays to break up the water stream into droplets. The air flow to the cooling tower is from the base and water falls from the top in a spray of droplets. To prevent escape of moisture with the leaving air, eliminators are provided at the top of the tower. The air and water mix causing cooling of the water by heat exchange with air. The cooled water is collected at the base, while the air exits from the top.

Cooling towers are classified as:

Natural draught or mechanical draught cooling towers according to type of air flow in the cooling towers.

The temperature difference between WBT of incoming air and outgoing temperature of water is known as the-cooling tower approach. Lowe, coaling tower approach is an indication of effective cooling in the cooling tower. Higher the quantity of water circulated, lower is the cooling tower approach. A pump is used to circulate water and its capacity dictates the flow rate of water.

**Classification of Cooling Towers Cooling Towers**



Marley Company is a leading manufacturer of cooling towers. We shall study each cooling tower in brief. **Natural Draught Towers**

In these cooling towers, air flows naturally in and out of the tower without the aid of any fan. They are less effective than the mechanical draught towers. They are classified as spray type, packed type or hyperbolic cooling tower.

### A] Spray Type Natural Draught Cooling Tower

In this tower, as shown in figure 17.14, air enters through the louvered sides and flows across the unit in a transverse direction. The air circulation through the tower depends on wind velocity. The capacity of tower varies from 50 to 100 litres/m$^3$ minute of water cooling.

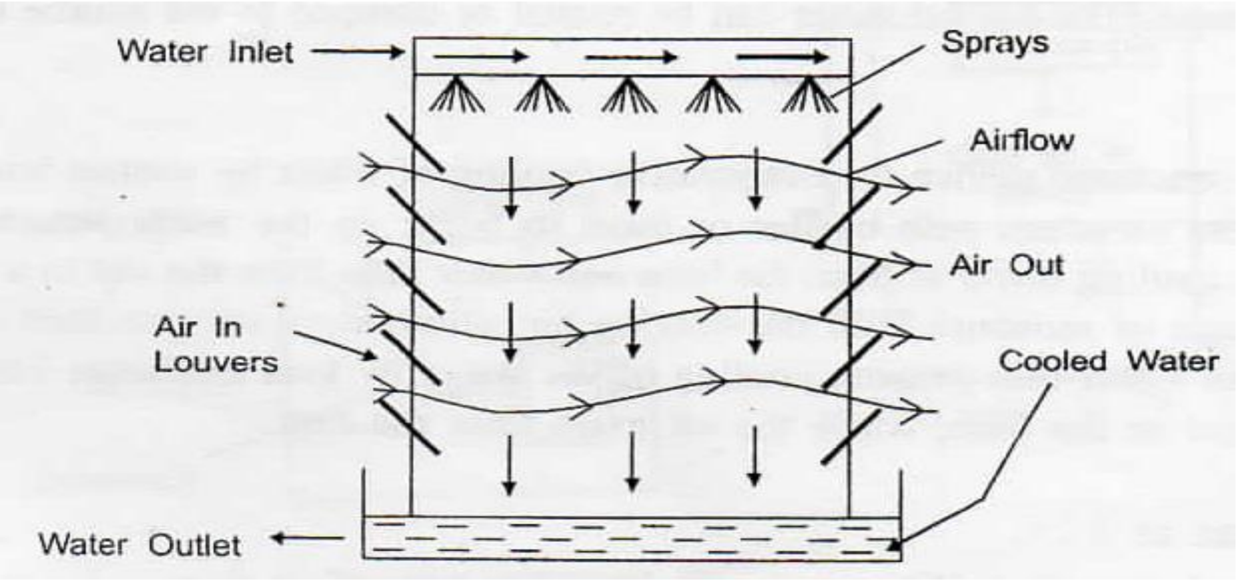


Fig. 14: Spray Type Natural Draught Cooling Tower

These towers are unsuitable for high-capacity power plants as their cooling range is limited and there is no control over outlet temperature of water. Hot water mixes with air and is cooled and finally collected at the base.

B] Packed Type Natural Draught Cooling Tower

The packed type cooling tower is similar to the spray type tower, except that water distributing packing or troughs are used to break water spray in droplets. The flow of air is crosswise to flow of water. The cold water is collected at the base. These towers are also not used in large installations due to high pumping head required.

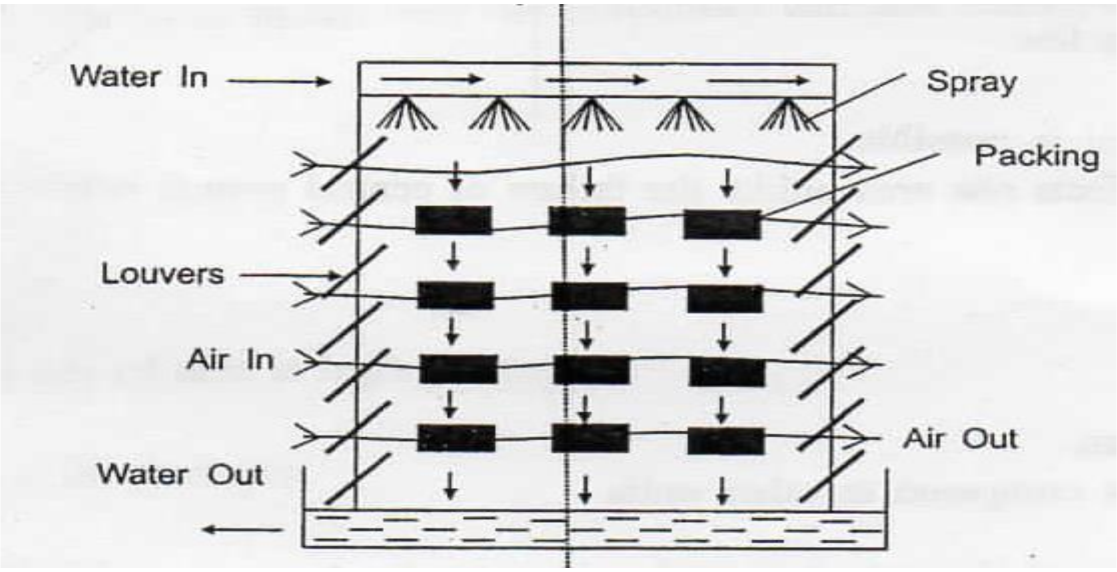


Fig. 15: Packed Tyr Natural Draught Cooling Tower

### C] Hyperbolic Cooling Tower

The first hyperbolic natural draught reinforced cooling tower made of concrete was designed and installed in 1916. Refer to figure .16 for the Construction of the tower. Air enters from the sides of the base.

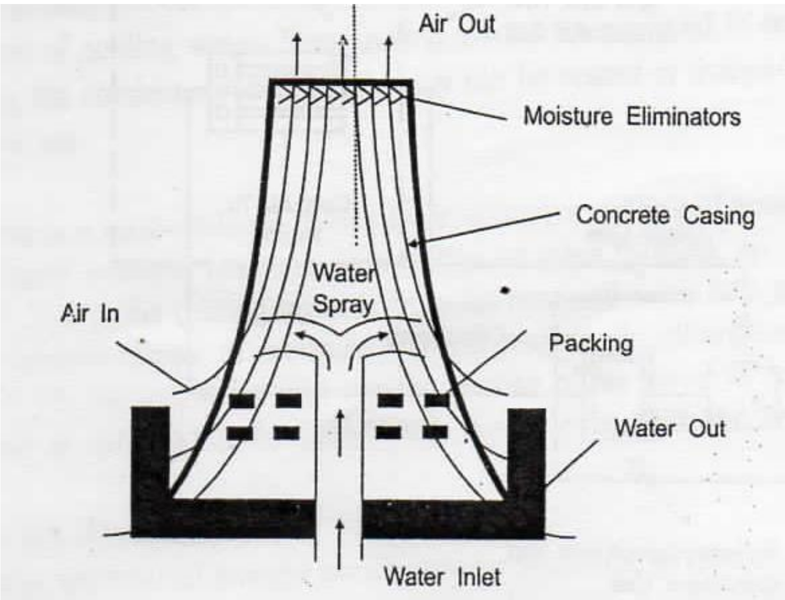


Fig. 16: Hyperbolic Cooling Tower

The tower is a steel-reinforced stack of concrete and hollow inside. It consists of a water inlet pipe and packing to break up water sprays as shown in figure .16. The shape of the stack is circular in plan and hyperbolic in profile and hence the name. Water enters through the inlet pipe and is sprayed. Sprays are broken into droplets by the packings. The heat of water is transferred to air stream. Some water evaporates causing evaporative doling effect: Warm air is exhausted from the top and cold water is re-circulated to the condenser. No fan is used, but flow of air is due to natural draught (chimney effect). The density difference of air at base and top creates the flow. Hyperbolic towers are cheap to operate and no electricity is required as fans are absent. They have long life and are self-supported structures. Ground fogging is eliminated. Drawbacks of the tower include its high initial cost, performance variation and large requirement of space due to height and width.

### Mechanical Draught Cooling Towers

Mechanics draught towers use mechanical means, such as fans, to achieve movement of air. They are more efficient, reduce wind age and spray losses and require less ground area. They require less piping and control over outlet water temperature is higher. They are preferred for large air-conditioning installations. These towers are classified as induced draught and forced draught. The induced draught is further classified as counter-flow or cross-flow cooling tower.

### A] Forced Draught Cooling Tower

Refer to figure 17 for construction of the tower.

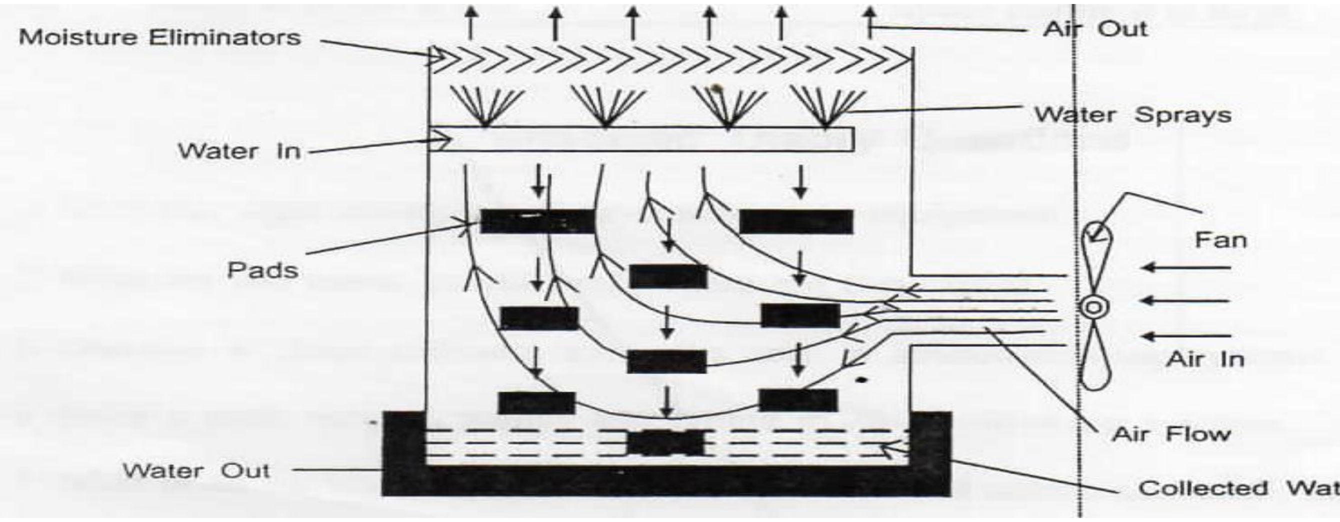


**Fig. 17: Forced Draught Cooling**

In the forced draught cooling tower, air is forced into the cooling tower by a fan as shown in figure 17. The fan is located at the base and air is blown in the upward direction. Water descends after being sprayed from a pipe. Any entrained water is removed by moisture eliminators located at the top of the tower. Water is collected at the base after being cooled by air and k removed. Hot air is exhausted from the top.

### B] Induced Draught Cooling Tower

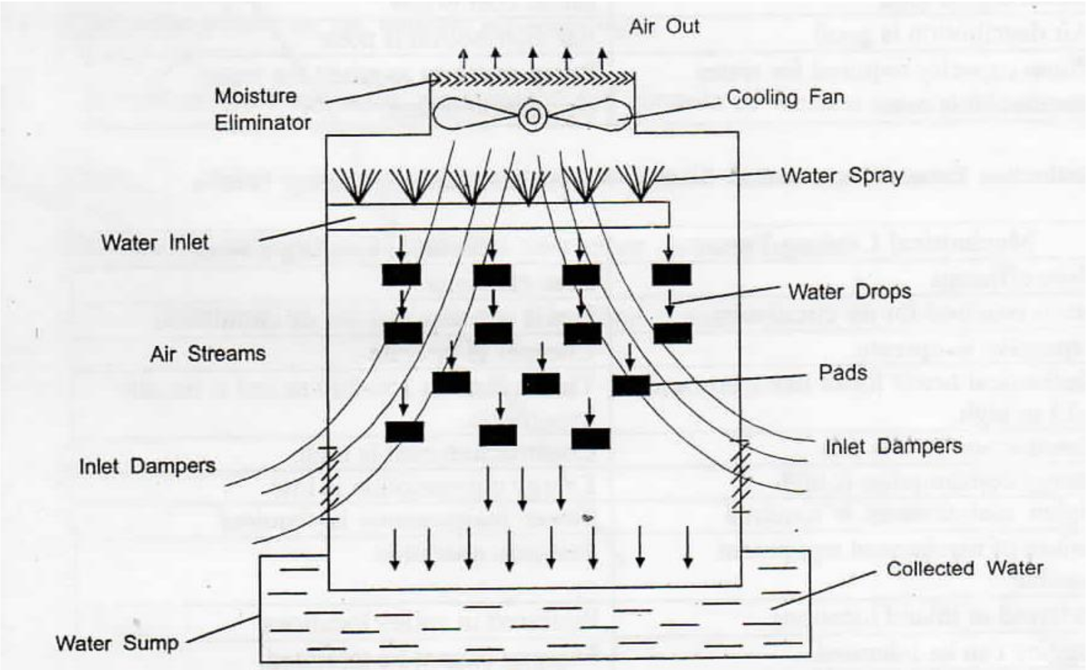


Fig. 18: induced Draught Cooling Tower

The induced draught ID tower uses induced draught fan (ID fan) to draw air into the tower. The ID fan is located at the top of the tower as shown in figure 17.18. The water is sprayed and is broken into droplets by pads and is collected at the base similar to the forced draught cooling tower. The inlet and outlet dampers control air flow. The moisture eliminators prevent escape of moisture to the environment. The static pressure loss is less in this tower. These can be counter -flow or cross-flow type, but counter-flow is more efficient as enthalpy difference is higher. The cooling of water depends on DBT, WBT of ambient air, water inlet temperature, height of tower, velocity of air, water distribution and quantity of air and type of draught used. Evaporative cooling also plays a vital role in cooling water.

### Distinction between Forced and Induced Draught Cooling Towers

| ***Sl.No.*** | ***Cooling Tower – Forced Draught*** | ***Cooling Tower – Induced Draught*** |
|---|---|---|
| 1 | More efficient | Less efficient |
| 2 | Vibration and noise minimum | Vibration and noise high |
| 3 | Fan at the base | Fan at the top |
| 4 | Blade erosion less | Blade erosion high |

|  |  |  |
|---|---|---|
| 5 | Safe operation | Unsafe operation |
| 6 | Re-circulated of air is a problem | Re-circulation of air is not a problem |
| 7 | Frost can be formed in fan housing during winter | No frost formation on fan housing |
| 8 | Fan size is limited by size of tower | Fan size is limited by weight of tower |
| 9 | Power consumption is high | Power consumption is low |
| 10 | Initial cost is high | Initial cost is low |
| 11 | Air distribution is good | Air distribution is poor |
| 12 | Pump capacity required for water circulation is more | Pump capacity required for water circulation is less. |

## Distinction between Mechanical Cooling Towers and Natural Cooling Towers

| ***Sl.No.*** | ***Mechanical Cooling Tower*** | ***Natural Cooling Tower*** |
|---|---|---|
| 1 | More efficient | Less efficient |
| 2 | Fan is required for air circulation | Fan is not required for air circulation |
| 3 | Expensive to operate | Cheaper to operate |
| 4 | Mechanical tower looks like a box and is 15m high | This is high as 150-200m and is usually hyperbolic |
| 5 | Construction cost is less | Construction cost is high |
| 6 | Energy consumption is high | Energy consumption is low |
| 7 | Higher maintenance is required | Lower maintenance is required |
| 8 | Failure of mechanical equipment possible | Fail-safe operation |
| 9 | Preferred in inland locations | Preferred in valley locations |
| 10 | Fogging can be tolerated | Fogging cannot be tolerated |
| 11 | Used for low-capacity power plants | Used for high-capacity power plants |

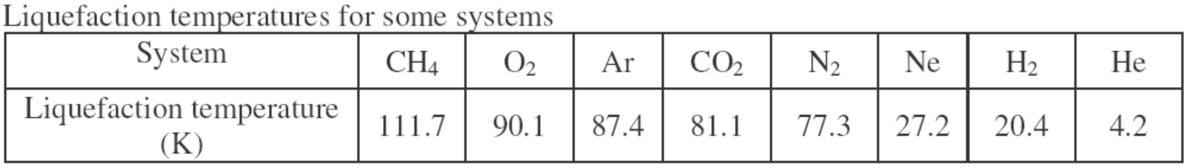
| | | |
|---|---|---|
| 12 | Space required is less | Space required is more. |

# UNIT V

# CRYOGENICS

## INTRODUCTION

The science and technology of producing a low-temperature environment is generally referred to as cryogenics. It is perhaps appropriate to start with some definitions. The word cryogenics has its origin in the Greek language where "cryos" means frost or cold and "gen" is a common root for the English verb to generate. Strictly speaking, cryogenics means to produce cold, yet the term has developed a more general connotation over years of usage by engineers and scientists. Today, the word cryogenics is associated with the production and study of low-temperature environments. Thus, a cryogenic engineer is a person who specializes in these areas. The expertise of a cryogenic engineer can vary considerably within this discipline. Expertise in cryogenic engineering is in demand in a wide variety of technical fields including advanced energy production and storage technologies, transportation and space programs, and a wide variety of physics and engineering research efforts. Over the years the word cryogenics has developed several common usages. A cryogenic fluid is one that is used in the production of cold, while cryogenic machinery is the hardware used in achieving low-temperature environments. At first it would appear that all machinery and fluids used in cooling would be identified as cryogenic. Generally, however, it is accepted that the word cryogenics is reserved for those processes that take place below about 100-150K. This distinction is established because it represents the point where permanent gases such as N2 and O2 begin to liquefy. Sometimes, cryogenics is used in reference to liquid natural gas and its liquifaction technology.



Liquefaction temperatures for some systems

| System | $CH_4$ | $O_2$ | Ar | $CO_2$ | $N_2$ | Ne | $H_2$ | He |
|---|---|---|---|---|---|---|---|---|
| Liquefaction temperature (K) | 111.7 | 90.1 | 87.4 | 81.1 | 77.3 | 27.2 | 20.4 | 4.2 |

## PRODUCTION OF LOW TEMPERATURES

### Joule – Thomson Effect

Most of the practical liquefaction systems utilize an expansion valve or Joule – Thomson value to produce low temperatures. If we apple the first law for steady flow to the expansion valve shown in Fig., k zero heat transfer and zero work transfer and for negligible kinetic – and potential – energy changes, we find that h1 = h2. Although the flow with the valve is irreversible and is not an isenthalpic process, the inlet an outlet states do lie on the same enthalpy curve. We could plot a series point of outlet conditions for given inlet conditions and obtain lines constant enthalpy. For a real gas, such a plot is shown in Fig.. It note that there is a region in which an expansion through the valve (crease in pressure) produces an increase in temperature, while in another region the expansion results in a decrease in temperature. Obviously, should want to operate the expansion valve in a liquefaction system in the region where a net decrease in temperature results. The curve with separates these two regions is called the inversion curve.

The effect of change in temperature for an isenthalpic change in pressure is represented by the Joule – Thomson coefficient $\mu_{JT}$, defined by

$$\mu_{JT} = \left(\frac{\partial T}{\partial p}\right)_h$$

Where the derivative is interpreted as the change in temperature due to the change in pressure at constant enthalpy. Note that the Joule – Thomson coefficient is the slope of the isenthalpic lines in Fig.. The Joule – Thomson coefficient is zero along the inversion curve, since a point of the inversion curve is one at which the slop of the isenthalpic line is zero. For a temperature increase during expansion, the Joule – Thomson coefficietion is negative; for a temperature decrease, the Joule – Thomson, coefficient positive.

From calculus, it can be shown that

$$\mu_{JT} = \left(\frac{\partial T}{\partial p}\right)^{h} = -\left(\frac{\partial T}{\partial h}\right)_p \left(\frac{\partial h}{\partial p}\right)_T$$

From basic thermodynamics (1), it can be shown that

$$dh = \left(\frac{\partial h}{\partial T}\right)_p dT + \left(\frac{\partial h}{\partial p}\right)_T dp = c_p dT + \left[v - T\left(\frac{\partial v}{\partial T}\right)_p\right] dp$$

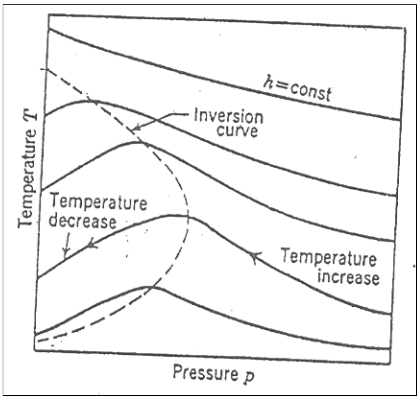
Where v is the specific volume of the material. By comparison of the coefficients of dT and dp in Eq. we see that

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p \quad \text{and} \quad \left(\frac{\partial h}{\partial p}\right)_T = v - T\left(\frac{\partial v}{\partial T}\right)_p$$

By combining the above Eqs. the Joule – Thomson coefficient may be expressed in terms of other thermodynamic properties as

$$\mu_{JT} = \frac{1}{c_p}\left[T\left(\frac{\partial v}{\partial T}\right)_P - v\right] = 0$$

A perfect gas would not experience any temperature change upon expansion through an expansion valve. To the gratification of cryogenics engineer gases are imperfect at low enough temperatures and high enough pressure.



We have seen that a perfect gas always has a zero Joule – Thomson coefficient; therefore, a positive or negative Joule – Thomson arise from the departure of negative Joule – Thomson coefficient varies from the departure of real gases from the perfect – gas behavior Enthalpy is defined as

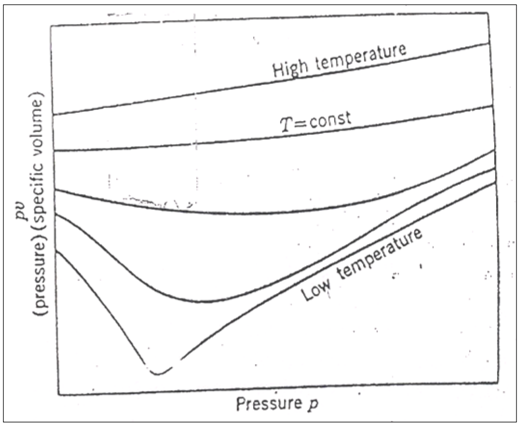
$$h = u + pv$$

Where u is the internal energy of the substance. Making this substitution in Eq.

$$\mu_{JT} = -\frac{1}{c_p}\left\{\left(\frac{\partial u}{\partial p}\right)_T + \left[\frac{\partial(pv)}{\partial p}\right]_T\right\}$$

The first term in above Eq. represents a departure from Joule law, which states that the internal energy of a gas is a function of temperature alone. If u = (T) = $c_v$ T, for example, then this term is zero. This term is always negative for real gases; thus it contributes to the production of a temperature decrease (positive $\mu_{JT}$). As molecules on moved farther apart in reducing the pressure in the expansion process their microscopic potential energy is increased. No external work or heat is added; therefore, this increase in microscopic potential energy must be offset by a decrease in microscopic kinetic energy. Temperature is one measure of microscopic kinetic energy, and any decrease in microscopic kinetic energy results in a decrease in temperature.

On the other hand, the second term in above Eq. may be either positive, negative, or zero. The second term represents a departure from Boyle’s law, which states that the product of pressure and volume for a gas is a function of temperature alone. If pv = f(T) = RT, for example then this term is always zero. A sketch of $p_v$ as a function of pressure is shown in below Figure. At low pressures and temperatures near the saturated vapor conditions, gases are more compressible than Boyle’s law predicts, since attractive forces are in action which try to condense the gas. This means that the second term in above Eq. is negative and contributes to the production of a temperature increase upon expansion. Whether the joule – Thomson coefficient is positive negative or zero for a real gas depends upon the relative magnitude of the terms.



One equation of state for gases which illustrates the behavior of real gases is the van der Waals equation of state,

$$\left(p+\frac{a}{v_2}\right)(v-b)=RT$$

Where a is a measure of the intermolecular forces and b is a measure of the finite size of the molecules. For a perfect gas, a = b = 0, since a perfect gas has no intermolecular forces, and the molecules are considered to be mass points with no volume. A van der Waals gas has molecules which are considered to be weakly attracting rigid spheres. If we calculate the Joule – Thomson coefficient for a van der Waals gas from Eqs. (3-12) and (3-15), we obtain

$$\mu_{JT}=\frac{(2a/RT)(1-b/v)^2-b}{cp\left[1-(2a/v)(1-b/v)^2\right]}\text{van der Waals gas}$$

For large values of the specific volume, Eq. (3-16) can be approximated by

$$\mu_{JT}=\frac{1}{c_p}\left(\frac{2a}{RT}-b\right)$$

Which shows that at some temperatures $\mu_{JT}$ is positive (for T < 2a/bR) and at other temperatures $\mu_{JT}$ is negative (for T > 2a/bR).

The inversion curve is represented by all points at which the Joule – Thomson coefficient is zero. For a vander Waals gas, this happen when (2a/RT) (1-b/v)2 = b, as can be seen from Eq. Solve for the inversion temperature T, for a van der Waals gas.

$$T1-\frac{2a}{bR}\left(1-\frac{b}{v}\right)^2$$ Van der Waals gas

Which can be related to the pressure through the equation of a state. The inversion curve at p=0,or $T_{imax}$ – 2a/bh maximum inversion temperatures for some gases are given in table .

We observe that all gases may be separated into two groups those for which the maximum inversion temperature is greater than room temperature which includes most gases; and those for which the maximum inversion temperature is below ambient temperature, which includes neon hydrogen, and helium. This difference in the level of the maximum version temperature determines whether a liquefaction system may use the Joule – Thomson effect alone in producing low temperatures or whether other methods, such as expansion engines or pre cooling, must also be use.

## ADIABATIC EXPANSION

The second method of producing low temperatures is the adiabatic expansion of the use gas through a work producing device, such as an expansion engine. In the idel case, the expansion would be reversible and adiabatic and therefore isentropic. In this case we can define the isentropic expansion coefficient use which express the temperature change due to a pressure change at constant entropy.

$$\mu_s=\left(\frac{\partial T}{\partial P}\right).$$

The isentropic expansion coefficient can be related to other properties of a gas in a manner similar to the one we used with the joule- Thomson coefficient.

$$\mu_s=\left(\frac{\partial T}{\partial P}\right)=-\left(\frac{\partial T}{\partial s}\right)_p\left(\frac{\partial s}{\partial P}\right)_T=+\frac{T}{C_p}\left(\left(\frac{\partial v}{\partial T}\right)\right)$$

The second factor in Eq (3-20) is the volumetric coefficient of thermal expansion multiplied by the specific volume, or $\beta v$, so $\mu_s$ is positive (a temperature decrease results from a pressure decrease) when the volumetric coefficient of expansion is positive. This is the case for all gases (although some substances, such as liquid water between 32 and 39˚F, have negative coefficients of expansion). For a perfect gas, $(\partial v/\partial T)_p=R/p=v/T$, so

$$\mu_e=\frac{v}{cp}$$ for a perfect gas

For a van der Waals gas, one can show that

$$\mu s=\frac{v}{c_p}\frac{1-b/v}{cp1-(2a/vRT)(1-b/v)^2}$$ van der Waals gas

Which is positive, since v > b.

We can make the observation that, for a gas, an isentropic expansion through an expander always results in a temperature decrease, whereas an expansion through an expansion valve may or may not result in a temperature decrease. The isentropic expansion process removes energy form of external work, so this method of low-temperature production is sometimes called the external – work method. Expansion through an expansion valve does not remove any energy from the gas, but moves the molecules farther apart under the influence of intermolecular forces, so this method is called the internal – work method.

It appears that expanding the gas through an expander would always be the most effective means of lowering the temperature of a gas, and cms is the case as far as the thermodynamics of the two process concerned. Between any two given pressures, an isentropic expander will always result in a lower final temperature than will an isenthalpic expansion from the same initial temperature. The practical problem associated with the expansion of a two – phase mixture in an expander make the use of an expansion valve necessary in all liquid faction systems.

## CRYOGENIC LIQUEFACTION SYSTEMS

### Introduction

System Performance Parameters

There are three payoff functions which we might use to indicate the performance of a liquefaction system:

1. Work required per unit mass of gas compressed – w/m
2. Work required per unit mass of gas liquefied – W/mf
3. Fraction of the total flow of gas which is liquefied, y = mf/m

The last two payoff functions are related to the first one by

$$\frac{-W}{m} = \frac{-W}{m_f} y$$

In any liquefaction system, we should want to minimize the work requirements and maximize the fraction of gas which is liquefied.

These payoff functions are different for different gases; therefore, we should also need another performance parameter which would allow the comparison of the same system using different fluids. The figure of merit for a liquefaction system is such a parameter. It is defined as the theoretical minimum work requirement divided by the actual would requirement for the system:

$$FOM = \frac{Wi}{W} = \frac{-W_i / m}{-W / m_f}$$

The figure of merit is a number between 0 and 1, and it gives a measure of how closely the system approaches the ideal operation.

There are several performance parameters which apply to the components of real systems. These include:

1. Compressor and expander adiabatic efficiencies
2. Compressor and expander mechanical efficiencies
3. Heat – exchanger effectiveness
4. Pressure drops through piping, heat exchangers, etc.
5. Heat transfer to the system from ambient surroundings

In our discussions of system performance, we shall not contain these component factors first but shall return to them when we disc the major components of the systems. Thus we shall first assume the all efficiencies and effectiveness's are 100 percent and that ire event pressure drops and heat in leaks are zero.

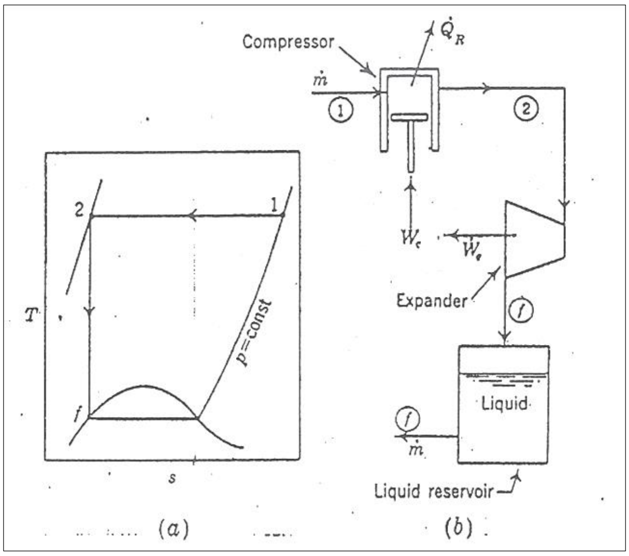
## The Thermodynamically Ideal System

In order to have a means of comparison of liquefaction through the figure of merit, we shall first analyze the thermodynamic ideal liquefaction system. This system is ideal in the thermodynamic sense, but it is not ideal as far as a practical system is concerned, as we shall see later. The perfect cycle in thermodynamics is the Carnot cycle. Liquefaction is essentially an open – system process; therefore, for the ideal liquefaction system, we shall choose the first two processes in the Carnot cycle: a reversible isothermal compression followed by a reversible isentropic expansion. The ideal cycle is shown on the T-s plane in fig. along with a schematic of the system.

The gas to be liquefied is compressed reversibly and isothermally from ambient conditions (point 1) to some high pressure (point 2). This high pressure is selected so that the gas will become saturated liquid upon reversible isentropic expansion in the expander (point f). The final condition at point f is taken at the same pressure as the initial pressure at point 1.

The pressure attained at the end of the isothermal compression is extremely high – on the order of $10^7$ psia for nitrogen. It is impossible to attain this pressure with existing compression equipment, which is the reason that it is not an ideal process for a practical system.

In the analysis of each of the liquefaction systems, we shall apply the first law of thermodynamics for steady flow, which may be written in general as where $Q_{net}$ is the net heat transfer to or from the system (heat transfer to the system is considered positive), $W_{net}$ is the net work done on a the system (work done by the system is considered positive), and the mation signs imply that we add the enthalpy terms (h), kinetic – enters (v2/2ge, where v is the fluid velocity and $g_c$ is the conversion of in Newton's second law; $g_c$ = 32.174$lb_m$-ft/lbf-sec$^2$ or 1 g-cm/ dyne-sec and potential – energy terms (zg/$g_c$ where z is the fluid elevation above datum plane and g is the local acceleration the to gravity; g=32 ft/sec$^2$ or 980 cm/ sec$^2$ at sea level) for all inlets and outlets of the system.



**The thermodynamically ideal liquefaction system**

$$Q_{net} - W_{net} = \sum_{outlets} m\left(h + \frac{v^2}{2g_c} + \frac{zg}{g_c}\right)$$

A system might consist of several different streams, which would result more than just one inlet and one outlet. In all our system analysis shall assume that the kinetic – and potential – energy changes are smaller than the enthalpy changes (not a bad assumption), and the energy terms may be neglected. Thus, in our special case, the first law steady flow may be written

$$Q_{net} - W_{net} = \sum_{outlete} mh - \sum_{inlete} mh$$

Applying the first law to the system shown in Fig. 3-1,

$$Q_R - W = m\left(h_f - h_1\right) = mT_1\left(h_1 - h_f\right)$$

The heat – transfer process is reversible and isothermal in the Carnot cycle thus

$$Q_R = mT_1\left(s_2 - s_1\right) = -mT_1\left(s_1 - s_f\right)$$

Since the process from point 2 to point f is isentropic, $s_2 = s_f$, where the entropy of the fluid. Substitution $Q_R$ from Eq. into Eq. we may determine the work requirement per unit mass compressed:

$$\frac{-W_1}{m} = T_1\left(s_1 - s_f\right) - \left(h_1 - h_f\right) = \frac{-W_1}{m_f}$$

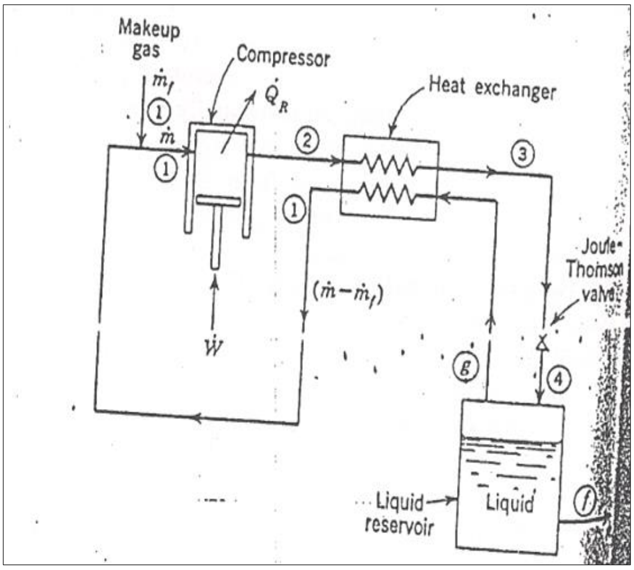
In the ideal system, 100 percent of the gas compressed is liquefied, m = mf, so that y = 1. Notice that a liquefaction system is a absorbing system; therefore, the net work requirement is negative (won done on the system), and the term – W1/m is a positive number.

Equations gives the minimum work requirement work requirement to liquefy a gas, so it is the value which we should try to approach in any practical system. Since was have made the final pressure at point 1, and point f is on the saturated – liquid curve, the ideal – work requirement depends only upon the pressure and temperature at point 1 and the type of gas liquefied. Ordinarily, we take point 1 at ambient conditions. Table 3-1 lists ideal – work requirements for some common gases, which point 1 taken as 1 atm (14.7 psia) and 70˚F.

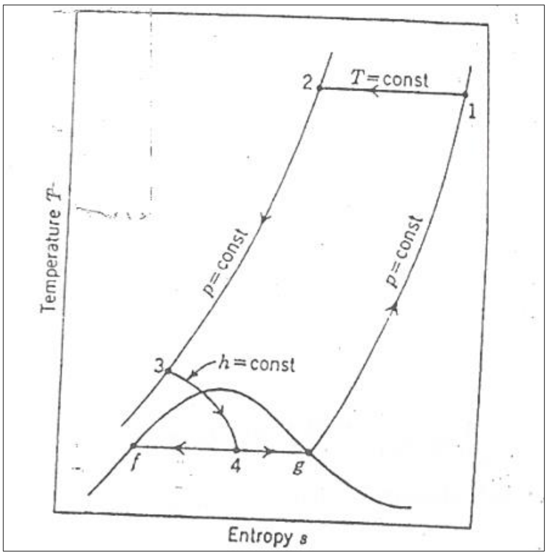
## Simple Linde – Hampson System

The Linde – Hampson system was the second used liquefy gases (the cascade system was the first), although it is the simple of all the liquefaction systems. A schematic of the Linde – Hampson system is shown in Fig. and the cycle is shown on the T-s plane in Fig.

In order to analyze the system, let us assume ideal condition no irreversible pressure drops (except for the expansion valve), no has in leaks from ambient, and 100 percent effective heat exchanger. This first compressed from ambient conditions at point 1 reversibly and isothermally to point 2. In a real system, Process 1 to 2 would actually be two process: an irreversible adiabatic or polytrophic compression followed by an after cooling to lower the gas temperature. The gas next passes through a constant pressure heat exchanger (ideally) in which it exchanges energy with the outgoing low – pressure stream to point3. From point 3 tp print 4, the gas expands through an expansion value to $p_4 = p_1$. At point 4, some of the gas stream is in the liquid state and is withdrawn at condition f( saturated – liquid condition). This cold gas is finally warmed to the initial temperature by absorbing energy at constant pressure (ideally) from the incoming high – pressure stream.



**Simple Linde – Hampson System**



T-S Diagram

Applying the first law for steady flow to the combined heat exchanger, expansion value, and liquid receiver obtain.

$0= (m-m_f)\, h_1+m_f h_f - m h_2$

Since there is no work or heat transfer to or from the surrounding for since these components. Solving for the fraction of the gas flow which is liquefied.

$$\frac{m_f}{m} = g = \frac{h_1 - h_2}{h_1 - h_f}$$

The fraction of gas liquefied thus depends upon (1) the pressure and temperature at ambient condirions (point 1 ), which fix $h_1$ and $h_f$; and (2) the pressure after the isothermal compression, which determines since the temperature at state point 1.

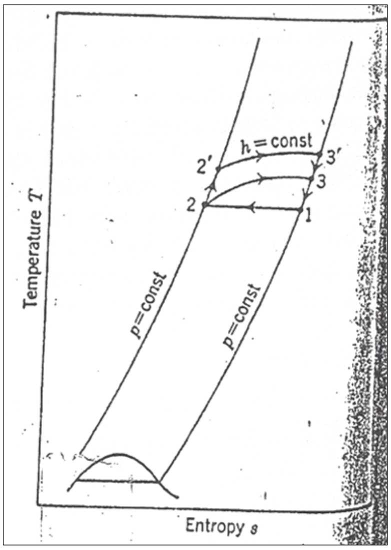
We do not have much freedom in choosing or varying ambient conditions; therefore points 1 and f are practically prescribed, and we at liberty to vary the performance of the system by varying the pressure at point 2 only. What would be the best pressure to pick for p2 from thermodynamic view point? Obviously, from a consideration of Eq. (3- 25) the best or optimum pressure at point 2 would be the one which minimize. Mathematically, for minimum $h_2$ we must have

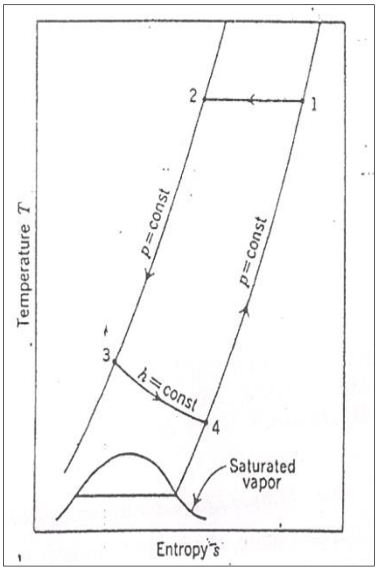
$$\left(\frac{\partial h}{\partial P}\right)_{T-T1} = 0$$

Referring to Eqs. (3-9) and (3-11), we see that this is equivalent to saying

$$\left(\frac{\partial h}{\partial P}\right)_{T-T1} = 0 = \mu_{[TCP]}T - T_1$$

Or that the pressure which minimizes h2 is the pressure for which $\mu_{JT} = 1$ or a temperature $T_1$. For optimum performance (maximum liquid yiek) of the linde – Hampson system, state point 2 should lie on the inversed. For air at 70˚F, the corresponding pressure is approximate





400 atm (5880 psia); however, actual systems commonly utilize pressures on the order of 200 atm.

The simple linde – hampson system shown in Fig. 3-5 would not work for such gases as neon, hydrogen, and helium for two reasons. First, the system could never get started, since the maximum inversion temperature for these gases is below room temperature. Referring to the T-s diagram of helium, for example we see that in first starting the system, there would be no heat exchange and points 2 and 3 would coincide. Expansion through the expansion value from the point 3 to point 4 would result in an increase in temperature, so that as the operation progressed, the gas entering the heat exchanger would be continually warmed rather than cooled; therefore, we could produce low temperatures. This is illustrated in Fig. 3-7. In the second place, from Eq. (3-23) we see that y is negative as long as h1 is smaller than h2, which is the case for helium., hydrogen, and neon at room temperature. This means that even if we could attain low temperatures, with the linde- Hampson system, the expansion through the expansion value at low temperatures would pass completely into the vapor region, and no gas would be liquefied. Beginning at ambient conditions, not enough energy can be exchanged in the heat exchanger to lower the gas temperature to the point at which liquid could be obtained after divided the liquefaction systems into two categories.

The work requirement for the linde- hampson system may be determined by applying the first law for steady flow to the isothermal compressor shown in

$$Q_R - W = m\,(h_2 - h_1)$$

Substituting for the reversible isothermal heat transfer from Eq.(3-6)

$$\frac{-W}{m} = T1(s_1 - s_2) - h_1 - h_2$$

The work requirement per unit mass liquefied is

$$\frac{-W}{m_f} = \frac{-W}{m}\frac{1}{y} = \frac{h_1 - h_f}{h_1 - h_2}\left[T1(s_1 - s_2) - h_1 - h_2\right]$$

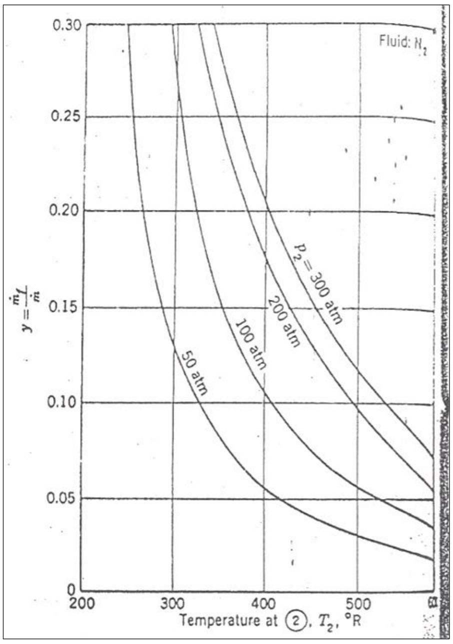
**Precooled Linde – Hampson System**

A plot of liquid yield y as a function of the temperature at the entrance of theheat exchanger(Point 2) for the simple Linde- Hampson system is shown in Fig. 3-9. It is apparent that the performance of a Linde – Hampson system could be improved if it were modified so that the gas entered the heat exchanger at a temperature lower that ambient. Such a modified ststem is shown in Fig 3-10, and the cycle is shown on the T-s Plane in Fig. A separate refrigeration system using a fluid such as carbon dioxide, ammonia, or a Freon compound is used to cool the main gas stream. The critical temperature of the auxiliary refrigerant must be above ambient temperature in order that the refrigerant can be condensed by exchanging heat with the atmosphere or with cooling water at ambient temperature.

For a 100 Percent effective heat exchanger, the temperatures at points 3 and 6 (see Fig 3-11) are the same. From a consideration of the second law of thermodynamics, T3 and T6 cannot be lower than the boiling point of the auxiliary refrigerant at point d; otherwise, we should be transferring heat "uphill" – from a low to a high temperature – without expending any work. These factors place restrictions on the performance of the precooled Linde – Hampson system.

Applying the first law for steady flow to the combined three – channel heat exchanger, primary – gas heat exchanger, liquid receiver and expansion value, and auxiliary refrigerant expansion value, for no work or heat transfers from the surroundings to these components,

$$0 = (m - mf)\, h_1 + m_r h_a + m_f h_f - mh2 - m_r h_d$$



If we define the refrigerant flow rate ratio t as

$$r= \frac{m_r}{m}$$

where $m_r$ is the mass flow rate of the auxiliary refrigerant, then Eq. (3-29) may be written as follows, solving for the liquid yield $y= m_{f/m.}$

$$y=\frac{h_1-h_2}{h_1-h_f}+r\,\frac{h_a-h_c}{h_1-h_f}$$

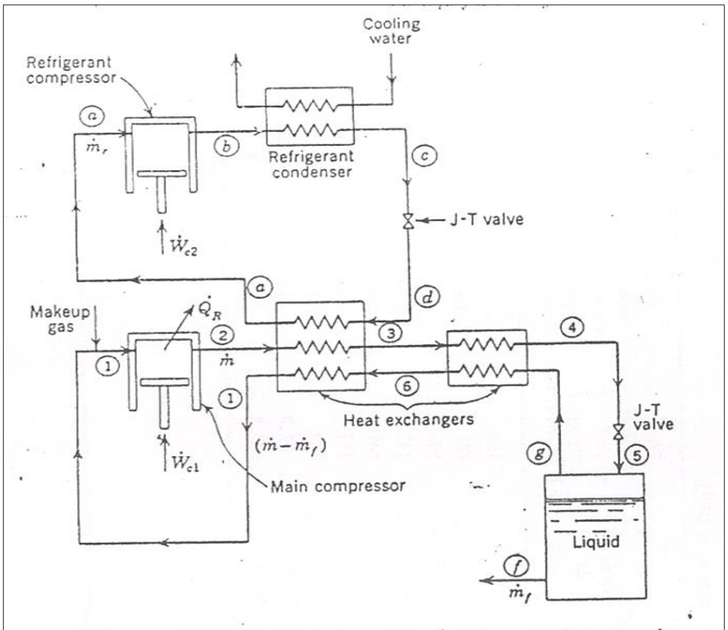
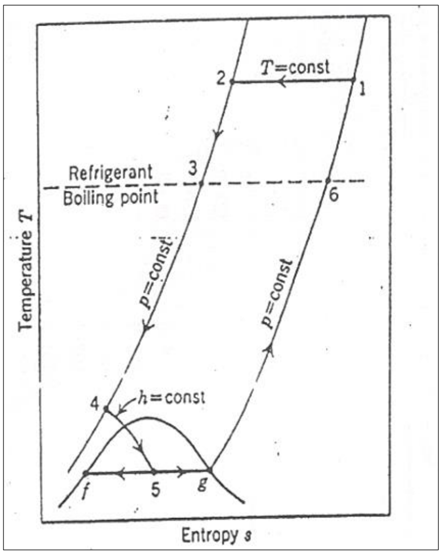


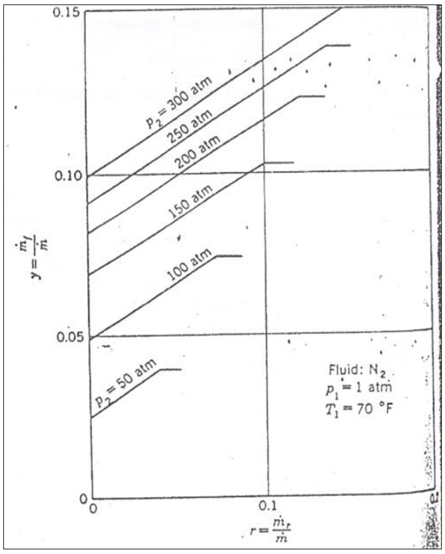
*Fig. 3-10* Precooled Linde-Hampson system. The precoolant may be any fluid which condenses at ambient temperature under pressure.

The first term on the right side of Eq. (3-31) represents liquid yield for the simple lindle – Hampson system operating under same conditions as the precooled system [refer to Eq. (3-23)]. Second term represents the improvement in liquid yield which is obtained through the use of precooling. As mentioned before, the temperature the auxiliary refrigerant does limit the liquid yield; otherwise, with a suitable value of the refrigerant flow – rate ratio r, Eq(3-31) could yield a value of 100 percent for the liquid yield y. the maximum liquid yield for the precooled system is



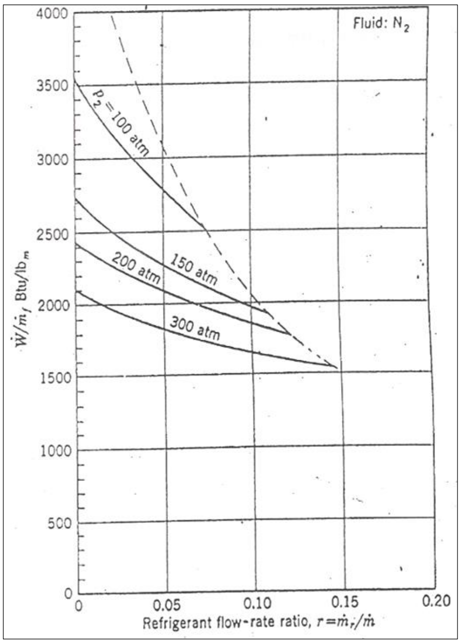
$$y_{max} = \frac{h6-h_3}{h6-h_f}$$

Where h3 and h6 are taken at the temperature of the boiling refrigerant at point d. another limiting factor is that if the refrigerant flow – rate ratio were too large, the liquid at point d would not be completely vaporized and liquid would enter the refrigerant compressor – not a very desirable



Situation. The variation of the liquid yield and the limiting liquid yield is shown. In Fig. 3-12 for nitrogen gas as the working fluid and Freon – 12 as the refrigerant, operating from 1 atm and b70F to 5.77 atm.

If the main compressor is reversible and isothermal and the auxiliary compressor is reversible and adiabatie the work requirement per unit mass of primary gas compressed is



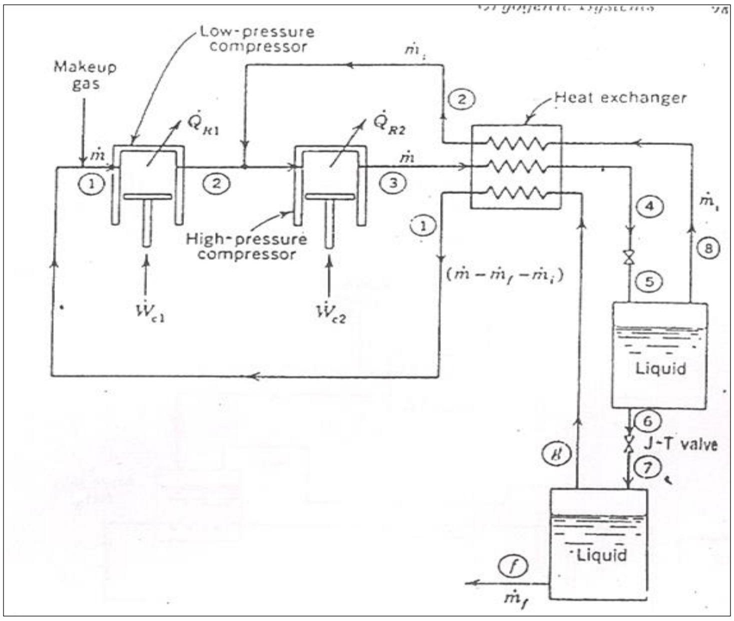
$$\frac{-W}{m} = T_1(S_1 - S_2) - (h_1 - h_2) + r(h_6 - h_a)$$

The last term represents the additional work requirement for the auxiliary compressor; therefore, the total work requirement for the precooled linde- Hampson system is greater than that for the simple system. The last term is usually approximately 10 percent, of the total work. The increase in liquid yield more than offsets the additional work requirement, however so that the work requirement per unit mass of gas liquefied is actually less for the precooled system than for the simple system, as indicated in Fig 3-13.

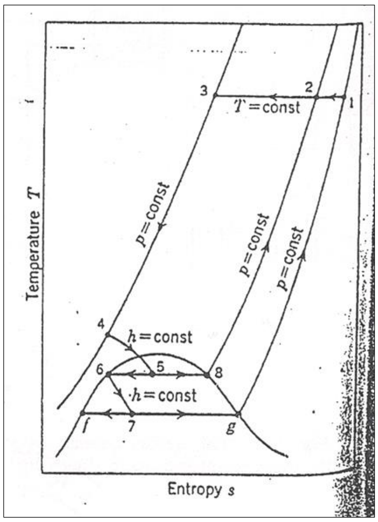
The simple Linde – Hampson system can be modified in another way to reduce the total work required, although this modification reduces the liquid yield somewhat. Since only a small portain of the gas which is compressed is liquefied in the simple system, we might modify it so that not all the gas is expanded to the lowest pressure but some is expanded to an intermediate pressure. The work requirement for an ideal isothermal compressor and a perfect gas would be $RT_1$ In $(p_2/p_1)$, so a reduction in the compression pressure ratio would decrease the work requirement. This is accomplished in the Linde dual – pressure system shown schematically in Fig 3-14. The cycle is Shown on the T-s Plane in Fig 3-15.

The gas is compressed first to an intermediate pressure and then to the high pressure of the cycle after a return stream has been added. The high – pressure gas is passed through a three – channel heat. Exchanger and expanded to the intermediate pressure at point 5, where some of the gas is liquefied. The saturated liquid and vapour are separated in a liquid receiver, and the vapor is returned to the second compressor through the three – channel heat exchanger while the liquid is expanded further to the low pressure of the cycle.

Applying the first law for steady flow to the exchanger the two liquid receivers, and the two expansion valves, we can determine the liquid yield for the Linde dual – pressure system:



**Schematic of the Linde dual – pressure system**



$$y = \frac{h_1 - h_2}{h_1 - h_f} - i\frac{h_i - h_2}{h_1 - h_f}$$

Where mi is the mass flow rate of the intermediate pressure stream at point 8 and m is the total mass flow rate through the high – pressure compressor. The second term represents the reduction in the liquid yield below that of the simple system because of splitting the flow at the intermediate – pressure – liquid receiver.

Applying the first law for steady flow to the two compressors, we find that the work requirement per unit mass of gas compressed in the high-pressure compressor is From Eq. (3-36) we see that the work requirement is reduced below that of the simple system by the amount of the second bracketed term.

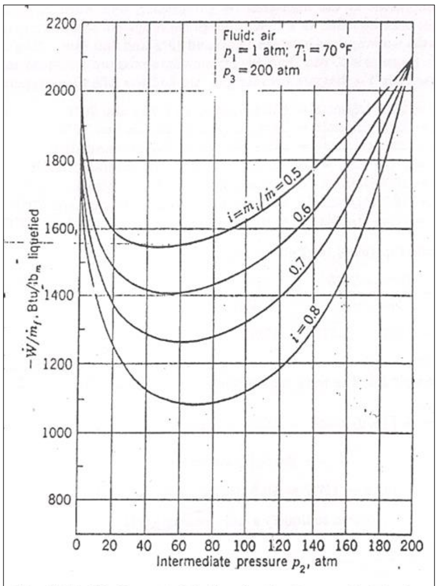


*Fig. 3-16* Work required to liquefy a unit mass of air in the Linde dual-pressure system.

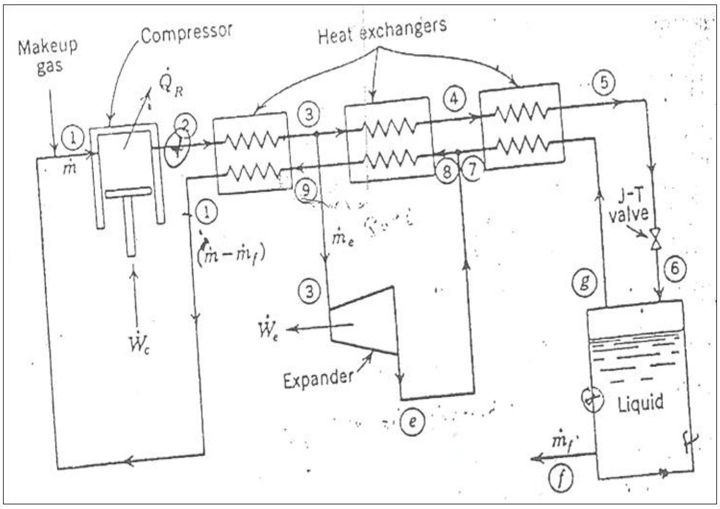
## Work required liquefying a unit mass of air in the Lind dual – pressure system

Practical liquefaction plants usually operate with I on the order of 0.8, so that the reduction in work requirement more than offsets the reduction in liquid yield; therefore, as for the precooled system, the work requirement to produce a unit mass of liquid is less for the dual - pressure system than for the simple system. Work requirements for air are illustrated in Fig. 3-16.

## Claude System

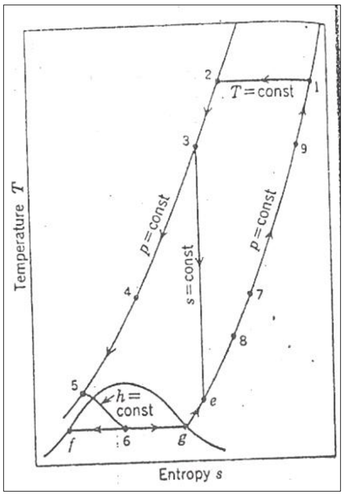
The expansion through an expansion valve is an irreversible process, thermodynamically speaking, so if we wish to come closer to the ideal performance, we must seek a better process to produce low temperatures. In the Claude system, shown in Fig. 3-18, energy is removed from the gas stream by allowing it to do some work in an expansion engine or expander. The Claude cycle is shown on the T-s plane in fig. 3-19. If the expansion engine is reversible and adiabatic (which we shall assume to be true for this analysis), the expansion process is isentropic, and a mush lower temperature is attained than for an isenthalpic expansion, as we saw in Sec. 3-4.

In the Claude system, the gas is first compressed to pressures on the order of 40 atm and then passed through the first heat exchanger. Approximately 80 percnt of the gas is then diverted from the mainstream, expanded through an expander, and reunited with the return stream below the second heat exchanger. The stream to be liquefied continues through the second and third heat exchangers and is finally expanded through an expansion valve to the liquid receiver. The cold vapor from the liquid receiver is returned through the heat exchangers to cool the incoming gas.



An expansion valve is still necessary in the Claude system, because much liquid cannot be tolerated in the expander in an actual system. The liquid has a much lower compressibility than the gas; there fore, if liquid were formed in the cylinder of an expansion engine (positive displacement type), high momentary stresses would result. Some rotary turbine expanders (axial – flow type) have been developed which can tolerate as much as 15 percent liquid by weight without damage to the turbine blades.

In some Claude systems, the energy output of the expanders is used to help compress the gas being liquefied. In most small – scale systems, the energy is dissipated in a brake or in an external air blower. Whether the energy is wasted or not does not affect the liquid yield; however, it does increase the compression work requirement when the expander work is not used.



Applying the first law of steady flow to the heat exchangers, the expansion engine, the expansion valve, and the liquid receiver as a unit, for no external heat transfer,

$$-W_c = \left(m - m_f\right)h_2 + m_f h_r - mh_2$$

Where $W_c$ is the work output of the expander. Applying the first law for steady flow to the expansion engine only,

$$-W_c = m_e\left(h_e - h_3\right)$$

If we define the fraction of the total flow which passes through the expander as x, then

$$x = \frac{m_e}{m}$$

Combining Eqs. (3-37) to (3-39), we obtain the liquid yield $y = m_f/m$ as

$$y = \frac{h_1 - h_2}{h_1 - h_f} + x\frac{h_3 - h_e}{h_1 - h_f}$$

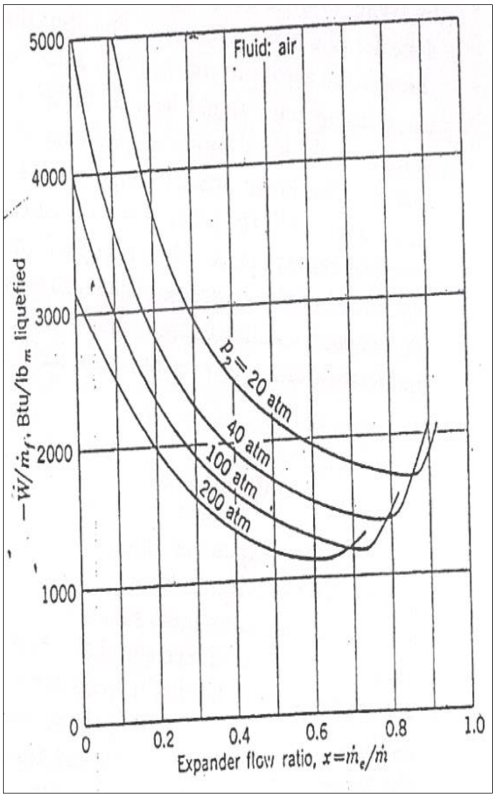
Again, we see that the second term represents the improvement in per formance over the simple Linde – Hampson system. Of course, x + y must be less than unity in Eq. (3-40).

The work requirement per unit mass compressed is exactly same as that for the Linde-Hampson system if the expander work is no utilized to help in the compression. This valve is given by Eq. (3-27). On the other hand, if the expander work is used to aid in compression then the net work is given by

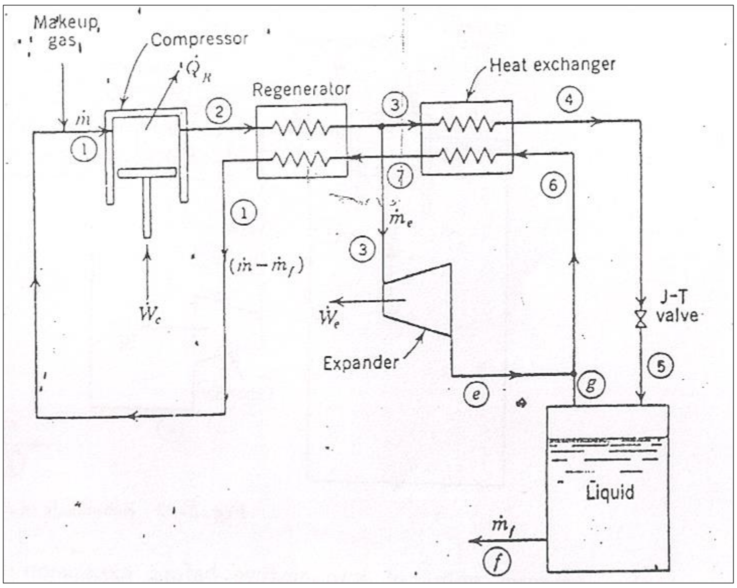
$$\frac{-W}{m} = \frac{-W_e}{m} - \frac{W_e}{m}$$
$$= \left[T_1(s_1 - s_2) - (h_1 - h_2)\right] - x(h_3 - h_e)$$

The last term in Eq. (3-41) is the reduction in energy requirement due to the utilization of the expander work.

For each value of high pressure (p2 or p3) and each value of expansion – engine flow x, one can find by series of calculations using the thermodynamic charts for the fluid used in the system that there is a finite temperature at point 3 (the condition upstream of the expander) which will yield the smallest work requirement. Energy requirements under this condition of astutely chosen T3 are shown in Fig. 3-20. We see that, for a given pressure level, there is also a value of x which makes the work requirement a minimum. As the high pressure is increased, minimum work requirement per unit mass of gas liquefied decreases.

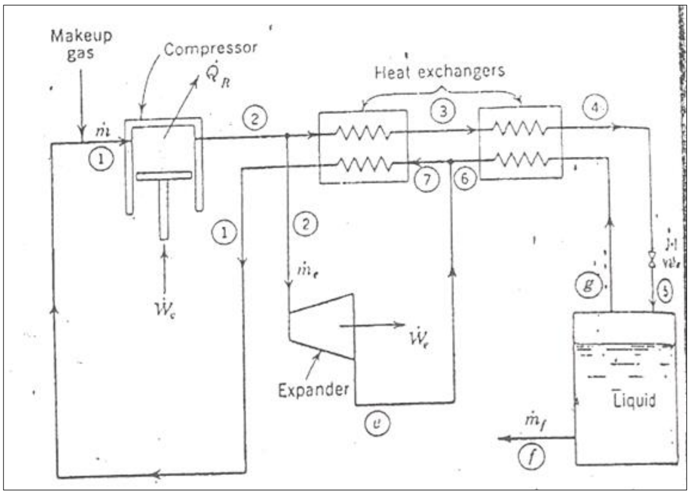


## Kapitza System



Kapitza (5) modified the basic Claude system by eliminating the third or low – temperature heat exchanger, as shown in Fig. 3-21. Several notable practical modifications were also introduced in this system. A rotary expansion engine was used instead of a reciprocating expander. The first heat exchanger in the Kapitza system (the high-temperature one) was actually two parallel regenerators, which combined the cooling process with the purification process. The incoming warm gas was cooled in one regenerator and impurities were deposited there, while the outgoing stream warmed up in the other regenerator and flushed out the frozen impurities deposited in it. After a few minutes, a valve was operated to cause the high – and low – pressure streams to exchange regenerators. The Kapitza system usually operated at relatively low pressures – on the order of 7 atm.

## Heylandt System

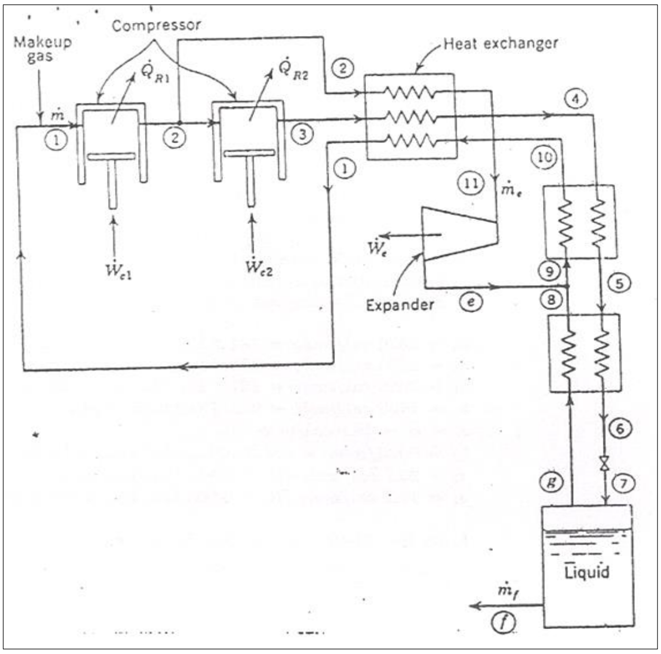


Heylandt noted that for a high pressure of approximately 200 atm and an expansion – engine flow – rate ratio of approximately 0.6, the optimum value of temperature before expansion through the a pander was about 70˚F or ambient temperature. Thus one could elisinate the first heat exchanger in the Claude system by compressing the gas to 200 atm. Such a modified Claude system is called the Heyland system, after its originator, and is used extensively in high –pressure liquefaction plants for air.

The advantage of the Heylandt system is that the lubricate problems in the expander are not difficult to overcome. In the air – liquefaction system, the gas enters at ambient temperature and leaves the pander at approximately 270˚R, so that light lubricants can be used. The Heylandt system, the expander and the expansion valve contribute nearly equally in producing low temperatures, whereas in the ordinary Claude system, the expander makes by far the largest contribution, as of will note from Example 3-4.

## Other Liquefaction Systems Using Expanders

There are many modification which one could use to improve the performance of the basic Claude system. In Fig. 3-23 is shown dual – pressure Claude system similar in principle to the Linde dual – press system. In this system, only the gas which is passed through the expansion valve is compressed to the high pressure. The gas which is circulate through the expander is compressed to some intermediated to some intermediate pressure the fore, the work requirement per unit mass of gas liquefied is reduced.



Maximum performance for nitrogen gas compressed between 1 and 35 atm is attained for this system when approximately 75 percent of the total flow is diverted through the expander.

Collins and Hughes (6) have developed a system for nitrogen liquefaction similar to the Kapitza system, except that reversing heat exchangers (similar in principle to regenerators, although not the same thing) are used instead of the first heat exchanger. These serve the dual purpose of cooling the incoming stream and purifying the gas at the same time.

**Comparison of Liquefaction Systems**

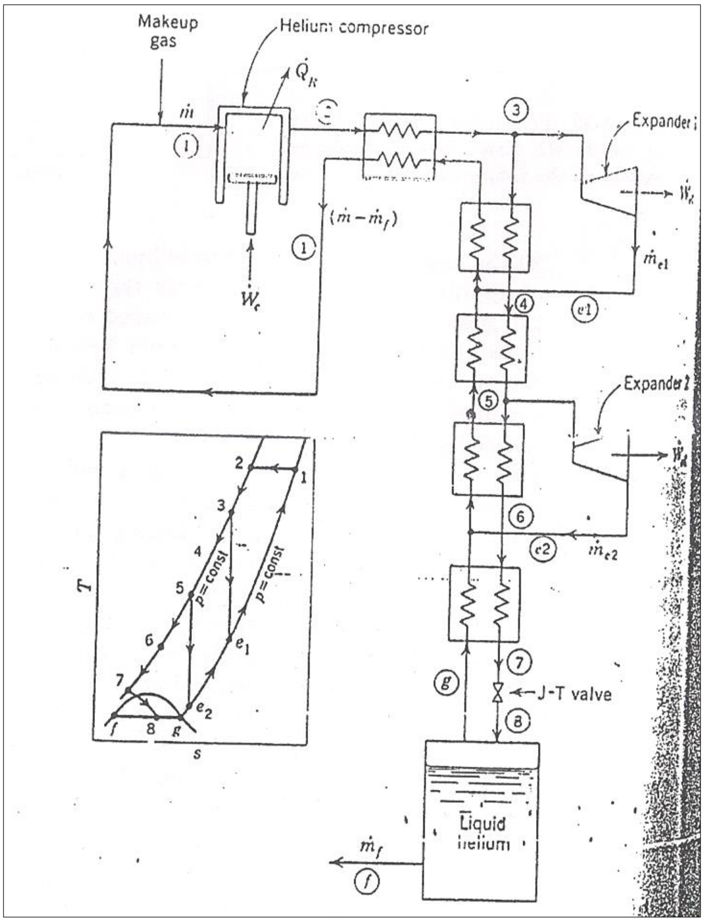
The performance parameters of the systems disused are given in Table

3-4 for air as the working fluid. The systems are assumed to

Every low – temperature laboratory has a helium liquefier where was either custom – made or commercially produced. Details of laboration type precooled Linde – Hampson helium liquiefiers are given in refs. 11 to 13

**Collins Helium –liquefaction System**

One of the milestones in cryogenic engineering is the design and development of a helium liqefier by S.C. Collins at the Massachusetu Institute of Technology. This liquefier is an extension of the Claud system, as shown in Fig. Depending upon the helium inlet pressure from two to five expansion engines are used in this system. This liquefier is produced commercially by Arthur D. Little Company, Cambridge, Massachusetts.



$$y = \frac{h_1 - h_2}{h_1 - h_2} + x1\frac{\Delta h_{c1}}{h_1 - h_1} + x2\frac{\Delta h_{e2}}{h_1 - h_f}$$

Where $x_1$ = me1/m

$x_2$ = me2/m

$\Delta$ $he_1$= enthalpy change of fluid passing through expander 1

$\Delta$ $he_2$= enthalpy change of fluid passing through expander 2

$Mc_1$, $mc_2$ = mass flow rates of fluid through expanders 1 and 2, respectively

For more than two engines, and additional term similar to the second term for each expander would be added to Eq. (3-47). For the two – expandor system, some typical parameters are as follows: x1 = 0.25 with an entrance temperature (to expander 1) of 108˚R and an exit temperature of 52˚R for an entrance pressure of 14 atm; x2 = 0.50 with an entrance temperature (to expander 2) of 27˚R and an exit temperature of 14˚R; y = 0.12.

The performance of the liquefier is increased (the liquid yield can be almost tripled) if a liquid – nitrogen precooling bath is used with the system; however, the preoolant bath is not required, since the system does not depend solcly upon the Joule – Thomson effect for the production of low temperature.

**Simon Helium – liquefaction System**

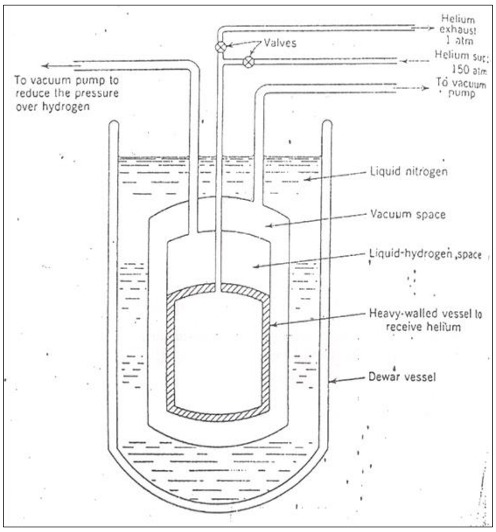
One of the methods used to liquefy small quantities of helium is the Simon liquefaction system. This does not operate as a steady – flow system, but produces liquid helium in a batch process. A schematic of the system is shown in Fig. 3-31, and the path followed by a unit mass of helium is shown on the T-s Plane in Fig 3-32.

The sequence of operaions for the simon system is as follows:

Process 1-2 Helium gas is introduced into the heavy – wall container at a pressure on the

Order of 150 atm and at ambient temperature.

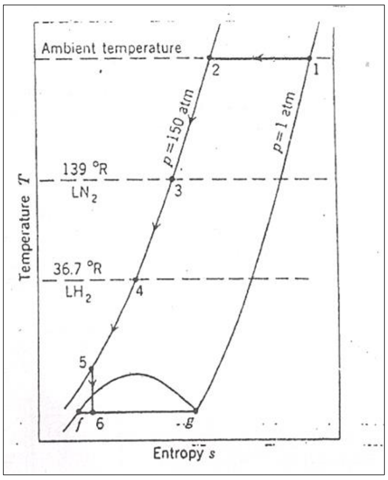
Process 2-3 Liquid nitrogen is introduced into the enclosing bath. This process cools the entrie container and its contents to liquid- nitrogen temperature (139 R). During this process, the vacuum space may be filled with helium gas at atmospheric



Pressure to act as a heat – transfer medium between the innet container and the liquid – nitrogen bath. At space between the two vessels is evacuated, thereby thermally isolating the inner vessel. The helium inlet value remains open during this process, and helium gas flows into the heavy – walled container to maintain a constant pressure of 150 atm.

Process 3-4 Liquid hydrogen is introduced into the upper part of inner container and the inner container, and the inner container and further cooled to liquid – hydrogen temperature (36.7R).

Process 4-5 The pressure above the liquid hydrogen is reduced approximately 1.7 mm Hg abs (0.033 Psia). During this process, the liquid hydrogen boils as the Pressure (and hence the liquid temperature) is lowered until the hydrogen solidified.



**Path followed by a unit mass of gas in the simon helium – liquefaction system on the temperature – entropy plane.**

At the end of the process, the solid hydrogen is at a temperature of approximately 18˚R, along with the whole inner vessel and the gaseous helium in the heavy – walled container, by a judicious choice of the amount of liquid hydrogen introduced, at the end of this process all the solid hydrogen will have been sublimed, and the heavy – walled container is thus thermally isolated at 18˚R.

Process 5-6 the Pressure of the gaseous helium is reduced from 150 to 1 atm by allowing the helium to be released to a gas holder external to the system. During this process, the helium which finally remains in the inner container does work against the helium, gas which is discharged. The discharged gas removes energy from the system, thereby lowering the temperature of the remaining helium. Heat is also transferred from the heavy – walled vessel to the remaining helium in the cool – down process. Since the specific heat of metals is so small in this temperature range, this heat transfer does not seriously penalize the system performance, however. At the end of this process, the heavy – walled container is usually from 80 to 100 percent full of liquid helium.

If we assue that the heat is transferred reversibly from the heavy – walled vessel and that the specific heat of the vessel material obeys the $T_3$ Debye expression Eq (2-0) than the liquid yield may be estimated for reversible heat transfer.

$dQ = m_c T_c\, ds_c = m_c C_c\, dT_c$

Where $m_c$ = mass of container

$T_c$= container temperature

$C_c$ = container specific heat

Substituting the debye expression for the container specific heat obtain for the differential entropy change of the container.

$$ds_c = 233.8 R_c T_c^{\,2} \frac{dT_c}{\theta D^3}$$

Integrating Eq (3-49) between the limits of point 5($T_5$) and point 6 (where the temperature is $T_f$), we obtain the total entropy change of the container.

$$\Delta sc = \frac{77.9 R_c}{\theta D^3}(Tf^3 - T_6^{\,3} = \frac{1}{3}(Cc_f - Cc_6)$$

Since there is practically no external heat transfer from the surroundings to the inner vessel, the total entropy change of the vessel and helium gas is given

by $(S_{out} - S_{in}) = \Delta S_{HC} + \Delta S_C$ or

$S_6(m_6 - m_5) - 0 = (m_6 s_6 - m_5 s_5) + m_c \Delta s_c$

Where $m_6$ is the total mass of helium remaining in the vessel after expansion (i.e., at point 6) m5 is the total mass of helium in the vessel before the expansion (at Point 5), and $s_5$ and $s_6$ are the entropy values at point 5 and 6 , respectively. Solving for $s_6$

$$s_6 = s_5 - \frac{m_c}{m_6}\Delta s_c$$

Note that if the entropy change of the container is very small, the entropy change of the helium is practically Zero.

We may also write $s_6$ in terms of the liquid yield $y = m_f/m_6$ as follows

$S_6 = y s_f + (1-y)\, s_g$

Where $s_f$ is the entropy of the saturated liquid at the final condition and $s_g$ is the entropy of the saturated vapor at the final condition. Equation the right– hand sides of Eqs. (3-52) and (3-53), we can solve for the liquid yield

$$Y = \frac{(s_g - s_5) + (m_c / m_6)\Delta s_c}{s_g - s_f}$$

$$y = \frac{s_g - s_5}{s_g - s_f}$$

Knowing the conditions before expansion allows us to determines s5 from a T-s diagram of helium, and with the final conditions we can then determine the percentage of the mass of helium (remaining in the container) which is in the liquid state.

Of interest also would be the fraction of the container volume which is filled with liquid. This ratio $V_f/V$, where V is the container volume, may be determined from

$$Y= \frac{m_f}{m_6} = \frac{\upsilon_6 / V}{\upsilon_f / V_f} = \frac{\upsilon_6 / \upsilon_f}{V_f / V}$$

Where $\upsilon$ is the fluid specific volume (the reciprocal of the density). We may also write the specific volume at point 6 as

$$\upsilon_6 = y\upsilon_f + (1-y)\ \upsilon_g = \upsilon_g - y\ (\upsilon_g - \upsilon_f)$$

Eliminating $\upsilon$6 between Eqs. (3-56) and (3-57) results in an expression for the volume ratio

$$\frac{V_f}{V} = \frac{y}{\upsilon_g / \upsilon_f - y(\upsilon_g / \upsilon_f - 1)}$$

Where y is determined from Eq(3-54). Unless the mass released from the vessel is measured, the calculation procedure is an interative one, since $m_6$ must be known to find y; however, for a given y, $m_6$ is fixed by Eq (3-57) and fact that m6 = V/$\upsilon_6$

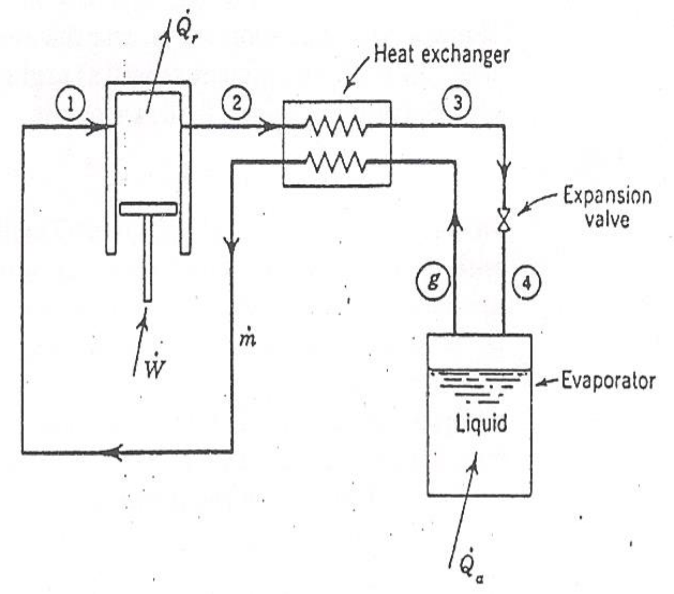
Cailletet observed a thick mist when oxygen gas at – 26 and high pressure was suddenly expanded in and apparatus similar to the simon liquefier; however, in general, the simon liquefier does not work well fot gases other than helium. There are two reasons for this fact (1) the ratio $\upsilon_s/\upsilon_f$ is approximately 0.6 for helium at 150 atm and 18 ˚**F** which means that more energy can be removed from the mass remaining in the vessel than for other gases under comparable conditions; and (2) the specific heat of metals is extremely small at helium temperatures, so or a small amount of cooling capacity is lost into the walls of the vessel for helium liquefaction.

The simon system is primarily a laboratory liquefier, since can produce liquid helium only in relatively small quantities. It is pecially well suited for experiments involving magnetic fields in which small space is available between the poles of a magnet. The liquid in heavy – walled container could serve as the refrigerant to cool a paramagnet substance, for example on which experiments would be carried out.

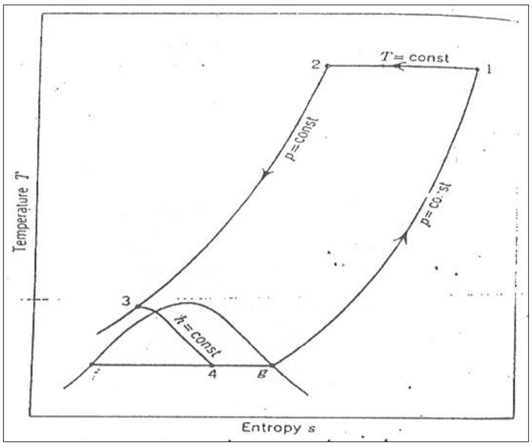
## JOULE – THOMSON REFRIGERATION SYSTEMS

Any of the liquefaction systems which do not use an expansion engine may be classed as Joule – Thomson refrigerators, since they depend upon the Joule – Thomson effect to produce low temperatures. Instead of withdrawing the liquid which is formed in a refrigerator, heat is absorbed from the low – temperature source to evaporate this liquid. A simple Linde – Hampson refrigerator is shown in Fig., and its cycle is shown on the temperature – entropy plane in Fig.

As mentioned in Chap. 3, the compression from point 1 to point 2 in Fig. would be isothermal in the ideal case; in practice, there is a small temperature difference because the effectiveness of the heat exchanger is less than unity. The compressed gas is passed through the heat exchanger, cooled to the low temperatures by heat exchange with the cold outgoing gas stream, and expanded through a Joule – Thomson valve into an evaporator. In the evaporator (which corresponds to the liquid receiver in the liquefaction system), the liquid formed after the expansion process is evaporated by absorbing heat from the space to be refrigerated. The vapor then returns through the heat exchanger, and the cycle is repeated.



**Linde – Hampson refrigerator**



**Thermodynamic cycle for the Linde – Hampson refrigerator**

If we apply the first law of thermodynamics to that exchanger, the expansion value, and the evaporator as a unit and assume no heat in leaks from ambient, as well as negligible Kinetic – and potential – energy changes of the working fluid, we obtain

$$Q_a = m (h_1 - h_2)$$

Immediately we see that the Joule – Thomson refrigerator cannot be used with neon, hydrogen, or helium as the working medium, unless there gases are first precolled below their maximum inversion temperatures. Since $Q_a$ is a positive quantity (considering the refrigerator as the thermodynamic system), h1 must be larger than $h_2$ in order to have a positive refrigeration effect ( $Q_a/m$). This condition is not true if the working fluid is at a temperature above its inversion temperature.

If we make the definition

$$Y = \frac{h_1 - h_2}{h_1 - h_f}$$

For a refrigeration system, the expression developed in chap 3. For liquefaction system may be directly applied for refrigeration systems. We interpret y for a refrigeration systems as the ratio if the refrigeration effect to the refrigeration which would be obtained if the refrigerant were heated from the saturated – liquid state to state point 1. In the liquefaction systems y was compressed. For example for simple for the simple linde – Hampson liquefaction system, from Eq (3-23),

$$Y = \frac{Q_a / m}{h_1 - h_f}$$

Using the definition of y for a refrigeration ststem, Eq (5-14),

$$\frac{Q_a / m}{h_1 - h_f} = y = \frac{h_1 - h_2}{h_1 - h_f}$$

Thus we obtain the correct expression for the refrigeration effect Eq (5-13):

$$\frac{Q_a}{m} = h_1 - h_2$$

Applying the first law to the compressor, we obtain the work requirement

$$\frac{-W}{m} = \frac{T_2(s_1 - s_2) - (h_1 - h_2)}{\eta_{c,o}}$$

Where $\eta_{c,o}$ is the overall efficiency of the compressor. From the definition of the coefficient of performance, Eq. (5-1),we find the COP for Linde – Hampson refrigerator to be

$$\text{COP} = -\frac{Qa/m}{W/m} = \frac{\eta_{c,o}(h_1 - h_2)}{T_2(s_1 - s_2) - (h_1 - h_2)}$$

Since the liquid in the evaporator boils at constant temperature, this refrigerator is of the isothermal – source type. Liquid nitrogen is a suitable refrigerant for maintaining temperature in the region between 120 R (67 K) and 200R (111 K). the temperature in the evaporator can be expansion – value setting. At 120 R the evaporator pressure would be 0.23 atm (3.38 Psia), and at 200 R the evaporator pressure would be 15.48 atm (227 Psia), using nitrogen as the refrigerant. The temperature range is limited on the lower end of the temperature scale by the triple point and also by the difficulty in maintaining low pressures with large mass flow rates. The pressure were lowered below the triple – point pressure, nitrogen snow would form in the evaporator, which would possible result in clogging of the expansion valve and would certainly result in poon heat exchange in the evaporator. The temperature is limited on the higher end of the temperature scale by the critical point. As the critical point approached, the heat of vaporization of the liquid approaches zero. The temperature range can be extended to temperatures above the critical temperature by using a cold – gas refrigerator instead of a boiling – liquid or isothermal – source, the temperature rang of the Linde – Hampson refrigerator using nitrogen s the working fluid may be considered as about 120˚R up to ambient temperatures. When we get into the temperature region above about 300˚R, however, other refrigerants such as the Freon compounds become more attractive as refrigeration media.

Geist and Lashmet (2) have investigated the use of argon as the refrigerant in immature Joule – Thomson refrigeration systems. Because of the thermal properties of argon compared with nitrogen, the heat exchanger in the refrigeration system using argon as the refrigerant can be made about 70 percent the size of the heat exchanger in a system using nitrogen as the working fluid. Argon has a higher Joule – Thomson coefficient than nitrogen at the same temperature and pressure. Since the difference in cost between argon and nitrogen is not a weighty factor in space – vehicle applications, argon would be the optimum refrigerant to furnish cooling for infrared detection devices operating at about 160˚R.

## CASCADE OR PRECOOLED JOULE – THOMSON – THOMSON REFRIGERATION SYSTEMS

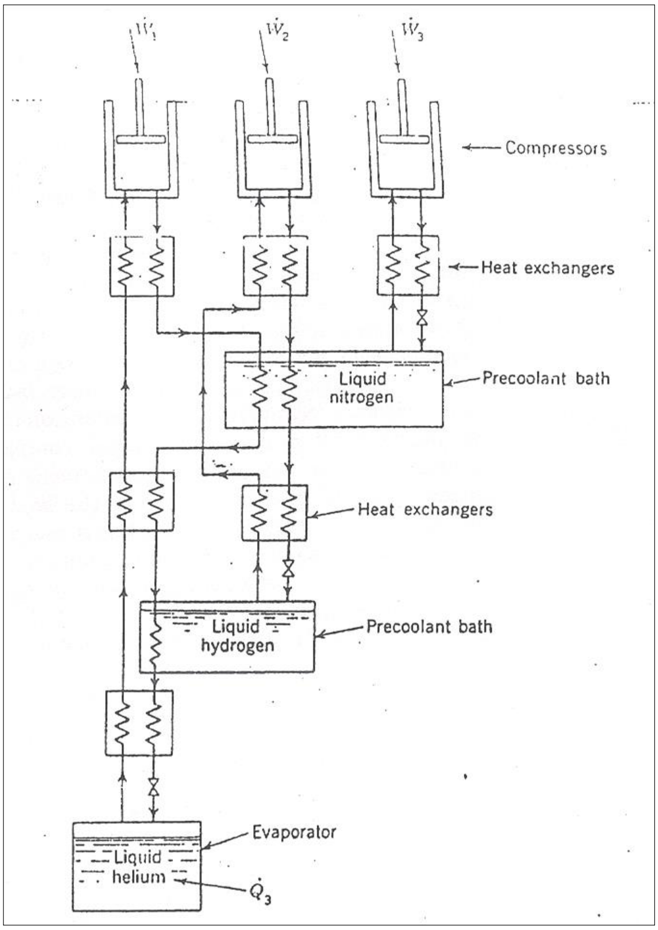
For temperature lower than those obtainable with liquid nitrogen, the only available working fluids are neon, hydrogen, and helium.

Ment per unit mass of main refrigerant is

$$-\frac{W_{net}}{m} = \frac{T_2\left(s_a - s_b\right) - \left(h_a - h_b\right)}{\eta_{c,a}} + z\left[\frac{T_b\left(s_a - s_b\right) - \left(h_a - h_b\right)}{\eta_{p,c}}\right]$$

Where $\eta_{c,a}$ is the overall efficiency of the main compressor and $\eta_{p,c}$ is the overall efficiency of the precoolant compressor. The coefficient of performance of the refrigerator may then be calculated by forming the parts of the heat absorbed by the liquid hydrogen or neon in the evaporator to the net work required.

Lower temperatures may be attained by using a three – stage refrigerator with nitrogen (or argon), hydrogen (or neon), and helium a the working fluids. Such a refrigerator has been developed (4) by air products and Chemicals, Inc., Allentown, Pennsylvania, and is shown schematically in Fig.. The nitrogen is compressed to about 200 atm passed through a heat exchanger, and expanded to 1.2 atm. The liquid nitrogen formed is used to precool both the hydrogen stream and the helium stream. The hydrogen is compressed to about 145 atm, passed through the heat exchangers and precooling bath, and expanded to 1.2 atm. The liquid hydrogen formed is used to precool the helium stream. The helium is compressed to 17.7 atm, passed through the heat exchangers and precooling baths, and expanded to 1.2 atm. The liquid helium formed is used to furnish refrigeration for the low – temperature region. In the air Products refrigerator, the mass flow rate of helium is 133.6 $\text{lb}_\text{m}$/hr. while the precoolant mass – flow – ratios are $m_{112}/mH_3 = 0.65$ and $mN2/mH_3 = 7.43$.



**Three – stage Joule – Thomson liquid – helium refrigerator**

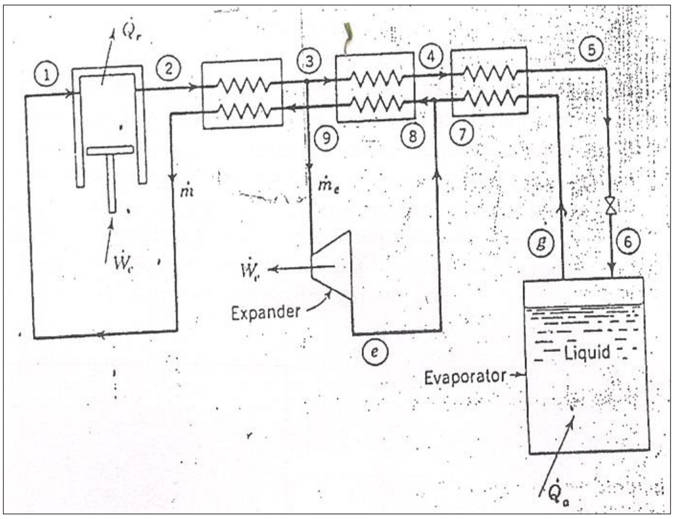
## EXPANSION – ENGINE REFRIGERATION SYSTEMS

The Claude liquefaction system or the Collins liquefaction system could be used as a refrigeration system. A schematic of a Claude refrigerator is shown in Fig. 5-12. If we apply the first law to the three heat exchangers, the expansion value, and the evaporator as a unit neglecting heat inleaks from ambient and kinetic and potential energy changes, we obtain the following for the heat absorbed by the refrigerant;

$$\frac{Q_n}{m} = (h_1 - h_2) + x(h_3 + h_e)$$

Where x= mc/m expander mass – flow – ratio, me = mass flow rate through the expander, m= mass flow rate through the compressor, and the subscripts on the enthalpy terms refer to the points given in Fig 5-13. The net work requirement, assuming that the expander work is utilized to help in the compression of the gas, is given by

$$-\frac{W_{net}}{m} = \frac{T_2\left(s_a - s_b\right) - \left(h_a - h_b\right)}{\eta_{c,a}} - x_{\eta c}, m(h_3 - h_c)$$



Claude refrigerator

Where $\eta_{c,a}$ is the overall efficiency of the compressor and $\eta_{c,m}$ is the mechanical efficiency of the expander.

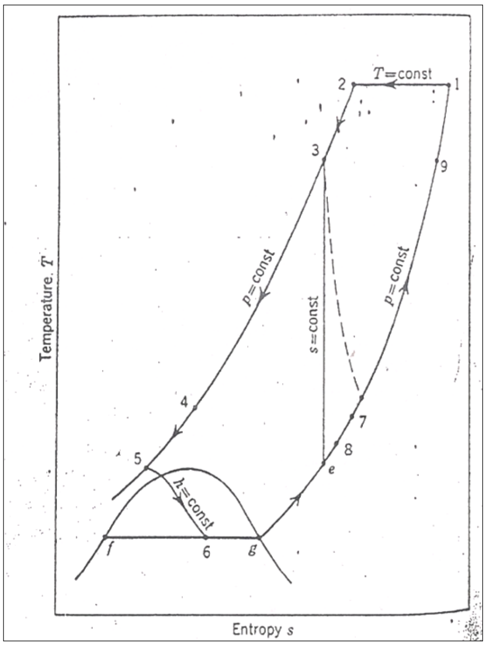
General electric corporation [5] has developed a miniature claude refrigerator which occupices only 1.5 ft3, weighs about 50 lbm, and maintains temperatures near 6.3 R by using helium as the working fluid. This refrigerator of the helium gas at the inlet to the expander is 27 R. the power developed by the expander is absorbed by a tiny alternator operating operating at speeds up 250,000rpm.

## COLD GAS REFRIGERATION SYSTEMS

In many applications, an isothermal source is not necessary, and only a certain amount of energy must be removed from a space. In this case, a gas such as helium, hydrogen, or neon can be used as the refrigeration medium, and the gas need not be condensed.

A basic cold gas refrigeration system is shown in Fig. 5-15, and the cycle for the refrigeration is shown on the temperature – entropy diagram in Fig 5-16. This refrigeration system has been used by the linde company, Tonawanda, New York n[6], and by Cryo Vac, Inc., Columbus, ohio, in providing refrigeration for the crypanels in spoace – simulation chambers.

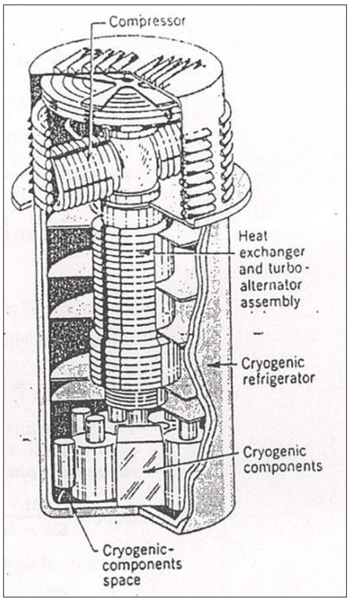
The refrigerant (helium is most frequently used) is first compressed to a moderatc pressure as nearly isothermally as possible. The maximum system pressure is determined to a large extent by the amount of refrigeration required. The gas is passé through a heat exchanger, a reversing heat exchanger may be used instead of the precooler - purifier which is usually cooled by liquid nitrogen from an auxiliary source. The denser cold gas is then expanded through an expansion engine to produce the required low temperatures. The gas is through the region to be cooled and back to the compressor through the heat exchangers.



The energy absorbed per unit mass of refrigerant or the refrigeration effect for the system shown in Fig. is given by

$$\frac{Q_a}{m} = h_7 - h_6$$

Where h6 is the enthalpy of the gas entering the space to be cooled and h7 is the enthalpy of the gas leaving the space to be cooled. Ideally, we should want the temperature at point 7 (leaving the cooling load) to be the same as the temperature of the gas at point 5 (leaving the cold end of the cold heat exchanger). If we let ha denote the enthalpy at point 8 in Fig. if the temperature Ta is equal to the temperature T4, then the enthalpy at point 7 is related to the effectiveness of the heat exchanger by the following



$$\varepsilon = \frac{h_4 - h_5}{h_6 - h_7}$$

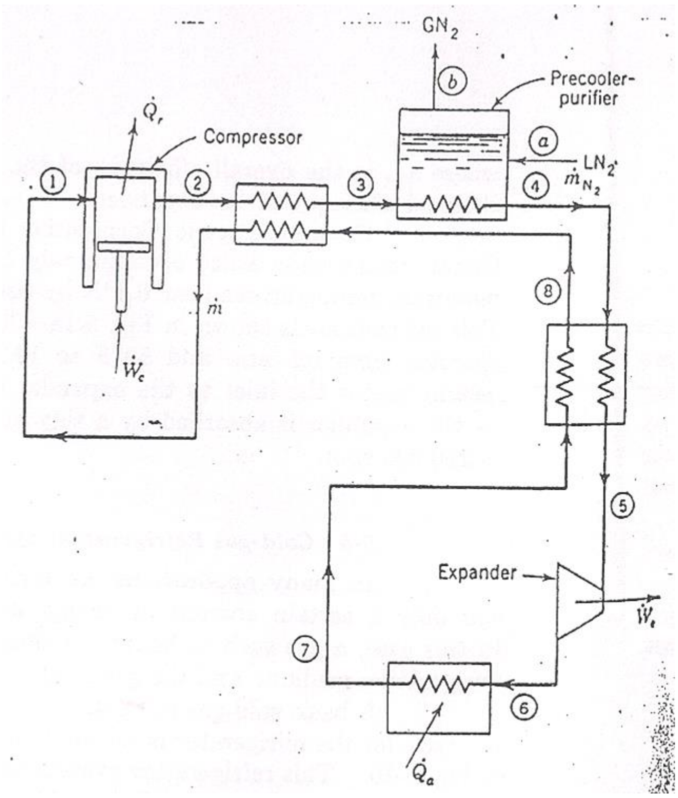
Therefore,

$$h_7 = h_8 - \frac{h_4 - h_5}{\varepsilon}$$

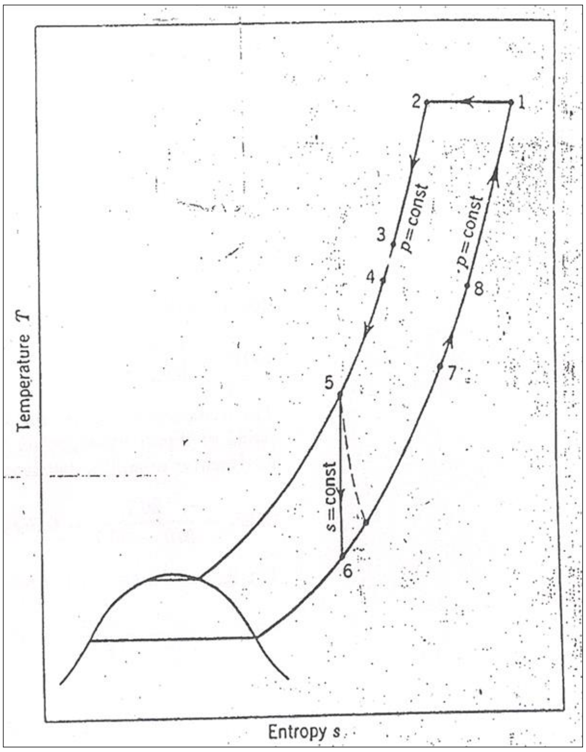
Notice that if the gas specific heat can be assumed constant and the process is isobaric in the heat exchanger (these conditions are only approximately met in actual systems), then $h_6 – h_7 = c_p(T_4 – T_7)$ and $h_4 – h_5 = c_p(T_4 – T_5)$, so the heat – exchanger effectiveness becomes

$$\varepsilon = \frac{T_4 - T_5}{T_4 - T_7}$$

From Eq. (5-29), we see that $T_7 = T_5$ for 100 percent effective heat exchanger.



Cold – gas refrigerator



Thermodynamic cycle for the cold – gas refrigerator

The refrigeration effect can also be written by applying the first law to all components of the system excluding the compressor and expander:

$$\frac{Q_a}{m} = \left(h_1 - h_2\right) + \frac{m_{N2}}{m}\left(h_b - h_a\right) + \left(h_5 - h_6\right)$$

The refrigeration effect is made up of the cooling supplied by the precooling fluid (the second term) and the cooling supplied by the expander (the third term). The first term is negative for gases such as helium, so that it represents a reduction in the refrigeration effect due to the inefficiencies of the heat exchangers and due to gas imperfections.

The total work requirement for this system, assuming that the

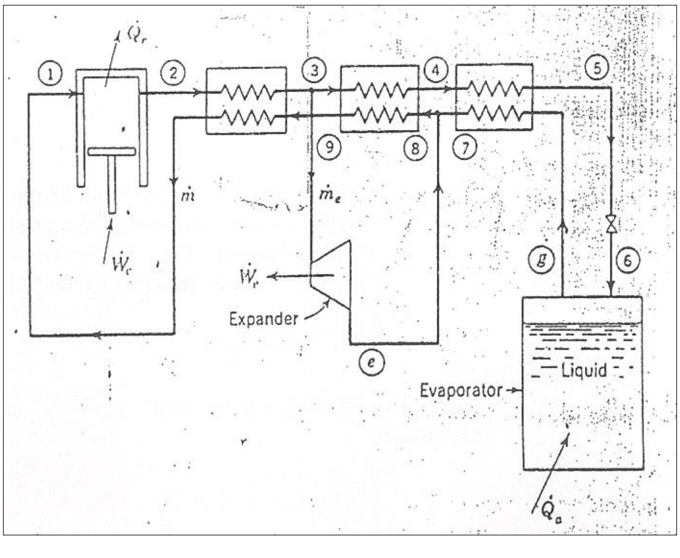
## Expansion – engine Refrigeration Systems

The Claude liquefaction system or the Collins liquefaction system could be used as a refrigeration system. A schematic of a Claude refrigerator is shown in Fig. 5-12. If we apply the first law to the three heat exchangers, the expansion valve, and the evaporator as a unit, neglecting heat inleaks from ambient and kinetic and potential – energy changes, we obtain the following for the heat absorbed by the refrigerant:

$$\frac{Q_a}{m} = \left(h_1 - h_2\right) + x\left(h_3 + h_e\right)$$

Where x =mc/m = expander mass – flow – rate ratio, mc = mass flow rate through the expander, m = mass flow rate through the compressor, and the subscripts on the enthalpy terms refer to the points given in Fig. 5-13. The net work requirement, assuming that the expander work is utilized to help in the compression of the gas, is given by

$$-\frac{W_{net}}{m} = \frac{T_2\left(s_1 - s_2\right) - \left(h_1 - h_2\right)}{\eta_{c.0}} - x\eta_{e,m}\left(h_3 - h_e\right)$$



Where $\eta_{c.0}$ is the overall efficiency of the compressor and $\eta_e, m$ is the mechanical efficiency of the expander.

General Electric Corporation (5) has developed a miniature Claude refrigerator which occupies only 1.5 ft$^3$, weighs about 50 lb$_m$, and maintains temperatures near 6.3˚C by using helium as the working fluid. This refrigerator is shown in Fig. 5-14. The General Electric refrigerator operates from 0.5 atm and 80˚F to 10 atm. The temperature of the nelium gas at the inlet to the expander is 27˚R. The power developed by the expander is absorbed by a tiny alternator operating at speeds up to 250,000 rpm.

## COLD – GAS REFRIGERATION SYSTEMS

In many applications, an isothermal source is not necessary and only a certain amount of energy must be removed from a space. In this case, a gas such as helium, hydrogen, or neon can be used as the refrigeration medium, and the gas need not be condensed.

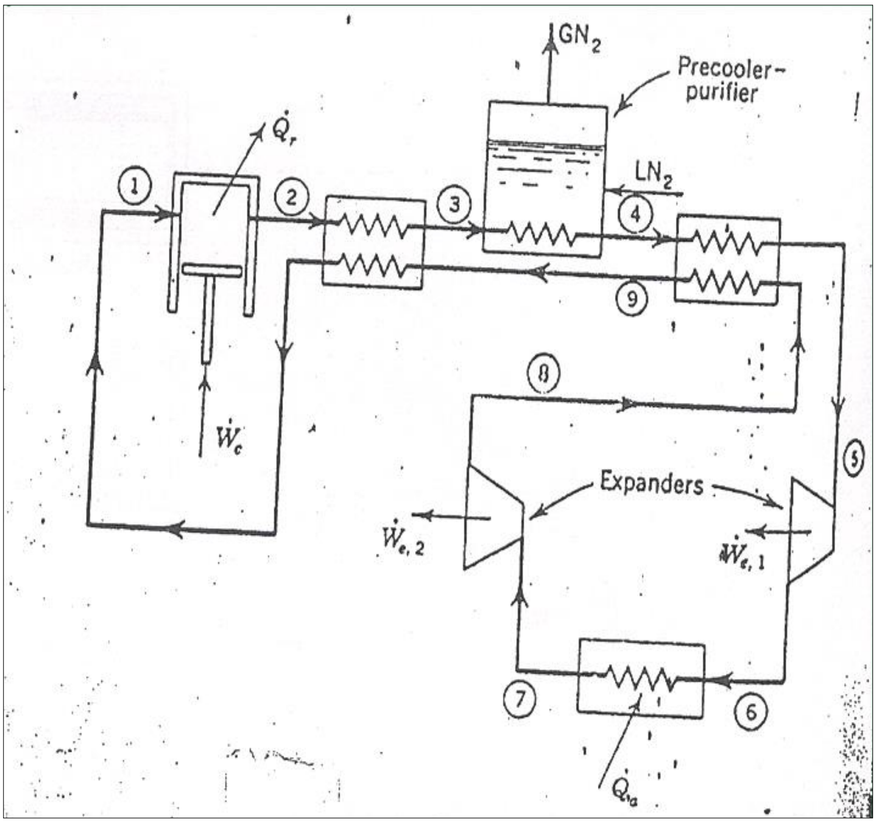
A basic cold – gas refrigeration system is shown in Fig and the cycle for the refrigerator is shown on the temperature – entropy diagram in Fig. This refrigeration system has been used by the Linde Company, Tonawanda, New York (6), and by Cryo Vac, Inc., Columbus, Ohio, in providing refrigeration for the cryopanels in space – simulation chambers. Expander work is utilized to aid in the compression of the gas, is given 1.

$$-\frac{W_{net}}{m} = \frac{T_2(s_1 - s_2 - (h_1 - h_2)}{\eta_{c.0}} - \frac{W_{N2}/m_{s2}}{m/m_{N2}} - \eta_{c,m}(h_5 - h_6)$$

Where Wn$_2$/mNs is the work per unit mass of nitrogen required to produc the liquid nitrogen used in the precooler – purifier. If a precooler – purifier not used, the second term is omitted; if the expander work is dissipates externally to the system and not utilized in the compression process, the third term is omitted. The nitrogen flow – rate ratio can be determined be applying the first law to the precooler – purifier.

$$\frac{m_{N2}}{m} = \frac{h_3 - h_4}{h_b - h_a}$$

There are many modifications which could be made on the basic system shown in Fig. 5-15. One could use two expanders to attain additional refrigeration, as Fig. 5-17. The space to be cooled acts as a reheater between the two expansion engines.



**PHILIPS REFRIGERATOR**

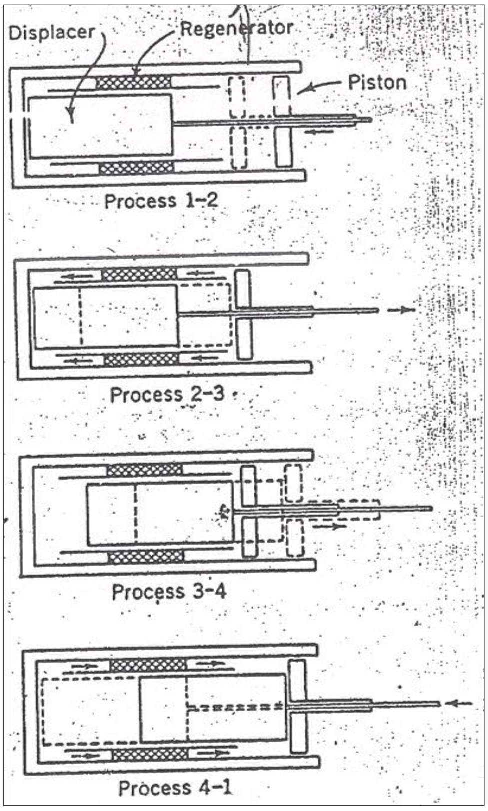
The Philips refrigerator operates on the Stirling cycle, which was invented in 1816 by a Scottish minister, Robert Striling, for use in hot – air engine. As carly as 1834, John Herschel (7) suggested that this engine could be used as a refrigerator. A schematic of the sequence of operations of the Philips refrigerator is shown in Fig. 5-18, and the cycle is shown on the temperature – entropy plane in Fig. 5-19.

The Philips refrigerator consists of a cylinder enclosing a piston, a displacer, and a regenerator. The piston compressed the gas, while the displacer simply moves the gas from one chamber to another without changing the gas volume. The heat exchange during the constant – volume process is carried out in the regenerator.

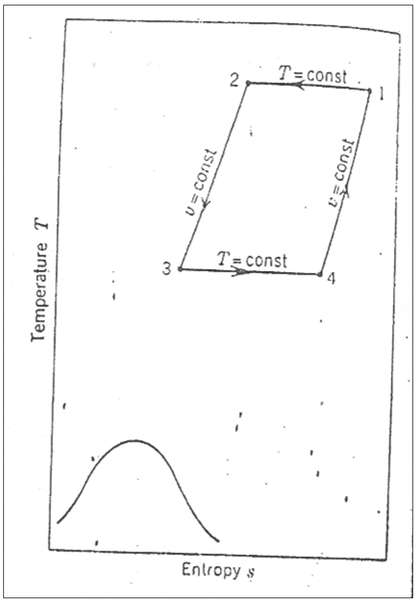
The sequence of operations for the system is as follows.

Process 1-2 The gas is compressed isothermally while rejecting heat to the high – temperature sink (surroundings).

Process 2-3 The gas is forced through the regenerator by the motion of the displacer. The gas is cooled at constant volume during this process. The energy removed from the gas is not transferred to the surroundings but is stored in the regenerator.



**Philips Refrigerator**



**Thermodynamic cycle for the Philips refrigerator**

Process 3-4 The gas is expanded isothermally while absorbing heat from the low – temperature source.

Process 4-1 The gas is forced through the regenerator by the motion of the displacer. The gas is heated at constant volume during this process. The energy which was stored during process 2-3 is transferred back to the gas during this process. In the ideal case (no heat inleaks), heat is transferred to the refrigerator only during process 3-4, and heat is rejected from the refrigerator only during process 1 – 2.

If we assume that the heat transfers to and from the refrigerator are reversible, the heat transferred may be determined by the second law of thermodynamics:

Heat rejected = $Q_r = T1(s_2 - s_1)$

Heat absorbed = $Q_a = mT_3\ (s_4 - s_3)$

Where m is the mass of gas in the refrigerator cylinder. By the first law $W_{net} = Q_r + Q_a$ for a cycle, so that the coefficient of performance of the ideal Philips refrigerator is

$$\text{COP} = -\frac{Q_a}{W_{net}} = \frac{T_3}{T_2(s_1 - s_2)/(s_4 - s_3) - T_3}$$

If the working fluid behaves as a perfeet gas we may write

$$S_1 - S_2 = C_p In\frac{T_1}{T_2} + RIN\frac{V_1}{V_2}$$

$$= RIn\frac{V_1}{V_2} = RIn\frac{V_4}{V3}$$

Since $T_1 = T_2$, $v_1$ = and $v_2 = v_3$, where v = specific volume of the gas From Eq. (5-33). We see that $s_1 - s_2 = s_4 - s_3$ for a perfect gas as the working fluid, so that the coefficient of performance for performance for the Philips refrigerator operating reversibly with a perfect gas as the refrigerant is

$$COP = \frac{T_2}{T_1 - T_2}$$

This is the same expression as that for the COP1 of a Carnot refrigerator; therefore, the ideal Philips refrigerator would have a figure of merit of unity. Mechanical losses in the drive unit used with the refrigerator, pressure losses in the regenerator, finite temperature differences during heat rejection and heat absorption, and finite temperature differences in the regenerator all tend to lower the figure of merit.

The Philips refrigerator is constructed by the Philips Research Laboratories, Eindhoven, Netherlands. It has been successfully used in the liquefaction of air in amounts of 11.6 $lm_b$/hr (5.51/hr), in gas – separation systems, and in general – purpose cold – box applications. The figure of merit for the actual system as contrasted to the ideal Philips refrigerator is about 0.3 when the source temperature is that of liquid air (142˚R or 79˚K).

## Importance of Regenerator Effectiveness for the Philips Refrigerator

The success of the Philips refrigerator depends to a large extent upon the effectiveness of the regenerator used in the system. Since the regenerator effectiveness is so important in many cryogenic refrigerators, we shall devote Sec. 5-11 to a discussion of the factors which influence the effectiveness of a regencreater. Briefly, a good regenerator should be constructed of a material with a large heat capacity, the period of operation should be small (i.e., the frequency of cycling the fluid through the regenerator should be large), the heat – transfer coefficient and heat – transfer area should be large, and the mass flow rate of gas through the regenerator should be small. In the Philips refrigerator, a light feltlike mass of fine wire is used as the regenerator material to attain the large heat capacity of the regenerator material and large heat – transfer area.

If the regenerator were less than 100 percent effective, the temperature of the gas leaving the regenerator at point 3 in fig. 5-15 would be somewhat higher than the sourced temperature. This means that some of the energy which could have been absorbed from the low – temperature source cannot be absorbed, because energy is wasted to cool the refrigerator gas down to the source temperature. The actual energy absorbed from the low – temperature source becomes.

$$Q_a = Q_0,_{ideal} = \Delta Q$$

Where $Q_{a,\ ideal}$ is the ideal heat absorbed and $\Delta Q$ is the heat which must be removed from the working fluid to overcome the fact that the regenerator effectiveness is less than unity. Since the regenerator effectiveness is defined as

$$\varepsilon = \frac{Q_{actual}}{Q_{ideal}} = \frac{Q_{2-3.ideal} - \Delta Q}{Q_{2-3.ideal}}$$

$$\text{Than } \Delta Q = (1-\varepsilon)Q_{2-3.ideal} = (1-\varepsilon)mc_v(T_2 - T_3)$$

Where $Q_{2-3.ideal}$ = ideal heat transferred from gas to regenerator during process 2 – 3.

m = mass of gas flowing through regenerator

$C_v$ = specific heat of gas flowing through regenerator

If we assume that the working fluid behaves as a perfect gas the energy removed from the cooling load in the ideal case is given by

$$Q_{a.ideal} == mT_3(s_4 - s_3) = mRT_3 In \frac{v_4}{v_3}$$

Or $$Q_{2-3.ideal} = (\lambda - 1)mcrT_3 In \frac{v_4}{v_3}$$

Where $\lambda$ = cp/cp = specific heat ratio for the refrigeration medium. If we divide Eq. (5-37) by Eq. (5-38), we obtain the fraction of the ideal refrigeration effect which is wasted because the effectiveness of the regenerator is less than unity:

$$\frac{\Delta Q}{Q_{a.ideal}} = \frac{1-\varepsilon}{\gamma - 1} \frac{T_2 / T_3 - 1}{In(v_4 / v_3)}$$

As a numerical example, suppose that we have helium as the working fluid, for which $\gamma$ = 1.67, and suppose that v4/v2 = 1.24. For refrigeration between 540˚R (300˚K) and 140˚R (77.8), the lost refrigeration effect becomes

$$\frac{\Delta Q}{Q_{a,ideal}} = \frac{1-\varepsilon}{0.67} \frac{(540/140)-1}{In1.24} = 19.87(1.\varepsilon)$$

If the regenerator has an effectiveness of 99 percent, than $\Delta Q / Q_{a.ideal} = 19.87$ percent, or almost 20 percent of the refrigeration effect is wasted because of a 1 percent decrease in the effectiveness from the ideal or 100 percent effectiveness. We see that, in this example, all the refrigeration would be wasted if the regenerator effectiveness were

$$\varepsilon \min = 1 - \frac{1}{19.87} = 1 - 0.0503 = 0.9497$$

Or about 0.5 percent. This example illustrates how critical the regenerator is in the performance of the whole refrigeration system.

## Solvay Refrigerator

The Solvay refrigerator was originally based on isentropic expansion and was utilized as early as in 1887 to produce a low temperature of 178 K. In 1959, McMahon and Gifford utilized this principle to design a refrigeration cycle using helium gas as the refrigerant to produce 55 K in a single stage using a miniature Solvay refrigerator. This has an open cylinder of diameter of 5.6 mm and a length of 51 mm for the operating pressures between 3.4 atm and 17 atm [5]. The Solvay refrigerator comprises a cylinder (sealed at the warm end) with a piston, a regenerator and a heat exchanger for absorbing heat at low temperature. The sequence of operation of the Solvay refrigerator may be schematically described by Figure 3 and corresponding the *T*-*s* diagram, as shown in Figure 4.

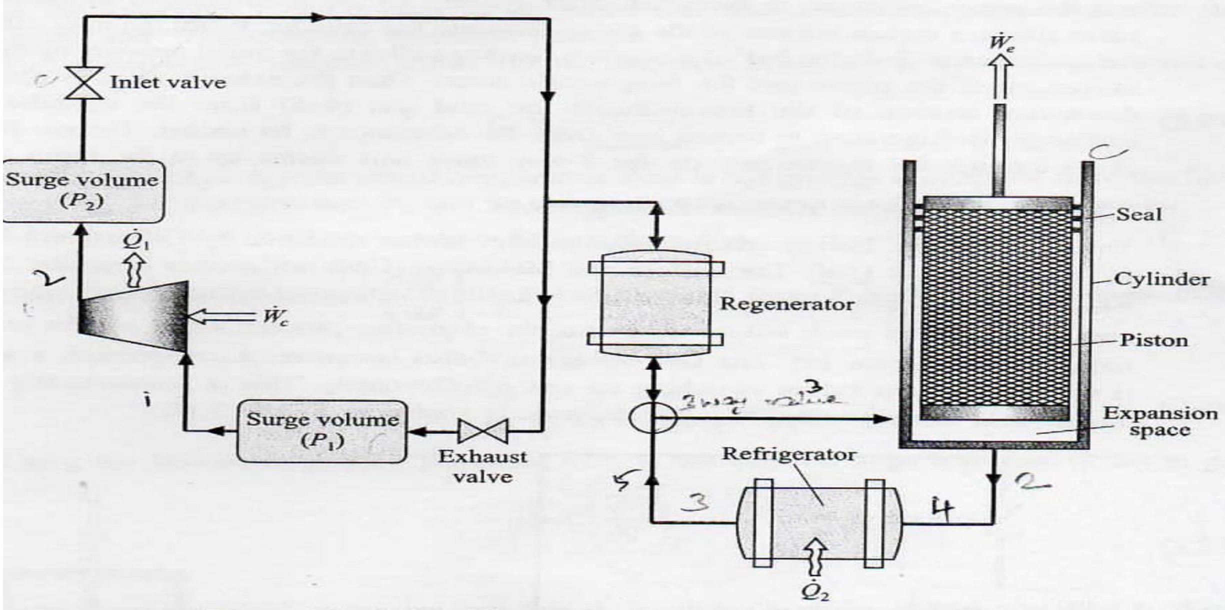


Figure Schematic representation of Solvay refrigerator

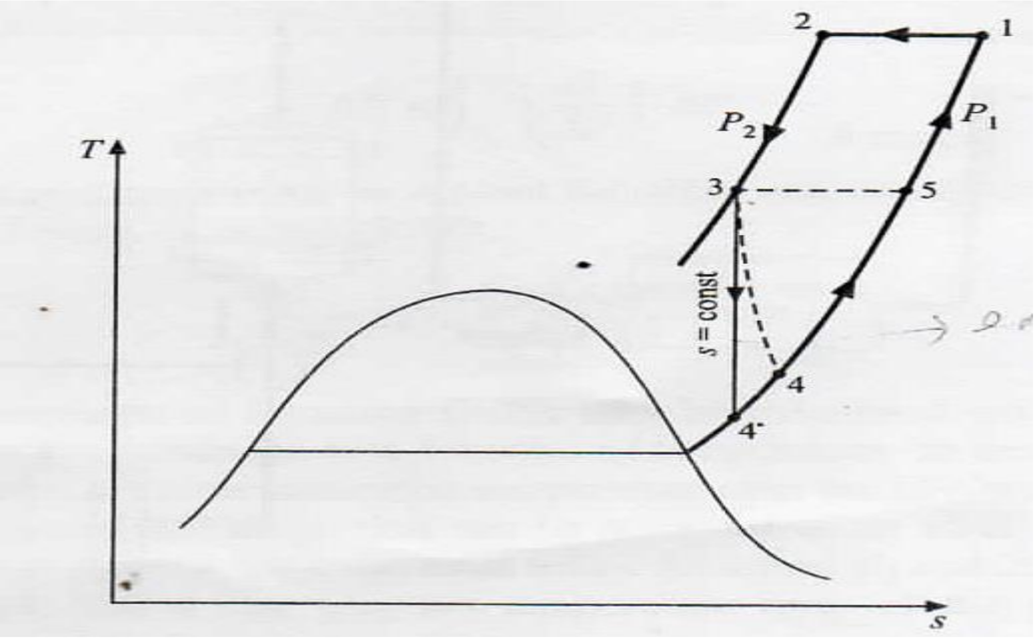
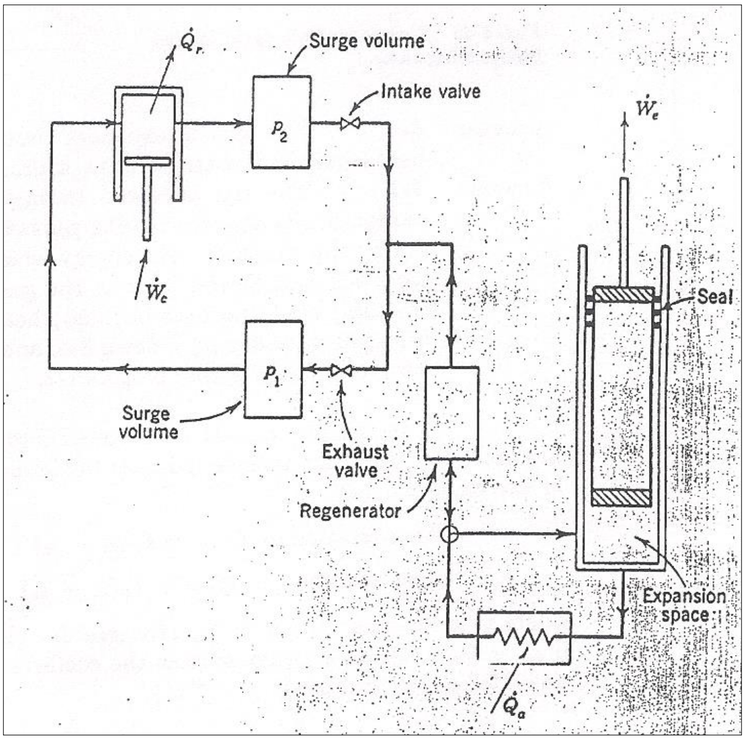


**Figure T-s diagram of Solvay refrigerator**

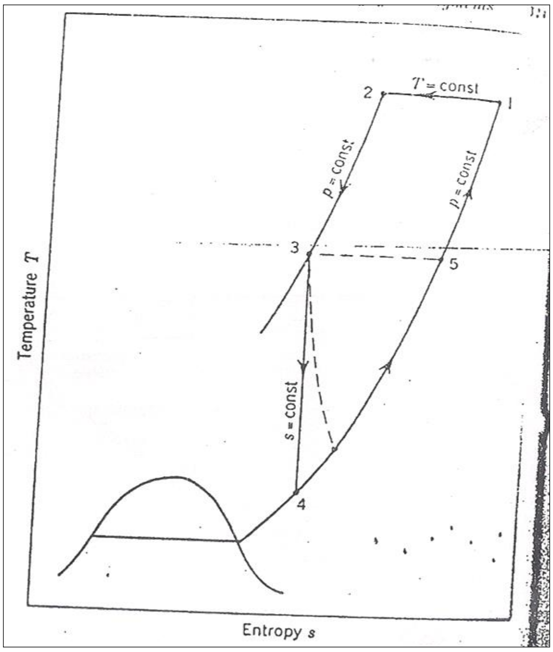
When the inlet valve opens, the gas is compressed (1-2) and the high-pressure gas enters the system. It then passes through the regenerator (2-3). During this process, the gas is cooled and the energy is stored in the regenerator system. As the piston moves upwards, the 3-way valve allows a certain volume of the gas to flow into the cylinder. When the inlet valve is closed, the gas expands (3-4) (or 3-4* if it expands isentropic ally) to the initial pressure by the downward movement of the piston and the temperature drops. Then the exhaust valve is opened and the downward motion of the piston forces the cold gas (4-5) from the cylinder via a heat exchanger (refrigerator) to absorb heat from the substance to be cooled. The gas finally passes (5-1) through the regenerator via the 3-way valve and warms up to the room temperature.

**A.D. Little Single – Single – volume Refrigerator**

In 1959 Gifford and MeMahon (8) of Arthur D. Little, Ine., reported two new refrigeration systems which were well which were well suited for miniaturezation.



**A.D. Little single – volume refrigerator**



**Path traced out by a unit mass of gas on the temperature**

These two systems were basically similar; one involved a single expansion volume, while the other utilized two expansion volume. The single – volume system is shown schematically in Fig. 5-20. If we were to consider a unit mass of gas as it flows through the system, it would trace out the path on the temperature – entropy plane as shown in Fig. 5-22. The volume coordinate refers to the volume of the expansion space plus the regenerator volume.

For the single – volume system, the sequence of operations is a follows.

Process 1 – 2 With the piston at the bottom of its stroke, the inlet valve is opened. The high – pressure gas flows into the regerator, in which the gas is cooled, and the system pressure as increased from a low pressure p1 to a higher pressure p2.

Process 2 – 3 With the inlet valve still open, the piston is raised to draw a volume of gas p3 – p2 into the cylinder. This gas has been cooled by passing through the regenerator.

Process 3 – 4 The inlet valve is closed and the gas within the cylinder is expanded (isentropically in the ideal case) to the initial pressure pv. As the gas expands, it does work on the piston, and energy is removed from the gas as work. The temperature of the gas therefore decreases.

Process 4 – 5 The exhaust valve is opened, and the piston is lowered to force the cold gas out of the cylinder. During this process, the cold gas passes through a heat exchanger to remove heat from the region to be cooled.

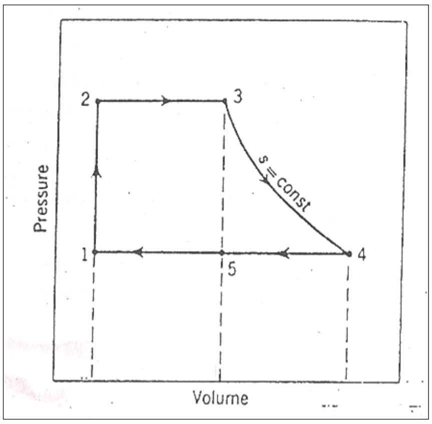
Process 5 – 1 The gas finally passes out through the regenerator, in which the cold gas is warmed back to room temperature.

Assuming that the work output during the expansion process is utilized in the compression process, the net work requirement for this system is given by

$$-\frac{W_{net}}{m}=\frac{T_2(s_1-s_2)-(h_1-h_2)}{\eta_{c,0}}-\eta_{c,m}\frac{m_c}{m}(h_3-h_4)$$

Where the first term represents the compressor work requirement and the second term represents the work output during the expansion process. The quantity me is the mass of gas expanded in the cylinder, and m is the mass of gas compressed. The energy removed from the low – temperature source is given by

$$\frac{Q_a}{m}=\frac{m_r}{m}(h_5-h_4)$$



From Eqs. (5-41) and (5-41), we see that the mass of comtned in the regenerator during the expansion process, m – m, m (1 – $m_r$/m), contributes to a decrease in the refrigeration effect of the system and an increase in the work requirements if the expansion work a utilized to aid in the compression. The regenerator volume must be kept small compared with the expansion – space volume if the coefficient performance is not to suffer.

The expansion piston in this system is similar to the Heyland piston shown in Fig. The piston is construefed of a poor has conductor so that it may be scaled at room temperature, which helps to avoid the problem of a low – temperature moving seal. This minimizes the gas leak as well as the heat inleak down the piston into the expansion volume.

## A.D. LITTLE DOUBLE – COLUME REFRIGERATOR

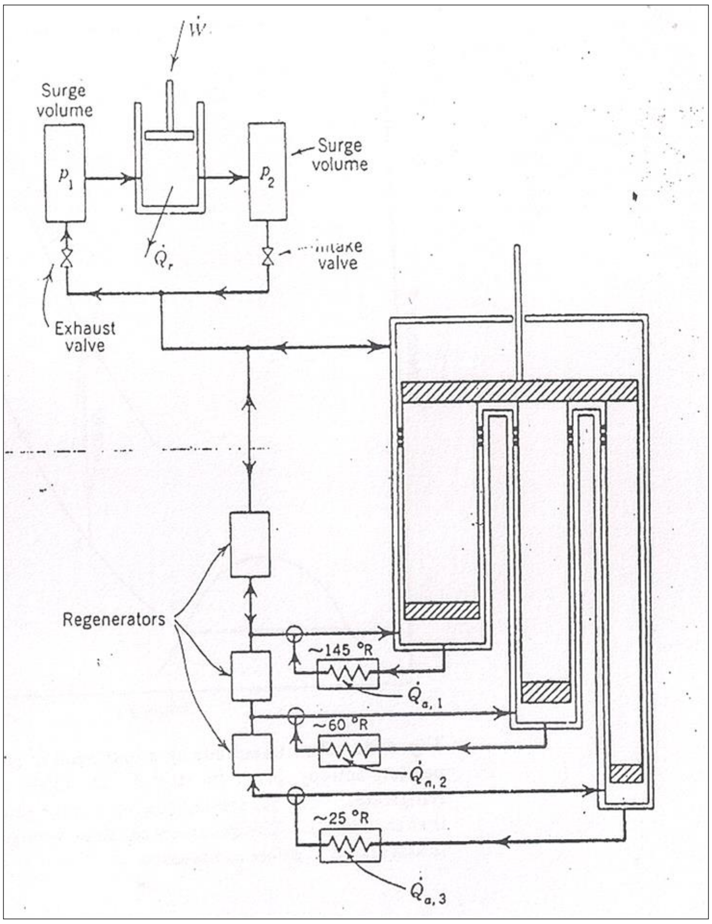
A schematic of the second novel refrigeration system developed by Arthur D. Little, Inc., the double – volume refrigerator, is shown is Fig. The path of a unit mass of working fluid is shown on the temperature – entropy plane in Fig. This system consists of a compressor, a cylinder closed at both ends, a displacer within the cylinder, and a regenerator.

The double – volume system differs from the single volume system in that no work is transferred from the system during the expansion process. The displacer serves the purpose of moving the gas from one expansion space to another and would do zero net work in the ideal case of zero pressure drop in the regenerator.

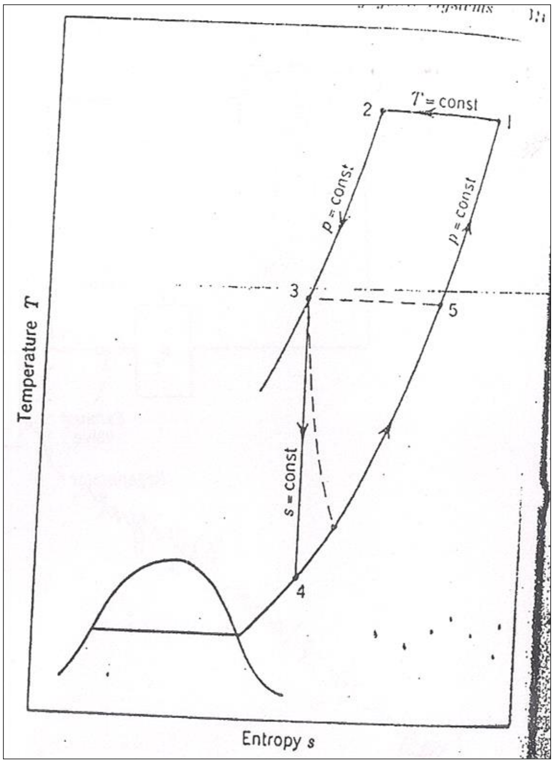
The sequence of operations for the double – volume system is as follows.

Process 1 – 2 With the displacer at the bottom of the cylinder, the inlet valve is opened and the pressure within the upper expansion space and the regenerator is increased from a low pressure p1 to p2 higher pressure p2. The volume of the lower expansion space is practically zero during this process since the displacer is at is lowest position.

Process 2 – 3 With the inlet valve still open and the exhaust valve closed, the displacer is moved to the top of the cylinder. The action moves the gas which was originally in the upper expansion space down through the regenerator to the lower expansion space. Since the gas is cooled s it passes through the regenerator, it will decrease in volume so that gas will be drawn in through the inlet valve during this process to maintain a constant pressure within the system.



Process 3 – 4 With the displacer at the top of the cylinder, the inlet valve is closed and the exhaust valve is opened, thus allowing the gas within the lower expansion space to expand to the initial pressure p1. The gas which is finally within the lower expansion space does work to push out the gas which leaves during this process; therefore, energy is removed as work from the gas finally left in the lower expansion space. This causes the gas within the lower expansion space. This causes the gas in the lower expansion space to drop to a low temperature. This process is similar to the expansion process in the Simon liquefier



Process 4 – 5 The low – temperature gas is forced out of the lower expansion space by moving the displacer downward to the bottom of the cylinder. This cold gas flows through a heat exchanger in which heat is transferred to the as from the low – temperature source to be cooled.

Process 5 – 1 The gas flows from the heat exchanger through the regenerator, in which the gas is warmed back to near ambient temperature.

The net work requirement for this system is given by

$$-\frac{W}{m} = T_1\left(s_1 - s_2 - (h_1 - h_2\right)$$

The energy removed from the low temperature source is given by

$$\frac{W_a}{m} = \frac{m_1}{m}\left(h_3 - h_4\right) = h_1 - h_2$$

Where mr is the mass of gas within the lower expansion space at the end of the expansion process 3 – 1 and m is the total mass compressed. Since the volume of the expansion space remains constant during the expansion process, the mass ratio m/m may be written in terms of the density ratio.

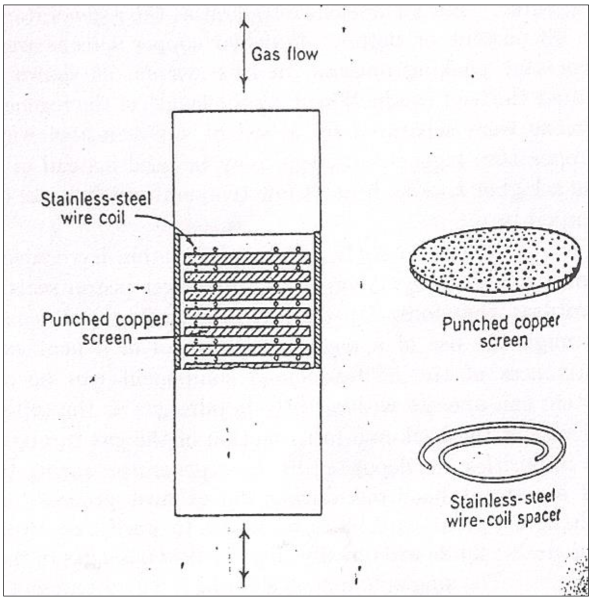
$$\frac{m_1}{m} = \frac{p_4}{p_3}$$

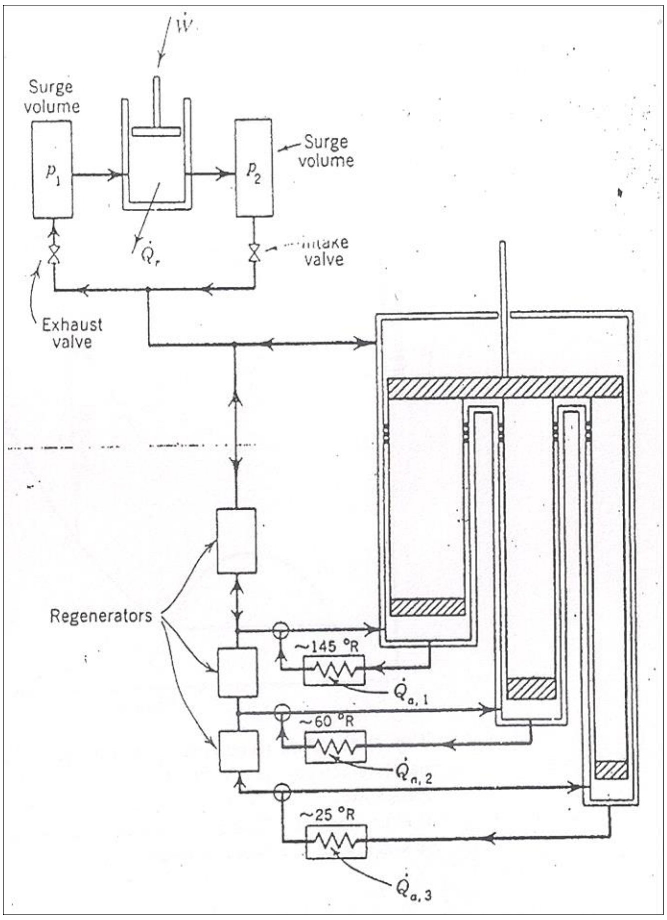
In both the single – volume and the double – volume refrigerator the regenerator is a critical component, as in the case of the Philips refrigerator. For an efficient refrigerator, the regenerator effectiveness must be 98 percent or better. Punched copper screens were used as the regenerator packing material in this system, as shown in Fig. 5-25. To reduce the heat conduction along the length of the regenerator, the punched screens were separated by a coil or stainless – steel wire. For very – low temperature regenerators, lead may be used instead of copper, since lead has a higher specific heat at low temperatures because of its lower Debye temperature.

The two A.D. Little refrigerators, have several advantages in common. The engine valves and displacer piston seals are at room temperature; therefore, low – temperature sealing problems are eliminated. Through the use of a regenerator instead of a heat exchanger, instead of a heat exchanger, high effectiveness of the heat – exchange component can be attained, and the system can operate with slightly impure gas as the refrigeration medium. Because of the back – and – forth motion of the gas through the regenerator, the impurities are deposited in the regenerator during the intake process and are swept back out during the exhaust process. In ordinary heat exchangers, great care must be taken to purify the working fluid, since condensable fluids will quickly clog the flow passages in the heat exchanger.

The single – volume system has two advantages over the double volume system: (1) The coefficient of performance of the single – volume system is inherently higher than that of the double – volume system, since more energy is removed from the working fluid by the work – producing process. (2) In the double – volume system, a small motor is required to move the displacer back and forth while the expanding gas moves the piston in the single – volume. On the other hand, the double – volume system has some advantages over the single – volume system: (1) There is practically no leakage past the displacer in the double – volume system because of the very small pressure difference across the displacer seal (2) The displacer and crank arm in the double – volume system need not I designed to support a large force therefore, the motion transmission system can be quite simple and subject to fewer problems with vibration.

One of the major advantages of the double – volume system is the move with which it can be adapted to multistage. A three – system frigerator is shown sechematically in Fig. 5-26. By using helium as the working fluid, refrigeration may be attained at three different temperature levels with only a slight increase in the complexity of the overall system. Compare the system shown in Fig. 5-26 with the caseade system shown in Fig. 5-11. And this advantage becomes more apparent. All the valves in the A.D. Little refrigerator still operate at room temperature, and the three displacers are operated by a single actuator. By multistaging temperatures near 25 or 30˚R can be attained with much less work than by using a single – stage system.


Gas flow
Stainless-steel wire coil
Punched copper screen
Punched copper screen
Stainless-steel wire-coil spacer


$\dot{W}$
Surge volume
$p_1$
$p_2$
Surge volume
$\dot{Q}_r$
Intake valve
Exhaust valve
Regenerators
~145 °R
$\dot{Q}_{a,1}$
~60 °R
$\dot{Q}_{a,2}$
~25 °R
$\dot{Q}_{a,3}$