Psychrometry

1. What is psychometry.

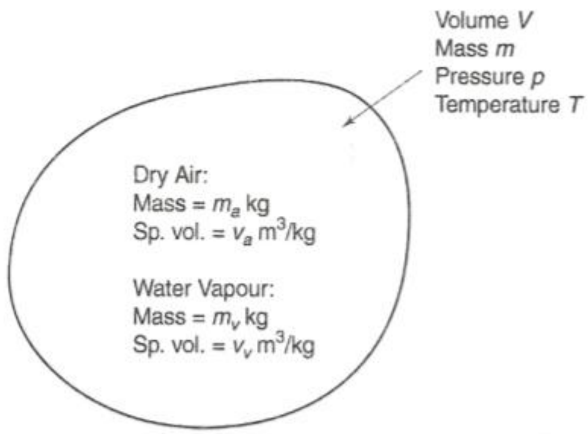
Atmospheric air makes up the environment in almost every type of air conditioning system. Hence a thorough understanding of the properties of atmospheric air and the ability to analyze various processes involving air is fundamental to air conditioning design.

Psychrometry is the study of the properties of mixtures of air and water vapour.

Atmospheric air is a mixture of many gases plus water vapour and a number of pollutants (Fig.27.1). The amount of water vapour and pollutants vary from place to place. The concentration of water vapour and pollutants decrease with altitude, and above an altitude of about 10 km, atmospheric air consists of only dry air. The pollutants have to be filtered out before processing the air. Hence, what we process is essentially a mixture of various gases that constitute air and water vapour. This mixture is known as *moist air*. The moist air can be thought of as a mixture of dry air and moisture.

2. Explain the Properties of Phychrometry

The properties of moist air are called psychrometric properties and the subject which deals with the behaviour of moist air is known as psychrometry. Moist air is a mixture of dry air and water vapour. They form a binary mixture. A mixture of two substances requires three properties to completely define its thermodynamic state, unlike a pure substance which requires only two. One of the three properties can be the composition. Water vapour is present in the atmosphere at a very low partial pressure. At this low pressure and atmospheric temperature, the water vapour behaves as a perfect gas. The partial pressure of dry air is also below one atmosphere which may also be considered to behave very much as a perfect gas. The Gibbs-Dalton laws of perfect gas mixture can be applied to the moist air. Since the water vapour part is continuously variable, all calculations in airconditioning practice are based on the dry air part. For calculating and defining the psychrometric properties, we may consider a certain volume V of moist air at pressure p and temperature T , containing ma kg of dry air and mv kg of water vapour as shown in Figure 6.3. The actual temperature t of moist air is called the dry bulb temperature (DBT). The total pressure p which is equal to the barometric pressure is constant.



Specific Humidity or Humidity Ratio Specific or absolute humidity or humidity ratio or moisture content is defined as the ratio of the mass of water vapour to the mass of dry air in a given volume of the mixture. It is denoted by the symbol ω.

$$\omega = \frac{m_v}{m_a} = \frac{V/v_v}{V/v_a} = \frac{v_a}{v_v} \qquad \ldots 6.10$$

where the subscripts *a* and *v* refer to dry air and water vapour respectively.

Now

$$p_a v_a = \frac{R}{M_a} T \qquad P_a V = m_a \frac{R}{M_a} T$$

$$p_v v_v = \frac{R}{M_v} T \qquad P_v V = m_v \frac{R}{M_v} T$$

Substituting for m_v and m_a from these expressions in Eq. (6.10) ,we obtain

$$\omega = \frac{M_v p_v}{M_a p_a} = \frac{18.016 p_v}{28.966 p_a} = 0.622 \frac{p_v}{p_a} \qquad \ldots 6.11$$

The units of ω are kg of water vapour per kg of dry air. Also, since *p* denotes the actual total atmospheric pressure, then from Dalton's law

$$p = p_a + p_v \qquad \ldots 6.12$$

So that

$$\omega = 0.622 \frac{p_v}{p - p_v} \qquad \ldots 6.13$$

Considering that the total atmospheric pressure remains constant at a particular locality, we can see that

$$\omega = f(p_v)$$

viz., the specific humidity is a function of the partial pressure of water vapour only.

<u>Dew Point Temperature</u> The normal thermodynamic state 1 as shown in the Figure 6.4 (a) of moist air is considered as unsaturated air. The water vapour existing at temperature T of the mixture and partial pressure pv of the vapour in the mixture is normally in a superheated state. If a sample of such unsaturated moist air containing superheated water vapour is cooled (at constant pressure),the mixture will eventually reach the saturation temperature td of water vapour corresponding to its partial pressure pv, at which point the first drop of dew will be formed, i.e., the water vapour in the mixture will start condensing. This temperature td is called the dew point temperature (DPT).

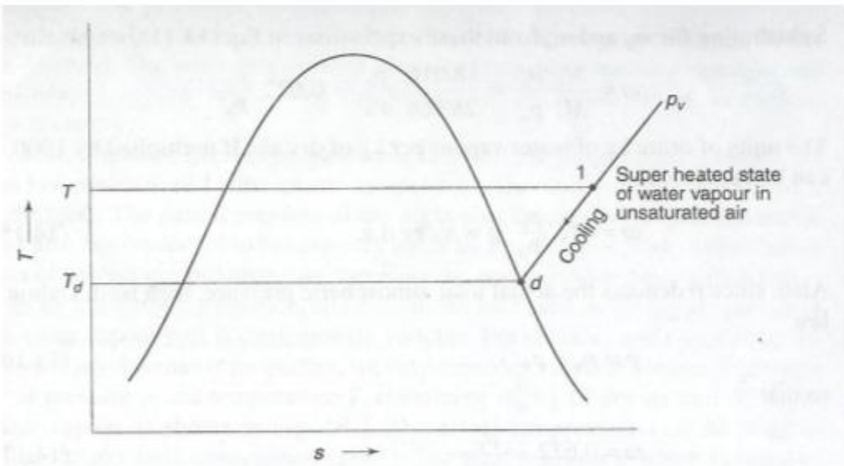
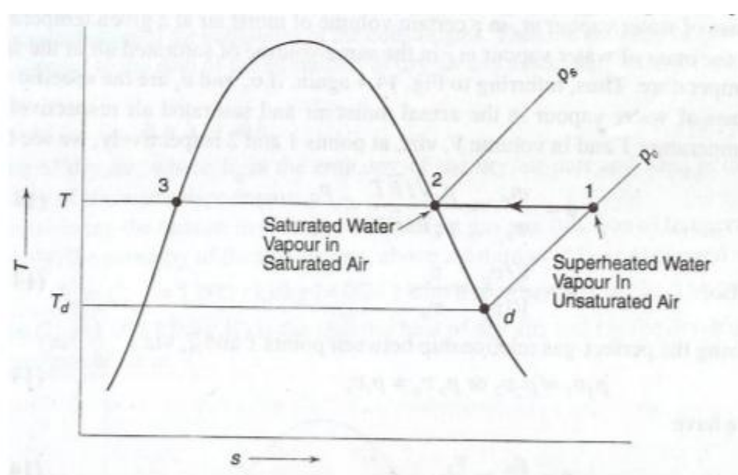


Figure 6.4(a) : Thermodynamic State of Water Vapour in Moist Air

Moisture can be removed from humid air by bringing the air in contact with a Properties of Moist Air cold surface or cooling coil whose temperature is below its dew point temperature. During the process of cooling, the partial pressure pv of water vapour and specific humidity ω remain constant until the vapour starts condensing.

<u>Degree of Saturation</u> Consider the water vapour in the super heated thermodynamic state 1 in unsaturated moist air representing the control volume V. the water vapour exists at the dry bulb temperature T of the mixture and partial pressure pv as shown in the Figure 6.4 (b).



: 6.4 (b) : An Imaginary Isothermal Process Showing the Change of State of Water Vapour

Now consider that more water vapour is added in this control volume V at temperature T itself. The partial pressure pv will go on increasing with the addition of water vapour until it reaches a value ps corresponding to state 2 in Figure 6.4 after which it cannot increase further as ps is the saturation pressure or maximum possible of water at temperature T. the thermodynamic state of water vapour is now saturated at point 2. the air containing moisture in such a state is called saturated air. In this state the air is holding the maximum amount of water vapour(the specific humidity being ωs, corresponding to the partial pressure ps) at temperature T of the mixture. The maximum possible specific humidity, ωs at temperature T is thus

$$\omega_s = 0.622 \frac{p_s}{p - p_s} \qquad \ldots 6.14$$

The ratio of the actual specific humidity ω to the specific humidity ω_s of saturated air at temperature T is termed as the *degree of saturation* denoted by the symbol μ. Thus

$$\mu = \frac{\omega}{\omega_s} = \frac{p_v}{p_s}\left[\frac{1 - p_s/p}{1 - p_v/p}\right] \qquad \ldots 6.15$$

Thus the degree of saturation is a measure of the capacity of air to absorb moisture.

Relative Humidity The relative humidity ϕ is defined as the ratio of the mole fraction of water vapour in moist air to mole fraction of water vapour in saturated air at the same temperature and pressure. From perfect-gas relationships another expression for ϕ is

$$\phi = \frac{\text{existing parial pressure of water vapor}}{\text{saturation pressure of pure water at same temperature}}$$

$$\phi = \frac{m_v}{m_{v_s}} = \frac{p_v v/RT}{p_s v/RT} = \frac{p_v}{p_s} \qquad \ldots 6.16$$

Also, $$\phi = \frac{V/v_v}{V/v_s} = \frac{v_s}{v_v} \qquad \ldots 6.17$$

When p_v is equal to p_s, ϕ is equal to unity, and the air is saturated and is considered to have 100 per cent RH.

From Eqs (6.14) and (6.17), we can write

$$\omega = 0.622\phi \frac{p_s}{p_a} \qquad \ldots 6.18$$

$$\phi = \frac{\omega}{0.622} \frac{p_a}{p_s} \qquad \ldots 6.19$$

Also we may write

$$\mu = \phi\left[\frac{1 - p_s/p}{1 - p_v/p}\right] \qquad \ldots 6.20$$

$$\phi = \frac{\mu}{1 - (1 - \mu)\, p_s/p} \qquad \ldots 6.21$$

3. Explain enthalpy of Moist air

According to Gibb's law, the enthalpy of a mixture of perfect gases can be obtained by the net summation of the enthalpies of the respective constituents. Therefore the enthalpy of the moist air h is equal to the summation of the enthalpies of dry air and of the water vapour associated with the air. Hence,

$$h = h_a + w h_v \qquad \ldots 6.22$$

Per kg of dry air, where h_a is the enthalpy of the dry air part and wh_v is the enthalpy of the water vapour part. The change in enthalpy of a perfect gas being considered as a function of temperature only, the enthalpy of the dry air part above a datum of 0^0C is expressed as:

$$h_a = C_{pa}\, t = 1.005\, t \text{ kJ/kg } (= 0.24\, t \text{ Btu/lbm where t is in } {}^0\text{F})$$

where C_p= 1.005 kJ/kg.K is the specific heat of dry air, and t is the dry-bulb temperature of air in ^{0}C.

Assuming the reference state enthalpy as zero for saturated liquid at 0^0C, the enthalpy of water vapour at point A in the above Figure can be expressed as:

$$h_v = h_A = C_{pw}\, t_d + (h_{fg})_d + C_{pv}\,(t - t_d) \text{ kJ/kg} \qquad \ldots 6.24$$

where C_{pw}= specific heat of liquid water

t_d = dew point temperature

$(h_{fg})_d$ = latent heat of vaporization at DPT

C_{pv}=specific heat of superheated vapour

Taking the specific heat of liquid water as 4.1868 kJ/kg K and that of water vapour as 1.88kJ/kg K, in the range 0 to 60^0C, we have

$$h_v = 4.1868\, t_d + (h_{fg})_d + 1.88\,(t - t_d)$$

At low pressure for an ideal gas, the enthalpy is a function of temperature only. Thus in Figure 6.5 the enthalpies at point B and C are also the same as the enthalpy at A. Accordingly, enthalpy of water vapour at A, at DPT of t_d and DBT of t, can be determined more conveniently by the following two methods:

(a) $h_A = h_C = (h_g)_t$ ...6.26

(b) $h_A = h_B = (h_{fg})_{0^oC} + C_{pv}(t - 0)$ ...6.27

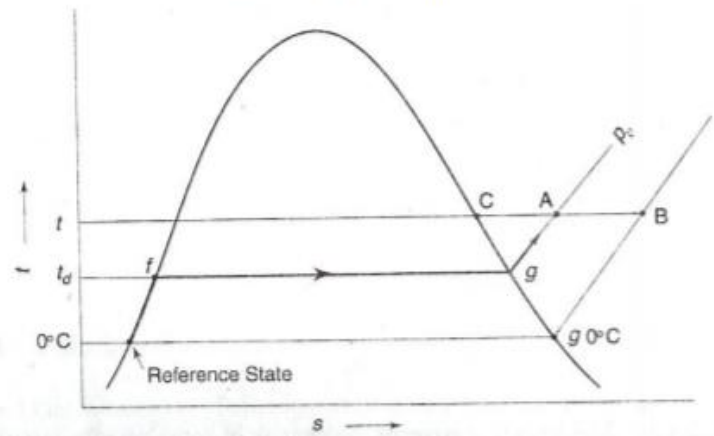


Figure 6.5: Evaluation of Enthalpy of Water Vapour Part

Using second expression and taking the latent heat of vaporization of water at 0^0C as 2501 k/kgK, we obtain the empirical expression for the enthalpy of the water vapour part

$$h_v = 2501 + 1.88t \text{ kJ/kg} \quad \text{...6.28}$$

And combining Eq 6.23 and 6.24, we have the enthalpy of moist air

$$h = 1.005t + \omega(2500 + 1.88t) \text{ kJ/kg d.a.} \quad \text{...6.29}$$

4. Explain enthalpy of moist air

The enthalpy of moist air can also be written as

$$h = (C_{pa} + \omega C_{pv})\, t + \omega (h_{fg})_{0^\circ C}$$

$$= C_p t + \omega (h_{fg})_{0^\circ C} \qquad ...6.30$$

where $C_p = C_{pa} + \omega C_{pv}$ ...6.31

$$= (1.005 + 1.88\,\omega)\ \text{kJ/(kg d.a.) (K)}$$

is termed as the *humid specific heat*. It is the specific heat of moist air $(1 + \omega)$ kg per kg of dry air.

5. Explain wet bulb temperature (WBT)

A psychrometer comprises of a dry bulb thermometer and a wet bulb thermometer. The dry bulb thermometer is directly exposed to the air and measures the actual temperature of air and is called dry bulb temperature. When the thermometer bulb is surrounded by a wet cloth exposed to the air. The temperature which is measured by the wick-covered bulb of such a thermometer indicates the temperature of liquid water in the wick and is called the wet bulb temperature. It is denoted by the symbol t' . The difference between the dry bulb and wet bulb temperatures is called wet bulb depression (WBD).

WBD = (t – t')

If the ambient air is saturated, i.e. the RH is 100 per cent, then there will be no evaporation of water on the bulb and hence WBT and DBT will be equal. The WBT is an indirect measure of the dryness of air.

6. Explain adiabatic saturation and thermodynamic wet bulb 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. 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 Figure 6.6.

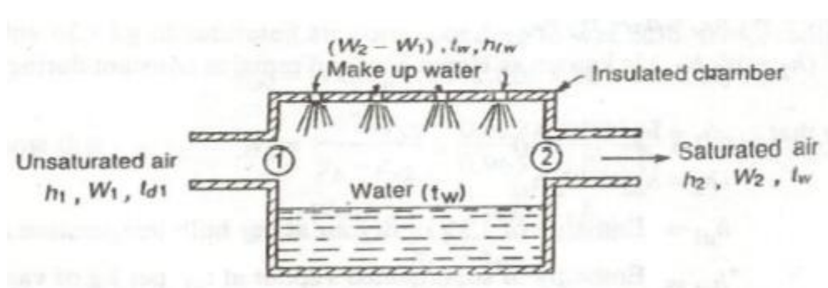


Figure 6.6: Adiabatic Saturation of Air

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 vaporise 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 Figure 6.7. During the adiabatic saturation process, the partial pressure of vapour increases, although the total pressure of the air-vapour mixture remains constant. The unsaturated air initially at dry bulb temperature td1 is cooled adiabatically to dry bulb temperature td2 which is equal to the adiabatic saturation temperature tw. It may be noted that the adiabatic saturation temperature is taken equal to the wet bulb temperature for all practical purposes.

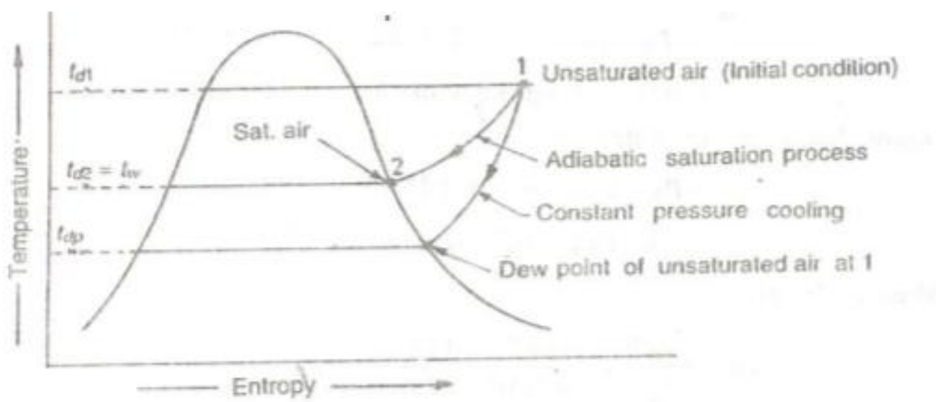


Figure 6.7: T-S Diagram for Adiabatic Saturation Process

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),

$$h_1 + (W_2 - W_1)\, h_{fw} = h_2 \qquad \ldots 6.32$$

$$h_1 - W_1\, h_{fw} = h_2 - W_2\, h_{fw} \qquad \ldots 6.33$$

The term $(h_2 - W_2\, h_{fw})$ is known as sigma heat and remains constant during the adiabatic process

We know that $h_1 = h_{a1} + W_1 h_{s1}$

$$h_2 = h_{a2} + W_2 h_{s2}$$

where

h_{a1} = Enthalpy of 1 kg of dry air at dry bulb temperature t_{d1},

h_{s1} = Enthalpy of superheated vapour at t 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 (6.33) may be written as:

$$(h_{a1} + W_1 h_{s1}) - W_1\, h_{fw} = (h_{a2} + W_2 h_{s2}) - W_2 h_{s2} \qquad \ldots 6.34$$

$$W_1(h_{s1} - h_{fw}) = W_2(h_{s2} - h_{fw}) + h_{a2} - h_{a1} \qquad \ldots 6.35$$

$$W_1 = \frac{W_2\left(h_{s2} - h_{fw}\right) + h_{a2} - h_{a1}}{h_{s1} - h_{fw}} \qquad \ldots 6.36$$

7. Explain Basic gas laws for moist air:

According to the *Gibbs-Dalton law* for a mixture of perfect gases, the total pressure exerted by the mixture is equal to the sum of partial pressures of the constituent gases. According to this law, for a homogeneous perfect gas mixture occupying a volume V and at temperature T, each constituent gas behaves as though the other gases are not present (i.e., there is no interaction between the gases). Each gas obeys perfect gas equation. Hence, the partial pressures exerted by each gas, $p_1, p_2, p_3 \ldots$ and the total pressure p_t are given by:

$$p_1 = \frac{n_1 R_u T}{V};\ p_2 = \frac{n_2 R_u T}{V};\ p_3 = \frac{n_3 R_u T}{V}\ \ldots\ldots$$
$$p_t = p_1 + p_2 + p_3 + \ldots\ldots$$

where $n_1, n_2, n_3, \ldots$ are the number of moles of gases 1,2,3,…

Applying this equation to moist air.

$$p = p_t = p_a + p_v$$

where $p = p_t$ = total barometric pressure

p_a = partial pressure of dry air

p_v = partial pressure of water vapour

Dry bulb temperature (DBT) is the temperature of the moist air as measured by a standard thermometer or other temperature measuring instruments.

Saturated vapour pressure (p_{sat}) is the saturated partial pressure of water vapour at the dry bulb temperature.

Relative humidity (Φ) is defined as the ratio of the mole fraction of water vapour in moist air to mole fraction of water vapour in saturated air at the same temperature and pressure.

$$\phi = \frac{\text{partial pressure of water vapour}}{\text{saturation pressure of pure water vapour at same temperature}} = \frac{p_v}{p_{sat}}$$

Relative humidity is normally expressed as a percentage. When Φ is 100 percent, the air is saturated.

Humidity ratio (W): The humidity ratio (or specific humidity) W is the mass of water associated with each kilogram of dry air1. Assuming both water vapour and dry air to be perfect gases2, the humidity ratio is given by:

$$W = \frac{\text{kg of water vapour}}{\text{kg of dry air}} = \frac{p_v V / R_v T}{p_a V / R_a T} = \frac{p_v / R_v}{(p_t - p_v)/R_a}$$

Substituting the values of gas constants of water vapour and air R_v and R_a in the above equation; the humidity ratio is given by:

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

8. Explain DPT

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

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

$$\mu = \left|\frac{W}{W_s}\right|_{t,P}$$

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

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

where c_p = specific heat of dry air at constant pressure, kJ/kg.K
c_{pw} = specific heat of water vapor, kJ/kg.K
t = Dry-bulb temperature of air-vapor mixture, ^oC
W = Humidity ratio, kg of water vapor/kg of dry air
h_a = enthalpy of dry air at temperature t, kJ/kg
h_g = enthalpy of water vapor3 at temperature t, kJ/kg
h_{fg} = latent heat of vaporization at 0^oC, kJ/kg
The unit of h is kJ/kg of dry air.
Humid specific heat: From the equation for enthalpy of moist air, the humid specific heat of moist air can be written as:

$$c_{pm} = c_p + W.c_{pw}$$

where c_{pm} = humid specific heat, kJ/kg.K
c_p = specific heat of dry air, kJ/kg.K
c_{pw} = specific heat of water vapor, kJ/kg
W = humidity ratio, kg of water vapor/kg of dry air
Since the second term in the above equation ($w.c_{pw}$) is very small compared to the first term, for all practical purposes, the humid specific heat of moist air, c_{pm} can be taken as 1.0216 kJ/kg dry air.K
Specific volume: The specific volume is defined as the number of cubic meters of moist air per kilogram of dry air.

$$v = \frac{R_a T}{p_a} = \frac{R_a T}{p_t - p_v} \qquad m^3/\text{kg dry air}$$

9. Explain various process in Psychrometric chart

A *Psychrometric chart* graphically represents the thermodynamic properties of moist air. Standard psychrometric charts are bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or humidity ratio (ordinate). The Left Hand Side of the psychrometric chart is bounded by the saturation line. Figure shows the schematic of a psychrometric chart. Psychrometric charts are readily available for standard barometric pressure of 101.325 kPa at sea level and for normal temperatures (0-50^{o}C). ASHRAE has also developed psychrometric charts for other temperatures and barometric pressures (for low temperatures: -40 to 10^{o}C, high temperatures 10 to 120^{o}C and very high temperatures 100 to 120^{o}C)

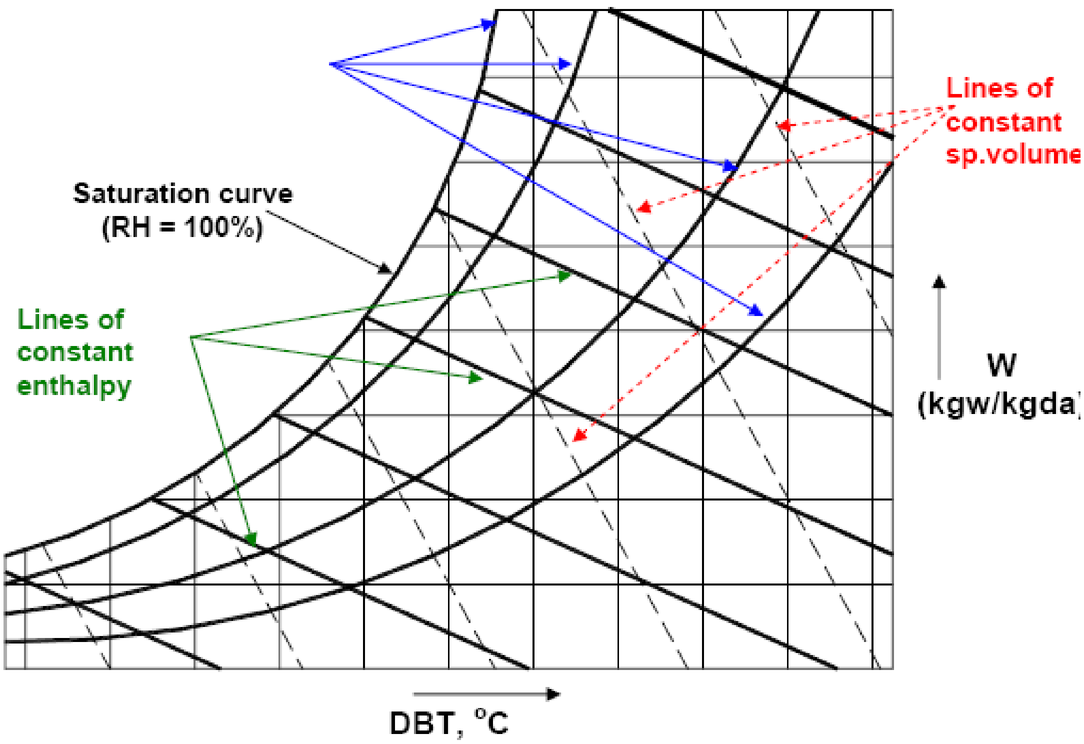


Fig.27.2: *Schematic of a psychrometric chart for a given barometric pressure*

10. Explain adiabatic saturation and thermodynamic wet bulb temperature:

Adiabatic saturation temperature is defined as that temperature at which water, by evaporating into air, can bring the air to saturation at the same temperature adiabatically. An adiabatic saturator is a device using which one can measure theoretically the adiabatic saturation temperature of air.

As shown in Fig., an adiabatic saturator is a device in which air flows through an infinitely long duct containing water. As the air comes in contact with water in the duct, there will be heat and mass transfer between water and air. If the duct is infinitely long, then at the exit, there would exist perfect equilibrium between air and water at steady state. Air at the exit would be fully saturated and its temperature is equal to that of water temperature. The device is adiabatic as the walls of the chamber are thermally insulated. In order to continue the process, make-up water has

to be provided to compensate for the amount of water evaporated into the air. The temperature of the make-up water is controlled so that it is the same as that in the duct.
After the adiabatic saturator has achieved a steady-state condition, the temperature indicated by the thermometer immersed in the water is the *thermodynamic wet-bulb temperature*. The thermodynamic wet bulb temperature will be less than the entering air DBT but greater than the dew point temperature.
Certain combinations of air conditions will result in a given sump temperature, and this can be defined by writing the energy balance equation for the adiabatic saturator. Based on a unit mass flow rate of dry air, this is given by:

$$h_1 = h_2 - (W_2 - W_1)h_f$$

where h_f is the enthalpy of saturated liquid at the sump or thermodynamic wet-bulb temperature, h_1 and h_2 are the enthalpies of air at the inlet and exit of the adiabatic saturator, and W_1 and W_2 are the humidity ratio of air at the inlet and exit of the adiabatic saturator, respectively.

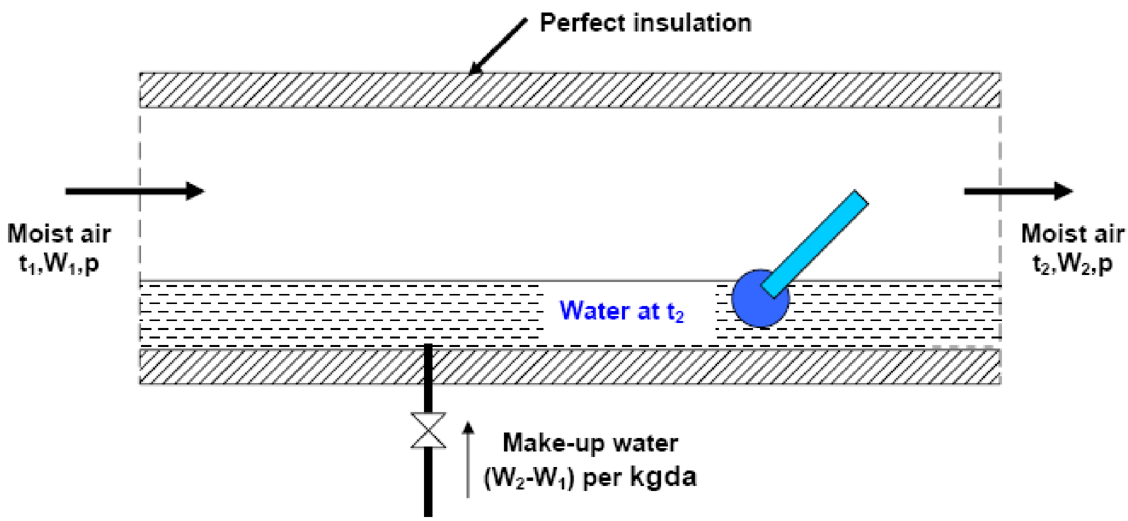
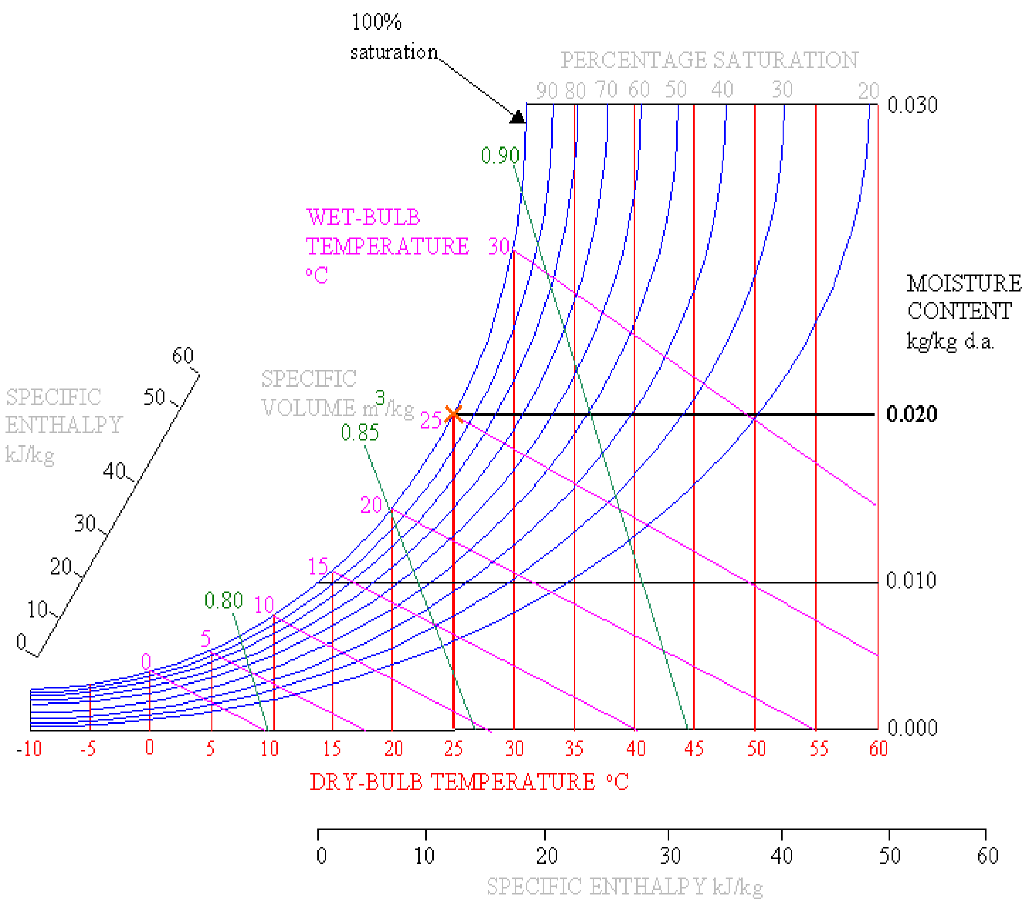


***Fig.27.4**: The process of adiabatic saturation of air*

Examples of Psychrometric Properties

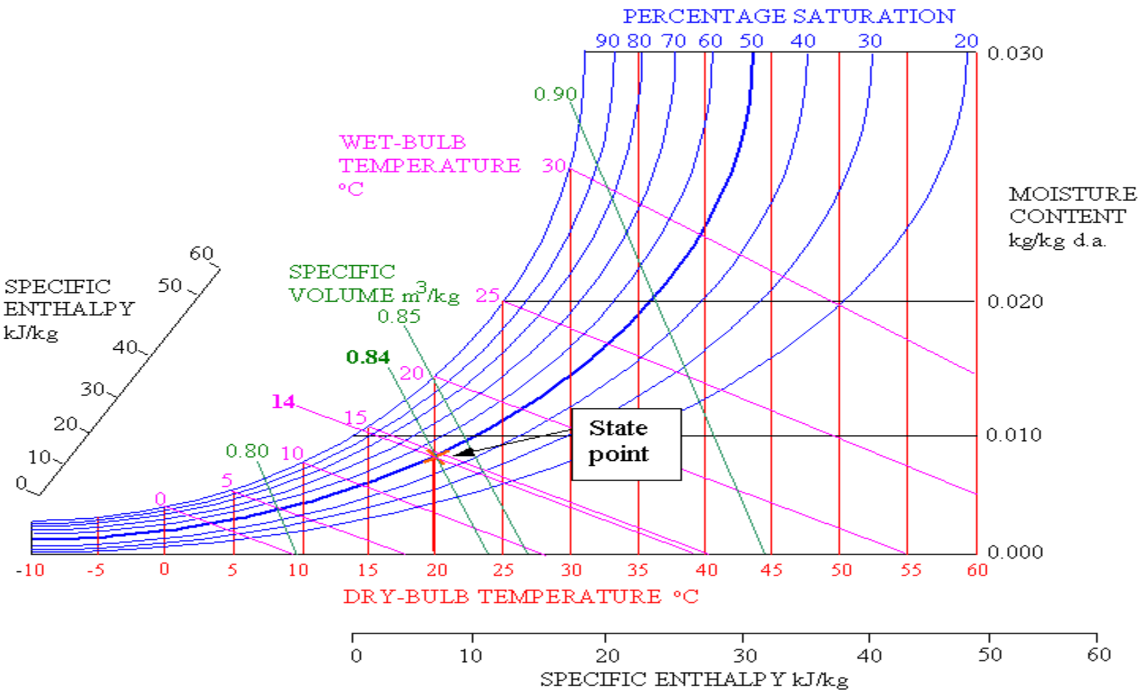
EXAMPLE 1

Find the moisture content of air at 25°C dry-bulb temperature and 25°C wet-bulb temperature.
Referring to the chart below, a vertical line is drawn upwards from 25°C dry-bulb temperature until it intersects at 25°C wet-bulb temperature.
This intersection point happens to be on the 100% saturation line.
The intersection point is highlighted and a horizontal line is drawn to the right to find the corresponding moisture content.
The moisture content is therefore 0.020 kg/kg dry air.



EXAMPLE 2

Find the specific volume and wet-bulb temperature of air at 20°C dry-bulb temperature and 50% saturation.
Referring to the chart below, a vertical line is drawn upwards from 20°C dry-bulb temperature until it intersects with the 50% saturation curve.
The intersection point is sometimes referred to as the state point.
The specific volume is found to be 0.84 m^3/kg and the wet-bulb temperature is 14°C



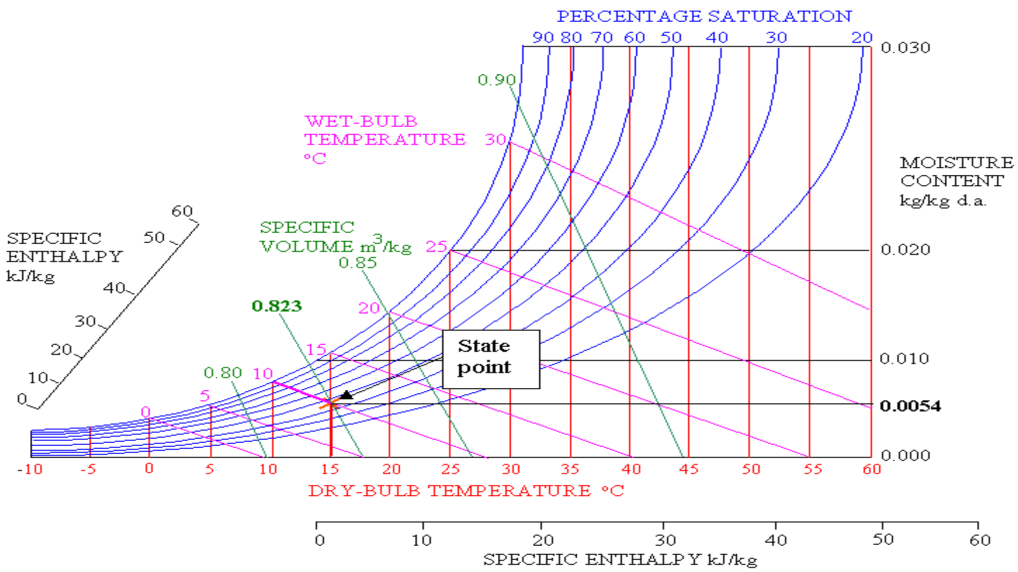
EXAMPLE 3

Find the specific volume, percentage saturation and moisture content of air at 15°C dry-bulb temperature and 10°C wet-bulb temperature.

Referring to the chart below, a vertical line is drawn upwards from 15°C dry-bulb temperature until it intersects with the 10°C wet-bulb temperature line.

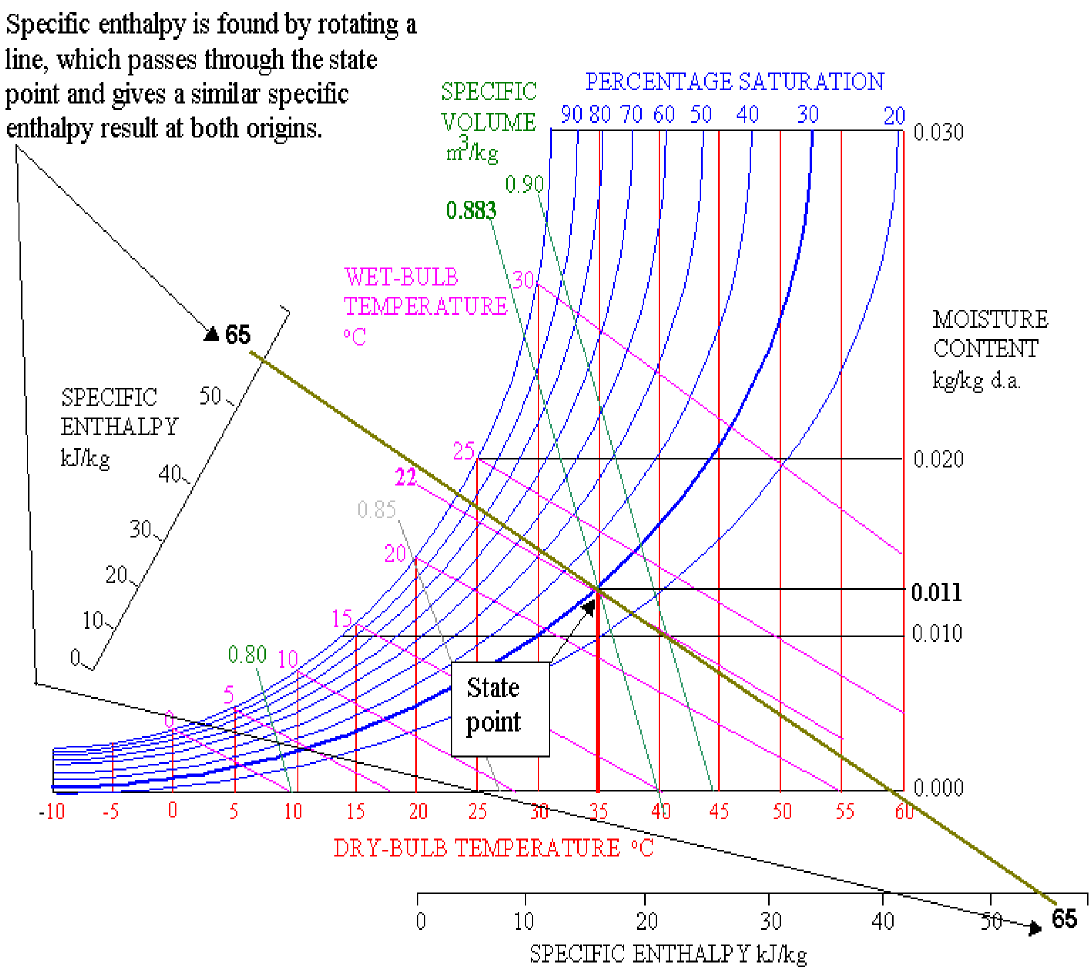
This intersection is the state point.

The specific volume is found to be 0.823 m^3/kg, the percentage saturation 52% and the moisture content 0.0054 kg/kg d.a.



EXAMPLE 4

Find the specific volume, wet-bulb temperature, moisture content and specific enthalpy of air at 35^{o}C dry-bulb temperature and 30% saturation.
Referring to the chart below, a vertical line is drawn upwards from 35^{o}C dry-bulb temperature until it intersects with the 30% saturation curve.
This intersection is the state point.
The specific volume is found to be 0.883 m^3/kg, the wet-bulb temperature is 22^{o}C, the moisture content 0.011kg/kg d.a. and the specific enthalpy 65 kJ/kg.



11. Explain Various Psychrometric Processes

In the design and analysis of air conditioning plants, the fundamental requirement is to identify the various processes being performed on air. Once identified, the processes can be analyzed by applying the laws of conservation of mass and energy. All these processes can be plotted easily on a psychrometric chart. This is very useful for quick visualization and also for identifying the changes taking place in important properties such as temperature, humidity ratio, enthalpy etc. The important processes that air undergoes in a typical air conditioning plant are discussed below.

a) **Sensible cooling:**

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant, the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. Figure shows the sensible cooling process O-A on a psychrometric chart. The heat transfer rate during this process is given by:

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

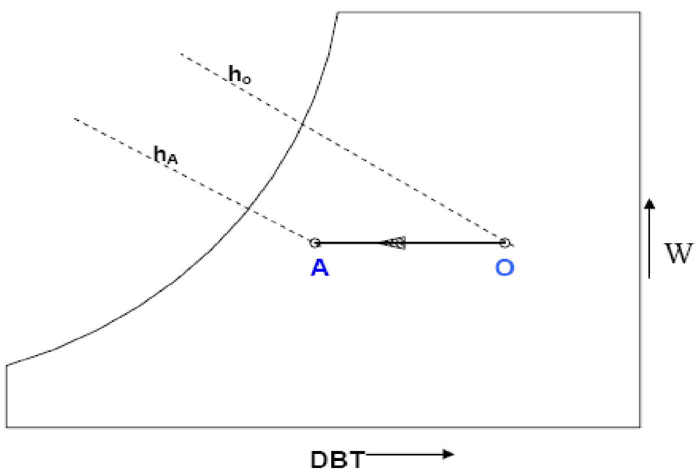


Fig. : Sensible cooling process O-A on psychrometric chart

b) Sensible heating (Process O-B):

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The heat transfer rate during this process is given by:

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

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

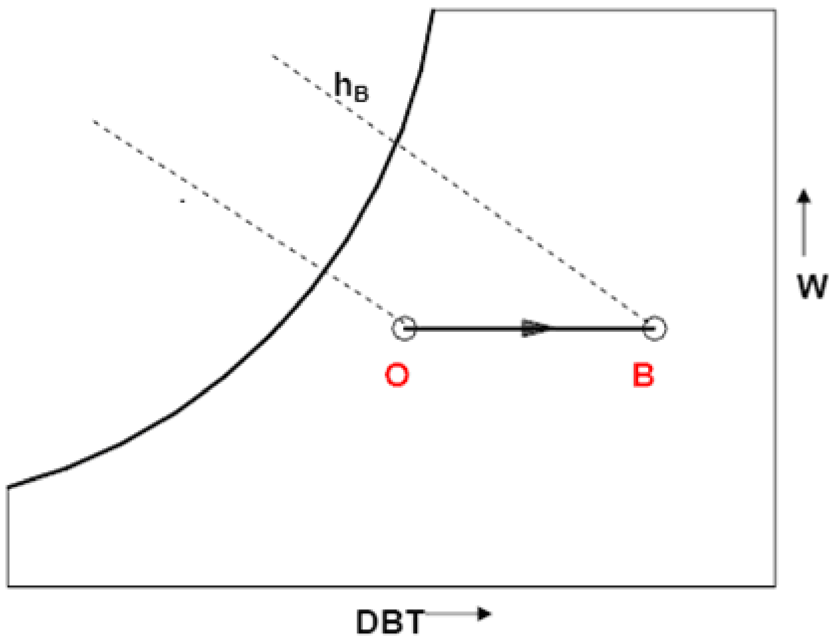


Fig : Sensible heating process on psychrometric chart

Cooling and dehumidification (Process O-C):

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in Fig., some of the water vapor in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. This is the process air undergoes in a typical air conditioning system. Although the actual process path will vary depending upon the type of cold surface, the surface temperature, and flow conditions, for simplicity the process line is assumed to be a straight line. The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations as given below:

By applying mass balance for the water:

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

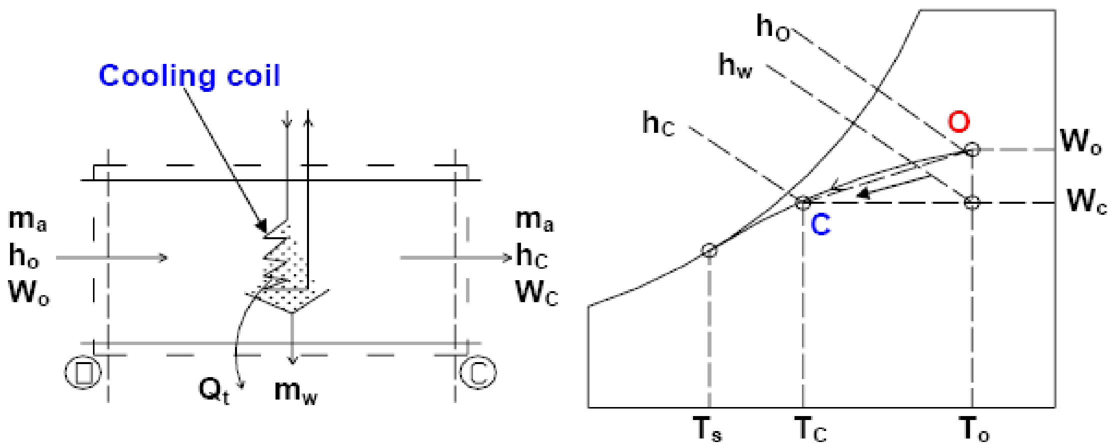


Fig. : Cooling and dehumidification process (O-C)

By applying energy balance:

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

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

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

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

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

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

$$Q_t = Q_l + Q_s$$

where $Q_l = m_a(h_O - h_w) = m_a.h_{fg}(w_O - w_C)$

$$Q_s = m_a(h_w - h_C) = m_a.c_{pm}(T_O - T_C)$$

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

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

<u>Heating and Humidification (Process O-D):</u>

During winter it is essential to heat and humidify the room air for comfort. As shown in Fig.28.5., this is normally done by first sensibly heating the air and then adding water vapour to the air stream through steam nozzles as shown in the figure.

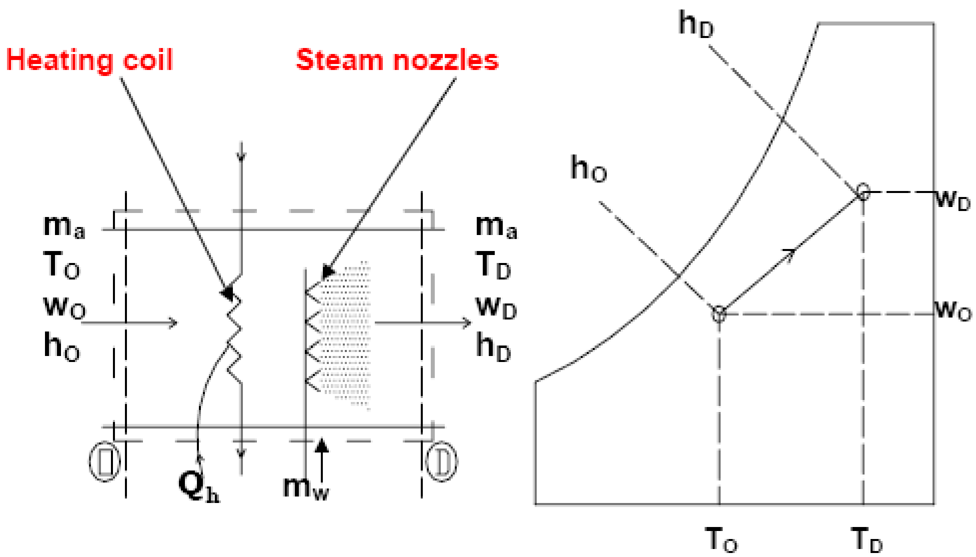


Fig. : *Heating and humidification process*

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

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

where m_a is the mass flow rate of dry air.

From energy balance:

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

where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam.

Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

<u>Cooling & humidification (Process O-E):</u>

As the name implies, during this process, the air temperature drops and its humidity increases. This process is shown in Fig. As shown in the figure, this can be achieved by spraying cool water in the air stream. The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew-point temperature to avoid condensation ($T_{DPT} < T_w < T_O$).

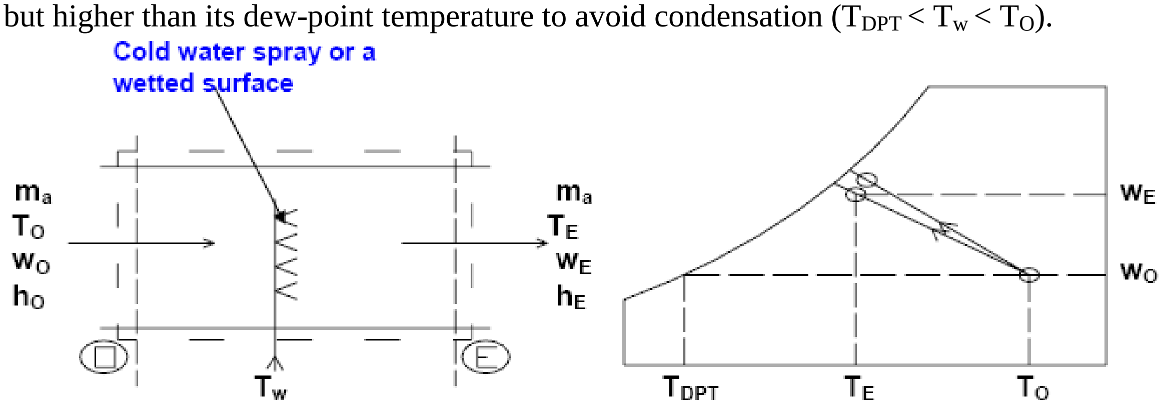


Fig. : *Cooling and humdification process*

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature. If the temperature of the water sprayed is equal to the wet-bulb temperature of air, then the net transfer rate will be zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water. Under a special case when the spray water is entirely recirculated and is neither heated nor cooled, the system is perfectly insulated and the make-up water is supplied at WBT, then at steady-state, the air undergoes an adiabatic saturation process, during which its WBT remains constant. This is the process of adiabatic saturation. The process of cooling and humidification is encountered in a wide variety of devices such as evaporative coolers, cooling towers etc.

Heating and de-humidification (Process O-F):

This process can be achieved by using a hygroscopic material, which absorbs or adsorbs the water vapor from the moisture. If this process is thermally isolated, then the enthalpy of air remains constant, as a result the temperature of air increases as its moisture content decreases as shown in Fig. This hygroscopic material can be a solid or a liquid. In general, the absorption of water by the hygroscopic material is an exothermic reaction, as a result heat is released during this process, which is transferred to air and the enthalpy of air increases.

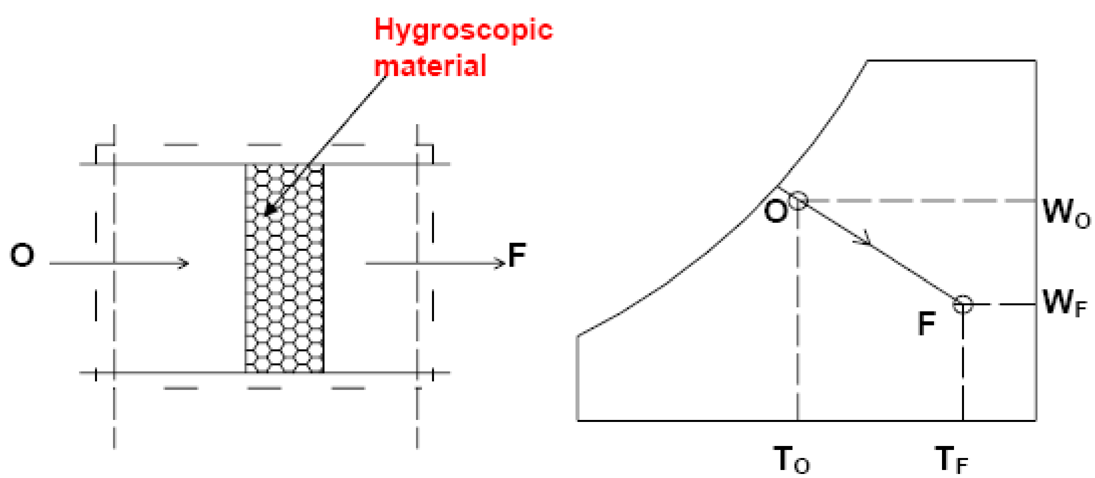


Fig. *Chemical de-humidification process*

Mixing of air streams:

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture.

i) Without condensation: Figure shows an adiabatic mixing of two moist air streams during which no condensation of moisture takes place. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance.

From the mass balance of dry air and water vapor:

$$m_{a,1}w_1 + m_{a,2}w_2 = m_{a,3}w_3 = (m_{a,1} + m_{a,2})w_3$$

From energy balance:

$$m_{a,1}h_1 + m_{a,2}h_2 = m_{a,3}h_3 = (m_{a,1} + m_{a,2})h_3$$

From the above equations, it can be observed that the final enthalpy and humidity ratio of mixture are weighted averages of inlet enthalpies and humidity ratios. A generally valid approximation is that the final temperature of the mixture is the weighted average of the inlet temperatures. With this approximation, the point on the psychrometric chart representing the mixture lies on a straight line connecting the two inlet states. Hence, the ratio of distances on the line, i.e., **(1-3)/(2-3)** is equal to the ratio of flow rates $\mathbf{m_{a,2}/m_{a,1}}$. The resulting error (due to the assumption that the humid specific heats being constant) is usually less than 1 percent.

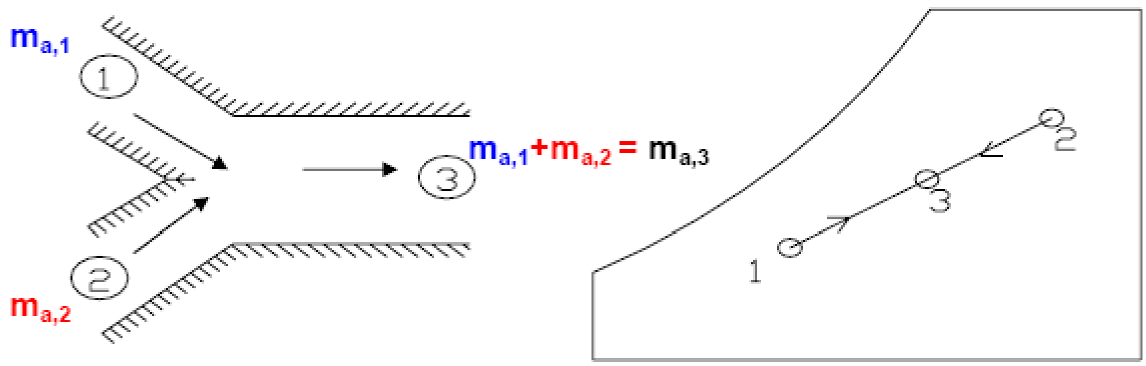


Fig. *. Mixing of two air streams without condensation*

12. Explain Air Washers:

An air washer is a device for conditioning air. As shown in Fig., in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes.

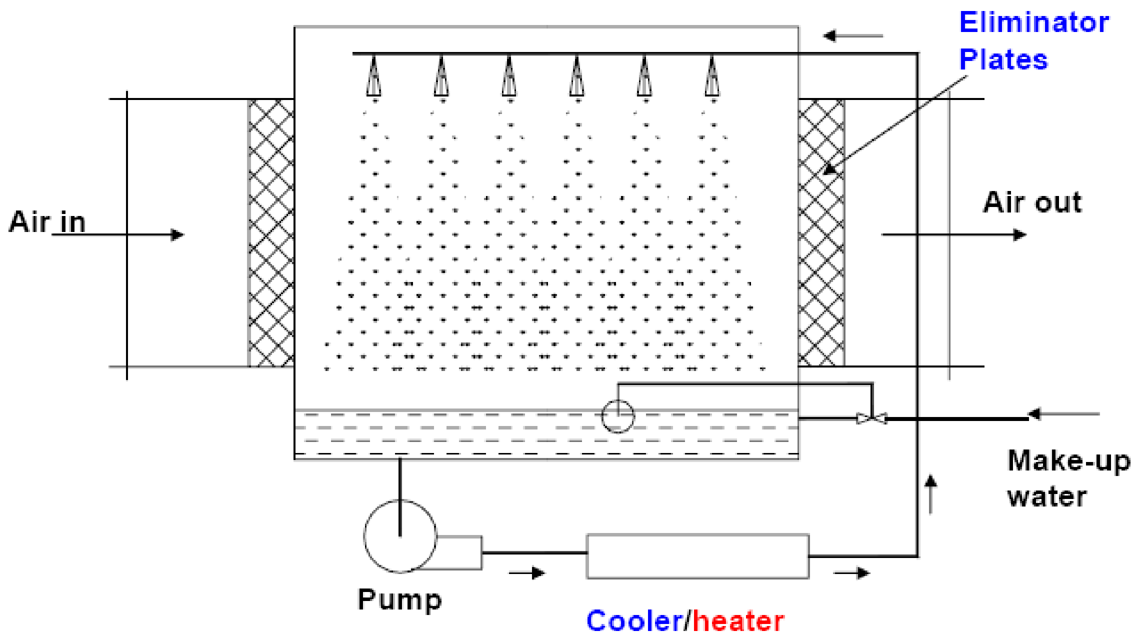


Fig. : Air washer

In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer. As a consequence of the 2nd law, the heat transfer between air and water droplets will be in the direction of decreasing temperature gradient. Similarly, the mass transfer will be in the direction of decreasing vapor pressure gradient. For example,

a) Cooling and dehumidification: tw < tDPT. Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled. Here both latent and sensible heat transfers are from air to water. This is shown by Process O-A in Fig..

b) Adiabatic saturation: tw = tWBT. Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or heating of water is required. That is this is a case of pure water recirculation. This is shown by Process O-B in Fig. This the process that takes place in a perfectly insulated evaporative cooler.

c) Cooling and humidification: tDPT < tw < tWBT. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from air to water, hence, water has to be cooled externally. This is shown by Process O-C in Fig.28.11.

d) Cooling and humidification: tWBT < tw < tDBT. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is shown by Process O-D in Fig.. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser.

e) Heating and humidification: tw > tDBT. Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally. This is shown by Process O-E in Fig..

Thus, it can be seen that an air washer works as a year-round air conditioning system. Though air washer is a and extremely useful simple device, it is not commonly used for comfort air conditioning applications due to concerns about health resulting from bacterial or fungal growth on the wetted surfaces. However, it can be used in industrial applications.

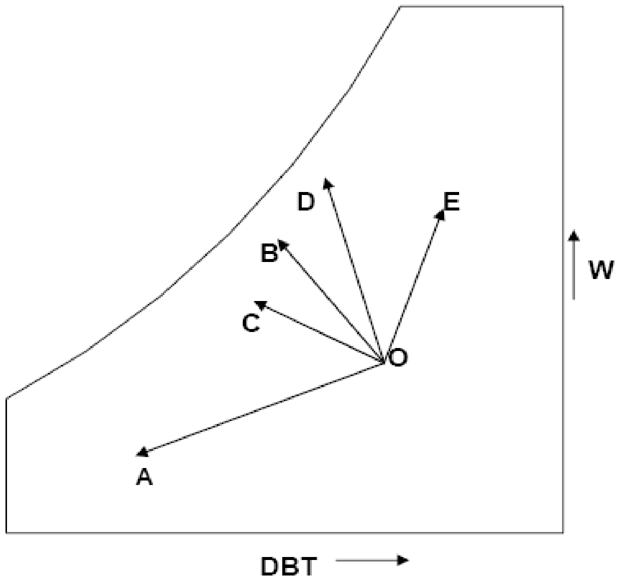


Fig. : Various psychrometric processes that can take place in an air washer

13. Describe Thermal comfort and factors affecting it:

Thermal comfort is defined as "that condition of mind which expresses satisfaction with the thermal environment". This condition is also some times called as "neutral condition", though in a strict sense, they are not necessarily same. A living human body may be likened to a heat engine in which the chemical energy contained in the food it consumes is continuously converted into work and heat. The process of conversion of chemical energy contained in food into heat and work is called as "metabolism". The rate at which the chemical energy is converted into heat and work is called as "metabolic rate". Knowledge of metabolic rate of the occupants is required as this forms a part of the cooling load of the air conditioned building. Similar to a heat engine, one can define thermal efficiency of a human being as the ratio of useful work output to the energy input. The thermal efficiency of a human being can vary from 0% to as high as 15-20% for a short duration. By the manner in which the work is defined, for most of the light activities the useful work output of human beings is zero, indicating a thermal efficiency of 0%. Irrespective of the work output, a human body continuously generates heat at a rate varying from about 100 W (e.g. for a sedentary person) to as high as 2000 W (e.g. a person doing strenuous exercise). Continuous heat generation is essential, as the temperature of the human body has to be maintained within a narrow range of temperature, irrespective of the external surroundings.

A human body is very sensitive to temperature. The body temperature must be maintained within a narrow range to avoid discomfort, and within a somewhat wider range, to avoid danger from heat or cold stress. Studies show that at neutral condition, the temperatures should be:

$$\text{Skin temperature, } t_{skin} \approx \mathbf{33.7^{o}C}$$
$$\text{Core temperature, } t_{core} \approx \mathbf{36.8^{o}C}$$

Heat balance equation for a human being:

The temperature of human body depends upon the energy balance between itself and the surrounding thermal environment. Taking the human body as the control volume, one can write the thermal energy (heat) balance equation for the human body as:

$$Q_{gen} = Q_{sk} + Q_{res} + Q_{st}$$

where Q_{gen} = Rate at which heat is generated inside the body

Q_{sk} = Total heat transfer rate from the skin

Q_{res} = Heat transfer rate due to respiration, and

Q_{st} = Rate at which heat is stored inside the body

The heat generation rate Q_{gen} is given by:

$$Q_{gen} = M(1-\eta) \approx M$$

where M = Metabolic rate, and

η = Thermal efficiency ≈ 0 for most of the activities

The total heat transfer rate from the skin Q_{sk} is given by:

$$Q_{sk} = \pm Q_{conv} \pm Q_{rad} + Q_{evp}$$

where Q_{conv} = Heat transfer rate due to convection (sensible heat)

Q_{rad} = Heat transfer rate due to radiation (sensible heat), and

Q_{evp} = Heat transfer rate due to evaporation (latent heat)

The convective and radiative heat transfers can be positive or negative, i.e., a body may lose or gain heat by convection and radiation, while the evaporation heat transfer is always positive, i.e., a body always looses heat by evaporation. Using the principles of heat and mass transfer, expressions have been derived for estimating the convective, radiative and evaporative heat transfer rates from a human body. As it can be expected, these heat transfer rates depend on several factors that influence each of the heat transfer mechanism.

Factors affecting thermal comfort:

Thermal comfort is affected by several factors. These are:

1. Physiological factors such as age, activity, sex and health. These factors influence the metabolic rate. It is observed that of these factors, the most important is activity. Other factors are found to have negligible effect on thermal comfort.

2. Insulating factor due to clothing. The type of clothing has strong influence on the rate of heat transfer from the human body. The unit for measuring the resistance offered by clothes is called as "**clo**". 1 clo is equal to a resistance of about 0.155 m^2.K/W. Typical clo values for different types of clothing have been estimated and are available in the form of tables. For example, a typical business suit has a clo value of 1.0, while a pair of shorts has a clo value of about 0.05.

3. Environmental factors. Important factors are the dry bulb temperature, relative humidity, air motion and surrounding surface temperature. Of these the dry bulb temperature affects heat transfer by convection and evaporation, the relative humidity affects heat loss by evaporation, air velocity influences both convective and evaporative heat transfer and the surrounding surface temperature affects the radiative heat transfer.

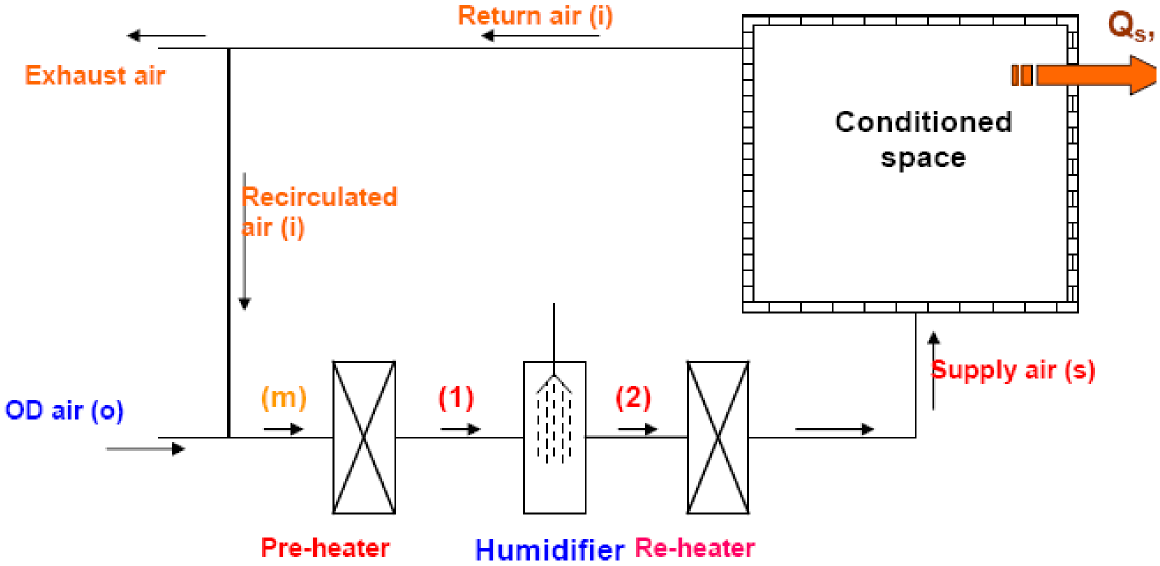
Apart from the above, other factors such as drafts, asymmetrical cooling or heating, cold or hot floors etc. also affect the thermal comfort. The objective of a comfort air conditioning system is to control the environmental factors so that comfort conditions prevail in the occupied space. It has no control on the physiological and insulating factors. However, wearing suitable clothing may help in reducing the cost of the air conditioning system.

14. Explain Winter Air Conditioning Systems

In winter the outside conditions are cold and dry. As a result, there will be a continuous transfer of sensible heat as well as moisture (latent heat) from the buildings to the outside. Hence, in order to maintain required comfort conditions in the occupied space an air conditioning system is required which can offset the sensible and latent heat losses from the building. Air supplied to the conditioned space is heated and humidified in the winter air conditioning system to the required level of temperature and moisture content depending upon the sensible and latent heat losses from the building. In winter the heat losses from the conditioned space are partially offset by solar and internal heat gains. Thus in a conservative design of winter A/C systems, the effects of solar radiation and internal heat gain are not considered.

Heating and humidification of air can be achieved by different schemes. Figure shows one such scheme along with the cycle on psychrometric chart. As shown in the figure, the mixed air (mixture of return and outdoor air) is first pre-heated (m-1) in the pre-heater, then humidified using a humidifier or an air washer (1-2) and then finally reheated in the re-heater (2-s). The reheated air at state ‘s’ is supplied to the conditioned space.

The flow rate of supply air should be such that when released into the conditioned space at state ‘s’, it should be able to maintain the conditioned space at state I and offset the sensible and latent heat losses (Q_s and Q_l). Pre-heating of air is advantageous as it ensures that water in the humidifier/air washer does not freeze. In addition, by controlling the heat supplied in the pre-heater one can control the moisture content in the conditioned space.



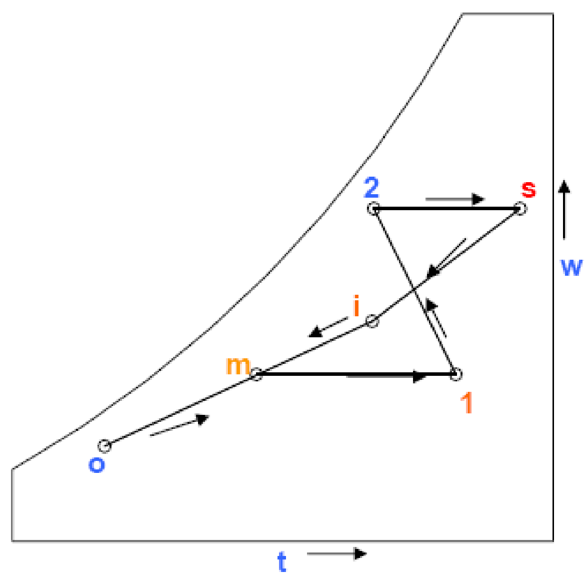


Fig. : A winter air conditioning system with a pre-heater

The humidification of air can be achieved in several ways, e.g. by bringing the air in contact with a wetted surface, or with droplets of water as in an air washer, by adding aerosol sized water droplets directly to air or by direct addition of dry saturated or superheated steam. Humidification by direct contact with a wetted surface or by using an air washer are not recommended for comfort applications or for other applications where people are present in the conditioned space due to potential health hazards by the presence of micro-organisms in water. The most common method of humidifying air for these applications is by direct addition of dry steam to air. When air is humidified by contact with wetted surface as in an air washer, then temperature of air decreases as its humidity increases due to simultaneous transfer of sensible and latent heat. If the air washer functions as an adiabatic saturator, then humidification proceeds along the constant wet bulb temperature line. However, when air is humidified by directly adding dry, saturated steam, then the humidification proceeds close to the constant dry bulb temperature line. The final state of air is always obtained by applying conservation of mass (water) and conservation of energy equations to the humidification process.

All year (complete) air conditioning systems:

Figure shows a complete air conditioning system that can be used for providing air conditioning throughout the year, i.e., during summer as well as winter. As shown in the figure, the system consists of a filter, a heating coil, a cooling & dehumidifying coil, a re-heating coil, a humidifier and a blower. In addition to these, actual systems consist of several other accessories such as dampers for controlling flow rates of re-circulated and outdoor (OD) air, control systems for controlling the space conditions, safety devices etc. Large air conditioning systems use blowers in the return air stream also. Generally, during summer the heating and humidifying coils remain inactive, while during winter the cooling and dehumidifying coil remains inactive. However, in some applications for precise control of conditions in the conditioned space all the coils may have to be made active. The blowers will remain active throughout the year, as air has to be circulated during summer as well as during winter. When the outdoor conditions are favourable, it is possible to maintain comfort conditions by using filtered outdoor air alone, in which case only the blowers will be running and all the coils will be inactive leading to significant savings in energy consumption. A control system is required which changes-over the system from winter operation to summer operation or vice versa depending upon the outdoor conditions.

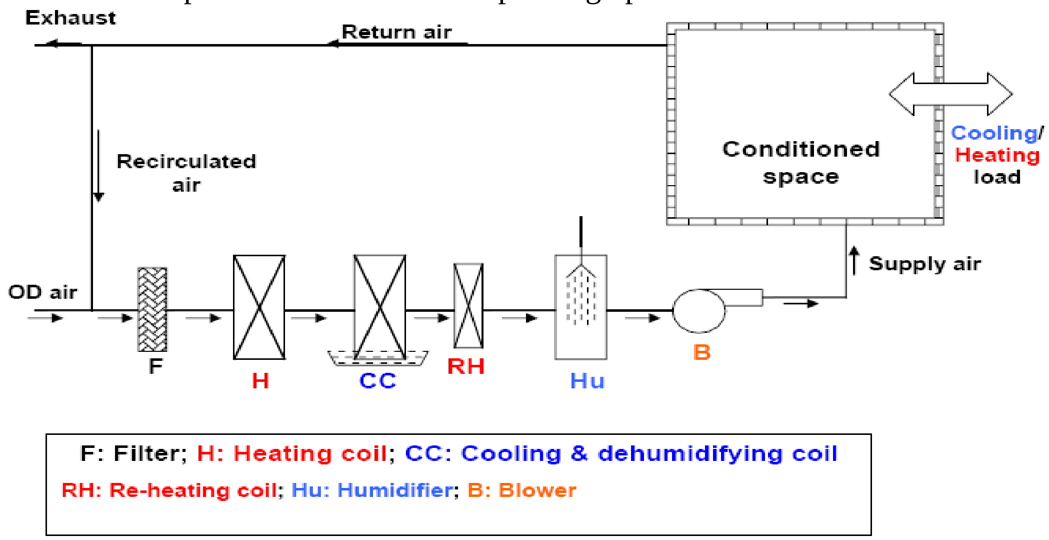


Fig. : An all year air conditioning system