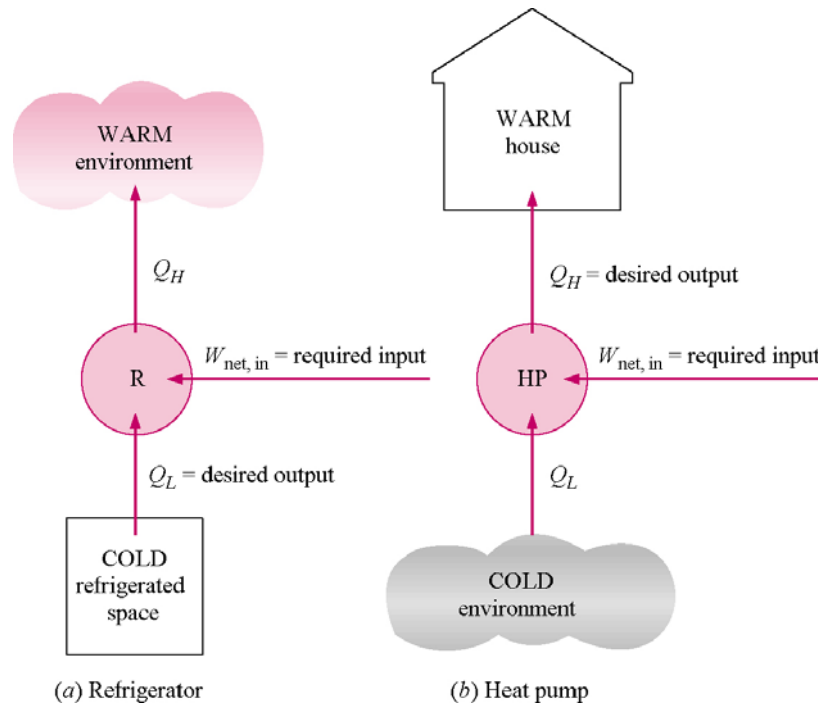# Chapter 10: Refrigeration Cycles

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

(a) Refrigerator   (b) Heat pump

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

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

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

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

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

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

Can you show this to be true?

Refrigerators, air conditioners, and heat pumps are rated with a SEER number or seasonal adjusted energy efficiency ratio. The SEER is defined as the Btu/hr of heat transferred per watt of work energy input. The Btu is the British thermal unit and is equivalent to 778 ft-lbf of work (1 W = 3.4122 Btu/hr). An EER of 10 yields a COP of 2.9.

Refrigeration systems are also rated in terms of tons of refrigeration. One ton of refrigeration is equivalent to 12,000 Btu/hr or 211 kJ/min. How did the term "ton of cooling" originate?

Reversed Carnot Refrigerator and Heat Pump

Shown below are the cyclic refrigeration device operating between two constant temperature reservoirs and the *T-s* diagram for the working fluid when the reversed Carnot cycle is used. Recall that in the Carnot cycle heat transfers take place at constant temperature. If our interest is the cooling load, the cycle is called the Carnot refrigerator. If our interest is the heat load, the cycle is called the Carnot heat pump.
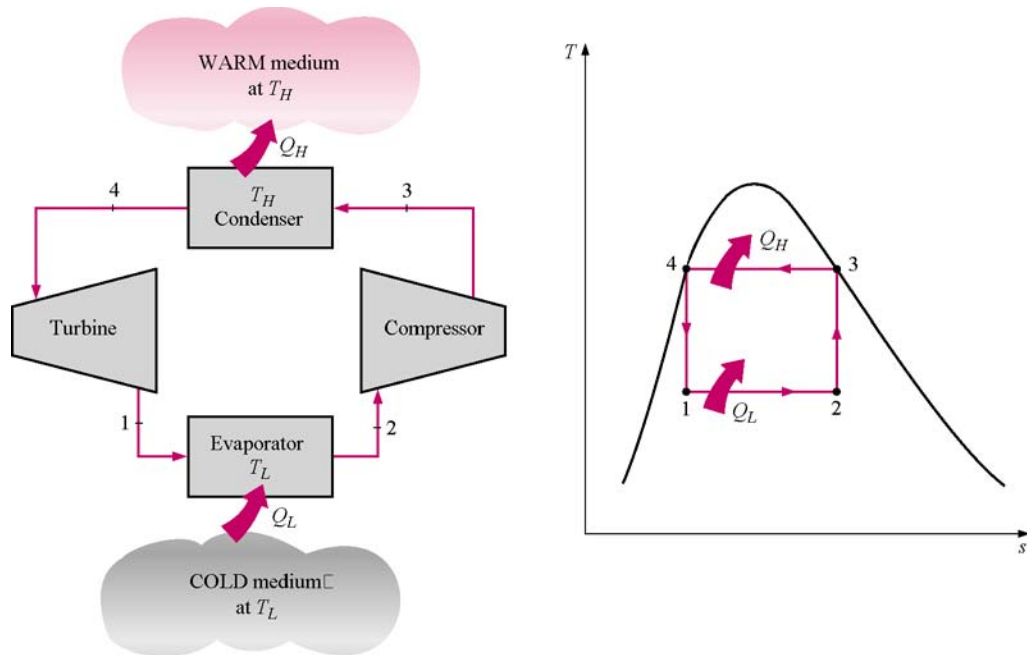
The standard of comparison for refrigeration cycles is the *reversed Carnot cycle.* A refrigerator or heat pump that operates on the reversed Carnot cycle is called a *Carnot refrigerator* or a *Carnot heat pump,* and their COPs are

$$COP_{R,Carnot} = \frac{1}{T_H / T_L - 1} = \frac{T_L}{T_H - T_L}$$

$$COP_{HP,Carnot} = \frac{1}{1 - T_L / T_H} = \frac{T_H}{T_H - T_L}$$

Notice that a turbine is used for the expansion process between the high and low-temperatures. While the work interactions for the cycle are not indicated on the figure, the work produced by the turbine helps supply some of the work required by the compressor from external sources.

Why not use the reversed Carnot refrigeration cycle?
• Easier to compress vapor only and not liquid-vapor mixture.
• Cheaper to have irreversible expansion through an expansion valve.

What problems result from using the turbine?

The Vapor-Compression Refrigeration Cycle

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

The ideal vapor-compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

Process	Description
1-2	Isentropic compression
2-3	Constant pressure heat rejection in the condenser
3-4	Throttling in an expansion valve
4-1	Constant pressure heat addition in the evaporator
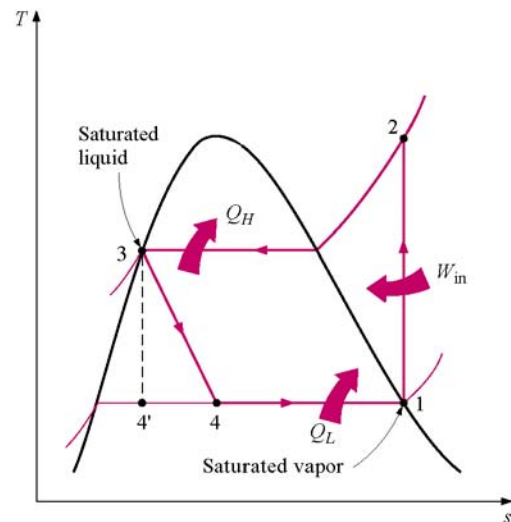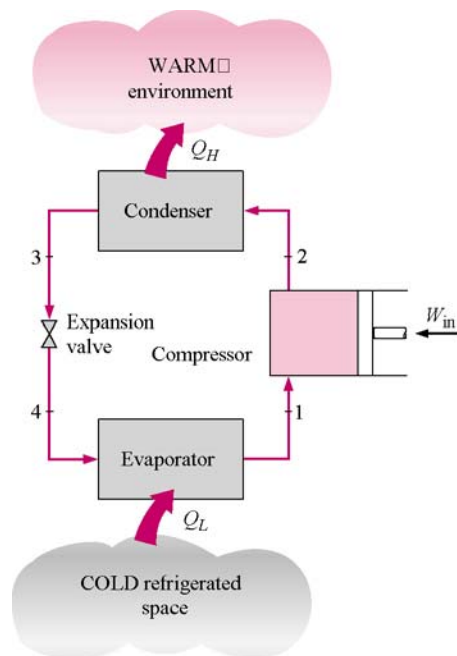
The *P-h* diagram is another convenient diagram often used to illustrate the refrigeration cycle.
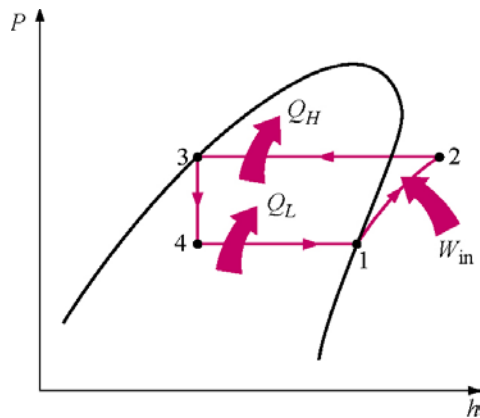
The ordinary household refrigerator is a good example of the application of this cycle.
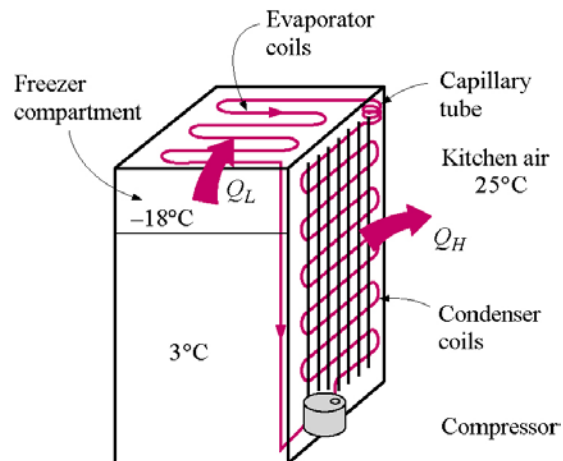
Results of First and Second Law Analysis for Steady-Flow

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

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

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

Example 10-1

Refrigerant-134a is the working fluid in an ideal compression refrigeration cycle. The refrigerant leaves the evaporator at -20°C and has a condenser pressure of 0.9 MPa. The mass flow rate is 3 kg/min. Find COP_R and $COP_{R, Carnot}$ for the same T_{max} and T_{min}, and the tons of refrigeration.

Using the Refrigerant-134a Tables, we have

State 1
Compressor inlet
$T_1 = -20^o C$
$x_1 = 1.0$

$h_1 = 235.31 \dfrac{kJ}{kg}$

$s_1 = 0.9332 \dfrac{kJ}{kg \cdot K}$

State 2
Compressor exit
$P_{2s} = P_2 = 900\,kPa$
$s_{2s} = s_1 = 0.9332 \dfrac{kJ}{kg \cdot K}$

$h_{2s} = 275.1 \dfrac{kJ}{kg}$

$T_{2s} = 44.74^o C$

State 3
Condenser exit
$P_3 = 900\,kPa$
$x_3 = 0.0$

$h_3 = 99.56 \dfrac{kJ}{kg}$

$s_3 = 0.3656 \dfrac{kJ}{kg \cdot K}$

State 4
Throttle exit
$T_4 = T_1 = -20^o C$
$h_4 = h_3$

$x_4 = 0.357$

$s_4 = 0.3670 \dfrac{kJ}{kg \cdot K}$

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

$$= \frac{(235.31 - 99.56)\dfrac{kJ}{kg}}{(275.1 - 235.31)\dfrac{kJ}{kg}}$$

$$= 3.41$$

The tons of refrigeration, often called the cooling load or refrigeration effect, are

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$

$$= 3\frac{kg}{min}(235.31 - 99.56)\frac{kJ}{kg}\frac{1\,Ton}{211\dfrac{kJ}{min}}$$

$$= 1.93\,Ton$$

$$COP_{R,\,Carnot} = \frac{T_L}{T_H - T_L}$$

$$= \frac{(-20 + 273)\,K}{(44.74 - (-20))K}$$

$$= 3.91$$

Another measure of the effectiveness of the refrigeration cycle is how much input power to the compressor, in horsepower, is required for each ton of cooling.

The unit conversion is 4.715 hp per ton of cooling.

$$\frac{\dot{W}_{net,in}}{\dot{Q}_L} = \frac{4.715}{COP_R}$$

$$= \frac{4.715}{3.41} \frac{hp}{Ton}$$

$$= 1.1 \frac{hp}{Ton}$$

Actual Vapor-Compression Refrigeration Cycle

Heat Pump Systems

HEAT PUMP OPERATION – HEATING MODE

Outdoor coil
Reversing valve
Indoor coil
Fan
Fan
Compressor
Expansion valve

— High-pressure liquid
— Low-pressure liquid–vapor
— Low-pressure vapor
— High-pressure vapor

HEAT PUMP OPERATION – COOLING MODE

Outdoor coil
Reversing valve
Indoor coil
Fan
Fan
Compressor
Expansion valve
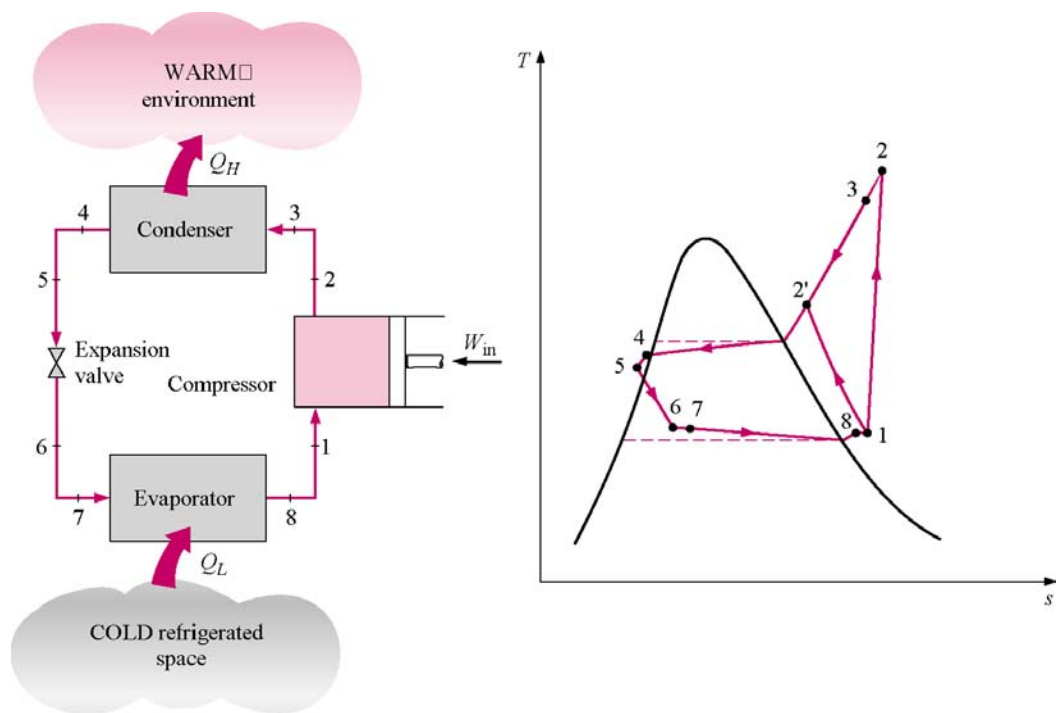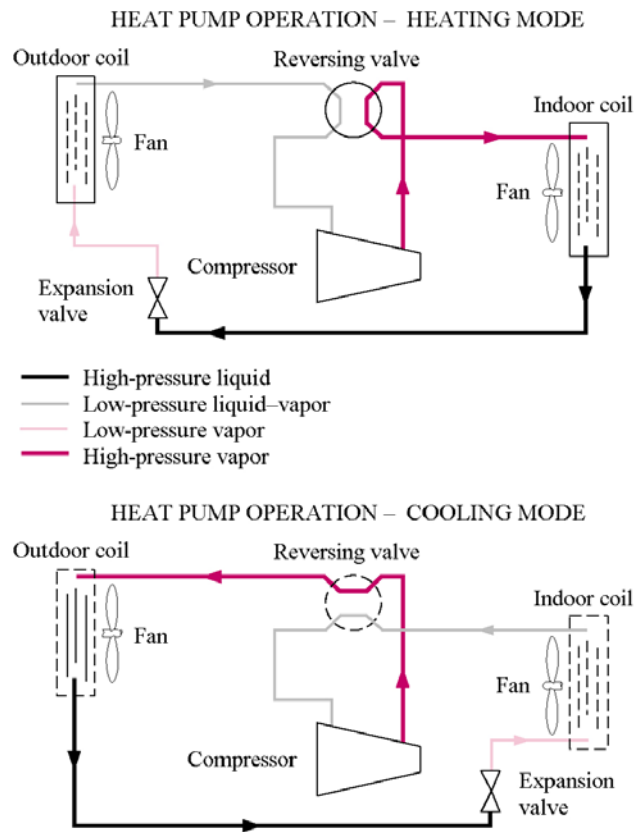
Other Refrigeration Cycles

Cascade refrigeration systems

Very low temperatures can be achieved by operating two or more vapor-compression systems in series, called *cascading*. The COP of a refrigeration system also increases as a result of cascading.
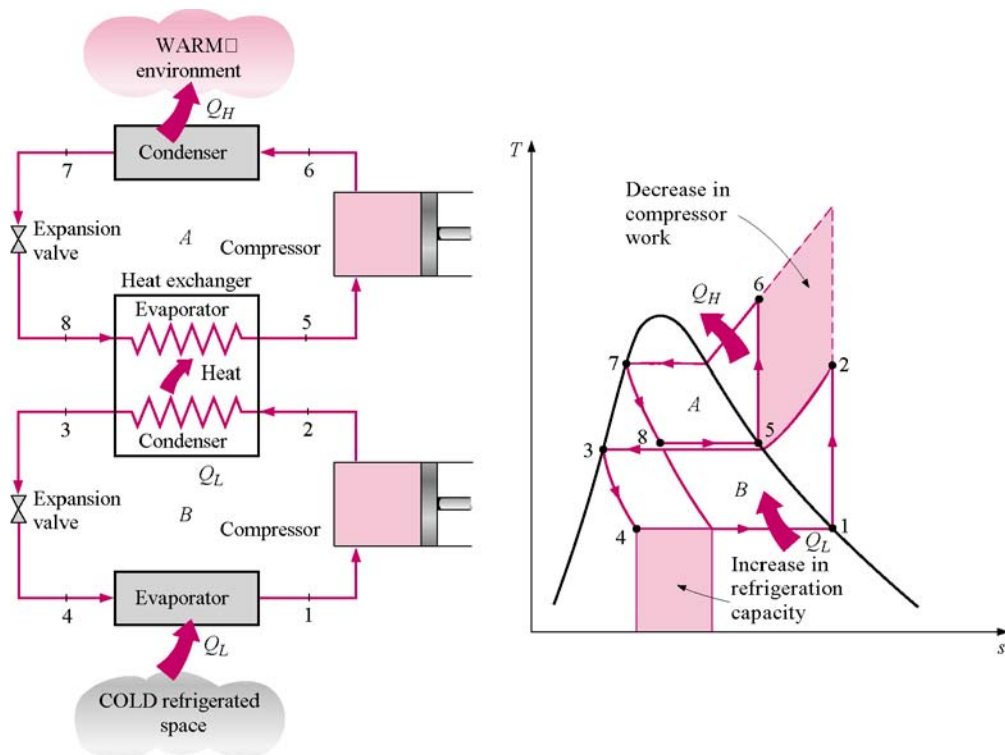
Multistage compression refrigeration systems
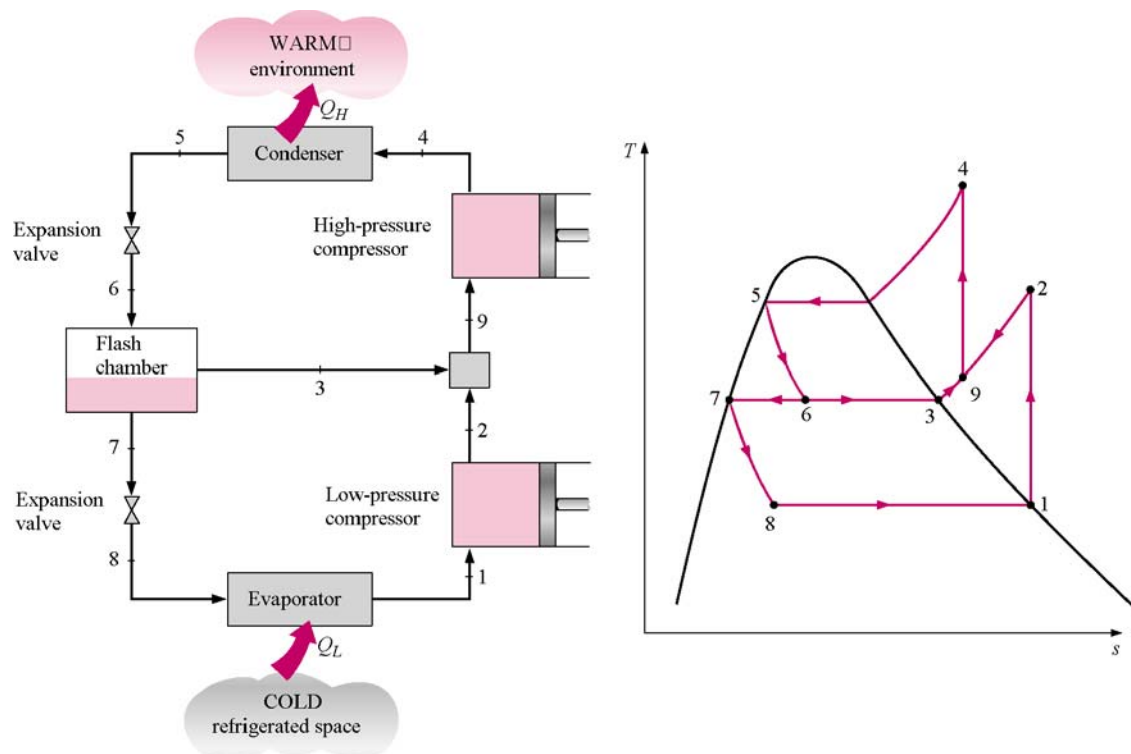
Multipurpose refrigeration systems

A refrigerator with a single compressor can provide refrigeration at several temperatures by throttling the refrigerant in stages.
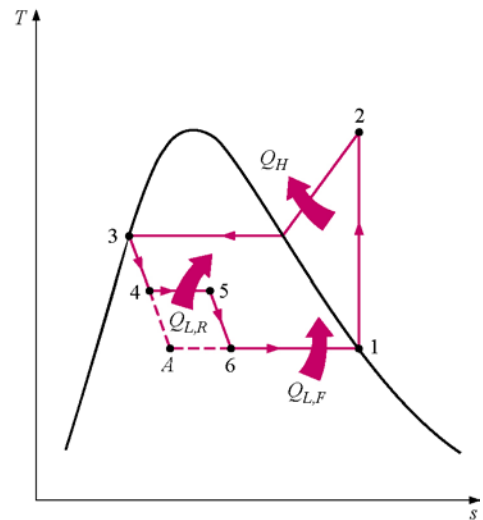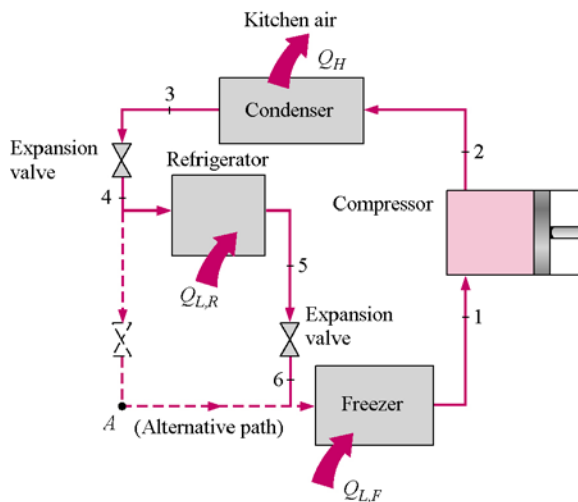
Liquefaction of gases

Another way of improving the performance of a vapor-compression refrigeration system is by using *multistage compression with regenerative cooling.* The vapor-compression refrigeration cycle can also be used to liquefy gases after some modifications.
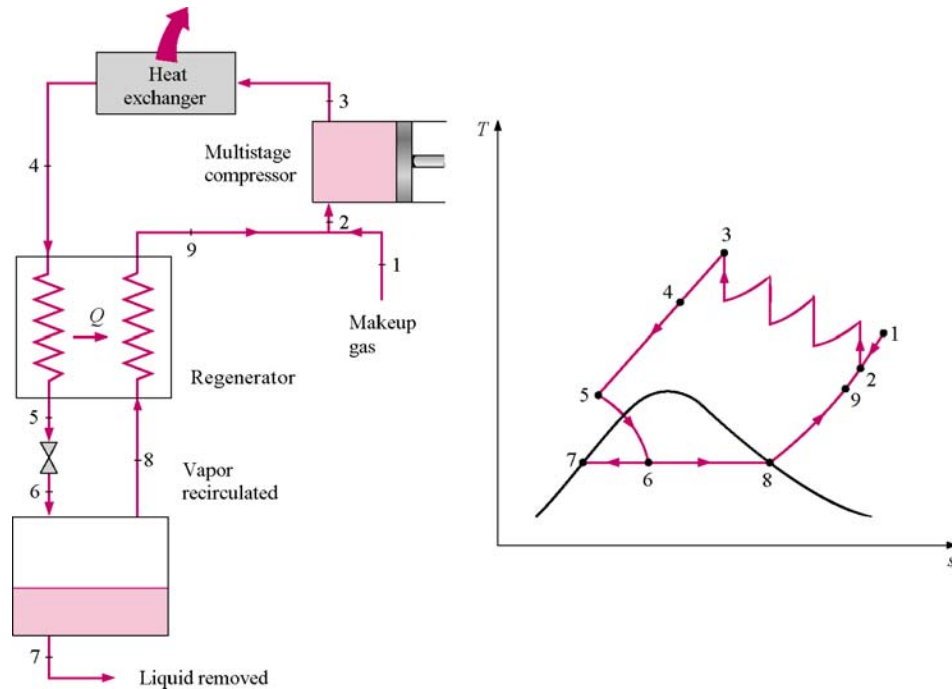
Gas Refrigeration Systems

The power cycles can be used as refrigeration cycles by simply reversing them. Of these, the *reversed Brayton cycle,* which is also known as the *gas refrigeration cycle,* is used to cool aircraft and to obtain very low (cryogenic) temperatures after it is modified with regeneration. The work output of the turbine can be used to reduce the work input requirements to the compressor. Thus, the COP of a gas refrigeration cycle is

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{q_L}{w_{comp,in} - w_{turb,out}}$$
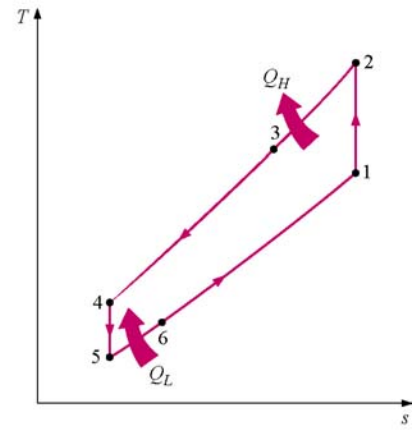
Absorption Refrigeration Systems

Another form of refrigeration that becomes economically attractive when there is a source of inexpensive heat energy at a temperature of 100 to 200°C is *absorption refrigeration,* where the refrigerant is absorbed by a transport medium and compressed in liquid form. The most widely used absorption refrigeration system is the ammonia-water system, where ammonia serves as the refrigerant and water as the transport medium. The work input to the pump is usually very small, and the COP of absorption refrigeration systems is defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{Q_{gen} + W_{pump,in}} \cong \frac{Q_L}{Q_{gen}}$$
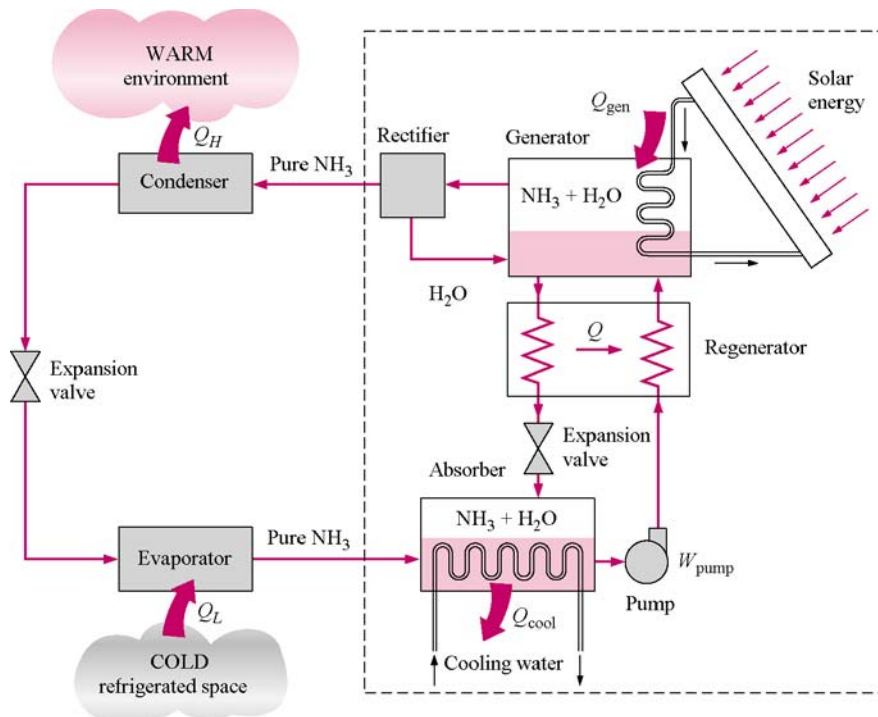
Thermoelectric Refrigeration Systems

A refrigeration effect can also be achieved without using any moving parts by simply passing a small current through a closed circuit made up of two dissimilar materials. This effect is called the *Peltier effect,* and a refrigerator that works on this principle is called a *thermoelectric refrigerator.*

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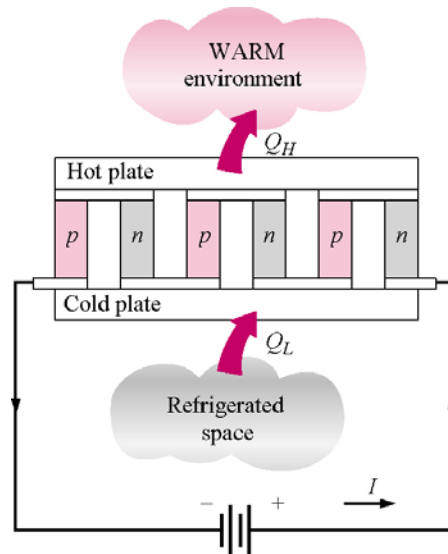
UNIT-III

PSYCHROMETRY

</div>

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:

<div align="center">

Table .1 Composition of dry air

</div>

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
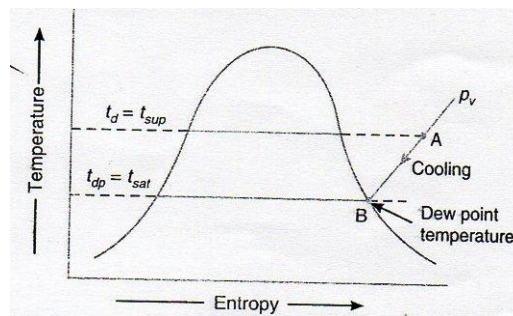

Fig.1. T-s diagram

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.
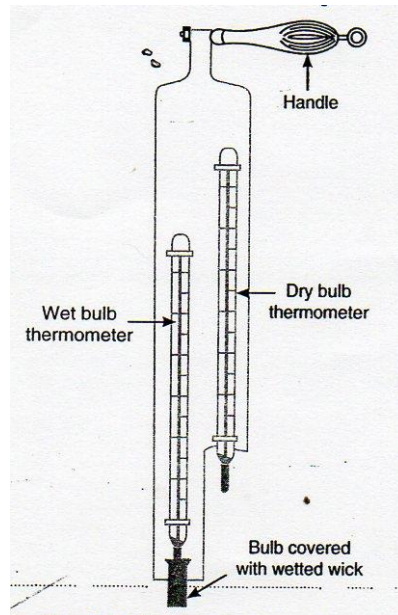
Fig.2, Sling psychrometer

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, T_a, 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 $\qquad\qquad\qquad v_a = v_v$

and $\qquad\qquad\qquad T_a = T_v, = T_d$ ... (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, $\qquad W = \dfrac{m_v}{m_a} = \dfrac{R_a\, P_v}{R_v\, P_a}$

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

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

$$\dots\; (\because\; p_b = p_a + p_v)$$
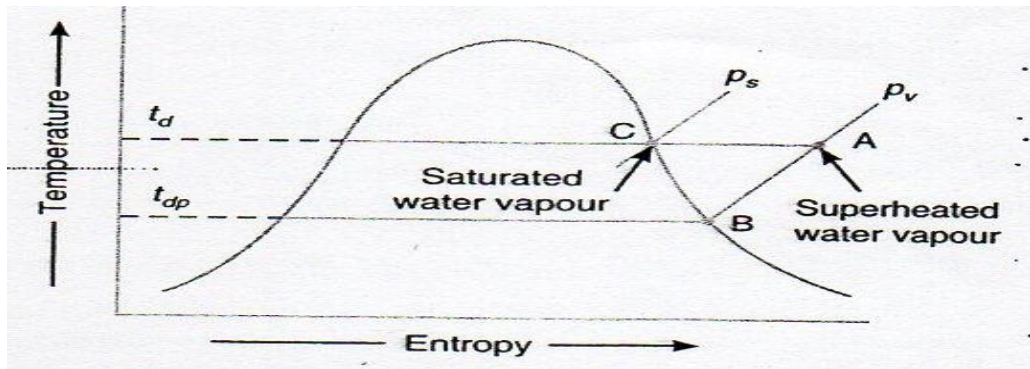
Fig.3 T-s diagram

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 μ. 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, $\emptyset$) = Pv / Ps is equal to zero, then the humidity ratio, W = 0, i.e. for dry air, $\mu = 0$.

(c) If the relative humidity, $\emptyset$) 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 $\emptyset$. 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 $\qquad\qquad\qquad\qquad\qquad Rv = Rs = 0.461 \text{ kJ/kg K}$

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

$\qquad\qquad P_b$ = Barometric pressure,

$$t_d = \text{Dry bulb temperature, and}$$

$$t_w = \text{Wet bulb temperature.}$$

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 m³ of dry air.

Let v_v = Volume of water vapour in m³/kg of dry air at its partial pressure,

v_a = Volume of dry air in m³/kg of dry air at its partial pressure,

ρ_v, = Density of water vapour in kg/m³ corresponding to its partial pressure and dry bulb

temperature t_d, and

ρ_a = Density of dry air in kg/m³ of dry air.

We know that mass of water vapour,

$$m_v = v_v \, \rho_v$$

and mass of dry air, $\qquad m_a = v_a \, \rho_a$

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 $\qquad p_a \, v_a = m_a \, R_a \, T_d$

Since $v_a = \dfrac{1}{\rho_a}$ 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 ρ_a in equation (*iii*), we have

$$\rho_v = \frac{W \, p_a}{R_a \, T_d} = \frac{W \, (p_b - p_v)}{R_a \, T_d} \qquad \dots (\because \ p_b = p_a + p_v)$$

where $\qquad p_a$ = Pressure of air in kN/m²,

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,

$$ph = 740 \text{ mm of Hg ... (Given)}$$

$$= 740 \text{ x } 133.3 = 98\ 642 \text{ N/m}^2 \ \dots \ (\because \text{ mm of Hg } = 133.3 \text{ N/m}^2)$$

$$= 0.986\ 42 \text{ bar} \qquad \dots. \ \because 1 \text{ bar} = 10^5 \text{ N/m}^2)$$

$\therefore$ Partial pressure of water vapour,

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

$$= 0.023\ 37 - \frac{(0.986\ 42 - 0.02337)(30 - 20)}{1544 - 1.44 \times 20}$$

$$= 0.023\ 37 - 0.006\ 36 = 0.017\ 01 \text{ bar}$$

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

$$t_{dp} = 15°C \text{ Ans}$$

2. Relative humidity

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30℃ is

$$Ps = 0.042\ 42 \text{ bar}$$

We know the relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{0.01701}{0.04242} = 0.40 \quad \text{or} \quad 40\% \quad \text{Ans.}$$

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,

$$\mu = \frac{W}{W_s} = \frac{0.010\,914}{0.027\,945} = 0.391 \quad \text{or} \quad 39.1\% \quad \text{Ans.}$$

Note : The degree of saturation (μ) may also be calculated from the following relation :

$$\mu = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right)$$

$$= \frac{0.01701}{0.04242}\left[\frac{0.98642 - 0.04242}{0.98642 - 0.01701}\right]$$

$$= 0.391 \quad \text{or} \quad 39.1\% \quad \text{Ans.}$$

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

$$h_{fgdp} = 2466.1 \text{ kJ/kg}$$

$\therefore$ 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°C and a wet bulb temperature of 18°C. 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.

$$p_w = 0.020\ 62\ bar = 0.020\ 62 \times 10^5 = 2062\ N/m^2$$

$$= \frac{2062}{133.3} = 15.47\ mm\ of\ Hg \qquad \dots(\because\ 1\ mm\ of\ Hg = 133.3\ N/m^2)$$

We know that $\qquad p_v = p_w - \dfrac{(p_b - p_w)(t_d - t_w)}{1544 - 1.44\ t_w}$

$$= 15.47 - \frac{(756 - 15.47)(30 - 18)}{1544 - 1.44 \times 18}\ mm\ of\ Hg$$

$$= 15.47 - 5.85 = 9.62\ mm\ of\ Hg$$

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 30°C is

$$p_s = 0.042\ 42\ bar = 0.042\ 42 \times 10^5 = 4242\ N/m^2$$

$$= \frac{4242}{133.3} = 31.8\ mm\ of\ Hg$$

We know that the relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{9.62}{31.8} = 0.3022\ or\ 30.22\%$$

Specific humidity

We know that specific humidity,

$$W = \frac{0.622\ p_v}{p_b - p_v} = \frac{0.622 \times 9.62}{756 - 9.62} = 0.008\ kg/kg\ of\ dry\ air\ \ \text{Ans.}$$

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,

$$h_{fgdp} = 2476.5 \text{ kJ/kg}$$

We know that enthalpy of air per kg of dry air,

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

$= 1.022 \text{ x } 30 + 0.008\ (2476.5 + 2.3 \text{ x } 10.6)$

$= 30.66 + 20 = 50.66 \text{ kJ/kg of dry air Ans.}$

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

$$= 0.8585 \times \frac{760}{746.38} = 0.8741 \text{ kg/kg of dry air} \quad \text{Ans.}$$

Note : The volume of mixture per kg of dry air may be calculated as discussed below :

We know that $\qquad v = v_a = \dfrac{R_a\ T_d}{p_a}$

where $\qquad R_a$ = 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

$$= P_b - P_v = 756 - 9.62 = 746.38 \text{ mm of Hg}$$

$$= 746.38 \times 133.3 = 994\ 92 \text{ N/m}^2$$

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 $\qquad 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
$$P_v = 12.16/0.638 = 19.06 \text{ mm of Hg}$$
$$= 19.06 \times 133.3 = 2540.6 \text{ N/m}^2 \text{ Ans.}$$

2. Relative humidity

From steam tables, we find that the saturation pressure of vapour corresponding to dry bulb temperature of 28`C is

$$P_s = 0.03778 \text{ bar} = 3778 \text{ N/m}^2$$

∴ Relative humidity,

$$\phi = \frac{p_v}{p_s} = \frac{2540.6}{3778} = 0.672 \text{ or } 67.2\% \text{ Ans.}$$

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/m² (0.025406 bar), the dew point temperature is,

$$t_{dp} = 21.1° \text{ C Ans.}$$

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.

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

$$= 1.022 \text{ x } 28 + 0.016 \, (2451.76 + 2.3 \text{ x } 21\_1)$$

$$= 28.62 + 40 \text{ - } 68.62 \text{ kJ/kg of dry air Ans.}$$

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.
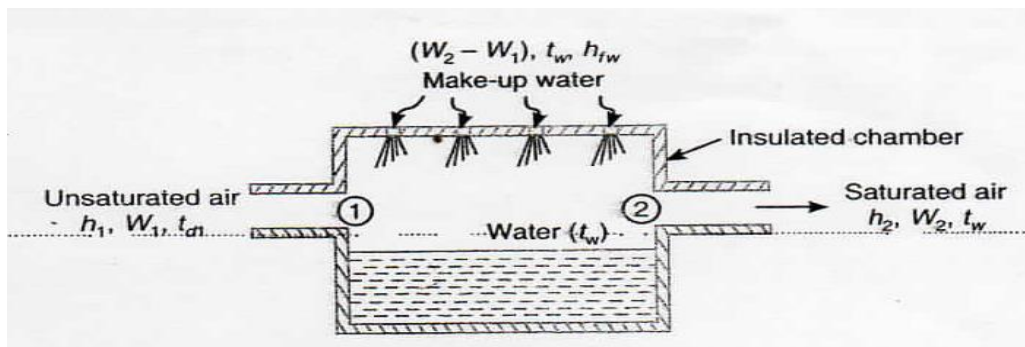

Fig.4 Adiabatic saturation of air.

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.*

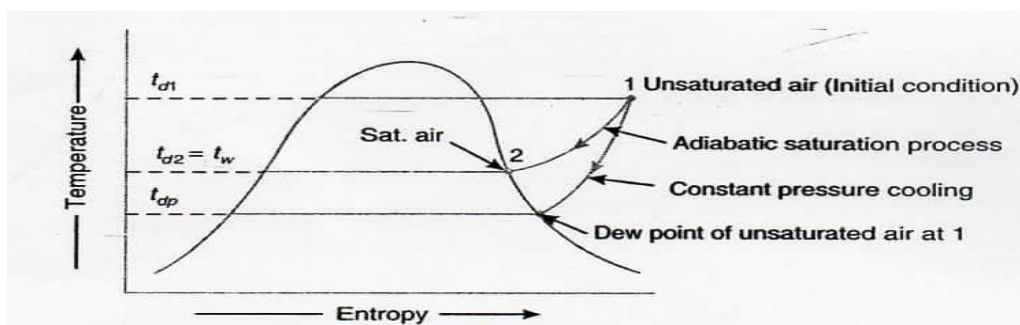The adiabatic saturation process can be represented on *T-s* diagram as shown by the curve 1-2 in Fig.5.

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),

$$h_1 + (W_2 - W_1) h_{fw} = h_2 \qquad \qquad \ldots (i)$$

or

$$h_1 - W_1 h_{fw} = h_2 - W_2 h_{fw} \qquad \qquad \ldots (ii)$$

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).
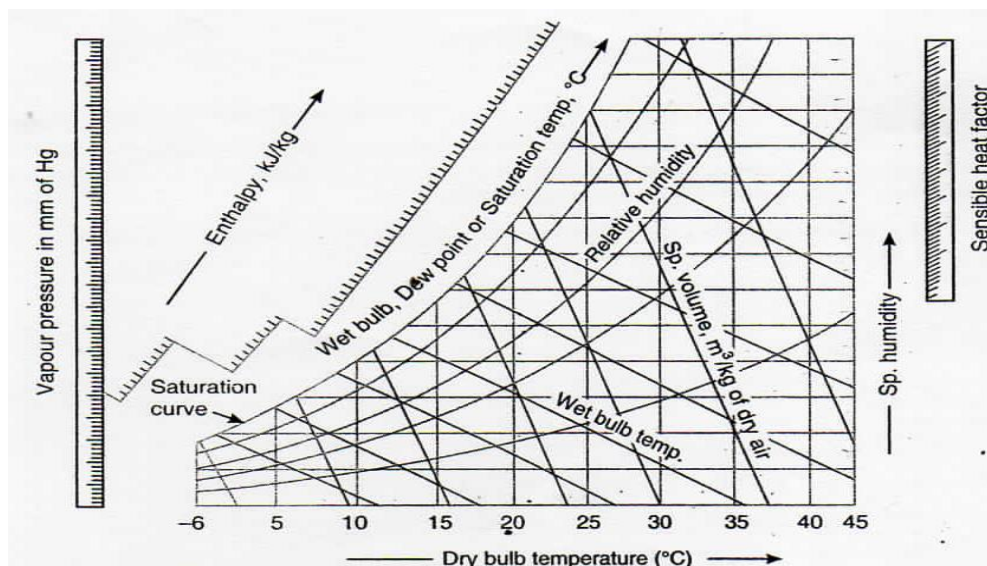

Fig. 6 Psychrometric chart.

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.
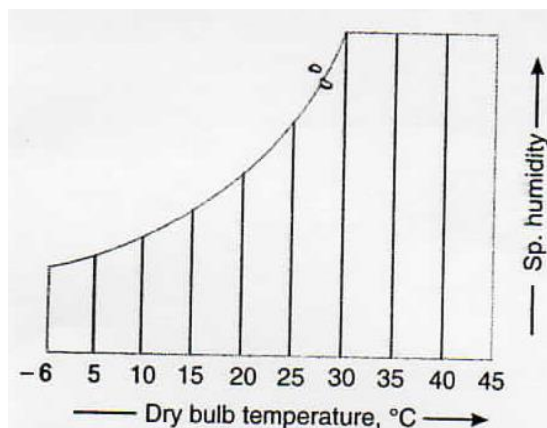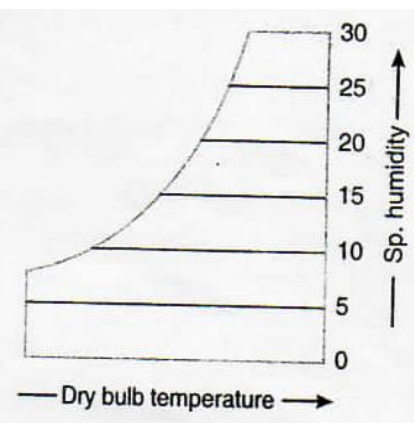

Fig.7. Dry bulb temperature lines.   Fig. 8. Specific humidity lines.

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.
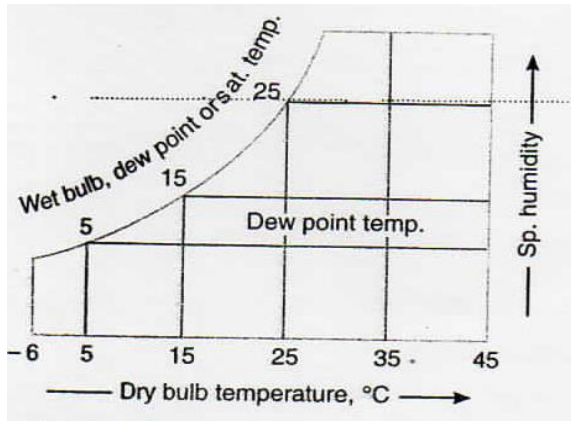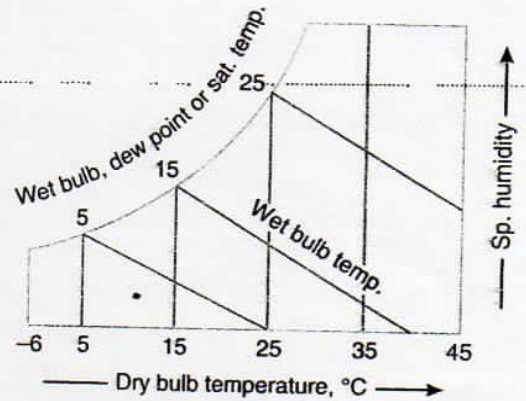
Fig. 9 Dew point temperature lines.   Fig.10 Wet bulb temperature lines.

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.
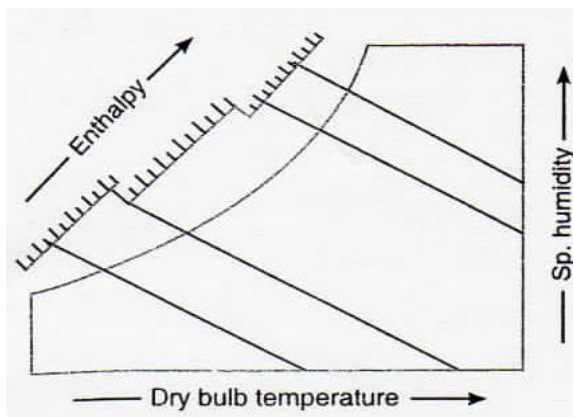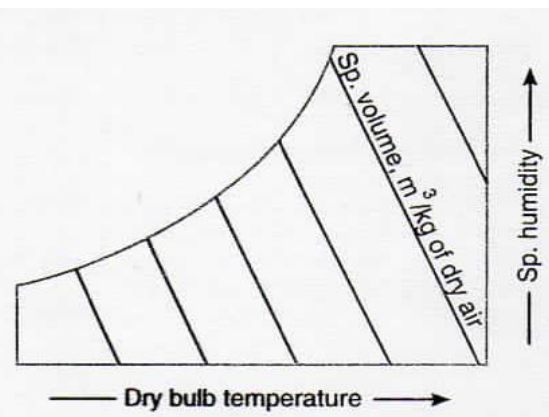

Fig. 11. Enthalpy lines.   Fig. 12. Specific volume lines.

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.
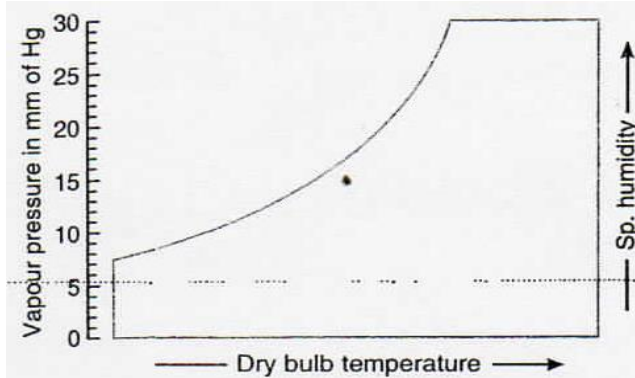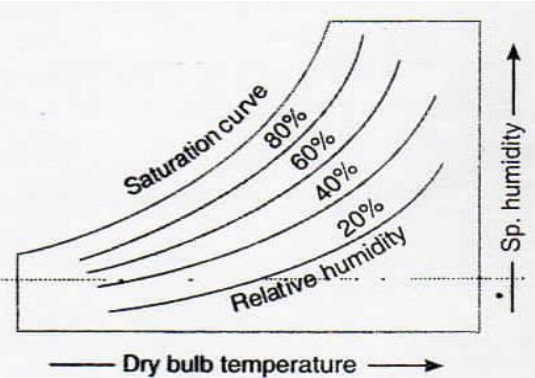

Fig. 13. Vapour pressure lines.   Fig. 14. Relative humidity lines.

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:

$$\begin{aligned}
\text{Heat added,} \quad q &= h_2 - h_1 \\
&= c_{pa}\,(t_{d2} - t_{d1}) + W\,c_{ps}\,(t_{d2} - t_{d1}) \\
&= (c_{pa} + W\,c_{ps})\,(t_{d2} - t_{d1}) = c_{pm}\,(t_{d2} - t_{d1})
\end{aligned}$$

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.

$$\therefore \quad \text{Heat added,} \quad q = 1.022\,(t_{d2} - t_{d1}) \text{ kJ/kg}$$
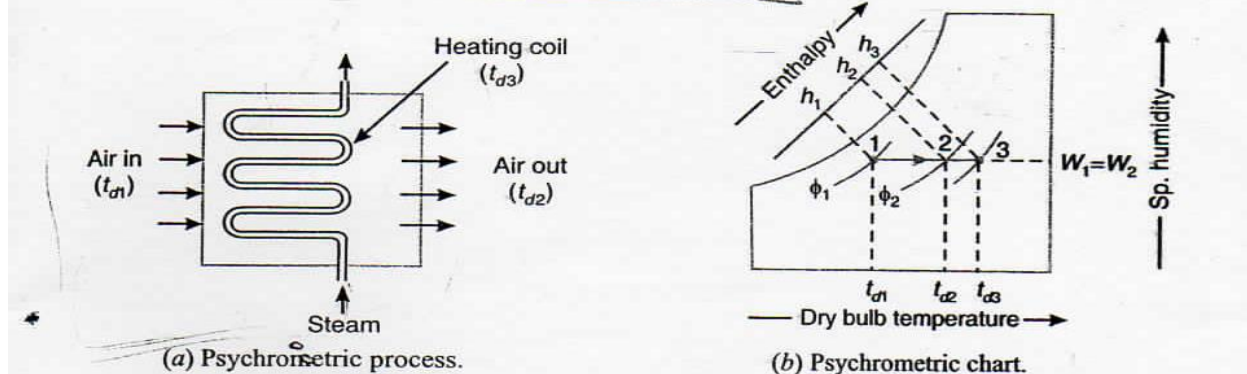
Fig.15 Sensible heating

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
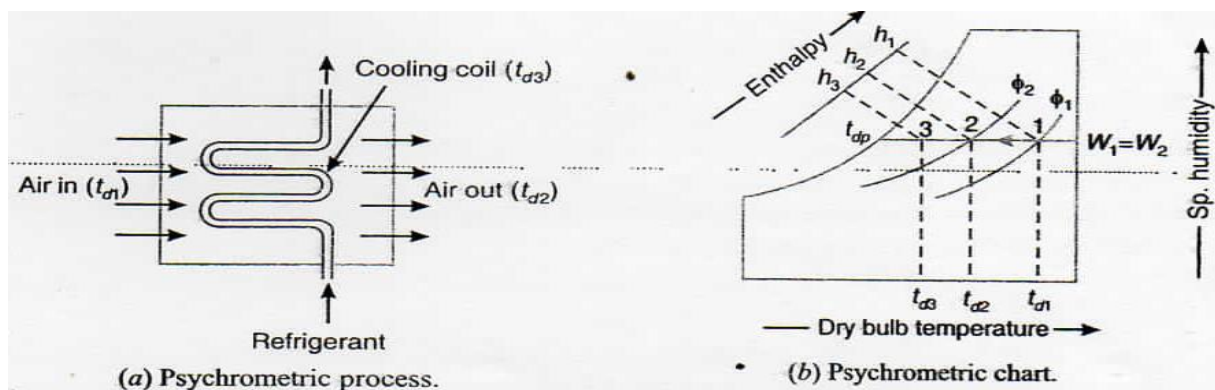
Fig. 16 Sensible cooling.

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, $\quad\quad\quad q = h_1 - h_2$

$$= C_{pa} (t_{d1} - t_{d2}) + W\ C_{ps}(t_{d1} - t_{d2})$$

$$= (C_{pa} + W\ C_{ps}) (t_{d1} - t_{d2}) = C_{pm} (t_{d1} - t_{d2})$$

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.

$\therefore$ Heat rejected, $\quad\quad q = 1.022\ (t_{d1} - t_{d2})$ kJ/kg

For air conditioning purposes, the sensible heat per minute is given as

$$SH = m_a\ C_{pm}\ \Delta t = v\rho\ C_{pm}\ \Delta t\ \text{kJ/min} \quad\quad ... (\because m = v\rho)$$

where $\quad\quad\quad\quad v = $ Rate of dry air flowing in m^3/min,

$\quad\quad\quad\quad\quad \rho = $ Density of moist air at 20° C and 50% relative humidity

$\quad\quad\quad\quad\quad = 1.2$ kg / m3 of dry air,

$\quad\quad C_{pm} = $ Humid specific heat = 1.022 kJ /kg K, and

$\quad\quad \Delta t = t_{d1} - t_{d2} = $ Difference of dry bulb temperatures between

$\quad\quad\quad\quad\quad$ the entering and leaving conditions of air in ° C.

Substituting the values of p and cp,,, in the above expression, we get

$$SH = v \times 1.2 \times 1.022 \times \Delta t = 1.2264\ v \times \Delta t\ \text{kJ/min}$$

$$= \frac{1.2264\ v \times \Delta t}{60} = 0.020\ 44\ v \times \Delta t\ \text{kJ/s or kW}$$
$$... (\because\ 1\ \text{kJ/s} = 1\ \text{kW})$$

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.
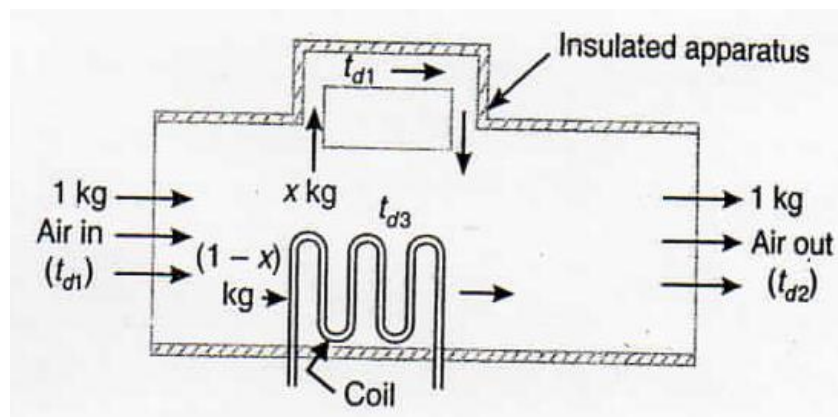
Fig.17. By-pass factor

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
$$x\,(t_{d3} - t_{d1}) = t_{d3} - t_{d2}$$

∴
$$x = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

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.

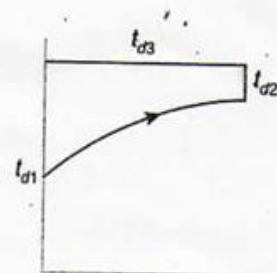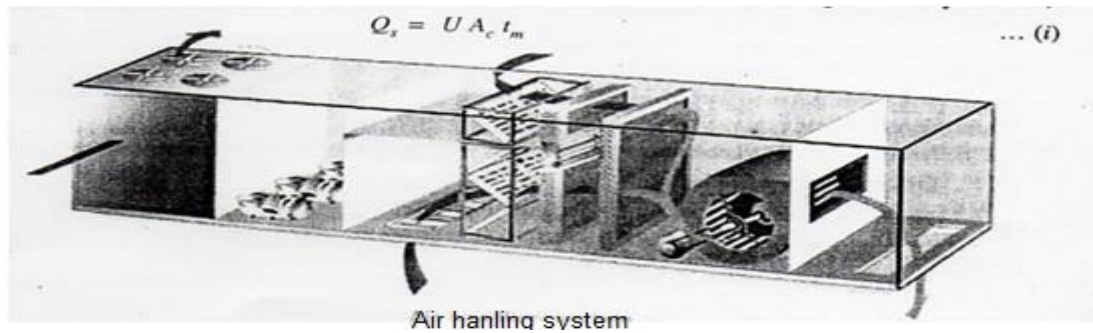$$Q_s = U A_c t_m \qquad \ldots (i)$$

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,

$$t_m = \frac{t_{d2} - t_{d1}}{\log_e\left[\dfrac{t_{d3} - t_{d1}}{t_{d3} - t_{d2}}\right]}, \text{ and } BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

∴
$$t_m = \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)}$$

Now the equation (i) may be written as

$$Q_s = U \times A_c \times \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)} \qquad \ldots (ii)$$

We have already discussed that the heat added during sensible heating,

$$Q_s = m_a c_{pm} (t_{d2} - t_{d1}) \qquad \ldots (iii)$$

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
$$\log_e (BPF) = -\frac{UA_c}{m_a c_{pm}}$$

∴
$$BPF = e^{-\left(\frac{UA_c}{m_a c_{pm}}\right)} = e^{-\left(\frac{UA_c}{1.022\, m_a}\right)} \qquad \ldots (iv)$$

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.
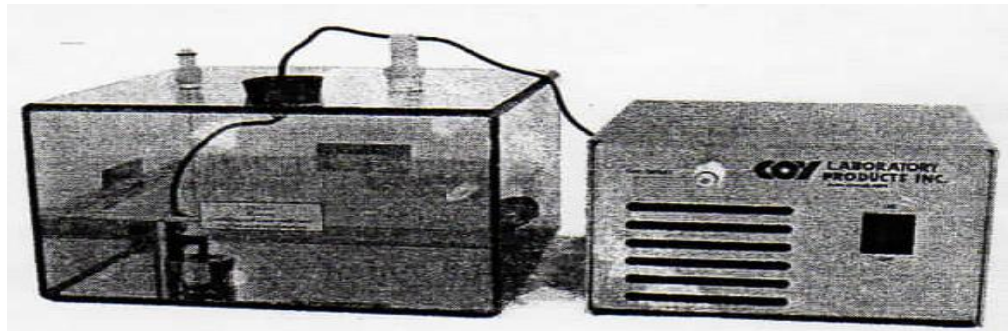
Ultrasonic humidification system

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.

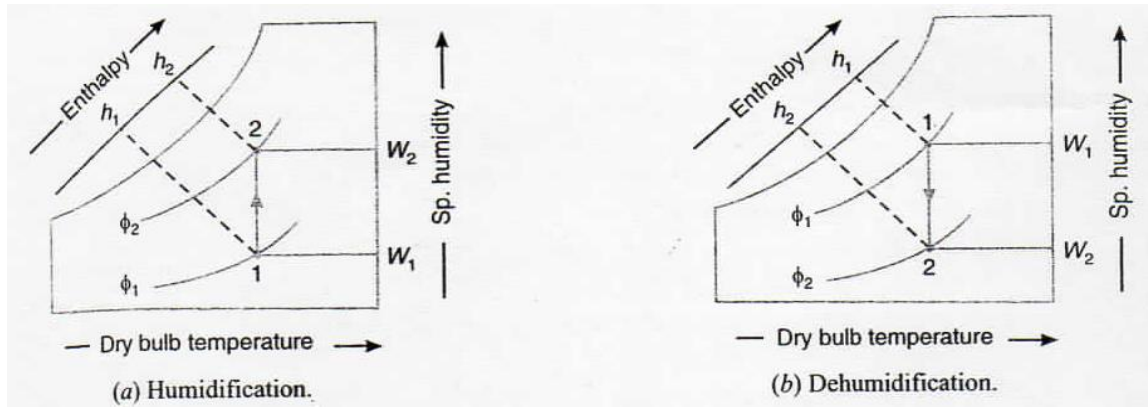Fig. 19 Humidification and dehumidification

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,
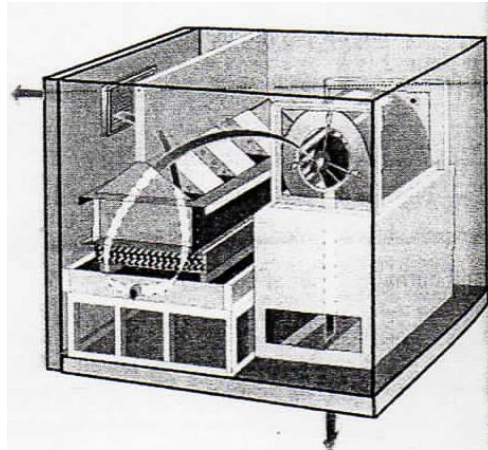
Multiple small plate dehumidification system

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:

$$LH = (h_1 - h_2) = h_{fg} (W_1 - W_2)$$

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

$$LH = m_a \, \Delta h = m_a \, h_{fg} \, \Delta W = v \, \rho \, h_{fg} \, \Delta W \qquad \qquad \dots (\because m_a = v \, \rho)$$

where   v = Rate of dry air flowing in m³/min,

ρ = Density of moist air = 1.2 kg/m³ 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

$$\text{LH} = v \times 1.2 \times 2500 \times \Delta W = 3000\, v \times \Delta W \text{ kJ/min}$$

$$= \frac{3000\, v \times \Delta W}{60} = 50\ v \times \Delta W \text{ kJ/s or kW}$$

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:
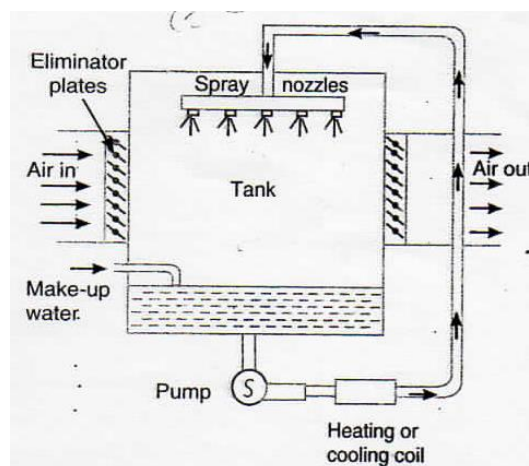

Fig. 20. Air-washer.

(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.

Fig. 21

Fig. 21 Cooling and dehumidification.
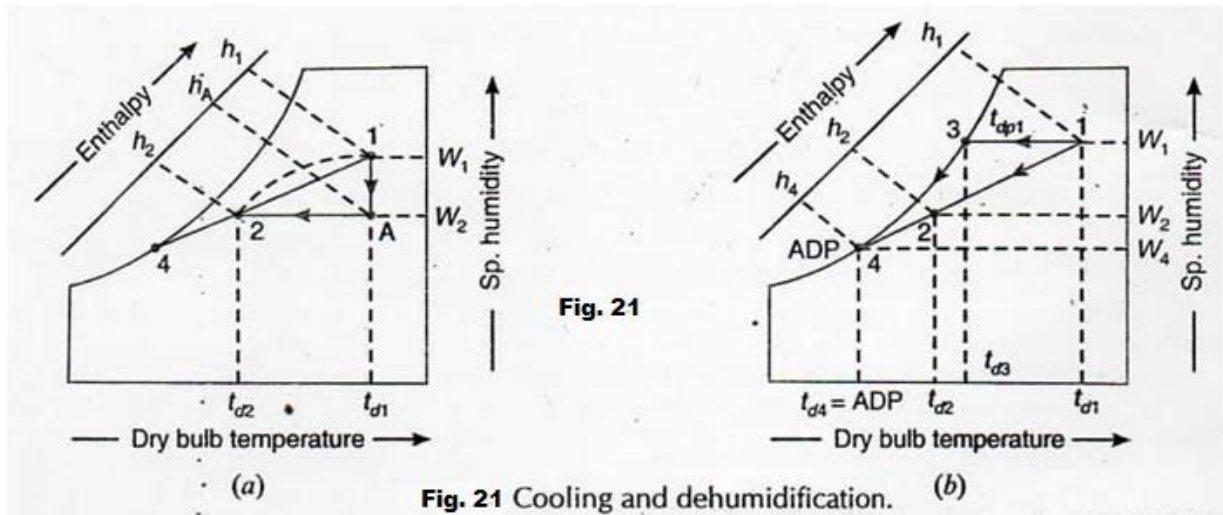
t_{d1} = Dry bulb temperature of air entering the coil,

t_{dp1} = 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 $\qquad$ LH = $h_1 - h_A$ = Latent heat removed due to condensation of vapour of the reduced moisture content ($W_1 - W_2$), and

$$SH = h_A - h_2 = \text{Sensible heat removed.}$$

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,

$$BPF = \frac{t_{d2} - t_{d4}}{t_{d1} - t_{d4}} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

$$0.15 = \frac{t_{d2} - 5}{35 - 5} = \frac{t_{d2} - 5}{30}$$

$$t_{d2} = 0.15 \times 30 + 5 = 9.5°C \text{ Ans.}$$
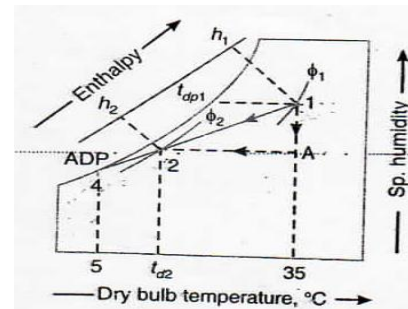
Fig.22

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature (t_{d2},) of 9.5°C on 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,

$$h_1 = 81 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 28 \text{ kJ/kg of dry air}$$

We know that cooling capacity of the coil

$$= m_a(h_1 - h_2) = 100 (81 - 28) = 5300 \text{ kJ/min}$$

$$= 5300/210 = 25.24 \text{ TR Ans.} \quad ....(\because 1 \text{ TR} = 210 \text{ kJ/min})$$

Example 2. 39.6 m³/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: $v_1 = 39.6 \text{ m}^3/\text{min}$; $t_{dt} = 31°C$; $t_{wt} = 18.5°C$; 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

$$h_1 = 52.5 \text{ kJ / kg of dry air}$$

Enthalpy at point 4,

$$h_4 = 17.7 \text{kJ/kg of dry air}$$

Specific humidity at point 1

$$W_1 = 0.0082 \text{ kg / kg of dry air}$$
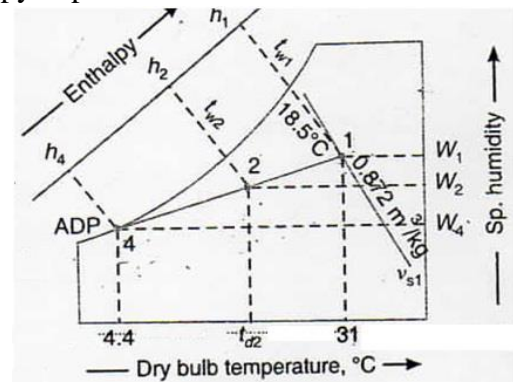
Specific humidity at point 4,

$$W_4 = 0.00525 \text{ kg / kg of dry air}$$

Specific volume at point

$$v_{s1}, = 0.872 \text{m}^3/ \text{ kg}$$

We know that mass flow rate of dry air at point 1,

Dry bulb temperature, °C →

$$m_a = \frac{v_1}{v_{s1}} = \frac{39.6}{0.872} = 44.41 \text{ kg/min}$$

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

$$t_{d2} = 18.5°C; \text{ and } t_{w2} = 12.5°C \text{ Ans.}$$

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.

Active desiccant system.
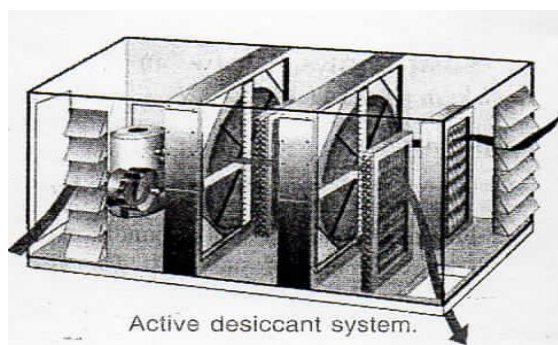
The air enters at condition 1 and leaves at condition 2. In this process, the dry bulb temperature as well as specific humidity of air increases. The final relative humidity of the air can be lower or higher than that of the entering air.
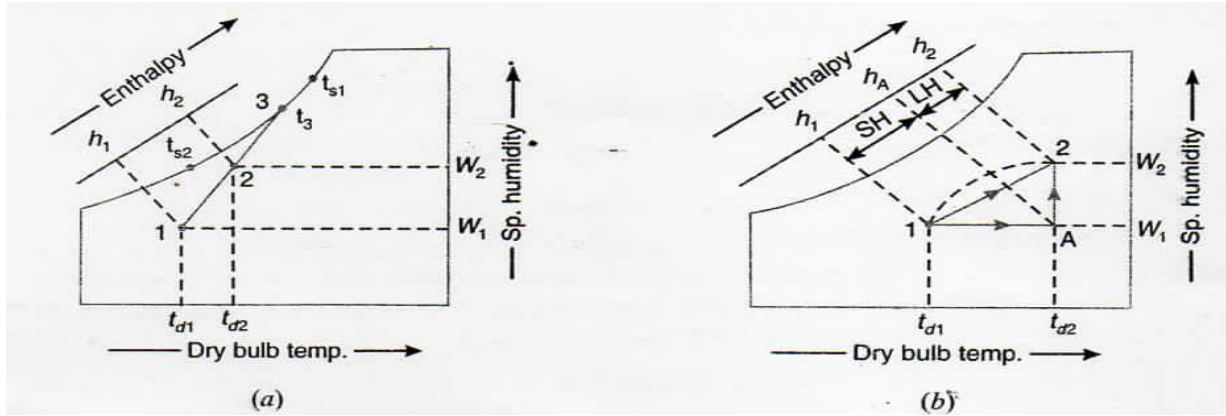
Fig.24 heating and humidification

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,

$$(m_{w1} - m_{w2}) = m_a (W_2 - W_1)$$

$$m_{w2} = m_{w1} - m_a (W_2 - W_1) \qquad \text{.... } (i)$$

or and for enthalpy balance,

$$m_{w1} \, hf_{w1} = m_{w2} \, h_{fw2} = m_a(h_2 - h_1) \qquad \text{....}(ii)$$

Substituting the value of mw2 from equation (i), we have

$$m_{w1} \, h_{fw1} - [\, m_{w1} - m_a (W_2 - W_1)] \, h_{fw2}$$

$$= m_a (h_2 - h_1)$$

$\therefore \qquad h_2 - h_1 = \dfrac{m_{w1}}{m_a} (h_{fw1} - h_{fw2}) + (W_2 - W_1) \, h_{fw2}$

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

$$q_s = (h_A - h_i) = \text{Sensible heat added}$$

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{q_S}{q} = \frac{q_S}{q_S + q_L} = \frac{h_A - h_l}{h_2 - h_l}$$

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.
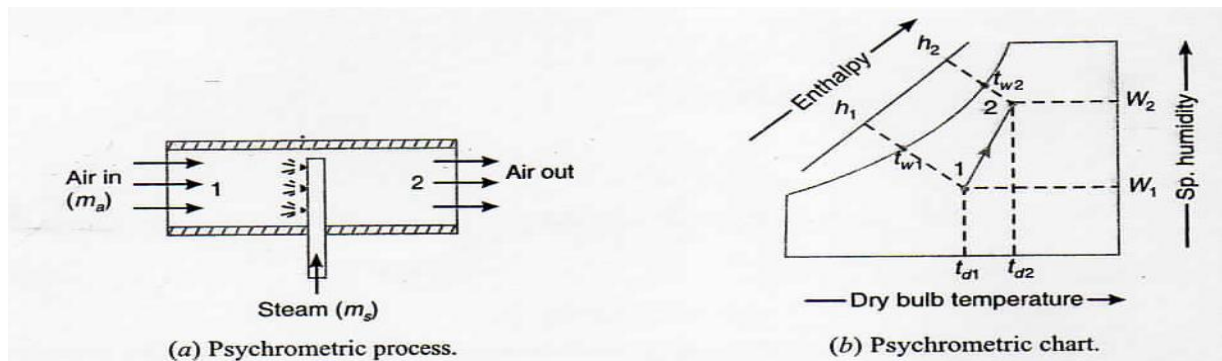
Fig.25 heating and humidification by steam injection

$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

$$h_s = \text{Enthalpy of steam injected into the air.}$$

Now for the mass balance,

$$W_2 = W_1 + \frac{m_s}{m_a} \qquad \text{....(i)}$$

and for the heat balance,

$$h_2 = h_1 + \frac{m_s}{m_a} \times h_s = h_1 + (W_2 - W_1) h_s \quad \text{... [From equation (i)]}$$

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; $\emptyset_1 = 25\%$; $t_{d2} = 30°$ 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,
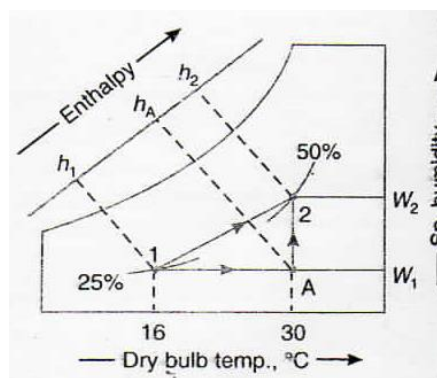
$$h_1 = 23 \text{ k1/kg of dry air}$$

Fig.26

Enthalpy of air at point A,

$$h_A = 38 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 64 \text{ la/kg of dry air}$$

∴ Heat added to the air

$$= h_2 - h_1 = 64 - 23 = 41 \text{ kJ/kg of dry air Ans.}$$

Moisture added to the air

From the psychrometric chart, we find that the specific humidity in the air at point 1,

$$W_1 = 0.0026 \text{ kg/kg of dry air}$$

and specific humidity in the air at point 2,

$$W_2 = 0.0132 \text{ kg /kg of dry air}$$

∴ Moisture added to the air

$$= W_2 - W_1 = 0.0132 - 0.0026 = 0.0106 \text{ kg/kg of dry air Ans.}$$

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 : $t_{dt} = 10°C$; $\emptyset_1 = 90\%$; $t_{d2} = 35°C$; $t_{w2} = 22.5°C$

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

$$= W_B - W_A = W_2 - W_1$$

$$= 0.0122 - 0.0068 = 0.0054 \text{ kg/kg of dry air Ans.}$$

4. Humidifying efficiency of the air washer

From the psychrometric chart, we find that

$$t_{dB} = 19.1°C \text{ and } t_{dB} = 18°C$$

We know that humidifying efficiency of the air washer,

$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{t_{dA} - t_{dB}}{t_{dA} - t_{dB'}}$$

$$= \frac{32.6 - 19.1}{32.6 - 18} = \frac{13.5}{14.6} = 0.924 \text{ or } 92.4\% \text{ Ans.}$$

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.
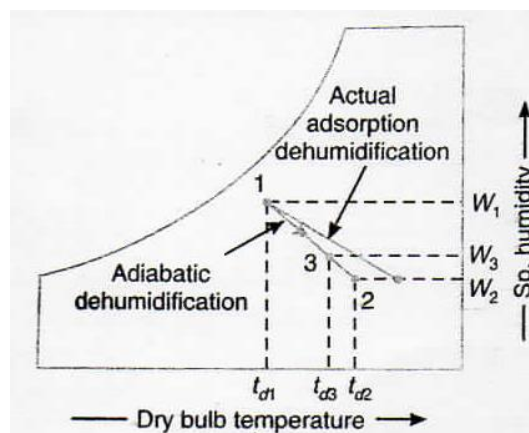

Fig.28

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: $t_{d1}, = t_{d3} = 21°C$; $\emptyset_2 = 20\%$

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.
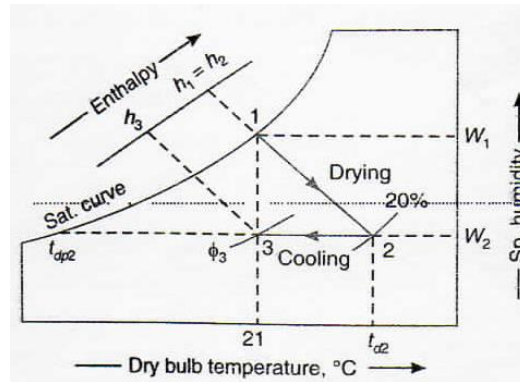
Fig.29

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.

$$h_2 = 61 \text{ kJ /kg of dry air}$$

and enthalpy of air at point 3,

$$h_3 = 43 \text{ kJ/kg of dry air}$$

∴ Heat rejected during the cooling process

$$= h_2 - h_3 = 61 — 43 = 18 \text{ kJ/kg of dry air A ns.}$$

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),

$$\emptyset_3 = 55\% \text{ Ans.}$$

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,

$$t_{dp2} = 11.6° \text{ C Ans.}$$

5. Moisture removed during the drying process

From the psychrometric chart, we find that moisture in air before the drying process at point 1,

$$W_1 = 0.0157 \text{ kg/kg of dry air}$$

and moisture in air after the drying process at point 2,

$$W_2 = 0.0084 \text{ kg/kg of dry air}$$