UNIT-III

PSYCHROMETRY

3.1 INTRODUCTION

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

3.2 PSYCHOMETRIC TERMS

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

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

Table .1 Composition of dry air

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

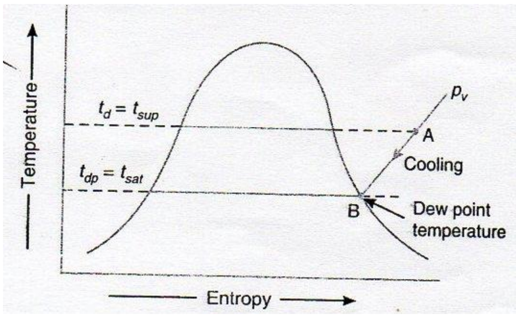


Fig.1. T-s diagram

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

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

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

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

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

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

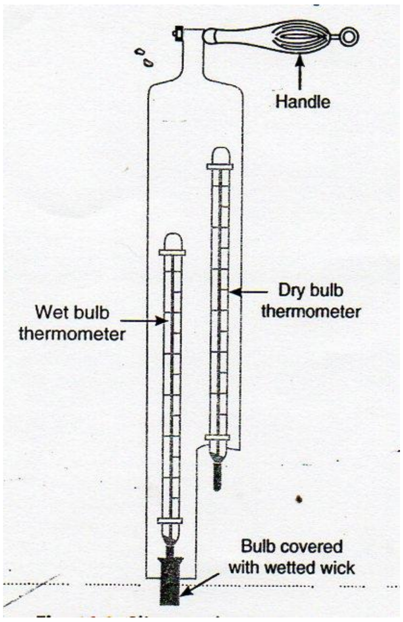


Fig.2, Sling psychrometer

3.3 DALTON'S LAW OF PARTIAL PRESSURES

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

$$P_b = P_a + P_v,$$

where P_a = Partial pressure of dry air, and

P_v= Partial pressure of water vapour.

3.4 PSYCHROMETRIC RELATIONS

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

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

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

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

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

$$P_a\, v_a = m_a\, R_a T_a$$

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

Also $\quad v_a = v_v$

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

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

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

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

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

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

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

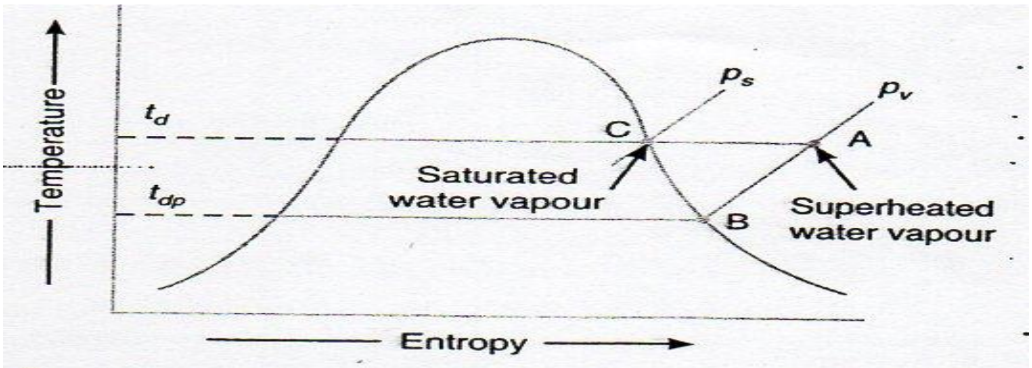


Fig.3 T-s diagram

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

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

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

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

2. ***Degree of saturation or percentage humidity***. We have already discussed that the degree of saturation is the ratio of vapour in a unit mass of water air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature (dry bulb temperature), it may be defined as the ratio of actual specific humidity to the specific humidity of saturated air at the same dry bulb temperature. It is, usually, denoted by 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,∅) = Pv / Ps is equal to zero, then the humidity ratio, W = 0, i.e. for dry air, $\mu = 0$.

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

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

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

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

water vapour in actual conditions, and

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

We know that for water vapour in actual conditions,

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

Similarly, for water vapour in saturated air,

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

According to the definitions,

$$v_v = v_s$$

$$Tv = Ts$$

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

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

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

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

The relative humidity may also be obtained as discussed below:

We know that degree of saturation,

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

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

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

P_b = Barometric pressure,

t_d = Dry bulb temperature, and

t_w = Wet bulb temperature.

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

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

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

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

temperature t_d, and

ρ_a = Density of dry air in kg/m^3 of dry air.

We know that mass of water vapour,

$$m_v = v_v \rho_v$$

and mass of dry air,

$$m_a = v_a \rho_a$$

Dividing equation (*i*) by equation (*ii*),

$$\frac{m_v}{m_a} = \frac{v_v \rho_v}{v_a \rho_a}$$

Since $v_a = v_v$, therefore humidity ratio,

$$W = \frac{m_v}{m_a} = \frac{\rho_v}{\rho_a} \quad \text{or} \quad \rho_v = W \rho_a$$

We know that $p_a v_a = m_a R_a T_d$

Since $v_a = \frac{1}{\rho_a}$ and $m_a = 1$ kg, therefore substituting these values we get

$$p_a \times \frac{1}{\rho_a} = R_a T_d \quad \text{or} \quad \rho_a = \frac{p_a}{R_a T_d}$$

Substituting the value of ρ_a in equation (*iii*), we have

$$\rho_v = \frac{W p_a}{R_a T_d} = \frac{W(p_b - p_v)}{R_a T_d} \quad \ldots (\because \; p_b = p_a + p_v)$$

where p_a = Pressure of air in kN/m^2,

R_a = Gas constant for air = 0.287 kJ/ kg K, and

T_d = Dry bulb temperature in K.

Example.1. The readings from a sling psychrometer are as follows ry bulb temperature = 30° C ; Barometer reading 740mm of Hg Using steam tables, determine : I. Dew point temperature ; 2. Relative humidity ; 3. Specific humidity ; 4. Degree of-saturation ; 5. Vapour density ; and 6. Enthalpy of mixture per kg of dry air.

Solution given: t_d = 30°C ; $t_{w.}$ = 20°C ; P_4= 740 mm of Hg

1.Dew point temperature

First of all, let us find the partial pressure of water vapour (P_v).

From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 20° C is

$$P_w = 0.023\ 37 \text{ bar}$$

We know that barometric pressure,

ph = 740 mm of Hg ... (Given)

= 740 x 133.3 = 98 642 N/m^2 … (∵ mm of Hg = 133.3 N/m^2)

= 0.986 42 bar …. ∵1 bar = 10^5 N/m^2)

∴ Partial pressure of water vapour,

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

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

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

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 Ans

2. Relative humidity

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

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

We know the relative humidity,

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

3. ***Specific humidity***

We know that specific humidity,

$$W = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622 \times 0.01701}{0.98642 - 0.01701}$$

$$= \frac{0.01058}{0.96941} = 0.010\,914 \text{ kg/kg of dry air}$$

$$= 10.914 \text{ g/kg of dry air} \quad \textbf{Ans.}$$

4. ***Degree of saturation***

We know that specific humidity of saturated air,

$$W_s = \frac{0.622\, p_s}{p_b - p_s} = \frac{0.622 \times 0.04242}{0.98642 - 0.04242}$$

$$= \frac{0.02638}{0.944} = 0.027\,945 \text{ kg/kg of dry air}$$

We know that degree of saturation,

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

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

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

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

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

5. ***Vapour density***

We know that vapour density,

$$\rho_v = \frac{W\,(p_b - p_v)}{R_a\, T_d} = \frac{0.010\,914\,(0.98642 - 0.01701)\,10^5}{287\,(273 + 30)}$$

$$= 0.012\,16 \text{ kg/m}^3 \text{ of dry air} \quad \textbf{Ans.}$$

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,

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

$$= 1.022 \times 30 + 0.010\,914\,[2466.1 + 2.3 \times 15]$$

$$= 30.66 + 27.29 = 57.95 \text{ kJ/kg of dry air} \quad \textbf{Ans.}$$

Example.2: On a particular day, the atmospheric air was found to have a dry bulb temperature of 30℃ and a wet bulb temperature of 18℃. The barometric pressure was observed to b 756mm of Hg. Using the tables of psychrometric properties of air, determine the relative humidity, the specific humidity, the dew point temperature, the enthalpy of air per kg of dry air and the volume of mixture per kg of dry air.

Solution: Given: $t_d = 30°C$; t_w - 18°C; $P_b = 756$ *mm* of *Hg*

Relative humidity

First of all, let us find the partial pressure of water vapour (p_v). From steam tables, we find that the saturation pressure corresponding to wet bulb temperature of 18°C is.

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

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

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

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

$$= 15.47 - 5.85 = 9.62 \text{ mm of Hg}$$

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

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

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

We know that the relative humidity,

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

Specific humidity

We know that specific humidity,

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

Dew point temperature

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour (P_v), therefore from steam tables, we find that corresponding to 9.62 mm of Hg or 9.62 x 133.3 = 1282.3 N/m^2 = 0.012 823 bar, the dew point temperature is,

$$t_{dp} = 10.6° \text{ C Ans.}$$

Enthalpy of air per kg of dry air

From steam tables, we also find that latent heat of vaporization of water at dew point temperature of 10.6°C,

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

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

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

$$= 1.022 \times 30 + 0.008\,(2476.5 + 2.3 \times 10.6)$$

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

Volume of the mixture per kg of dry air

From psychrometric tables, we find that specific volume of the dry air at 760 mm of Hg and 30°C dry bulb temperature is 0.8585 m^3/kg of dry air. We know that one kg of dry air at a partial pressure of (756 — 9.62) mm of Hg occupies the same volume as W = 0.008 kg of vapour at its partial pressure of 9.62 mm of Hg. Moreover, the mixture occupies the same volume but at a total pressure of 756 mm of Hg.

∴ Volume of the mixture (v) at a dry bulb temperature of 30°C and a pressure of 9.62 mm of Hg

= Volume of 1 kg of dry air (v_a) at a pressure of (756 — 9.62) or

746.38 mm of Hg

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

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

We know that $$v = v_a = \frac{R_a\, T_d}{p_a}$$

where R_a = Gas constant for air = 287 J/kg K

T_d = Dry bulb temperature in K

= 30 + 273 = 303 K, and

p_a = Pressure of air in N/m^2

$= P_b - P_v = 756 - 9.62 = 746.38$ mm of Hg

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

Substituting the values in the above equation,

$$v = \frac{287 \times 303}{994\,92} = 0.8741 \text{ m}^3\text{/kg of dry air} \quad \text{Ans.}$$

Example.3. The humidity ratio of atmospheric air at 28°C dry bulb temperature and 760 mm of mercury is 0.016 kg / kg of dry air. Determine: 1. partial pressure of Water vapour; 2.relative humidity; 3. dew point temperature; 4. specific enthalpy; and 5. vapour density.

Solution: Given: $t_d = 28$°C ; $P_b = 760$ mm of Hg ; W = 0.016 kg/ kg of dry air

1.Partial pressure of water vapour

Let P_v = Partial pressure of water vapour.
We know that humidity ratio (W),

$$0.016 = \frac{0.622\,p_v}{p_b - p_v} = \frac{0.622\,p_v}{760 - p_v}$$

$12.16 - 0.016\,P_v = 0.622\,P_v$ or $0.638\,P_v = 12.16$

$P_v = 12.16/0.638 = 19.06$ mm of Hg

$= 19.06 \times 133.3 = 2540.6$ N/m^2 Ans.

2. Relative humidity

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

$P_s = 0.03778$ bar $= 3778$ N/m^2

∴ Relative humidity,

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

3. Dew point temperature

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour (P_v), therefore from steam tables, we find that corresponding to a pressure of 2540.6 N/m^2 (0.025406 bar), the dew point temperature is,

$t_{dp} = 21.1°$ C Ans.

4. Specific enthalpy

From steam tables, latent heat of vaporization of water corresponding to a dew point temperature of 21.1° C,

$$h_{fgdp} = 2451.76 \text{ kJ/kg}$$

We know that specific enthalpy.

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

$$= 1.022 \times 28 + 0.016\,(2451.76 + 2.3 \times 21\_1)$$

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

5. Vapour density

We know that vapour density,

$$\rho_v = \frac{W\,(p_b - p_v)'}{R_a\,T_d} = \frac{0.016\,(760 - 19.06)\,133.3}{287\,(273 + 28)}$$

$$= 0.0183 \text{ kg/m}^3 \text{ of dry air.}$$

3.5 THERMODYNAMIC WET BULB TEMPERATURE OR ADIABATIC SATURATION TEMPERATURE

The thermodynamic wet bulb temperature or adiabatic saturation temperature is the temperature at which the air can be brought to saturation state, adiabatically, by the evaporation of water into the flowing air.

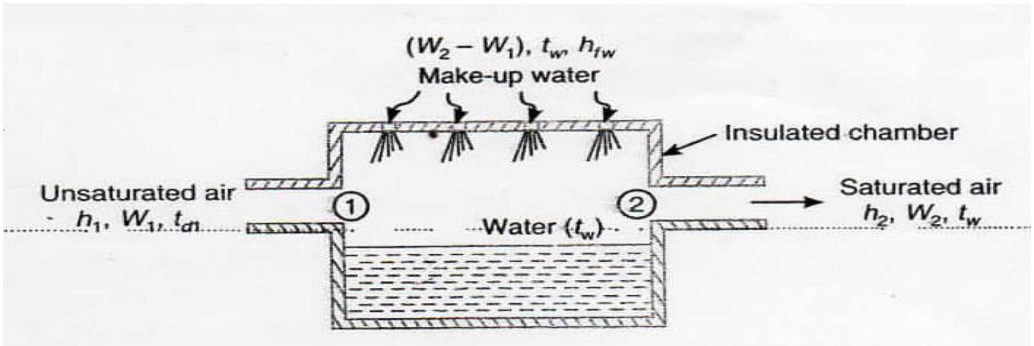


Fig.4 Adiabatic saturation of air.

The equipment used for the adiabatic saturation of air, in its simplest form, consists of an insulated chamber containing adequate quantity of water. There is also an arrangement for extra water (known as make-up water) to flow into the chamber from its top, as shown in Fig.4.

Let the unsaturated air enters the chamber at section 1. As the air passes through the chamber over a long sheet of water, the water evaporates which is carried with the flowing stream of air, and the specific humidity of the air increases. The make-up water is added to the chamber at this temperature to make the water level constant. Both the air and water are cooled as the evaporation takes place. This process continues until the energy transferred from the air to the water is equal to the energy required to vaporize the water. When steady conditions are reached, the air flowing at section 2 is saturated with water vapour. The temperature of the saturated air at section 2 is known as *thermodynamic wet bulb temperature or adiabatic saturation temperature.*

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

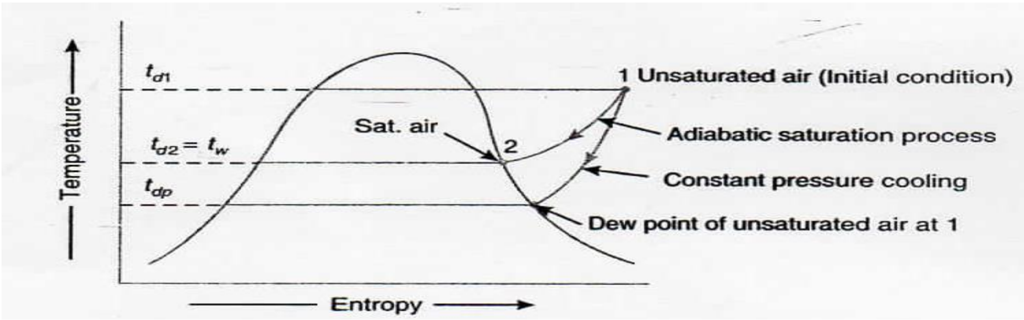


Fig.5. T-s diagram for adiabatic saturation process

During the adiabatic saturation process, the partial pressure of vapour increases, although the total ressure of the air-vapour mixture. The unsaturated air initially at dry bulb temperature t_{d2}, is coo e adiabatically to dry bulb temperature td, which is equal to the adiabatic saturation temperature t_w. It may be noted that the adiabatic saturation temperature is taken equal to the wet bulb temperature for all practical purposes.

Let h_1 = Enthalpy of unsaturated air at section 1,

W_1 = Specific humidity of air at section 1,

h_2, W_2 = Corresponding values of saturated air at section 2, and

h_{fw}= Sensible heat of water at adiabatic saturation temperature.

Balancing the enthalpies of air at inlet and outlet (i.e. at sections 1 and 2),

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

or

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

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

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

and $\quad h_2 = h_{a2} + W_2\, h_{s2}$

where $\quad h_{a1}$ = 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).

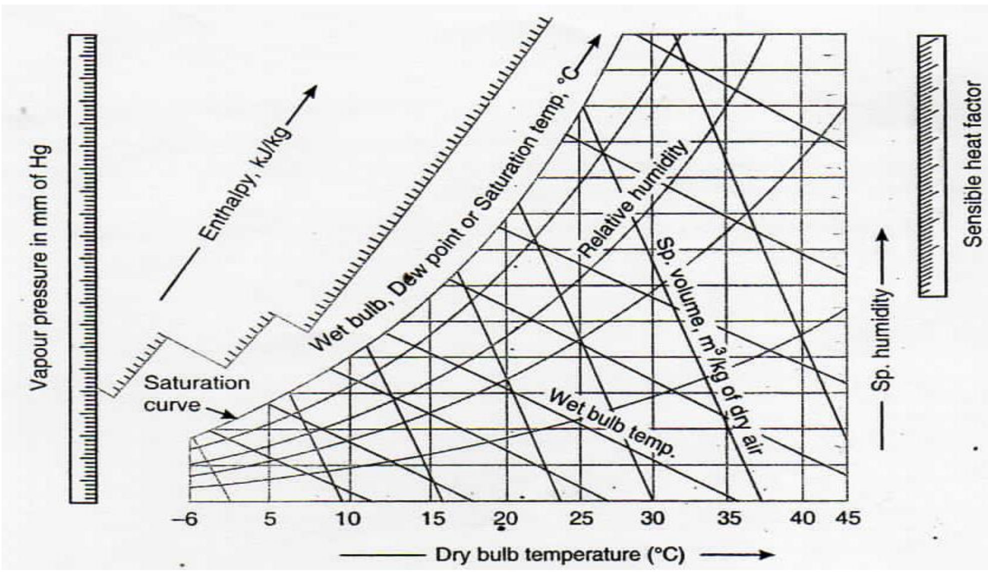


Fig. 6 Psychrometric chart.

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.

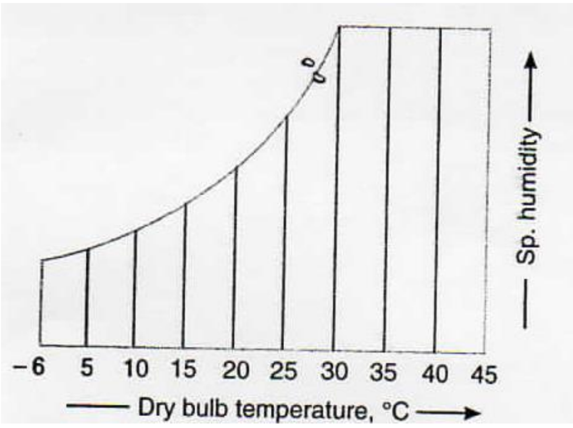


Fig.7. Dry bulb temperature lines.

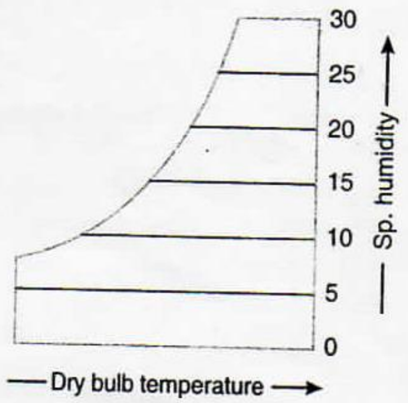
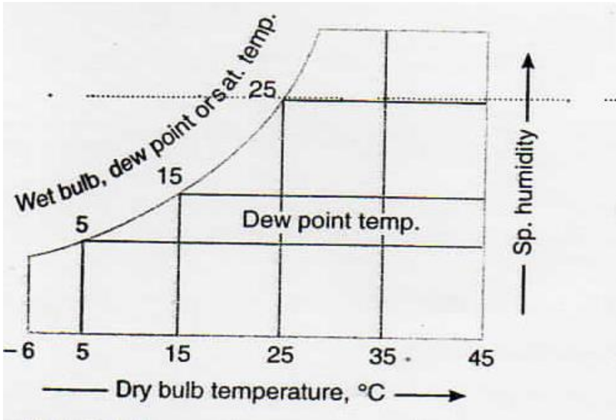


Fig. 8. Specific humidity lines.

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.



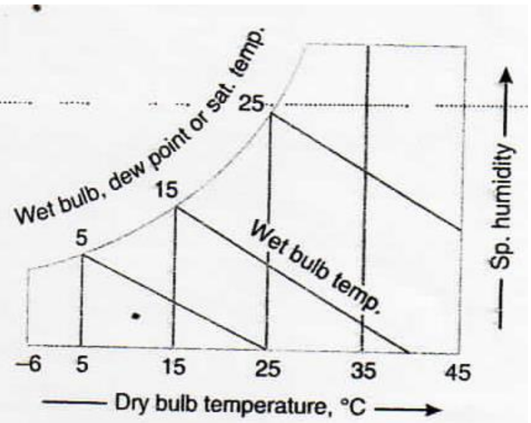


Fig. 9 Dew point temperature lines. Fig.10 Wet bulb temperature lines.

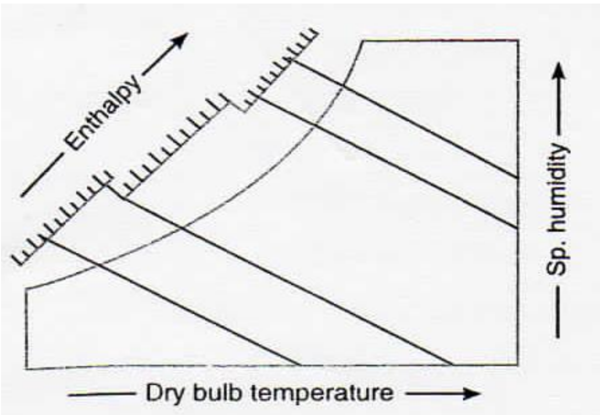
4. ***Wet bulb temperature lines***. The wet bulb temperature lines are inclined straight lines and non-uniformly spaced as shown in Fig.10. At any point on the saturation curve, the dry bulb and wet bulb temperatures are equal.

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.



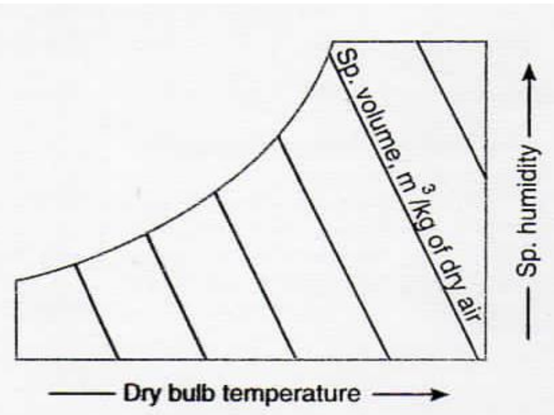


Fig. 11. Enthalpy lines. Fig. 12. Specific volume lines.

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.

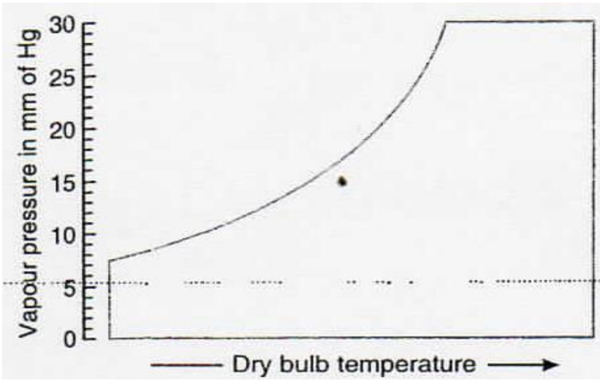


Fig. 13. Vapour pressure lines.

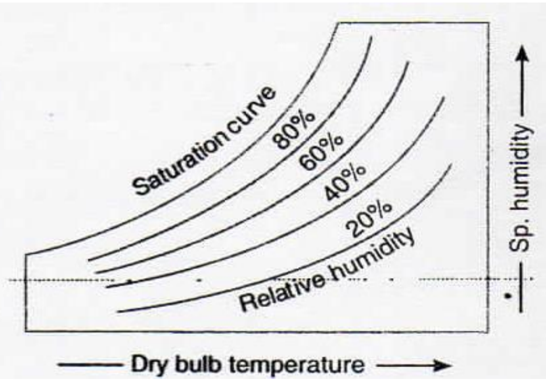


Fig. 14. Relative humidity lines.

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.

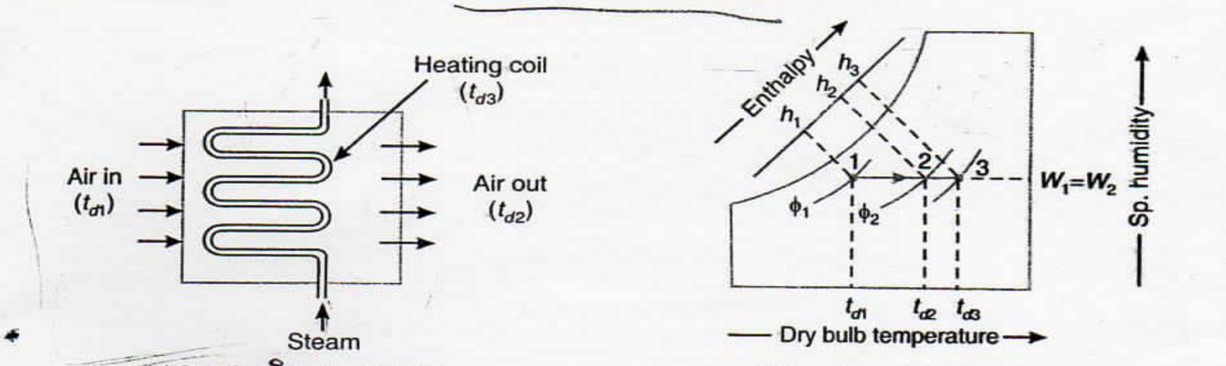
The heat absorbed by the air during sensible heating may be obtained from the psychrometric chart by the enthalpy difference (h_2 - h_1) as shown in Fig. 15 (b). It may be noted that the specific humidity during the sensible heating remains constant (i.e. $W_1 = W_2$).

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.

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



(*a*) Psychrometric process. (*b*) Psychrometric chart.

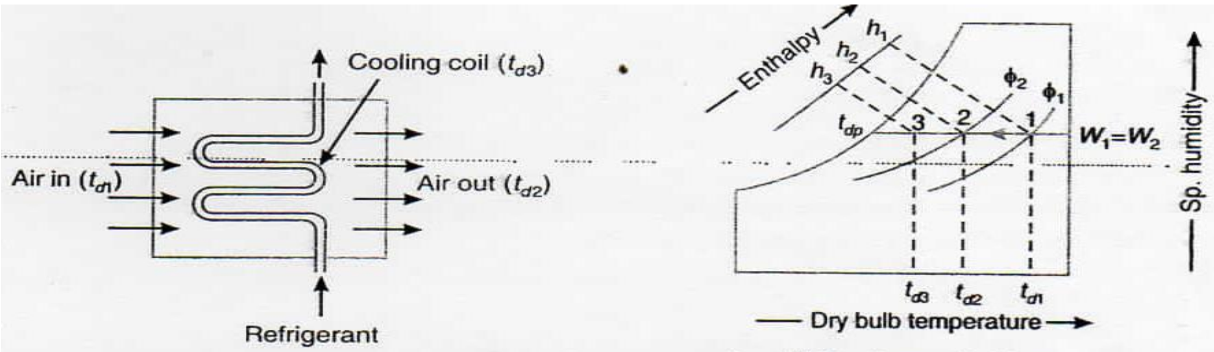
Fig.15 Sensible heating

Notes: 1. For sensible heating, steam or hot water is passed through the heating coil. The heating coil may be electric resistance coil.

2. The sensible heating of moist air can be done to any desired temperature.

3.72 Sensible Cooling

The cooling of air without any change in its specific humidity, is known as sensible cooling. Let air at temperature rd, passes over a cooling coil of temperature t_{d3} as shown in Fig. 16 (a). It may be noted that the temperature of air leaving the cooling coil (t_{d2}) will be more than t_{d3}. The process of sensible cooling, on the psychrometric chart, is shown by a horizontal line 1-2 extending from right to left as shown in Fig. 16



(*a*) Psychrometric process. (*b*) Psychrometric chart.

Fig. 16 Sensible cooling.

The heat rejected by air during sensible cooling may be obtained from the psychrometric chart by the enthalpy difference (h1 — h,) as shown in Fig. 16(b).

It may be noted that the specific humidity during the sensible cooling remains constant (i.e.$W_1 = W_2$). The dry bulb temperature reduces from t_{d1} to t_{d2} and relative humidity increases from $\emptyset_1$ to $\emptyset_2$ as shown in Fig. 16(b). The amount of heat rejected during sensible cooling may also be obtained from the relation:

Heat rejected, $q = h_l - h_2$

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

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

The term (C_{pa} + W C_{ps}) is called humid specific heat (C_{pm}) and its value is taken as 1.022 kJ /kg K.

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

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

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

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

ρ = Density of moist air at 20° C and 50% relative humidity

= 1.2 kg / m3 of dry air,

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

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

the entering and leaving conditions of air in ° C.

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

$SH = v \times 1.2 \times 1.022 \times \Delta t = 1.2264\, v \times \Delta t$ kJ/min

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

3.73 By-pass Factor of Heating and Cooling Coil

The temperature of the air corning out of the apparatus (t_{d2}) will be less than *t_{d3} in case the coil is a heating coil and more than t_{d3} in case the coil is a cooling coil.

Let 1 kg of air at temperature t_{d1} is passed over the coil having its temperature (i.e. coil surface temperature) t_{d3} as shown in Fig. 17.

A little consideration will show that when air passes over a coil, some of it (say x kg) just by-passes unaffected while the remaining $(1 - x)$ kg comes in direct contact with the coil. This by-pass process of air is measured in terms of a by-pass factor. The amount of air that by-passes or the by-pass factor depends upon the following factors :

1. The number of fins provided in a unit length i.e. the pitch of the cooling coil fins ;

2. The number of rows in a coil in the direction of flow; and

3. The velocity of flow of air.

It may be noted that the by-pass factor of a cooling coil decreases with decrease in fin spacing and increase in number of rows.

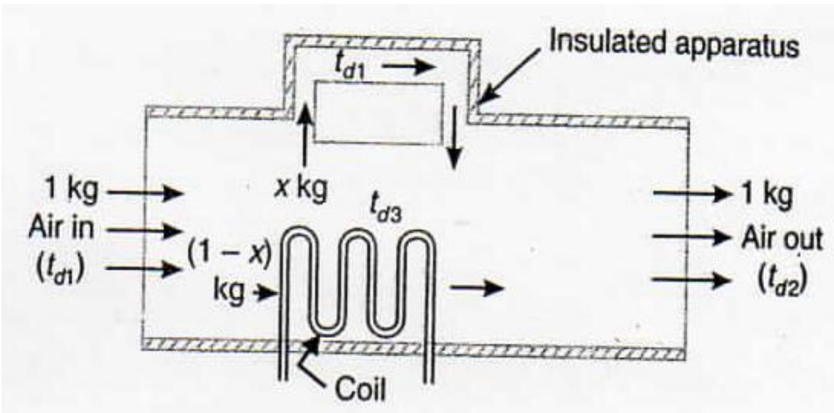


Fig.17. By-pass factor

Balancing the enthalpies, we get

$$x\, c_{pm}\, t_{d1} + (1 - x)\, c_{pm}\, t_{d3} = 1 \times c_{pm}\, t_{d2} \quad \text{... (where } c_{pm} = \text{Specific humid heat)}$$

or

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

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

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,

$$BPF = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

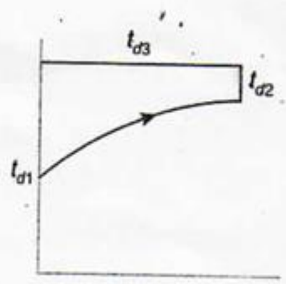


Fig. 18.

The by-pass factor for heating or cooling coil may also be obtained as discussed below :

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,

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

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

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

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

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

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,

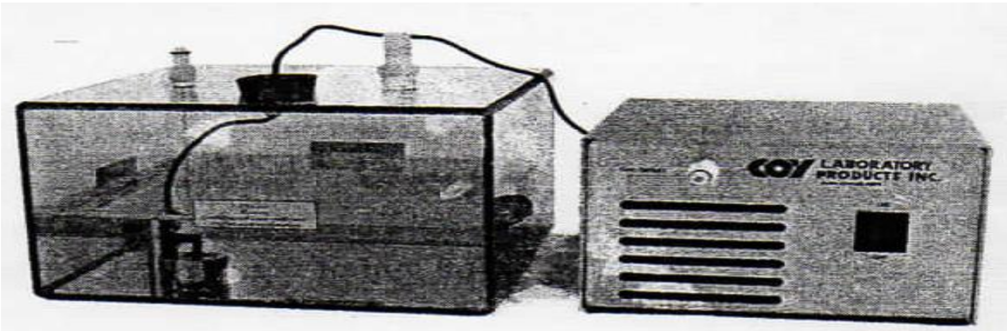
$$\eta_H = 1 - BPF = 1 - \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}} = \frac{t_{d2} - t_{d1}}{t_{d3} - t_{d1}}$$

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 ϕ_1 to ϕ_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 ϕ_1 to ϕ_2 and specific humidity also decreases from W_1 to W_2 as shown in Fig. 19 (b).

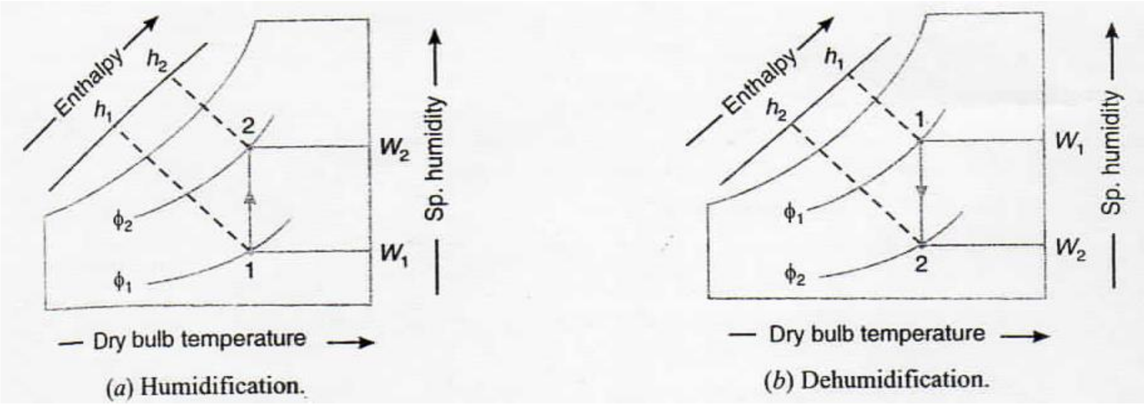
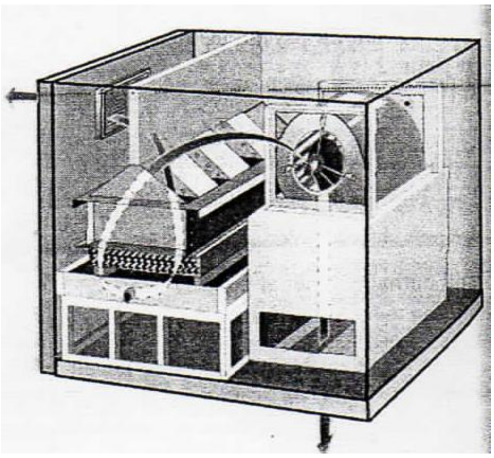


Fig. 19 Humidification and dehumidification

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

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

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

ρ = Density of moist air = 1.2 kg/m^3 of dry air,

h_{fg} = Latent heat of vaporization = 2500 kJ /kg, and

AW = Difference of specific humidity between the entering and leaving conditions of

air = $(W_2 - W_1)$ for humidification and $(W_1 – W_2)$ for dehumidification.

Substituting these values in the above expression, we get

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

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

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:

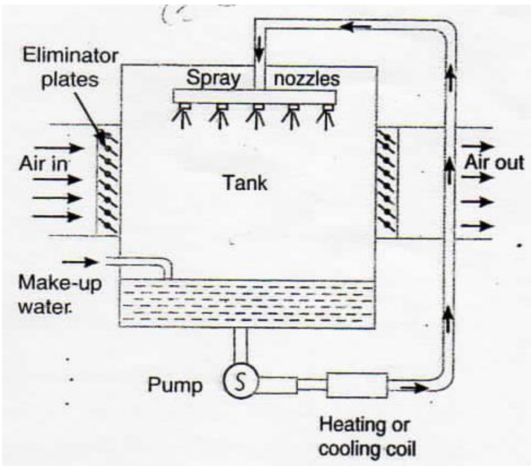


Fig. 20. Air-washer.

(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,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{SH + LH}$$

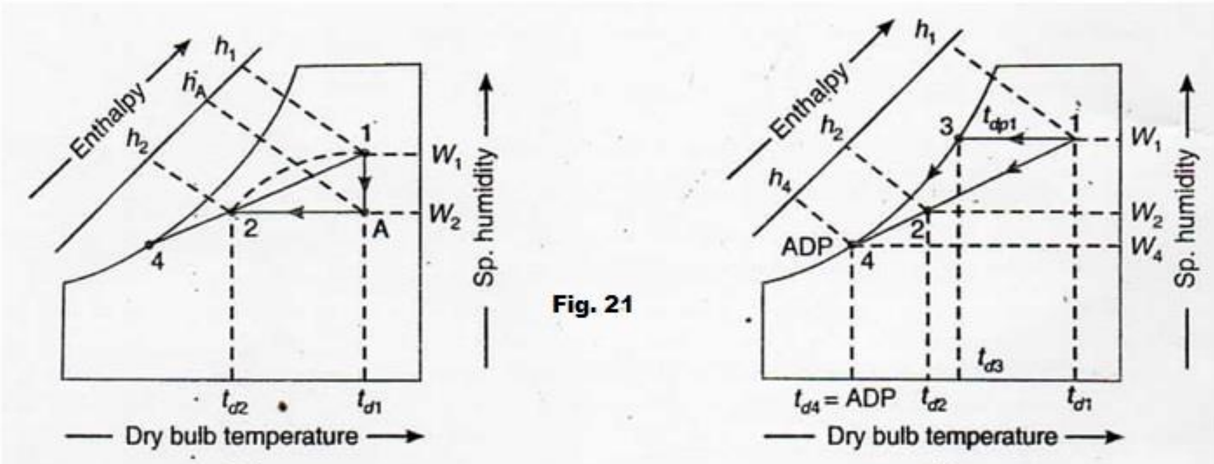
where SH = Sensible heat, and

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.



(a) **Fig. 21** Cooling and dehumidification. (b)

t_{d1} = Dry bulb temperature of air entering the coil,

t_{dpl} = Dew point temperature of the entering air = t_{d3} and

t_{d4} = Effective surface temperature or ADP of the coil.

Under ideal conditions, the dry bulb temperature of the air leaving the cooling coil (i.e. t_{d4}) should be equal to the surface temperature of the cooling coil (i.e. ADP), but it is never possible due to inefficiency of the cooling coil. Therefore, the resulting condition of air coming out of the coil is shown by a point 2 on the straight line joining the points 1 and 4. The by-pass factor in this case is given by

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

Also

$$BPF = \frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$$

Actually, the cooling and dehumidification process follows the path as shown by a dotted curve in Fig. 21(a), but for the calculation of psychrometric properties, only end points are important. Thus the cooling and dehumidification process shown by a line 1-2 may be assumed to have followed a path 1-A (i.e. dehumidification) and A-2 (i.e. cooling) as shown in Fig. 21 (a). We see that the total heat removed from the air during the cooling and dehumidification process is

$$q = h_1 - h_2 = (h_1 - h_A) + (h_A - h_2) = LH + SH$$

where LH = $h_1 - h_A$ = Latent heat removed due to condensation of vapour of the reduced moisture content ($W_1 - W_2$), and

SH = $h_A - h_2$ = Sensible heat removed.

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{LH + SH} = \frac{h_A - h_2}{h_1 - h_2}$$

Note: The line 1-4 (i.e. the line joining the point of entering air and the apparatus dew point) in Fig. 21 (b) is known as sensible heat factor line.

Example 1: In a cooling application, moist air enters a refrigeration coil at the rate of 100 kg of dry air per minute at 35° C and 50% RH. The apparatus dew point of coil is 5° C and by-pass factor is 0.15. Determine the outlet state of moist air and cooling capacity of coil in TR.

Solution Given: m_a = 100 kg/min; t_{dt} = 35°C; ∅ = 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,

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

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

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

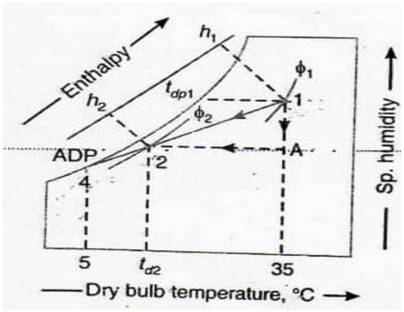


Fig.22

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature (t_{d2},) of 9.5°Con the line 1-4 is $\emptyset_2$ = 99%. Ans.

Cooling capacity of the coil

The resulting condition of the air coming out of the coil is shown by point 2, on the line joining the points 1 and 4, as shown in Fig. 22. The line 1-2 represents the cooling and dehumidification process which may be assumed to have followed the path 1-A (i.e. dehumidification) and A-2 (i.e. cooling). Now from the psychrometric chart, we find that enthalpy of entering air at point 1,

h_1= 81 kJ/kg of dry air

and enthalpy of air at point 2,

h_2= 28 kJ/kg of dry air

We know that cooling capacity of the coil

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

= 5300/210 = 25.24 TR Ans. ….(∵ 1 TR = 210 kJ/min)

Example 2. 39.6 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.

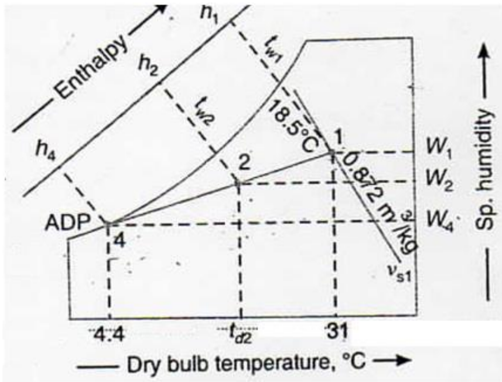
Solution: Given: v_1= 39.6 m³/min; t_{dt} = 31°C; t_{wt} = I8.5°C; ADP= t_{d4}= 4.4°C; Q= 12.5 kW = 12.5 kJ/s = 12.5 x 60 kJ/min

Dry and wet bulb temperature of the air leaving the coil

Let t_{d2} and t_{w2} = Dry and wet bulb temperature of the air leaving the coil.

First of all, mark the initial condition of air, i.e. 31°C dry bulb temperature and 18.5°C wet bulb temperature on the psychrometric chart at point 1, as shown in Fig. 23. Now mark the effective surface temperature (ADP) of the coil at 4.4°C at point 4.

From the psychrometric chart, we find that enthalpy at point 1



h_1 = 52.5 kJ / kg of dry air

Enthalpy at point 4,

h_4 = 17.7kJ/kg of dry air

Specific humidity at point 1

W_1 = 0.0082 kg / kg of dry air

Specific humidity at point 4,

W_4= 0.00525 kg / kg of dry air

Specific volume at point

v_{s1}, = 0.872m³/ kg

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

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

and cooling capacity of the coil,

$$Q = m_a (h_1 - h_2)$$

or

$$h_1 - h_2 = \frac{Q}{m_a} = \frac{12.5 \times 60}{44.41} = 16.89 \text{ kJ / kg of dry air}$$

$\therefore$

$$h_2 = h_1 - 16.89 = 52.5 - 16.89 = 35.61 \text{ kJ / kg of dry air}$$

The equation for the condition line 1–2–4 is given as

$$\frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$$

$$\frac{W_2 - 0.005\,25}{0.0082 - 0.005\,25} = \frac{35.61 - 17.7}{52.5 - 17.7}$$

$\therefore$

$$W_2 = 0.006\,77 \text{ kg / kg of dry air}$$

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

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

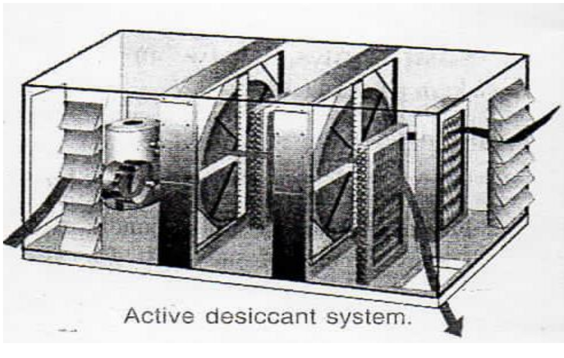
By-pass factor

We know that by-pass factor,

$$BPF = \frac{h_2 - h_4}{h_1 - h_4} = \frac{35.61 - 17.7}{52.5 - 17.7} = 0.5146 \text{ Ans.}$$

3.10 Heating and Humidification

This process is generally used in winter air conditioning to warm and humidify the air. It is the reverse process of cooling and -- dehumidification. When air is passed through a humidifier having spray water temperature higher than the dry bulb temperature of the entering air, the unsaturated air will reach the condition of saturation and thus the air becomes hot. The heat of vaporization of water is absorbed from the spray water itself and hence it gets cooled. In this way, the air becomes heated and humidified. The process of heating and humidification is shown by line 1-2 on the psychrometric chart as shown in Fig. 24.



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

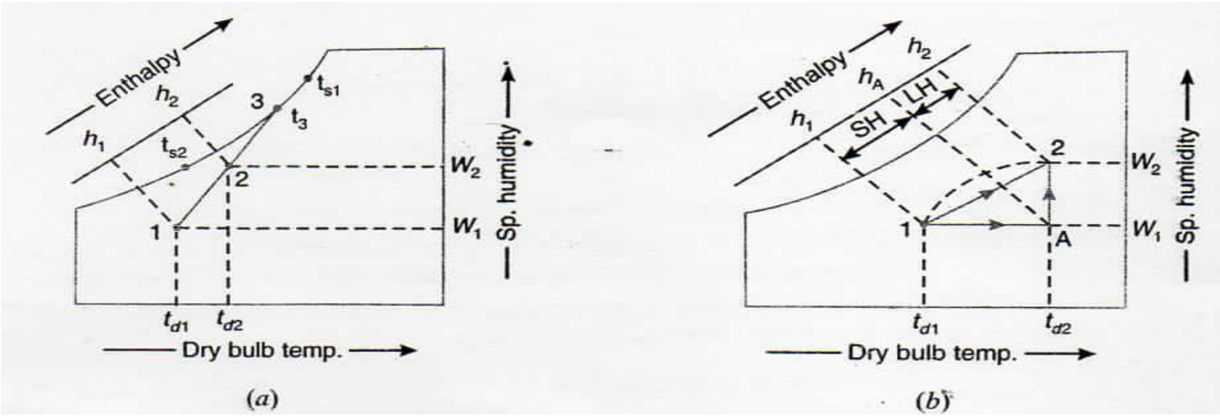


Fig.24 heating and humidification

Let m_{w1} and m_{w2} = Mass of spray water entering and leaving the humidifier in kg,

h_{fw1} and h_{fw2} = Enthalpy of spray water entering and leaving the humidifier in kJ/kg,

W_1 and W_2 = Specific humidity of the entering and leaving air in kg/kg of dry

h_1 and h_2 = Enthalpy of entering and leaving air in kJ/kg of dry air, and

m_a = Mass of dry air entering in kg.

For mass balance of spray water,

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

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

or and for enthalpy balance,

$$m_{w1}\, h_{fw1} = m_{w2}\, h_{fw2} = m_a(h_2 - h_1) \qquad \text{....}(ii)$$

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

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

$$= m_a (h_2 - h_1)$$

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

The temperatures t_{s1} and t_{s2} shown in Fig. 24 (a) denote the temperatures of entering and leaving spray water respectively. The temperature 13 is the mean temperature of the spray water which the entering air may be assumed to approach.

Actually, the heating and humidification process follows the path as shown by dotted curve in Fig. 24(b), but for the calculation of psychrometric properties, only the end points are important. Thus, the heating and humidification process shown by a line 1-2 on the psychrometric chart may be assumed to have followed the path 1-A (i.e. heating) and A-2 (i.e. humidification), as shown in Fig. 24(b). We see that the total heat added to the air during heating and humidification is

$$q = h_2 - h_1 = (h_2 - h_1) + (h_A - h_i) = q_t + q_s$$

where $q_t = (h_2 - h_A)$ = Latent heat of vaporization of the increased moisture

content ($W_2 - W_1$), and

$q_s = (h_A - h_i)$ = Sensible heat added

We know that sensible heat factor,

$$SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{q_S}{q} = \frac{q_S}{q_S + q_L} = \frac{h_A - h_1}{h_2 - h_1}$$

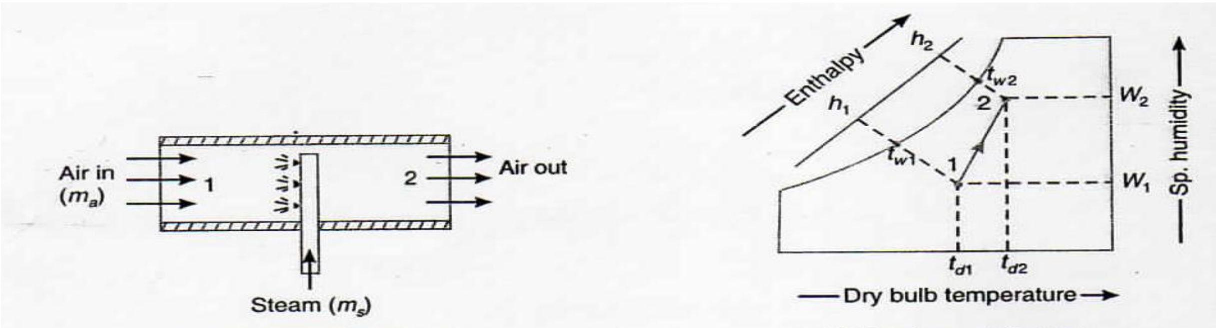
Note: The line 1-2 in Fig. 24 (b) is called sensible heat factor line.

3.11 Heating and Humidification by Steam Injection

The steam is normally injected into the air in order to increase its specific humidity as shown in Fig. 25 (a). This process is used for the air conditioning of textile mills where high humidity is to be maintained. The dry bulb temperature of air changes very little during this process, as shown on the psychrometric chart in Fig. 25 (b).

Let m_s = Mass of steam supplied,

m_a = Mass of dry air entering,



(*a*) Psychrometric process. (*b*) Psychrometric chart.

Fig.25 heating and humidification by steam injection

W_1= Specific humidity of air entering,

W_2 = Specific humidity of air leaving,

h_1 = Enthalpy of air entering,

h_2 = Enthalpy of air leaving, and

h_s = Enthalpy of steam injected into the air.

Now for the mass balance,

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

and for the heat balance,

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

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 k1/kg of dry air

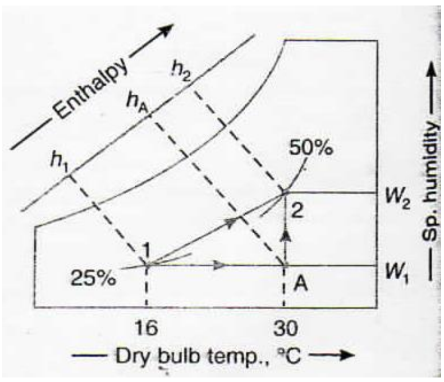


Fig.26

Enthalpy of air at point A,

h_A = 38 kJ/kg of dry air

and enthalpy of air at point 2,

h_2 = 64 la/kg of dry air

∴ Heat added to the air

= h_2- h_1= 64 - 23 = 41 kJ/kg of dry air Ans.

Moisture added to the air

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

W_1 = 0.0026 kg/kg of dry air

and specific humidity in the air at point 2,

W_2 = 0.0132 kg /kg of dry air

∴ Moisture added to the air

= $W_2 - W_1$ = 0.0132 - 0.0026 = 0.0106 kg/kg of dry air Ans.

Sensible heat factor of the process

We know that sensible heat factor of the process,

$$SHF = \frac{h_A - h_1}{h_2 - h_1} = \frac{38 - 23}{64 - 23} = 0.366 \text{ Ans.}$$

Example 4: Air at 10°C dry bulb temperature and 90% relative humidity is to be ated and humidified to 35°C dry bath temperature and 22.5°C wet bulb temperature. The air is pre-heated sensibly before passing to the air washer in which water is re-circulated. The relative humidity of the air coming out of the air washer is 90%. This air is again reheated sensibly to obtain the final desired condition. Find: 1. the temperature to which the air should be preheated. 2. the total heating required; 3. the makeup water required in the air washer ; and 4. the humidifying efficiency of the air washer.

Solution: Given : t_{dt} = 10°C; $\emptyset_1$ = 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.

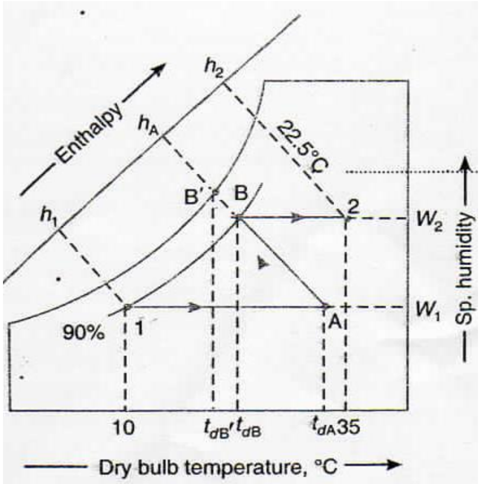


Fig.27

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

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

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

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,

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

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

3.12 Heating and Dehumidification -Adiabatic Chemical Dehumidification

This process is mainly used in industrial air conditioning and can also be used for some comfort air conditioning installations requiring either a low relative humidity or low dew point temperature in the room.

In this process, the air is passed over chemicals which have an affinity for moisture. As the air comes in contact with these chemicals, the moisture gets condensed out of the air and gives up its latent heat. Due to the condensation, the specific humidity decreases and the heat of condensation supplies sensible heat for heating the air and thus increasing its dry bulb temperature.

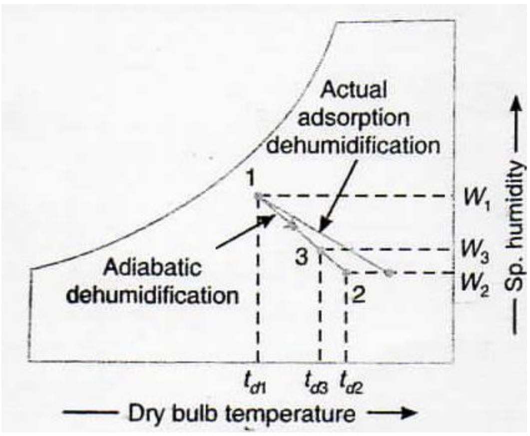


Fig.28

The process, which is the reverse of adiabatic saturation process, is shown by the line 1-2 on the psychrometric chart as shown in Fig. 28. The path followed during the process is along the constant wet bulb temperature line or-constant enthalpy line.

The effectiveness or efficiency of the dehumidifier is given as

$$\eta_H = \frac{\text{Actual increase in dry bulb temperature}}{\text{Ideal increase in dry bulb temperature}} = \frac{t_{d3} - t_{d1}}{t_{d2} - t_{d1}}$$

Notes: 1. In actual practice, the process is accompanied with a release of heat called heat of adsorption, which is very large. Thus the sensible heat gain of air exceeds the loss of latent heat and the process is shown above the constant wet bulb temperature line in Fig. 28.

2. Two types of chemicals used for dehumidification are absorbents and adsorbents. The absorbents are substances which can take up moisture from air and during this process change it chemically, physically or in both respects. These include water solutions or brines of calcium chloride, lithium chloride, lithium bromide and ethylene glycol. These are used as air dehydrators by spraying or otherwise exposing a large surface of the solution in the air stream.

The adsorbents are substances in the solid state which can take up moisture from the air and during this process do not change it chemically or physically. These include silca gel (which is a form of silicon dioxide prepared by mixing fused sodium silicate and sulphuric acid) and activated alumina (which is a porous amorphous form of aluminum oxide).

Example 5: Saturated air at 21° C is passed through a drier so that its final relative humidity is 20%. The drier uses silica gel adsorbent. The air is then passed through a cooler until its final temperature is 21° C without a change in specific humidity. Determine : 1. the temperature of air at the end of the drying process; 2. the heat rejected during the cooling process ; 3. the relative humidity at the end of cooling process; 4. the dew point temperature at the end of the drying process ; and 5. the moisture removed during the drying process.

Solution: Given: t_{d1}, = t_{d3} = 21°C; ϕ_2= 20%

1.Temperature of air at the end of drying process

First of all, mark the initial condition of air i.e. at 21°C dry bulb temperature upto the saturation curve (because the air is saturated) on the psychrometric chart at point 1, as shown in Fig. 29. Since the drying process is a chemical dehumidification process, therefore-. it follows a path along-the-constant wet bulb temperature or the constant enthalpy line as shown by the line 1- 2 in Fig. 29. Now mark the point 2 at relative humidity of 20%. From the psychrometric chart, the temperature at the end of drying process at point 2, t_{d2} = 38.5°C Ans.

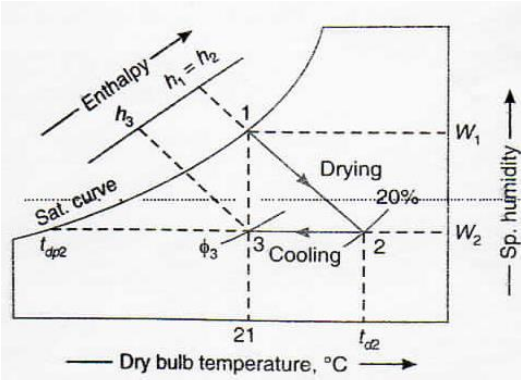


Fig.29

2. Heat rejected during the cooling process

The cooling process is shown by the line 2-3 on the psychrometric chart as shown in Fig.29. From the psychrometric chart. we find that enthalpy of air at point 2.

h_2 = 61 kJ /kg of dry air

and enthalpy of air at point 3,

h_3 = 43 kJ/kg of dry air

∴ Heat rejected during the cooling process

= h_2 - h_3 = 61— 43 = 18 kJ/kg of dry air A ns.

3. Relative humidity at the end of cooling process

From the psychrometric chart, we find that relative humidity at the end of cooling process (i.e. at point 3),

$\emptyset_3$ = 55% Ans.

4. Dew point temperature at the end of drying process

From the psychrometric chart, we find that the dew point temperature at the end of the drying process,

t_{dp2} = 11.6° C Ans.

5. Moisture removed during the drying process

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

W_1 = 0.0157 kg/kg of dry air

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

W_2 = 0.0084 kg/kg of dry air

∴ Moisture removed during the drying process

$$= W, — W, = 0.0157\ 0.0084 = 0.0073 \text{ kg/kg of dry air Ans.}$$

AIR CONDITIONING SYSTEMS

3.13 Introduction

The air conditioning is that branch of engineering science which deals with the study of conditioning of air i.e., supplying and maintaining desirable internal atmosphere conditions for human comfort, irrespective of external conditions. This subject, in its broad sense, also deals with the conditioning of air for industrial purposes, food processing storage of food and other materials.

3.14 Factors affecting comfort Air Conditioning

The four important factors for comfort air conditioning are discussed as below:

1. Temperature of air: In air conditioning, the control of temperature means the maintenance of any desired temperature within an enclosed space even though the temperature of the outside air is above or below the desired room temperature. This is accomplished either by the addition or removal of heat from the enclosed space as and when demanded. It may be noted that a human being feels comfortable when the air is at 21°C with 56% relative humidity.

2. Humidity of air: The control of humidity of air means the decreasing or increasing of moisture contents of air during summer or winter respectively in order to produce comfortable and healthy conditions. The control of humidity is not only necessary for human comfort but it also increases the efficiency of the workers. In general, for summer air conditioning, the relative humidity should not be less than 60% whereas for winter air conditioning it should not be more than 40%.

3. Purity of air: It is an important factor for the comfort of a human body. It has been noticed that people do not feel comfortable when breathing contaminated air, even if it is within acceptable temperature and humidity ranges. It is thus obvious that proper filtration, cleaning and purification of air is essential to keep it free from dust and other impurities.

4. Motion of air: The motion or circulation of air is another important factor which should be controlled, in order to keep constant temperature throughout the conditioned space. It is, therefore, necessary that there should be equi-distribution of air throughout the space to be air conditioned.

3.15 Air Conditioning System

We have already discussed in Art. the four important factors which affect the human comfort. The system which effectively controls these conditions to produce the desired effects upon the occupants of the space is known as an *air conditioning system*.

3.16 Equipments Used in an Air Conditioning System

Following are the main equipments or parts used in an air conditioning system:

1. Circulation fan. The main function of this fan is to move air to and from the room.

2. Air conditioning unit. It is a unit which consists of cooling and dehumidifying processes for summer air conditioning or heating and humidification processes for winter air Conditioning.

3. Supply duct. It directs the conditioned air from the circulating fan to the space to be air conditioned at proper point

4. Supply outlets. These are grills which distribute the conditioned air evenly in the room.

5. Return outlets. These are the openings in a room surface which allow the room ait to enter the return duct.

6. Filters. The main function of the filters is to remove dust, dirt and other harmful bacteria from the air.

3.17 Classification of Air Conditioning Systems

The air conditioning systems may be broadly classified as follows:

1. *According to the purpose*

   (a) Comfort air conditioning system, and

   (b) Industrial air conditioning system.

2. *According to season of the year*

   (a) Winter air conditioning system,

   (b) Summer air conditioning system, and

   (c) Year-round air conditioning system.

3. *According to the arrangement of equipment*

   (a) Unitary air conditioning system, and

   (b) Central air conditioning system.

In this chapter, we shall discuss all the above mentioned air conditioning systems one by one.

3.18 Comfort Air Conditioning System

In comfort air conditioning, the air is brought to the required dry bulb temperature and relative humidity for the human health, comfort and efficiency. If sufficient data of the required condition is not given, then it is assumed to be 21°C dry bulb temperature and 50% relative humidity. The sensible heat factor is, generally, kept as following:

For residence or private office = 0.9

For restaurant or busy office = 0.8

Auditorium or cinema hall = 0.7

Ball room dance hall etc. = 0.6

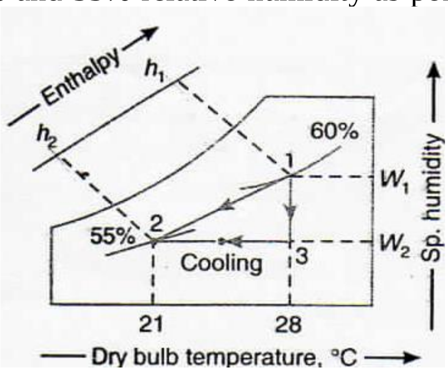
The comfort air conditioning may be adopted for homes, offices, shops, restaurants, theatres, hospitals, schools etc.

Example 1: ***An air conditioning plant is required to supply 60 m of air per minute at a DBT of 21°C and 55% RH. The outside air is at DBT of 28°C and 60% RH. Determine the mass of water drained and capacity of the cooling coil. Assume the air conditioning plant first to dehumidify and then to cool the air.***

Solution: Given $v_2 = 60\ m^3/min$; $t_{d2} = 21°C$; $Ø_2 = 55\%$; $t_{dt} = 28°C$; $Ø_1 = 60\%$

Mass of water drained

First of all, mark the initial condition of air at 28°C dry bulb temperature and 60% relative humidity on the 1 h psychrometric chart as point 1, as shown in Fig. 1. Now mark the final condition of air at 21°C dry bulb temperature % and 55% relative humidity as point 2. From the psychrometric chart, we find that



Specific humidity of air at point 1,

$W_1 = 0.0141$ kg/kg of dry air

Specific humidity of air at point 2,

$W_2 = 0.0084$ kg / kg of dry air

and specific volume of air at point 2,

$v_{s2} = 0.845$ m3/ kg of dry air

We know that mass of air circulated,

$$m_a = \frac{v_2}{v_{s2}} = \frac{60}{0.845} = 71 \text{ kg / min}$$

∴ Mass of water drained

$= m_a (W_1 - W_2) = 71(0.0142 — 0.0084) = 0.412$ kg / min

$= 0.412 \times 60 = 24.72$ kg / h Ans.

Capacity of the cooling coil

From the psychrometric chart, we find that

Enthalpy of air at point 1,

$h_1 = 64.8$ kJ / kg of dry air

and enthalpy of air at point 2,

$h_2 = 42.4$ kJ / kg of dry air

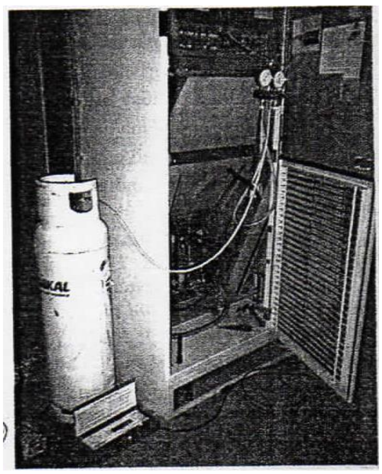
$\therefore$ Capacity of the cooling coil

$= m_a (h_1 - h_2) = 71(64.8 - 42.4) = 1590.4$ kJ / min

$= 1590.4 / 210 = 7.57$ TR Ans.

3.19 Industrial Air Conditioning System

It is an important system of air conditioning these days in which the inside dry bulb temperature and relative humidity of the air is kept constant for proper working of the machines and for the proper research and manufacturing processes. Some of the sophisticated electronic and other machines need a particular dry bulb temperature and relative humidity. Sometimes, these machines also require a particular method of psychrometric processes. This type of air conditioning system is used in textile mills, paper mills, machine-parts manufacturing plants, tool rooms, photo-processing plants etc.



Industrial air-conditioning system

Example 2: ***Following data refers to an air conditioning system to be designed for an industrial***

process for hot and wet climate:

Outside conditions = 30° C DBT and 75% RH

Required inside conditions = 20° C DBT and 60% RH

The required condition is to be achieved first by cooling and dehumidifying and then by heating. If 20 nil of air is absorbed-by the plant every minute, find : 1. capacity of the cooling coil in tones of refrigeration; 2. capacity of the heating coil in kW; 3. amount of water removed per hour; and 4. By-pass factor of the heating coil, if its surface temperature is 35°C.

Solution: Given t_{dt} = 30°C; $\emptyset_1$= 75%; t_{d3}= 20°C; $\emptyset_3$ = 60%; v_l = 20 m^3/ min; t_{d4}= 35°C

1.Capacity of the cooling coil in tones of refrigeration

First of all, mark the initial condition of air at 30°C dry bulb temperature and 75% relative humidity on the psychrometric chart as point 1, as shown in Fig. 2. Then mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity oil the chart as point 3.

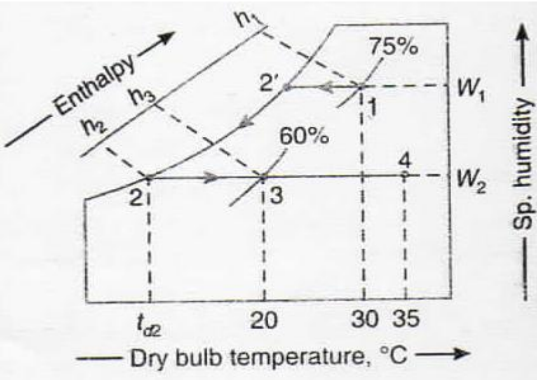


Fig.2

Now locate the points 2' and 2 on the saturation curve by drawing horizontal lines through points 1 and 3 as shown in Fig. 2. On the chart, the process 1-2' represents the sensible cooling. 2'-2 represents dehumidifying process and 2-3 represents the sensible heating process. From the psychrometric chart, we find that the specific volume of air at point 1.

v_{x1} = 0.886 m3/ kg of dry air

Enthalpy of air at point 1,

h_1 = 81.8 kJ/kg of dry air

and enthalpy of air at point 2,

h_2 = 34.2 kJ / kg of dry air

We know that mass of air absorbed by the plant,

$$m_a = \frac{v_1}{v_{s1}} = \frac{20}{0.866} = 22.6 \text{ kg / min}$$

∴ Capacity of the cooling coil

$= m_a (h_1 - h_2) = 22.6 (81.8 - 34.2) = 1075.76$ kJ / min

$= 1075.76 / 210 = 5.1$ TR Ans.

2. Capacity of the heating coil in kW

From the psychrometric chart, we find that enthalpy of air at point 3,

$h_3 = 42.6$ kJ / kg of dry air

∴ Capacity of the heating coil

$= m_a (h_1 - h_2) = 22.6 (42.6 - 34.2) = 189.84$ Id / min

$= 189.84/ 60 = 3.16$ kW Ans.

3. Amount of water removed per hour

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

$W_1 = 0.0202$ kg / kg of dry air

and specific humidity of air at point 2,

$W_2 = 0.0088$ kg / kg of dry air

∴ Amount of water removed per hour

$= m_a (W_1 - W_2) = 22.6 (0.0202 — 0.0088) = 0.258$ kg / min

$= 0.258 \times 60 = 15.48$ kg / h Ans.

4. By-pass factor of the heating coil

We know that by-pass factor,

$$BPF = \frac{t_{d4} - t_{d3}}{t_{d4} - t_{d2}} = \frac{35 - 20}{35 - 12.2} = 0.65\dot{8} \textbf{ Ans.}$$

... [From psychrometric chart, t_{d2}, = 12.2°C]

3.20 Winter Air Conditioning System

In winter air conditioning, the air is heated, which is generally -accompanied by humidification. The schematic arrangement of the system is Damper shown in Fig. 3.

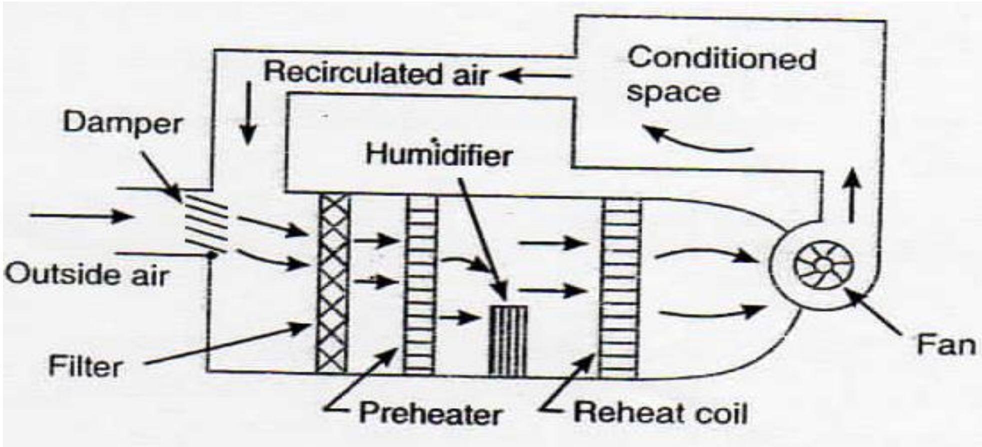


Fig.3 Winter air conditioning system

The outside air flows through a damper and mixes up with the Outside air re-circulated air (which is obtained Fan from the conditioned space). The Filter mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a preheat coil in order to prevent the possible freezing of water and to control the evaporation of water in the humidifier. After that, the air is made to pass through a reheat coil to bring the, air to the designed dry bulb temperature. Now, the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (known as re-circulated air) is again conditioned as shown in Fig.3.

The outside air is sucked and made to mix with re-circulated air, in order to make for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

3.21 Summer Air Conditioning System

It is the most important type of air conditioning, in which the air is cooled and generally dehumidified. The schematic arrangement of a typical summer air conditioning system is shown in Fig. 4.

The outside air flows through the damper, and mixes up with re-circulated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry bulb temperature of the air in the conditioned space. The cooled air passes through a perforated membrane and loses its moisture in the condensed form which is collected in a sump. After that, the air is made to pass through a heating coil which heats up the air slightly. This is done to bring the air to the designed dry bulb temperature and Damper relative humidity.

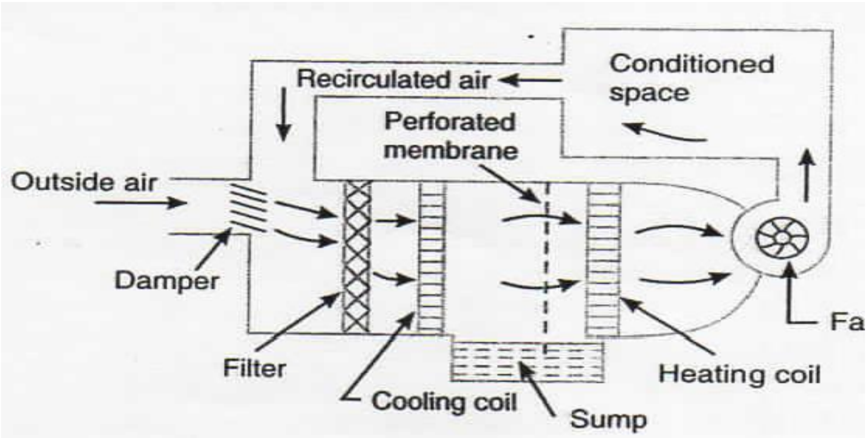


Fig 4 summer air conditioning system

Now the conditioned air is supplied to the conditioned space by a fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (known as re-circulated air) is again conditioned -as shown in Fig. 4: The outside air is sucked and made-I6 mix with the re-circulated air in order to make up for the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.

Example 3: The amount of air supplied to an air conditioned hall is 300m^3/min. The atmospheric conditions are 35°C DBT and 55% RH. The required conditions are 20°C DBT and 60% RH. Find out the sensible heat and latent heat removed from the air per minute. Also find sensible heat factor for the system.

Solution: Given v_1 = 300 m^3/min; t_{dt}, = 35°C ; $\emptyset_1$ = 55%; t_{d2} = 20°C; $\emptyset_2$ = 60%

First of all, mark the initial condition of air at 35°C dry bulb temperature and 55% relative humidity on the psychrometric chart at point 1, as shown in Fig. 5. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Locate point 3 on the chart by drawing horizontal line through point 2 and vertical line through point 1. From the psychrometric chart, we find that specific volume of air at point 1,

$$v_{s1} = 0.9 \text{ m}^3\text{/kg of dry air}$$

∴ Mass of air supplied,

Sensible heat removed from the air

From the psychrometric chart, we find that enthalpy of air at point 1,

h_1= 85.8 kJ/kg of dry air

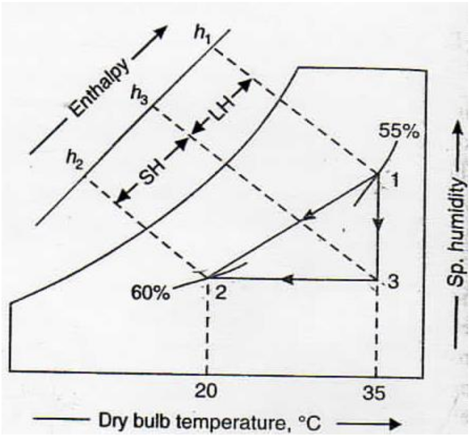


Fig.5

Enthalpy of air at point 2,

$$h_2 = 42.2 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 3,

$$h_3 = 57.4 \text{ kJ/kg of dry air}$$

We know that sensible heat removed from the air,

$$\text{SH} = m_a (h_3 - h_2)$$

$$= 333.3 (57.4 - 42.2) = 5066.2 \text{ kJ/min Ans.}$$

Latent heat removed from the air

We know that latent heat removed from the air,

$$\text{LH} = m_a (h_1 - h_3)$$

$$= 333.3 (85.8 - 57.4) = 9465.7 \text{ kJ/min Ans.}$$

Sensible heat factor for the system

We know that sensible heal factor for the system,

$$SHF = \frac{SH}{SH + LH} = \frac{5066.2}{5066.2 + 9465.7} = 0.348 \text{ Ans.}$$

Example 4: An air handling unit in an air conditioning plant supplies a total of 4500 m^3/min of dry air which comprises by mass 20% of fresh air at 40°C DBT and 27°C WBT and 8,%, re-circulated air at 25°C DBT and 50% RH. The air leaves the cooling coil at 13°C saturated. Calculate the total cooling load and room heat gain. The following data can be used:

Condition	DBT °C	WBT °C	RH %	Sp. Humidity (% of water vapour / Kg of dry air)	Enthalpy kJ/kg of dry air
Outside	**40**	**27**	**--**	**17.2**	**85**
Inside	**25**	**--**	**50**	**10.0**	**51**
ADP	**13**	**--**	**100**	**9.4**	**36.8**

Specific volume of air entering the cooling coil is 0.869 m^3*/ kg of dry air*.

Solution: Given v_3= 4500 m^3/min; t_{dt}, = 40°C; = 27°C; t_{d2}, = 25°C; $\emptyset_2$ = 50%; t_{d4}= ADP = 13°C; W^1 = 17.2 g / kg of dry air = 0.0172 kg/kg of dry air; W_2 = 10g / kg of dry air =0.01 kg/kg of dry air; W_4=9.4g/kg of dry air = 0.0094 kg/kg of dry air; h_1=85 kJ/kg of dry air; h_2 = 51 kJ/kg of dry air; h_4 = 36.8 kJ/kg of dry air ; v_{s3} = 0.869 ni3/kg of dry air.

First of all, mark the condition of fresh air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 6. Now mark the condition of re-circulated air at 25°C dry bulb temperature and 50% relative humidity as point 2. The condition of air entering the cooling coil *(point 3) is marked on the line 1-2, such that the specific volume of air at this point is 0.869 m^3/kg of dry air. The point 4 represents the condition of air leaving the cooling coil at 13°C on the saturation curve.

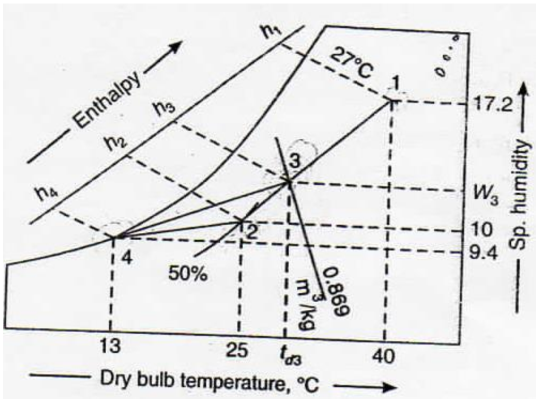


Fig.6

From the psychrometric chart, we find that enthalpy of air entering the cooling coil at point 3,

$$h_3 = 57.8 \text{ kJ / kg of dry air}$$

Specific humidity of air entering the cooling coil at point 3,

$$W_3 = 0.0116 \text{ kg / kg of dry air}$$

and dry bulb temperature of air entering the cooling coil at point 3,

$$t_{d3} = 28.3°C$$

Total cooling load

We know that mass of air entering the cooling coil,

$$m_{a3} = \frac{v_3}{v_{s3}} = \frac{4500}{0.869} = 5178 \text{ kg / min}$$

Total cooling load = m_{a3} (h_3 - h_4) = 5178 (57.8 — 36:8) = 108 738 kJ / min

= 108 738 / 210 = 517.8 TR Ans.

Room heat gain

Since the total mass of air (m_a, = 5178 kg / min) comprises 20% of fresh air, therefore mass of fresh air supplied at point 1.

$$m_{a1} = 0.2 \text{ x } 5178 = 1035.6 \text{ kg / min}$$

and fresh air load = m_{a1} ($h1 – h_2$) = 1035.6 (85 — 51) = 35 210 la / min

= 35 210 / 210 = 168 TR Ans.

∴ Room heat gain = Total cooling load Fresh air load

= 517.8 — 168 = 349.8 TR Ans.

Example 5: A conference room of 60 seating capacity is to be air conditioned for comfort conditions of 22°C dry bulb temperature and 55% relative humidity. The outdoor conditions are 32°C dry bulb temperature and 22°C wet bulb temperature. The quantity of air supplied is 0.5m^3/min/person. The comfort conditions are achieved first by chemical dehumidification and by cooling coil. Determine I. Dry bulb temperature of air at exit of dehumidifier; 2. Capacity of dehumidifier; 3. Capacity and surface temperature of cooling coil, if the by-pass factor is 0.30.

Solution: Given: Seating capacity = 60; t_{d2}= 22°C; $\emptyset_2$ = 55%; t_{dt} = 32°C; t_{wt} = 22°C; = 0.5 m^3/min / person = 0.5 x 60 = 30 m^3/min; BPF = 0.3.

First of all, mark the outdoor conditions of air re. at 32°C dry bulb temperature and 22°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.8. Now mark the required comfort conditions of air i.e. at 22°C dry bulb temperature and 55% relative humidity, as point 2. In order to find the condition of air leaving the dehumidifier, draw a constant wet bulb temperature line from point 1 and a constant specific humidity line-from point 2. Let these two lines intersect at point 3. The line 1-3 represents the chemical dehumidification and the line 3-2 represents sensible cooling.

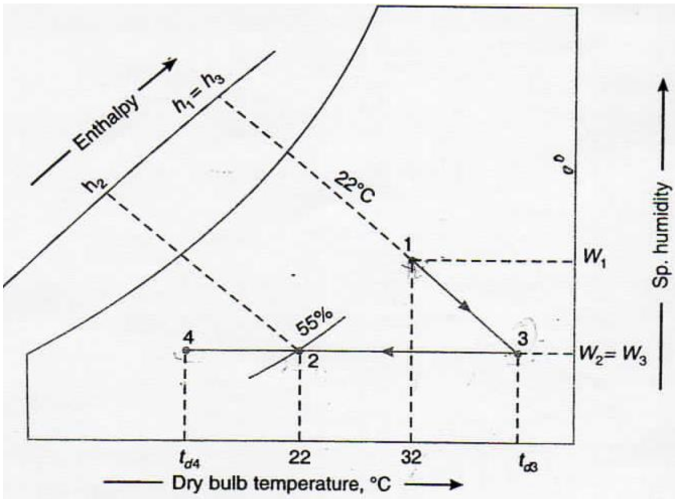


Fig.7

1. Dry bulb temperature of air at exit of dehumidifier

From the psychrometric chart, we find that dry bulb temperature of air at exit of dehumidifier i.e. at point3,

$$t_{d3} = 41°C \text{ Ans.}$$

2. Capacity of dehumidifier

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = = 64.5 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 45 \text{ kJ / kg of dry air}$$

Specific humidity of air at point 1,

$$W_1 = 0.0123 \text{ kg / kg of dry air}$$

Specific humidity of air at point 3,

$$W_3 = W_2 = 0.0084 \text{ kg/kg of dry air}$$

and specific volume of air at point 1,

$$v_{s1} = 0.881 \text{ .m3/ kg of dry air}$$

We know that mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{30}{0.881} = 34.05 \text{ kg / min}$$

∴ Capacity of the dehumidifier

$= m_a (W_1 - W_3)$

$= 34.05 (0:0123 - 0.0084) = 0.1328$ kg / min

$= 0.1328 \times 60 = 7.968$ kg / h Ans.

3. Capacity and surface temperature of cooling coil

We know that capacity of the cooling coil

$= m_a (h_3 - h_2) = 34.05 (64.5 — 45) = 664$ kJ/min

$= 664 / 210 = 3.16$TR Ans. .... ($\because$ 1TR = 210 kJ/min)

Let t_{d4} = Surface temperature of the cooling coil.

We know that by-pass factor (BPF),

$$0.3 = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}} = \frac{22 - t_{d4}}{41 - t_{d4}}$$

$$0.3 (41 - t_{d4}) = 22 - t_{d4} \quad \text{or} \quad 12.3 - 0.3\, t_{d4} = 22 - t_{d4}$$

$$t_{d4} = \frac{22 - 12.3}{0.7} = 13.86°\text{C} \ \ \text{Ans.}$$

Example 6: ***The following data refer to air conditioning of a public hall:***

Outdoor conditions = 40°C DBT, 20°C WBT
Required comfort conditions = 20°C DBT, 50% RH
Seating capacity of hall = 1000
Amount of outdoor air supplied = 0.3 m³/min/person
If the required condition is achieved first by adiabatic humidifying and then cooling, find: 1. The capacity of the cooling coil and surface temperature of the coil if the by-pass factor is 0.25; and 2. The capacity of the humidifier and its efficiency.

Solution: Given $t_{dt} = 40°C$; $t_{w1} = 20°C$; $t_{d2} = 20°C$; $\emptyset_2 = 50\%$; Seating capacity = 1000; $v_1 = 0.3$ m^3/min/person = 0.3 x 1000 = 300 m3/min; BPF = 0.25

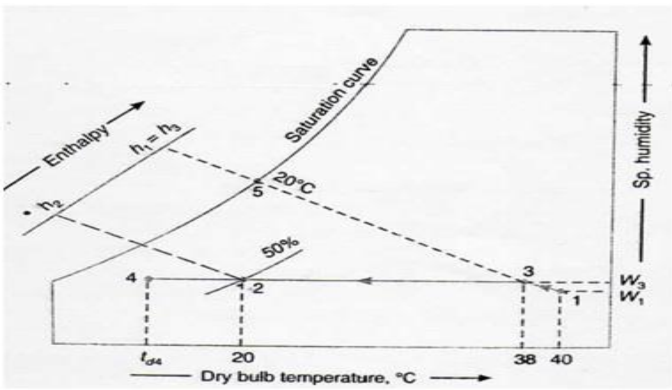


Fig.8

First of all, mark the outdoor conditions of air i.e., at 40°C dry bulb temperature and 20°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 8. Now mark the required comfort conditions of air i.e. at 20°C dry bulb temperature and 50% relative humidity, as point 2. From point 1, draw a constant wet bulb temperature line and from point 2 draw a constant specific humidity line. Let these two lines intersect at point 3. The line 1-3 represents adiabatic humidification and the line 3-2 represents sensible cooling.

From the psychrometric chart, we find that specific volume of air at point 1,

$$v_{s1}, = 0.896 \text{ m}^3/\text{ kg of dry air}$$

∴ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{300}{0.896} = 334.8 \text{ kg / min}$$

1. Capacity of the cooling coil and surface temperature of the coil

From the psychrometric chart, we find that enthalpy of air at point 3,

$$h_3 = 57.6 \text{ k)1 kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 39 \text{ / kg of dry air}$$

Dry bulb temperature of air after humidification i.e., at point 3.

$$t_{d3} = 38°\text{C}$$

We know that capacity of the cooling coil

$$= m_a (h_3 - h_2) = 334.8 (57.6 - 39) = 6227 \text{ kJ/min}$$

$$= 6227 / 210 = 29.6 \text{ TR Ans.}$$

Let t_{d4} = Surface temperature of the coil

We know that by-pass factor (BPF),

$$0.25 = \frac{t_{d2} - t_{d4}}{t_{d3} - t_{d4}} = \frac{20 - t_{d4}}{38 - t_{d4}}$$

$$0.25\,(38 - t_{d4}) = 20 - t_{d4} \quad \text{or} \quad 9.5 - 0.25\,t_{d4} = 20 - t_{d4}$$

$$t_{d4} = \frac{20 - 9.5}{0.75} = 14°\text{C Ans.}$$

2. Capacity of the humidifier and its efficiency

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

$$W_1 = 0.0064 \text{ kg / kg of dry air}$$

Specific humidity at point 3,

$$W_3 = 0:0074 \text{ kg / kg of dry air}$$

and dry bulb temperature at point 5,

$$t_{d5} = 20°C$$

We know that capacity of the humidifier

$$= m_a (W_3 - W_1) = 334.8 (0.0074 - 0.0064) = 0.3348 \text{ kg / min}$$

$$= 0.3148 \times 60 = 20.1 \text{ kg / h Ans.}$$

and efficiency of the humidifier,

$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{t_{d1} - t_{d3}}{t_{d1} - t_{d5}}$$

$$= \frac{40 - 38}{40 - 20} = 0.10 \text{ or } 10\% \text{ Ans.}$$

3.22 Year-Round Air Conditioning System

The year-round air conditioning system should have equipment for both the summer and winter air conditioning. The schematic arrangement of a modern summer year-round air conditioning system is shown in Fig.9.

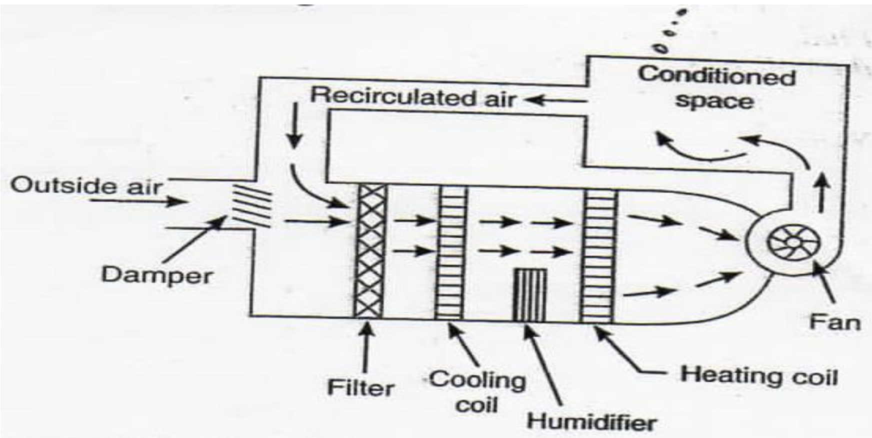


Fig.9 Year-round air conditioning system

The outside air flows through the damper and mixes up with the Damper re-circulated air (which is obtained from the conditioned space). The mixed air passes through a filter to remove dirt, dust and other impurities. In summer air conditioning, the cooling coil operates to cool the air to the desired value. The dehumidification is obtained by operating the cooling coil at a temperature lower than the dew point temperature (apparatus dew point). In winter, the cooling coil is made inoperative and the heating coil operates to heat the air. The spray type humidifier is also made use of in the dry season to humidify the air.

Example 7: *An air conditioning plant is to be designed for a small office for winter conditions with the following data:*

Outdoor conditions = 10°C DBT and 8°C WBT

Required indoor conditions = 20°C DBT and 60% RH

Amount of air circulation = 0.3 m³/min/person

Seating capacity of the office = 50 persons

The required condition is achieved first by heating and then by adiabatic humidifying. Find: I. Heating capacity of the coil in kW and the surface temperature, if the by-pass factor of the coil is 0.32 and 2. capacity of the humidifier.

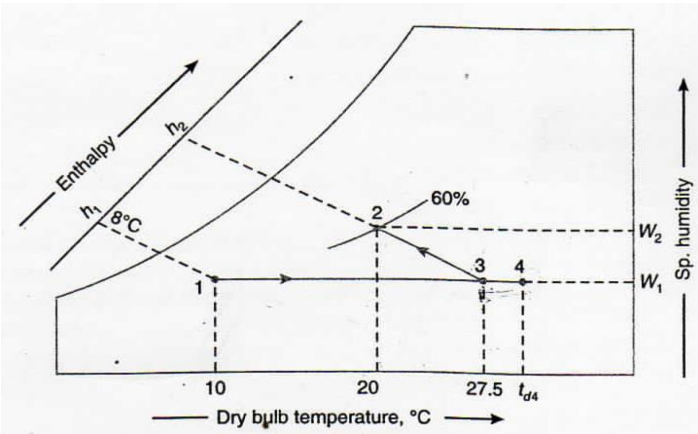


Fig.10

Solution: Given t_{dt}, = 10°C; t_{wl} = 8°C; t_{d2} = 20°C; $\emptyset_2$= 60%; seating capacity = 50 persons; v_1 = 0.3 m^3/min/person = 0.3 x 50 = 15 m^3/min; BPF = 0.32

First of all, mark the initial condition of air at 10°C dry bulb temperature and 8°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.11. Now mark the final condition of air at 20°C dry bulb temperature and 60% relative humidity on the chart as point 2. Now locate point 3 on the chart by drawing horizontal line through point I and constant enthalpy line through point 2, From the psychrometric chart, we find that the specific volume at point 1,

$$v_{sl} = 0.81 \text{ m}^3/\text{kg of dry air}$$

∴ Mass of air supplied per minute,

$$m_a = \frac{v_1}{v_{s1}} = \frac{15}{0.81} = 18.52 \text{ kg / min}$$

1. Heating capacity of the coil in kW and the surface temperature

From the psychrometric chart, we find that enthalpy at point 1,

h_1 = 24.8 kJ / kg of dry air

and enthalpy at point 2, h2= 42.6 kJ / kg of dry air

We know that heating capacity of the coil

= m_a (h_2 - h_1) = 18.52 (42.6 - 24.8) = 329.66 kJ/min

= 329.66 /60 = 5.5 kW Ans.

Let t_{d4} = Surface temperature of the coil.

We know that by-pass factor (*BPF*),

$$0.32 = \frac{t_{d4} - t_{d3}}{t_{d4} - t_{d1}} = \frac{t_{d4} - 27.5}{t_{d4} - 10}$$ ….[From psychrometric chart, t_{d3}, = 27.5°C]

or 0.32 (t_{d4} – 10) = t_{d4} – 27.5 or 0.32 t_{d4} – 3.2 = t_{d4} – 27.5

t_{d4} = 24.3 / 0.68 = 35.7°C Ans.

2. Capacity of the humidifier

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

W_1 = 0.0058 kg / kg of dry air

and specific humidity at point 2,

W_2 = 0.0088 kg / kg of dry air

We know that capacity of the humidifier,

= m_a (W_2 - W_1) = 18.52 (0.0088 — 0.0058) = 0.055 kg / min

= 0.055 x 60 = 3.3 kg / h Ans.

Example 8: *A small office hall of 25 person capacity is provided with summer air conditioning system with the following data:*

Outside conditions = 34°C DBT and 28°C WBT

Inside conditions = 24°C DBT and 50% RH

Volume of air supplied = 0.4 m3 / min / person

Sensible heat load in room = 125 600 k / h

Latent heat load in the room = 42 000 kJ / h

Find the sensible heat factor of the plant.

Solution: Given Seating capacity = 25 persons; t_{dt} = 34°C; t_{wl} = 28°C; t_{d2}, = 24°C ; $\emptyset_2$ = 50% ; v_1 = 0.4 m^3/min/person = 0.4 x 25 = 10 m3/min; S.H. load = 125 600 kJ / h; L.H. load = 42 000 kJ / h

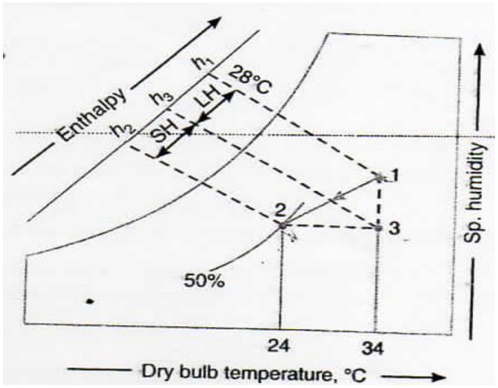


Fig.11

First of all, mark the initial condition of air at 34°C dry bulb temperature and 28°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.12. Now mark the final condition of air at 24°C dry bulb temperature and 50% relative humidity on the chart as point 2. Now locate point 3 on the chart by drawing horizontal line through point 2 and vertical line through point 1. From the psychrometric chart, we find that specific volume at point 1,

v_{s1} = 0.9 m3/ kg of dry air

Enthalpy of air at point 1,

h_1 = 90 kJ / kg of dry air

Enthalpy of air at point 2,

h_2 = 48 ld / kg of dry air

and enthalpy of air at point 3,

h_3 = 58 / kg of dry air

We know that mass of air supplied per min,

$$m_a = \frac{v_1}{v_{s1}} = \frac{10}{0.9} = 11.1 \text{ kg / min}$$

and sensible heat removed from the air

$$= m_a (h_3 - h_2) = 11.1(58 - 48) = 111 \text{ kJ / min}$$

$$= 111 \text{ x } 60 = 6660 \text{ kJ/h}$$

Total sensible heat of the room,

$$SH = 6660 + 125\,600 = 132\,260 \text{ kJ / h}$$

We know that latent heat removed from the air

$$= m_a (h_1 - h_3) = 11.1(90 - 58) = 355 \text{ kJ / min}$$

$$= 355 \text{ x } 60 = 21\,300 \text{ kJ / h}$$

∴ Total latent heat of the room,

$$LH = 21\,300 + 42000 = 63300 \text{ kJ / h}$$

We know that sensible heat factor,

$$SHF = \frac{SH}{SH + LH} = \frac{132\,260}{132\,260 + 63\,300} = 0.676 \text{ Ans.}$$

Example 9: A restaurant with a capacity of 100 persons is to be air-conditioned with the following conditions:

Outside conditions	: 30°C DBT and 70% RH
Desired inside conditions	: 23°C DBT and 55% RH
Quantity of air supplied	: 0.5 rn3 / min / person

The desired conditions are achieved by cooling, dehumidifying and then heating. Determine: I. Capacity of cooling coil in tones of refrigeration; 2. Capacity of heating coil; 3. Amount of water removed by dehumidifier; and 4. By-pass factor of the heating coil if its surface temperature is 35°C.

Solution: Given: Number of persons = 100; t_{dt} = 30°C; $\emptyset_1$ = 70%; t_{d4} = 23°C; $\emptyset_4$ = 55% ; v_l = 0.5 m^3/ min / person = 0.5 x 100 = 50 m^3/train

First of all, mark the outside conditions of air at 30°C dry bulb temperature and 70% relative humidity on the psychrometric chart as point 1, as shown in Fig. 18.13. Now mark the desired inside conditions of air at 23°C dry bulb temperature and 55% relative humidity on the chart as point 4. The process 1-2 represents the sensible cooling, process 2-3 represents dehumidification and the process 3-4 represents the sensible heating.

From the psychrometric chart, we find that the specific volume at point 1.

$$v_{s1} = 0.885 \text{ m}^3\text{/kg of dry air}$$

∴ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{50}{0.885} = 56.5 \text{ kg / min}$$

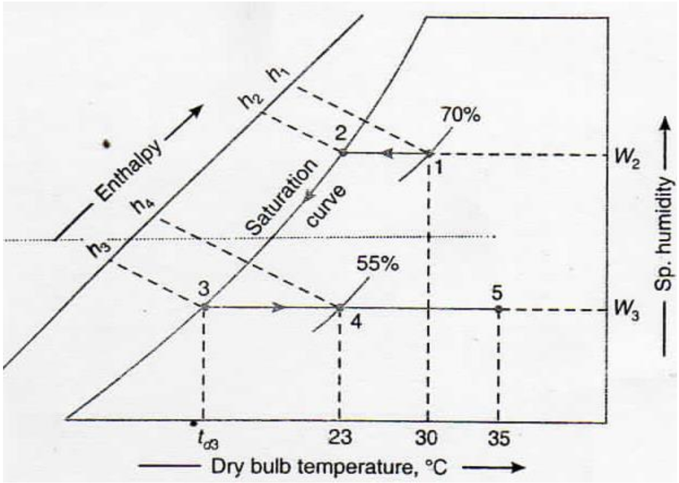


Fig.12

1. Capacity of cooling coil in tones of refrigeration

From the psychrometric chart, we find that enthalpy of air at point 1,

h_1= 78.5 kJ/kg of dry air

Enthalpy of air at point 3,

h_3 = 37.8 kJ/kg of dry air

∴ Capacity of the cooling coil

= m_a (h_1- h_3) = 56.5 (78.5 — 37.8) = 2300 kJ/min

= 2300 / 210 = 10.95 TR Ans.

2. Capacity of heating coil

From the psychrometric chart, we find that enthalpy of air at point 4,

h_4 = 47.6 kJ / kg of dry air

∴ Capacity of the heating coil

= m_a (h_4 – h_3) = 56.5 (47.6 — 37.8) = 554 kJ/min

= 554 / 60 = 9.23 kW Ans.

3. Amount of water removed by dehumidifier

From the psychrometric chart, we find that specific humidity at point 2,

$$W_2 = 0.0188 \text{ kg / kg of dry air}$$

and specific humidity at point 3,

$$W_3 = 0.0095 \text{ kg / kg of dry air}$$

∴ Amount of water removed by dehumidifier

$$= m_a (W_2 - W_3) = 56.5\,(0.0188 - 0.0095) = 0.525 \text{ kg I min}$$

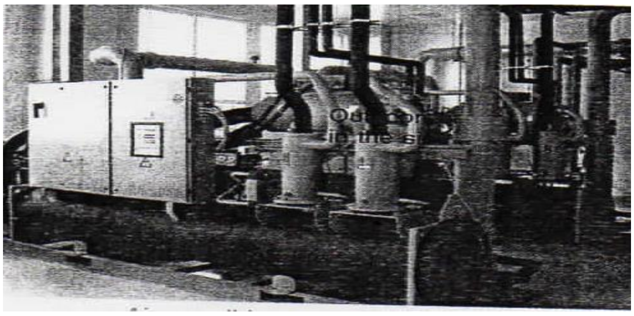
$$= 0.525 \times 60 = 31.5 \text{ kg / h} \quad \text{Ans. 4.}$$

4. By-pass factor of the heating coil

Let t_{d5} = Surface temperature of the heating coil = 35°C …..(Given)

From the psychrometric chart, we find that dry bulb temperature at point 3,

$$t_{d3} = 13.5°\text{C}$$



Air conditioning system

We know that by-pass factor of the heating coil,

$$BPF = \frac{t_{d5} - t_{d4}}{t_{d5} - t_{d3}} = \frac{35 - 23}{35 - 13.5} = 0.558 \text{ Ans.}$$

Room Sensible Heat Factor

It is defined as the ratio of the room sensible heat to the room total heat. Mathematically, room sensible heat factor,

$$RSHF = \frac{RSH}{RTH} = \frac{RSH}{RSH + RLH}$$

where RSH = Room sensible heat,

RLH = Room latent heat, and

RTH = Room total heat.

The conditioned air supplied to the room must have the capacity to take up simultaneously both the room sensible heat and room latent heat loads. The point S on the psychrometric chart, as shown in Fig. 18.14, represents the supply air condition and the point R represents the required final condition in the room (i.e., room design condition). The line SR is called the room sensible heat factor line (RSHF line). The slope of this line gives the ratio of the room sensible heat (RSH) to the room latent heat (RLH). Thus the supply air having its conditions given by any point on this line will satisfy the requirements of the room with adequate supply of such air. In other words, the supply air having conditions marked by points S_1, S_2, S_3, S_4 etc., will satisfy the requirement but the quantity of air supplied will be different for different supply air points. The supply condition at S requires minimum air and at point 54, it is maximum of all the four points.

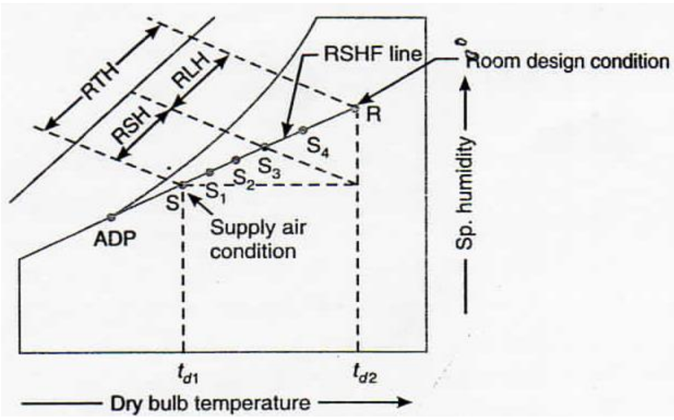


Fig. 13. Representation of supply air condition and room design condition.

When the supply air conditions are not known, which in fact is generally required to be found out, the room sensible heat factor line may be drawn from the calculated value of room sensible heat factor (RSHF), as discussed below:

1. Mark point a on the sensible heat factor scale given on the right hand corner of the psychrometric chart as shown-in Fig.14. The point a represents the calculated value of *RSHF*.

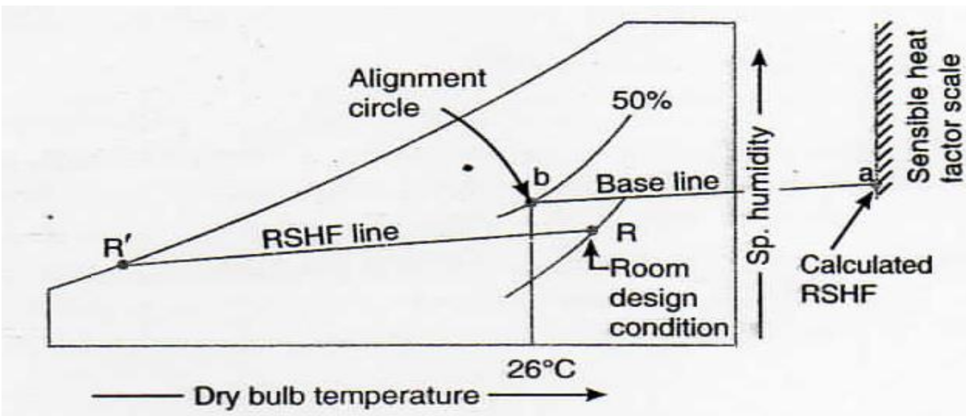


Fig. 14 Room sensible heat factor (RSHF) line.

2. Join point *a* with the *alignment circle or the reference point *b*. The line *ab* is called base line.

3. Mark point *R* on the psychrometric chart to represent the room design conditions.

4. Through point *R* draw a line *RR*' parallel to the base line *ab*. This line is the required room sensible

heat factor line.

Note: In a cooling and dehumidification process, the temperature at which the room sensible heat factor line intersects the saturation curve is called *room apparatus dew point* (*ADP*).

3.23 Grand Sensible Heat Factor

It is defined as the ratio of the total sensible heat to the grand total heat which the cooling coil or the conditioning apparatus is required to handle. Mathematically, grand sensible heat factor,

$$GSHF = \frac{TSH}{GTH} = \frac{TSH}{TSH + TLH} = \frac{RSH + OASH}{(RSH + OASH) + (RLH + OALH)}$$

where TSH = Total sensible heat = RSH + OASH

TLH = Total latent heat = RLH + OALH

GTH = Grand total heat = TSH + TLH = RSH + RLH + OATH

= RSH + RLH + (DASH + OALH)

Let v_1 = Volume of outside air or ventilation in m^3/min,

t_{d1} = Dry bulb temperature of outside air in °C,

W_1 = Specific humidity of outside air in kg / kg of dry air,

h_i = Enthalpy of outside air in kJ / kg of dry air,

t_{d2} = Dry bulb temperature of room air in °C,

W_2 = Specific humidity of room air in kg / kg of dry air, and

h_2 = Enthalpy of room air in kJ / kg of dry air.

∴ Outside air sensible heat,

$$OASH = 0.020\,44\, v_1 (t_{d1} - t_{d2}) \text{ kW}$$

Outside air latent heat,

$$OALH = 50\ v_1\ (W_1 - W_2)\ \text{kW}$$

and outside air total heat,

$$OATH = OASH + OALH$$

The outside air total heat may also be calculated from the following relation:

$$\text{OATH} = 0.02\ v_1(h_1 - h_2)\ \text{kW}$$

Generally, the air supplied to the air conditioning plant is a mixture of fresh air (or outside air or ventilation) and the re-circulated air having the properties of room air. On the psychrometric chart, as shown in Fig. 15, the point 1 represents the outside condition of air, the point 2 represents the• room air condition and the point 3 represents the mixture condition of air entering the cooling coil. When the mixture condition enters the cooling coil or conditioning apparatus, it is cooled and dehumidified. The point 4 shows the supply air or leaving condition of air from the cooling coil or conditioning apparatus. When the point 3 is joined with the point 4, it gives a grand sensible heat factor line (*GSHF* line) as shown in Fig. 5. This line, when produced up to the saturation curve, gives apparatus dew point (*ADP*).

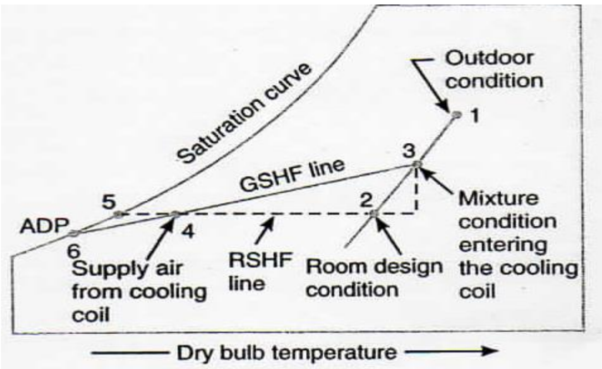


Fig.15 Grand sensible heat factor

If the mixture condition entering the cooling coil or conditioning apparatus and the grand sensible heat factor (*GSHF*) are known, then the *GSHF* line may be drawn on the psychrometric chart in the similar way as discussed for RSHF line. The point 4, as shown in Fig. 15, is the intersection of *GSHF* line and *RSHF* line. This point gives the ideal conditions for supply air to the room.

3.24 Effective Room Sensible Heat Factor

It is defined as the ratio of the effective room sensible heat to the effective room total heat. Mathematically, effective room sensible heat factor,

$$ERSHF = \frac{ERSH}{ERTH} = \frac{ERSH}{ERSH + ERLH}$$

Where $ERSH$ = Effective room sensible heat = $RSH + OASH \times BPF$

$= RSH + 0.020\,44\, v_1\, (t_{dt} - t_{d2})\, BPF$

$ERLH$ = Effective room latent heat = $RLH + OALH \times BPF$

$= RLH + 50\, v_1\, (W_1 - W_2)\, BPF$

$ERTH$ = Effective room total heat = $ERSH + ERLH$

BPF = By-pass factor

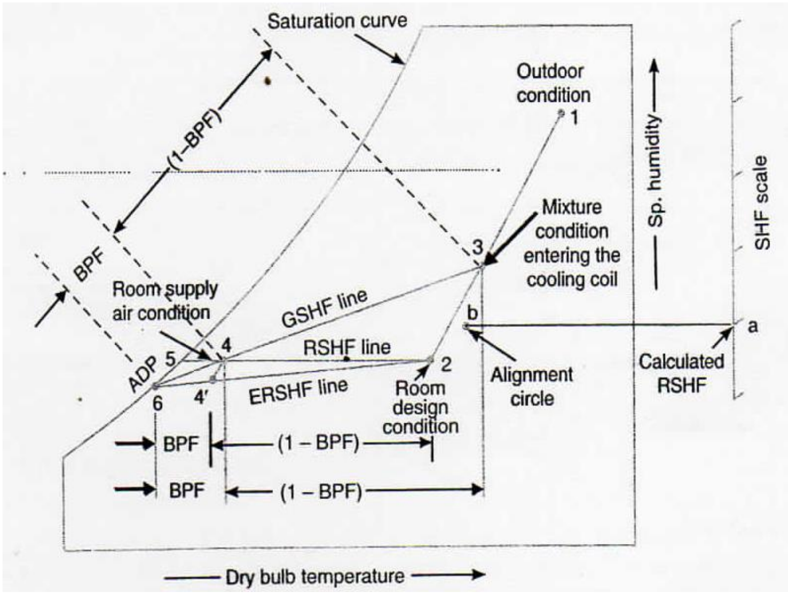


Fig. 16. Effective room sensible heat factor.

The line joining the point 2 and point 6 i.e. ADP, as shown in Fig. 18.17, gives the effective room sensible heat factor line (*ERSHF* line). From point 4, draw 4.4' parallel to 3-2. Therefore from similar triangles 6-4-4' and 6-3-2,

$$BPF = \frac{\text{Length 4-6}}{\text{Length 3-6}} = \frac{\text{Length 4}'\text{-6}}{\text{Length 2-6}}$$

The by-pass factor is also given by,

$$BPF = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4}' - ADP}{t_{d2} - ADP}$$

Notes:1. The effective room sensible heat (*ERSH*), effective room latent heat (*ERLH*) and effective room total heat (ERTH) may also be obtained from the following relations:

$ERSH = 0.020\,44\, v_d (t_{d2} - ADP)\, (1 - BPF)$ kW

$ERLH = 50\, v_d\, (W_2 - W_{ADP})\, (1 - BPF)$ kW

and $ERTH = 0.02\, v_d (h_2 - h_{ADP})\, (1 - BPF)$ kW

where v_d = Volume of dehumidified air to room or space in m^3/min, *ADP* = Apparatus dew point in °C,

W_{ADP} = Specific humidity at apparatus dew point in kJ / kg of dry air, and

h_{ADP} = Enthalpy at apparatus dew point in kJ / kg of dry air.

2. The mass of dehumidified air is given by

$$m_d = \frac{\text{Room total heat}}{h_2 - h_4}$$

where h_2 = Enthalpy of air at room condition, and

h_4 = Enthalpy of supply air to room from the cooling coil.

Example 10: *In an air conditioning system, the inside and outside conditions are dry bulb temperature 25°C, relative humidity 50% and dry bulb temperature 40°C, wet bulb temperature 27°C respectively. The room sensible heat factor is 0.8. 50% of the room air is rejected to atmosphere and an equal quantity of fresh air added before air enters the air conditioning apparatus. If the fresh air added is 100 m³/min, determine:*

1. Room sensible and latent heat load;

2. Sensible and latent heat load due to fresh air;

3. Apparatus dew point;

4. Humidity ratio and dry bulb temperature of air entering air conditioning apparatus.

Assume by-pass factor as zero, density of air as 1.2 kg / m³ at a total pressure of 1.01325 bar. **Solution:** Given t_{dt} = 40°C; t_{w1}, = 27°C; t_{d2}= 25°C; $\emptyset_2$ = 50% ; RSHF= 0.8; v_1 = 100 m³/min; ρ_a =1.2 kg/m³

The flow diagram for the air conditioning system is shown in Fig. 17, and it is represented on the psychrometric chart as discussed below:

First of all, mark the outside condition of air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18. Now mark the inside condition of air at 25°C dry bulb temperature and 50% relative humidity as point 2. Since 50% of the room air and 50% of fresh air is added before entering the air conditioning apparatus, therefore mark point 3 on the line 1-2 such that

$$\text{Length 2-3} = \frac{\text{Length 1-2}}{2}$$

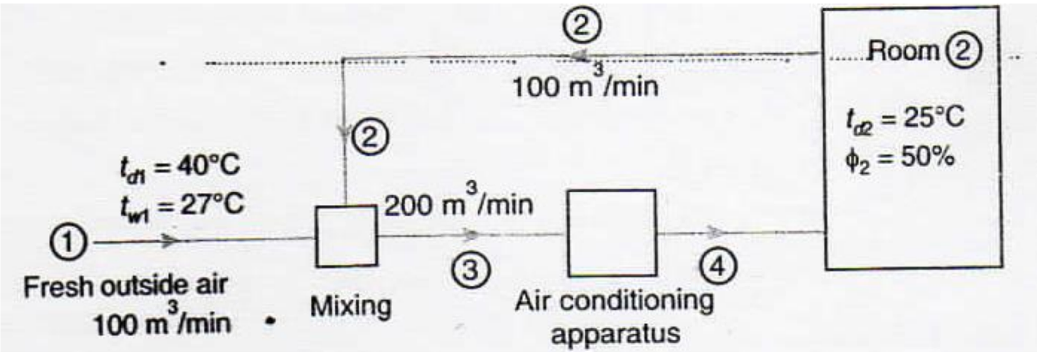


Fig.17

Now mark the given value of RSHF (i.e. 0.8) on the room sensible heat factor scale and join this with the alignment circle (i.e. 26°C DBT and 50% RH). From point 2, draw a line 2-4 parallel to this line. This line is called RSHF line. The point 4 represents the apparatus dew point (*ADP*). From the psychrometric chart, we find the enthalpy of air at point 1,

$$h_1 = 85.2 \text{ kJ / kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 50 \text{ kJ / kg of dry air}$$

and enthalpy of air at point 4,

$$h_4 = 33 \text{ kJ / kg of dry air}$$

1. Room sensible and latent heat load

We know that mass of air supplied to the room,

$m_a = v_3 \times \rho_a = (100 + 100)1.2 = 240$ kg/min

∴ Room sensible heat load,

$$\text{RSH} = m_a C_{pm} (t_{d2} - t_{d4})$$

$$= 240 \times 1.022 (25 - 11.8) = 3238 \text{ kJ / min}$$

$$= 3238/60 = 53.96 \text{ kJ/s or kW}$$ … [∵ From psychrometric chart, $t_{d4} = 11.8$°C]

and room total heat load,

$$RTH = m_a (h_2 - h_4) = 240 (50 - 33) = 4080 \text{ kJ/min}$$

$$= 4080/60 = 68 \text{ kJ/s or kW}$$

∴ Room latent heat load,

RLH = RTH - RSH

= 68 — 53.96 = 14.04 kW

2. Sensible and latent heat load due to fresh air

We know that mass of fresh air supplied,

$m_F = v_1 \times \rho_a = 100 \times 1.2 = 120$ kg / min

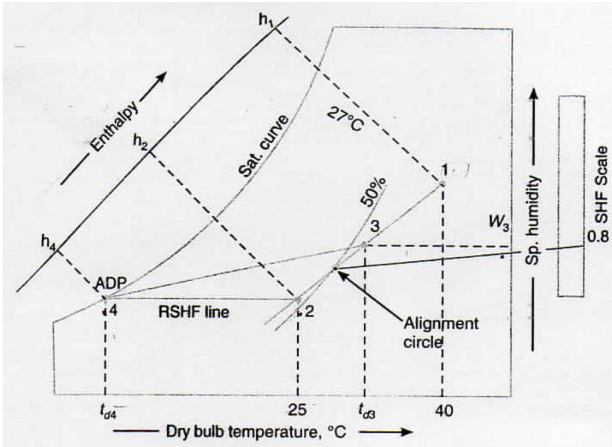


Fig.18

∴ Sensible heat load due to fresh air

$= m_F C_{pm} (t_{d1} - t_{d2})$

= 120 x 1.022 (40 - 25) = 1840 kJ/min

= 1840 / 60 = 30.67 kJ/s or kW Ans.

and total heat load due to fresh air

$= m_F (h_1 - h_2)$ = 120 (85.2 - 50) = 4224 kJ/min

= 4224 / 60 = 70.4 kJ/s or kW

∴Latent heat load due to fresh air

= Total heat load — Sensible heat load

= 70.4 — 30.67 = 39.73 kW Ans.

3. Apparatus dew point

From the psychrometric chart, we find that apparatus dew point (*ADP*) corresponding to point 4 is

$$t_{d4} = 11.8°C \text{ Ans.}$$

4. Humidity ratio and dry bulb temperature of air entering air conditioning apparatus

The air entering the air conditioning apparatus is represented by point 3 on the psychrometric chart as shown in Fig. 18. From the psychrometric chart, we find that humidity ratio corresponding to point 3,

$$W_3 = 0.0138 \text{ kg / kg of dry air Ans.}$$

and dry bulb temperature corresponding to point 3.

$$t_{d4} = 32.5°C \text{ Ans.}$$

Example 11: *An air conditioned space is maintained at 27°C dry bulb temperature and 40% relative humidity. The ambient conditions are 40°C dry bulb temperature and 27°C wet bulb temperature. The space has a sensible heat gain of 14kW. The air is supplied to the space at 7°C saturated. Calculate:*

1. Mass of moist air supplied to the space in kg / h; 2. Latent heat gain of space in kW; and 3. Cooling load of air-washer in kW if 30 per cent of air supplied to the space is fresh, the remainder being re-circulated.

Solution: Given t_{d2}=.27°C; $\emptyset_2$ = 50%; t_{dt} = 40°C; t_{w1} = 27°C; Qs = 14 kW = 14 kJ/s = 14x3600 kJ/ h; t_{d4} = 7°C

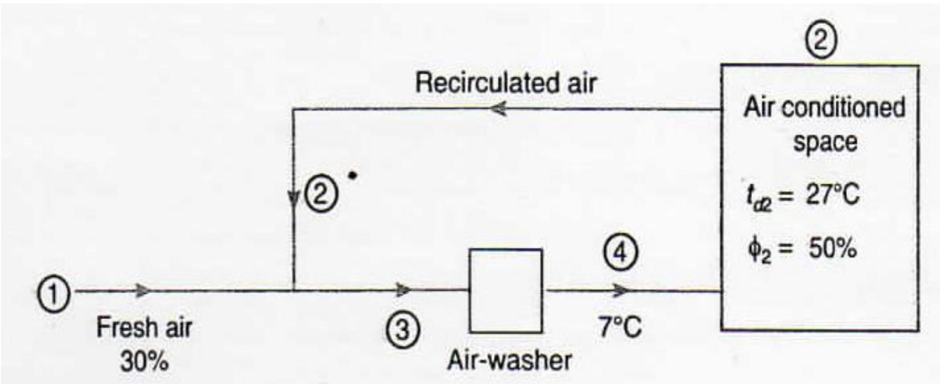


Fig. 19

The line diagram for the air conditioned space is shown in Fig. 19 and it is represented on the psychrometric chart as discussed below:

First of all, mark the ambient (outside) conditions of air at 40°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 20. Now mark the inside conditions of the space at 27°C dry bulb temperature and 50% relative humidity. Since the air is supplied to the space at 7°C saturated, therefore mark point 4 on the saturation curve at 7°C. Also 30 per cent of air supplied to the space (i.e. at point 2) is fresh, therefore mark point 3 on the line 2-1, such that

Length 2-3 = 0.3 x Length 2- I

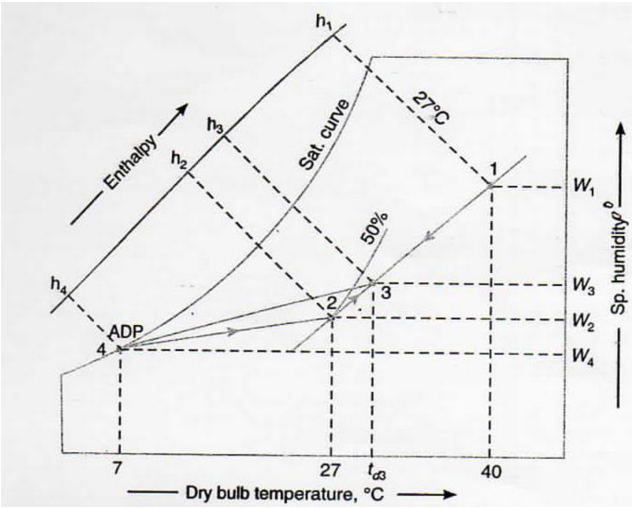


Fig.20

Now from the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 85 \text{ kJ / kg of dry air}$$

Specific humidity of air at point 1,

$$W_1 = 0.0172 \text{ kg / kg of dry air}$$

Enthalpy of air at point 2,

$$h_2 = 56 \text{ kJ / kg of dry air}$$

Specific humidity of air at point 2,

$$W_2 = 0.0112 \text{ kg / kg of dry air}$$

Enthalpy of air at point 4,

$$h_4 = 23 \text{ kJ / kg of dry air}$$

Specific humidity of air at point 4,

$$W_4 = 0.0062 \text{ kg / kg of dry air}$$

1. Mass of moist air supplied to the space in kg / h

We know that mass of dry air supplied to the space,

$$m_a = \frac{Q_S}{c_{pm}(t_{d2} - t_{d4})} = \frac{14 \times 3600}{1.022\,(27-7)} = 2465.75 \text{ kg / h}$$

…. [$\because$ C_{pm} = Humid specific heat = 1.022 kJ / kg K]

∴ Mass of moist air supplied to the space ,

$= ma (1 + W_4) = 2465.75 (1 + 0.0062)$

$= 2481$ kg / h Ans.

2. Latent heat gain of space in kW

We know that latent heat gain of space,

$Q_L = m_a (W_2 - W_4) h_{fg}$

$= 2465.75 (0.0112 — 0.0062) 2500 = 308\ 22$ kJ/h

$= 308\ 22 / 3600 = 8.56$ kJ/s or kW Ans.

3. Cooling load of air-washer in kW

From the psychometric chart, we find that dry bulb temperature of air at point 3,

$*t_{d3} = 31°C$

and enthalpy of air at point 3,

$h_3 = 64.6$ kJ / kg of dry air

We know that cooling load of air-washer

$= m_a (h_3 - h_4) = 2465.75 (64.6 - 23) = 102\ 575$ kJ/h

$= 102\ 575 / 3600 = 28.5$ k-J/s or kW Ans

Example 12: Air flowing at the rate of 100m^3/min at 40°C dry bulb temperature and 50% relative humidity is mixed with another stream flowing at the rate of 20m^3/min at 26°C dry: bulb temperature and 50% relative humidity. The mixture flows over a cooling coil whose apparatus dew point temperature is 10°C and by-pass factor is 0.2. Find dry bulb temperature and relative humidity of air leaving the coil. If this air is supplied to an air-conditioned room where dry bulb temperature of 26°C and relative humidity of 50% are maintained, estimate 1. Room sensible heat factor; and 2. Cooling load capacity of the coil in tones of refrigeration

Solution: Given v_1 =100 m^3/min; t_{dt} = 40°C; $\emptyset_1$ = 50%; v_2 = 20 m^3/min ; t_{d2} = 26°C ; $\emptyset_2$ = 50% ; *ADP* =10°C ; *BPF* = 0.2

The flow diagram for an air-conditioned room is shown in Fig 21 and it is represented on the psychrometric chart as discussed below:

First of all, mark the initial condition of air at 40°C dry bulb temperature and 50% relative humidity on the psychrometric chart as point 1, as shown in Fig. 22. Now mark the room condition of air at 26°C dry bulb temperature and 50% relative humidity as point 2. From the psychrometric chart, we find that enthalpy of air at point 1,

h_1 = 99.8 kJ / kg of dry air

Enthalpy of air at point 2,

h_2 = 53.5 Id / kg of dry air

Specific volume of air at point 1,

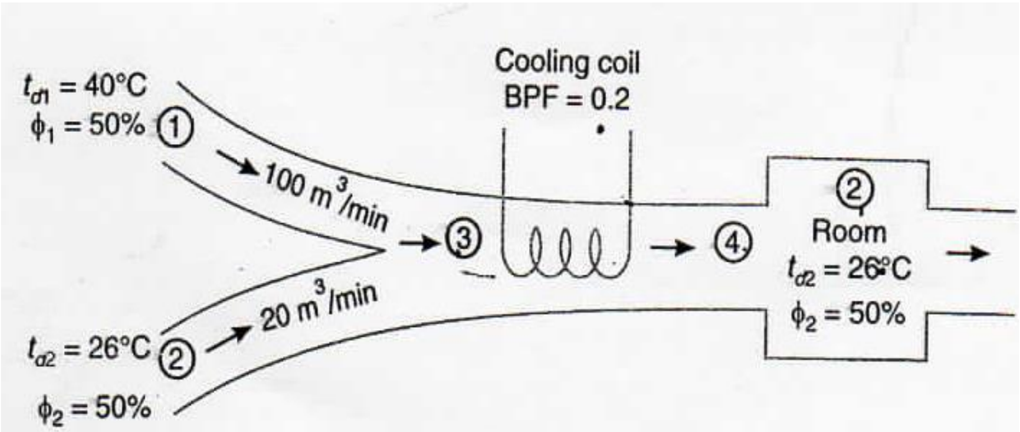
v_{s1}= 0.92 m3/ kg of dry air

and specific volume of air at point 2,

v_{s2} = 0.862 m^3/ kg of dry air

We know that mass of air supplied at point 1,

$$m_{a1} = \frac{v_1}{v_{s1}} = \frac{100}{0.92} = 108.7 \text{ kg / min}$$



Fig,21

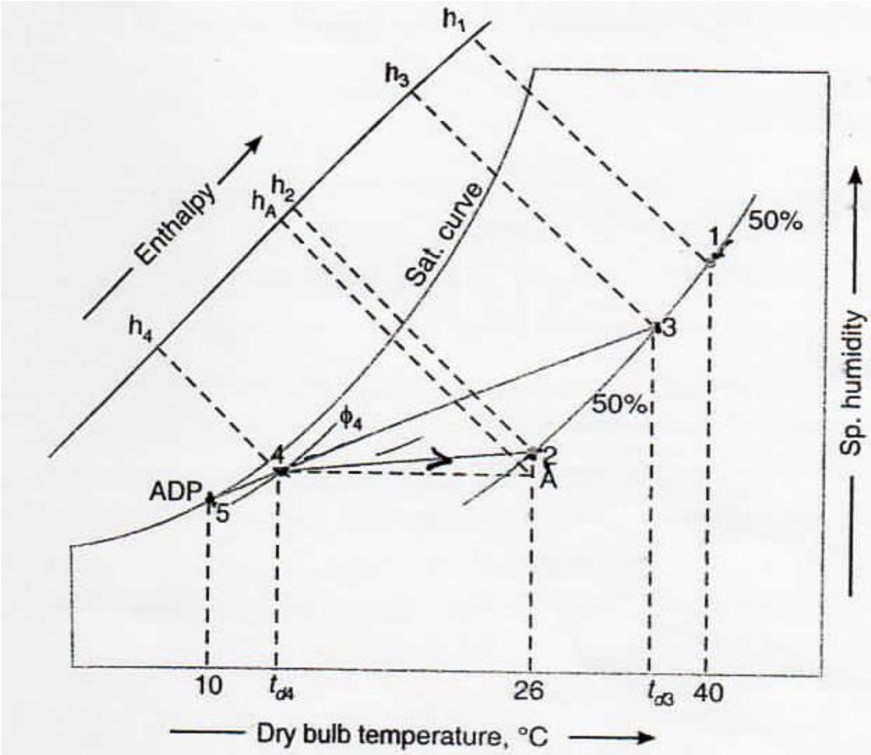


Fig.22

and mass of air supplied at point 2,

$$m_{a2} = \frac{v_2}{v_{s2}} = \frac{20}{0.862} = 23.2 \text{ kg / min}$$

∴ Mass of air flowing through the cooling coil at point 3.

$$m_{a3} = m_{al} + m_{a2} = 1083 + 212 = 131.9 \text{ kg / min}$$

For the energy balance,

$$m_{a1}h_1 + m_{a2}\, h_2 = m_{a3}\, h_3$$

$$\therefore \qquad h_3 = \frac{m_{a1}\, h_1 + m_{a2}\, h_2}{m_{a3}} = \frac{108.7 \times 99.8 + 23.2 \times 53.5}{131.9}$$

$$= 91.65 \text{ kJ / kg of dry air}$$

Now draw a constant enthalpy line corresponding to h3 = 91.65 k7 / kg of dry air which intersects the line 1-2 at point 3. From the psychrometric chart, we find that dry bulb temperature of air entering the cooling coil at point 3 is

$$t_{d3} = 37.6°C$$

Mark point 5 on the saturation curve such that ADP = 10°C, and draw a line 3-5. The point 4 lies on this line.

Dry bulb temperature and relative humidity of air leaving the coil

Let t_{d4} = Dry bulb temperature of air leaving the coil.

We know that by-pass factor (BPF),

$$0.2 = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4} - 10}{37.6 - 10}$$

$\therefore t_{d4} = 15.52°C$ Ans.

From the psychrometric chart, we find that relative humidity of air leaving the coil at point 4 is

$\emptyset_4 = 92\%$ Ans.

1. Room sensible heat factor

From the psychrometric chart, we find that enthalpy of air at point 4, h

$h_4 = 42$ kJ / kg of dry air.

and enthalpy of air at point A (which is the intersection of horizontal line from point 4 and vertical line from point 2),

$h_A = 52.5$ kJ / kg of dry air

We know that room sensible heat factor,

$$RSHF = \frac{h_A - h_4}{h_2 - h_4} = \frac{52.5 - 42}{53.5 - 42} = 0.913$$ Ans

2. Cooling load capacity of the coil

We know that cooling load capacity of the coil

$= m_{a3}(h_3 - h_4) = 131.9\ (91.65 - 42) = 6548.8$ kJ/min

$= 6548.8 / 210 = 31.185$ TR Ans.

… ($\because$ 1 TR= 210 1J/min)

Example 13: *An air conditioned auditorium is to be maintained at 27°C dry bulb temperature and 60 % relative humidity. The ambient condition is 40°C dry bulb temperature and 30°C wet bulb temperature. The total sensible heat load is 100 000 kJ/h and the total latent heat load is 40 000 kJ/h. 60% of the return air is re-circulated and mixed with 40% of make-up air after the cooling coil. The condition of air leaving the cooling coil is at 18°C.*

Determine: 1. Room sensible heat factor; 2. The condition of air entering the auditorium; 3. The amount of make-up air; 4. Apparatus dew point; and 5. By-pass factor of the cooling coil. Show the processes on the psychrometric chart.

Solution: Given t_{d4} =27°C; $\emptyset_4$ = 60%; t_{dt}, = 40°C; t_{w1}= 30°C; RSH= 100 000kJ / h; RLH = 40 000 kJ / h; t_{d2} = 18°C

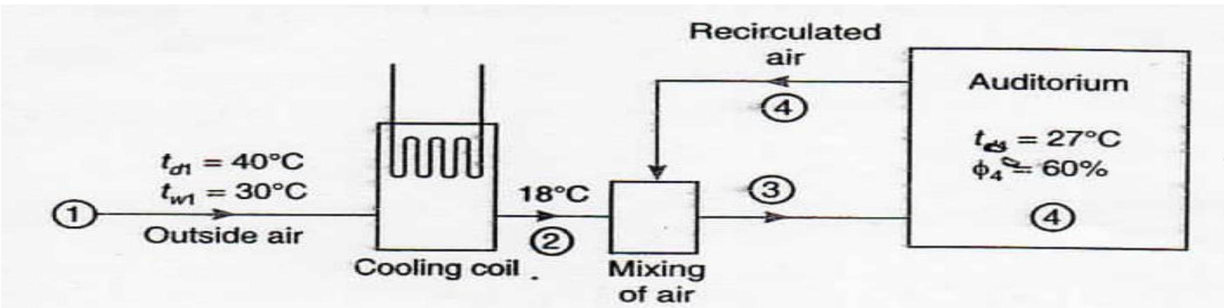
1. Room sensible heat factor

We know that room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{100\,000}{100\,000 + 40\,000} = 0.714 \text{ Ans.}$$

2. Condition of air entering the auditorium

The line diagram for processes involved in the air conditioning of an auditorium is shown in Fig. 23. These processes are shown on the psychrometric chart as discussed below



:

Fig.23

First of all, mark the ambient condition of air (outside air) i.e. at 40°C dry bulb temperature and 30°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.26. Now mark the condition of air in the auditorium, i.e. at 27°C dry bulb temperature and 60% relative humidity, as point 4.

Mark the calculated value of *RSHF* = 0.714 on the sensible heat factor scale as point *a* and join with point b which is the alignment circle (i.e. 26°C *DBT* and 50% *RH*) as shown in Fig. 24. Now from point 4, draw a line 4-5 (known as *RSHF* line) parallel to the line *ab*. Since the condition of air leaving the cooling coil is at 18°C, therefore, mark point 2 such that t_{d2}= 18°C. Join points 1 and 2 and produce up to point 6 on the saturation curve. The line 1-2-6 is the *GSHF* line. It is given that 60% of the air from the auditorium is re-circulated and mixed with 40% of the make-up air after the cooling coil. The mixing condition of air is shown at point such that

$$\frac{\text{Length } 2-3}{\text{Length } 2-4} = 0.6$$

The condition of air entering the auditorium is given by point 3. From the psychrometric chart, we find that at point 3,

Dry bulb temperature, t_{d3} = 23°C Ans.

Wet bulb temperature, t_{w3}= 19.5° C Ans.

and relative humidity, $\emptyset_3$ = 72% Ans.

3. Amount of make-up air

From the psychrometric chart, we find that enthalpy of air at point 4,

h_4 = 61 kJ/kg of dry air

and enthalpy of air at point 3, h_3 = 56 kJ/kg of dry air

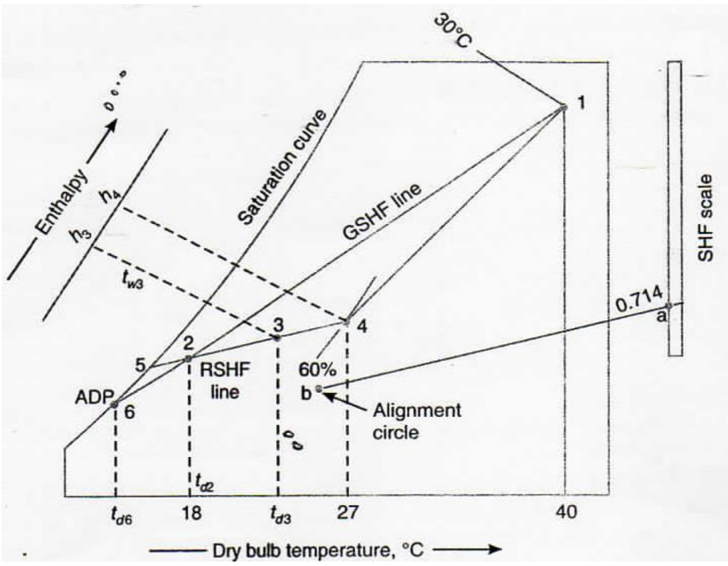


Fig.24

We know that mass of supply air to the auditorium,

$$m_S = \frac{\text{Room total heat}}{h_4 - h_3} = \frac{RSH + RLH}{h_4 - h_3}$$

$$= \frac{100\,000 + 40\,000}{61 - 56} = 28\,000 \text{ kg / h}$$

Since the make-up air is 40% of supply air, therefore mass of make-up air

= 0.4 x 28 000= 11 200 kg / h Ans.

4. Apparatus dew point

From the psychromettic chart, we find that the apparatus dew point of the cooling coil at point 6 is

ADP = *td6*= 13° C Ans. 3

5. By-pass factor .of the coaling coil

We know that by-pass factor of the cooling coil,

$$BPF = \frac{t_{d2} - ADP}{t_{d1} - ADP} = \frac{18 - 13}{40 - 13} = \frac{5}{27} = 0.185 \text{ Ans.}$$

Example 14: *An air conditioned hall is to be maintained at 27°C dry bulb temperature and 21°C wet bulb temperature. It has a sensible heat load of 46.5 kW and latent heat load of 17.5 kW. The air supplied from outside atmosphere at 38°C dry bulb temperature and 27°C wet bulb temperature is 25m^3/min, directly into the room through ventilation and infiltration. Outside air to be conditioned is passed through the cooling coil whose apparatus dew point is 15°C. The quantity of re-circulated air from the hall is 60%. This quantity is mixed with the conditioned air after the cooling coil. Determine : 1. condition of air after the coil and before the re-circulated air mixes with it; 2. condition of air entering the hall, i.e. after mixing with re-circulated air; 3. mass of fresh air entering the cooler; 4. by-pass factor of the cooling coil: and 5. refrigerating load on the cooling coil.*

Solution: Given t_{d4} = 27°C; t_{w4} = 21°C; Q_{s4} = 46.5 kW; Q_{L4} = 17.5 kW; t_{dt}, = 38°C; t_{w1}= 27°C ; v_1 = 25m^3/min; ADP = 15°C

The line diagram for the processes involved in the air conditioning of a hall is shown in Fig. 18.27. These processes are shown on the psychrometric chart as discussed below:

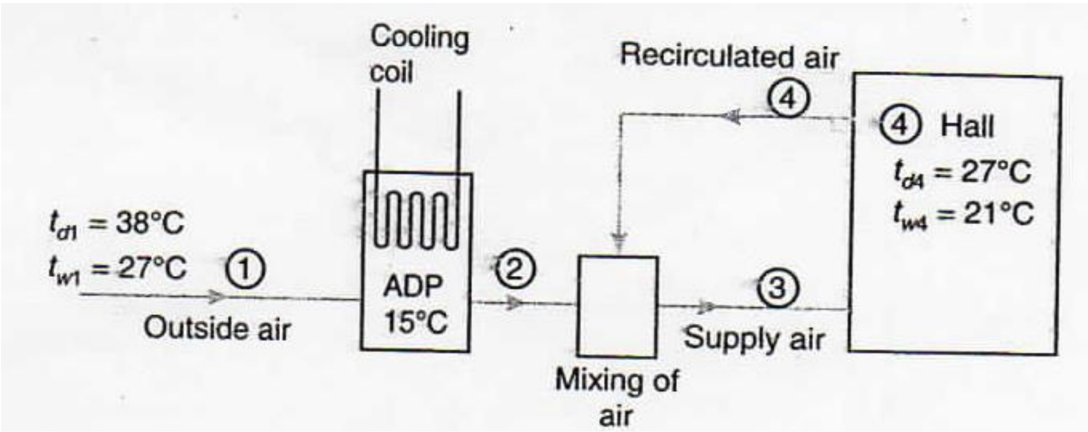


Fig. 25

First of all, mark the condition of outside air i.e. at 38°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.28. Now mark the condition of air in the hall, i.e. at 27°C dry bulb temperature and 21°C wet bulb temperature, at point 4. Mark point A by drawing vertical and horizontal lines from points 1 and 4 respectively. Since 25 m3/min of outside air at Id, = 38°C and t_{w1} = 27°C is supplied directly into the room through ventilation and infiltration, therefore the sensible heat and latent heat of 25 m^3/min infiltrated air are added to the hall in addition to the sensible heat load of 46.5 kW and latent heat load of 17.5 kW.

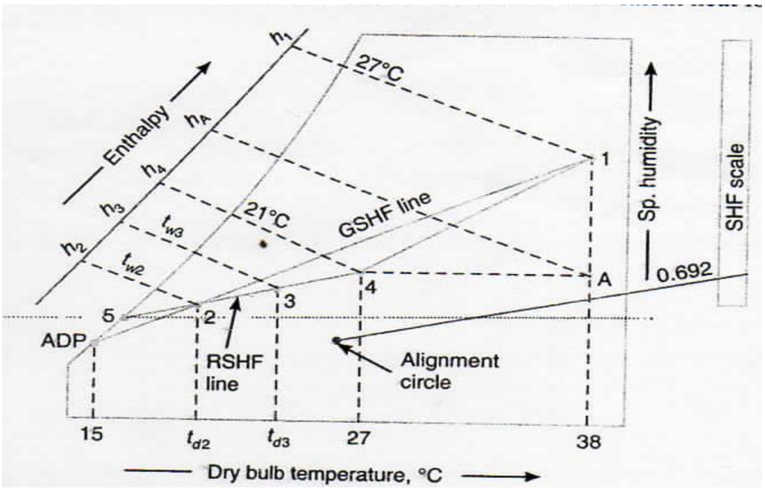


Fig.26

From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 85 \text{ kJ/kg of dry air}$$

Enthalpy of air at point 4

$$H_4 = 61 \text{ kJ/kg of dry air}$$

and enthalpy of air at point A ,

$$h_A = 72.8 \text{ kJ / kg of dry air}$$

Also specific volume of air at point 1,

$$v_{s1}, = 0.907 \text{ m}^3\text{/ kg of dry air}$$

∴ Mass of air infiltrated into the hall,

$$m_a = \frac{v_1}{v_{s1}} = \frac{25}{0.907} = 27.56 \text{ kg / min}$$

Sensible heat load due to the infiltrated air,

$$Q_{s1} = m_a (h_A - h_4) = 27.56 (72.8 - 61) = 325.21 \text{ kJ/min}$$

$$= 325.21/60 = 5.42 \text{ kW}$$

and latent heat load due to the infiltrated air,

$$Q_{L1} = m_a (h_1 - h_A) = 27.56 (85 — 72.8) = 336.23 \text{ kJ/min}$$

$= 336.23/60 = 5.6$ kW

∴ Total room sensible heat load,

$$RSH = Q_{s4} + Q_{s1} = 46.5 + 5.42 = 51.92 \text{ kW}$$

and total room latent heat load

$$RLH = Q_{L4} + Q_{L1} = 17.5 + 5.6 = 23.1 \text{ kW}$$

We know that room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{51.92}{51.92 + 23.1} = 0.692$$

Now mark this calculated value of *RSHF* on the sensible heat factor scale and join with the alignment circle (i.e. 26°C *DBT* and 50% *RH*) as shown in Fig. 26. From point 4, draw a line 4-5 (known as RSHF line) parallel to this line. Since the outside air marked at point 1 is passed through the cooling coil whose *ADP* = 15°C, therefore join point 1 with *ADP* = 15°C on the saturation curve. This line is the *GSHF* line and intersects the RSHF line at point 2, which represents the condition of air leaving the cooling coil. Also 60% of the air from the hall is re-circulated and mixed with the conditioned air after the cooling coil. The mixing condition of air is shown at point 3 such that

$$\frac{\text{Length 2-3}}{\text{Length 2-4}} = 0.6$$

1. Condition of air after the coil and before the re-circulated air mixes with it

The condition of air after the coil and before the re-circulated air mixes with it is shown by point 2 on the psychrometric chart, as shown in Fig. 26. At point 2, we find that

Dry bulb temperature, t_{d2} = 19°C Ans.

Wet bulb temperature, t_{w2} = 17.5°C Ans.

2. Condition of air entering the hall, i.e. after mixing with re-circulated air

The condition of air entering the hall, i.e. after mixing with re-circulated air, is shown by point 3 on the psychrometric chart, as shown in Fig. 26. At point 3, we find that

Dry bulb temperature, t_{d3} = 24°C Ans.

Wet bulb temperature, t_{w3}= 19.8°C Ans.

3. Mass of fresh air entering the cooler

The mass of fresh air passing through the cooling coil to take up the sensible and latent heat of the hall is given by

$$m_F = \frac{\text{Total heat removed}}{h_{\dot{5}} - h_2} = \frac{RSH + RLH}{h_4 - h_2}$$

$$= \frac{51.92 + 23.1}{61 - 49} = 6.25 \text{ kg/s} = 6.25 \times 60 = 375 \text{ kg / min Ans.}$$

... (From psychrometric chart, h_2 = 49 kJ / kg of dry air)

4. By-pass factor of the cooling coil

We know that by-pass factor of the cooling coil,

$$BPF = \frac{t_{d2} - ADP}{t_{d1} - ADP} = \frac{19 - 15}{38 - 15} = 0.174 \text{ Ans.}$$

5. Refrigerating load on the cooling coil

We know that the refrigerating load on the cooling coil

$= m_f (h_1 - h_2) = 375(85 — 49) = 13\ 500$ kJ/min

$= 13\ 500/210 = 64.3$ TR Ans.

Example 15: *The room sensible and latent heat loads for an air conditioned space are kW and 5 kW respectively. The room condition is 25°C dry bulb temperature and 50% relative humidity. The outdoor condition is 40°C dry bulb temperature and 50% relative humidity. The ventilation requirement is such that on mass flow rate basis 20% of fresh air is introduced and 80% of supply air is re-circulated. The by-pass factor of the cooling coil is 0.15.*

Determine: I. supply airflow rate; 2. outside air sensible heat; 3. outside air latent heat; 4. grand total heat; and 5. effective room sensible heat factor.

Solution: Given *RSH* = 25kW; *RLH* = 5kW; t_{d2}= 25°C; $\emptyset_2$ = 50%; t_{dt} = 40°C; $\emptyset_1$ = 50%; *BPF* = 0.15

The flow diagram for the air conditioned space is shown in Fig. 18.29 and it is represented on the psychrometric chart as discussed below:

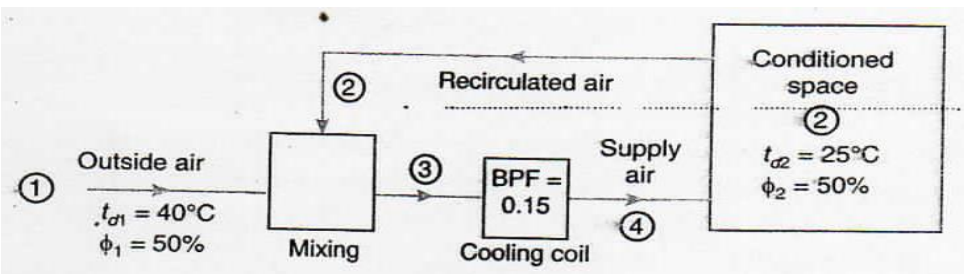


Fig.27

First of all, mark the initial condition of air at 40°C dry bulb temperature and 50% relative humidity on the psychrometric chart as point 1, as shown in Fig. 28. Now mark the room, condition of air at 25°C dry bulb temperature and 50% relative humidity as point 2. We know that i room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{25}{25+5} = 0.833$$

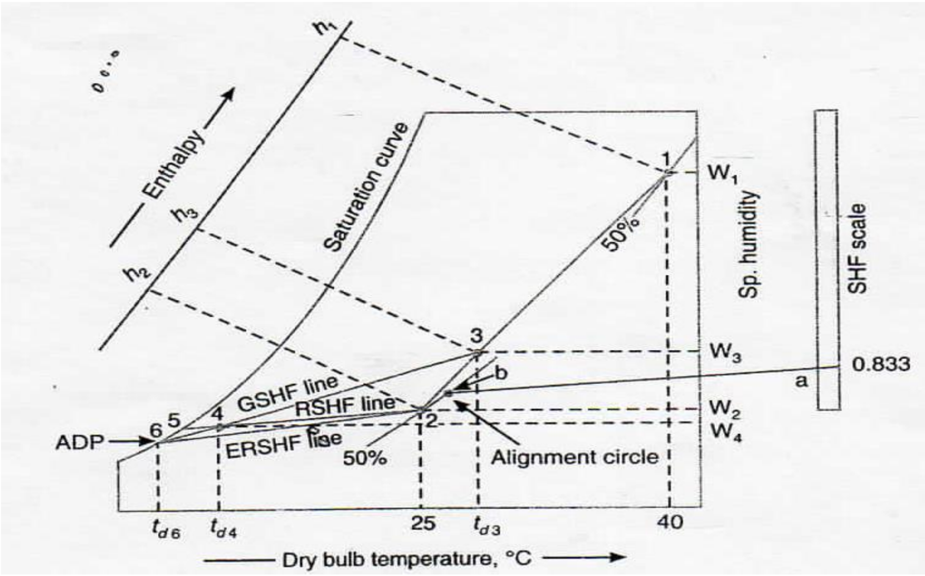


Fig. 28

Now mark this calculated value of *RSHF*=0.833 on the sensible heat factor scale as point *a* and join with point *b* which is the alignment circle (i.e. 26°C dry bulb temperature and 50% relative humidity). From point 2, draw a line 2-5 parallel to the line *ab*. The line 2-5 is called *RSHF* line. Since 20% of fresh or outside air is mixed with 80% of supply air, therefore the condition of air entering the cooling coil after mixing process is marked on the line 1-2 by point 3, such that

Length 2-3 = Length 1-2 x 0.2

Through point 3,-draw a line 3-6 (known as *GSHF* line) intersecting the *RSHF* line at point 4 and the saturation curve at point 6, such that

$$\frac{\text{Length 4-6}}{\text{Length 3-6}} = BPF = 0.15$$

1. Supply air flow rate

Let v = Supply air flow rate in m3/min,

t_{d4} =Dry bulb temperature of air leaving the cooling coil, and

t_{d6} = Apparatus dew point (ADP).

From the psychrometric chart, we find that dry bulb temperature of air entering the cooling coil at point 3,

$$t_{d3} = 28°C$$

We know that by-pass factor (BPF)

$$0.15 = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4} - t_{d6}}{28 - t_{d6}}$$

By trial and error, we find that

$$t_{d4} = 13.72° \text{ C and } t_{d6} = 11.2°C$$

We know that room sensible heat load,

$$RSH = 0.020\ 44\ v\ (t_{d2} - t_{d4})$$

$$25 = 0.020\ 44\ v\ (25 - 13.72) = 0.23\ v$$

$$v = 25 / 0.23 = 108.7 \text{ m3/min Ans.}$$

2. Outside air sensible heat

Since the outside air is 20% of the supply air, therefore outside air flow rate,

$$v_0 = 0.2\ v = 0.2 \times 108.7 = 21.74 \text{ m}^3\text{/min}$$

We know that outside air sensible heat,

$$\text{OASH} = 0.020\ 44\ v_0\ (t_{dt} - t_{d2})$$

$$= 0.020\ 44 \times 21.74\ (40 - 25) = 6.66 \text{ kW Ans.}$$

3. Outside air latent heat

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

$$W_1 = 0.0236 \text{ kg / kg of dry air}$$

and specific humidity of room air at point 2,

$$W_2 = 0.0098 \text{ kg / kg of dry air}$$

We know that outside air latent heat,

$$\text{OALH} = 50\ v_0\ (W_1 - W_2)$$

$$= 50 \times 21.74\ (0.0236 - 0.0098) = 15 \text{ kW Ans.}$$

4. Grand total heat

We know that total sensible heat,

$$TSH = RSH + OASH = 25 + 6.66 = 31.66 \text{ kW}$$

and total latent heat, $TLH = RLH + OALH = 5 + 15 = 20 \text{ kW}$

∴ Grand total heat,

$$GTH = TSH + TLH = 31.66 + 20 = 51.66 \text{ kW Ans.}$$

Note: The total sensible heat (*TSH*) and total latent heat (*TLH*) may also be calculated as follows:

From psychrometric chart, we find that specific humidity at point 3,

$$W_3 = 0.0127 \text{ kg / kg of dry air}$$

and specific humidity at point 4.

$$W_4 = 0.009 \text{ kg / kg of dry air}$$

We know that total sensible heat,

$$TSH = 0.020\ 44\ v\ (t_{d3} - t_{d4})$$

$$= 0.020\ 44 \times 108.7\ (28 - 13.72) = 31.7 \text{ kW}$$

and total latent heat, $TLH = 50\ v\ (W_3 - W_4)$

$$= 50 \times 108.7\ (0.0127 - 0.009) = 20.1 \text{ kW}$$

5. Effective room sensible heat factor

We know that effective room sensible heat,

$$ERSH = RSH + OASH \times BPF = 25 + 6.66 \times 0.15 = 26 \text{ kW}$$

3.25 COMFORT CONDITIONS

3.26 Introduction

Strictly speaking, the human comfort depends upon physiological and psychological condition. Thus it is difficult to defang the term 'human comfort'. There are many definitions given for this term by different bodies. But the most accepts definition, from the subject point of view, is given by the American Society of Heating, Refrigeration and air Conditioning Engineers (ASHRAE) which states : human comfort is that conditions of mind, which expressed satisfaction with the thermal environment

3.27 Thermal Exchanges of Body with Environment

The human body works best at a certain temperature, like any other machine, but it cannot tolerate wide range of variations in their environment temperatures like machines. The human body maintains its thermal equilibrium with the environment by means of three modes of hat transfer i.e evaporation, radiation and convection. The way in which the individual's body maintains itself in comfortable equilibrium will be by its automatic use of one or more of the three models of heat transfer. A human body feels comfortable when the heat produced by metabolism of human body is equal to the of the heat dissipated to the surroundings and the heat stared inhuman body by rising the temperature of body tissues. This phenomenon may be represented by the following equation.

It may be noted that

1. The metabolic heat produced ($Q_{M)}$ depends upon the rate of food energy consumption in the body. A fasting, weak or sick man, will have less metabolic heat production.
2. The heat loss by evaporation is always positive. It depends upon the vapour pressure difference between the skin surface and the surrounding air. The heat loss of evaporation ($Q_{E)}$ is given by

$$Q_E = C_d A (Ps - Pv) h_{fg} C_C$$

C_d = Diffusion coefficient in kg of water evaporated per Unit surface area and pressure difference per hour.

A = Skin surface area = 1.08 m2 for normal man,

Ps = Saturation vapour pressure corresponding to skin temperature

P_v = Vapour pressure of surrounding air,

H_{f8} = Latent heat of vaporization = 2450 kJ/kg,

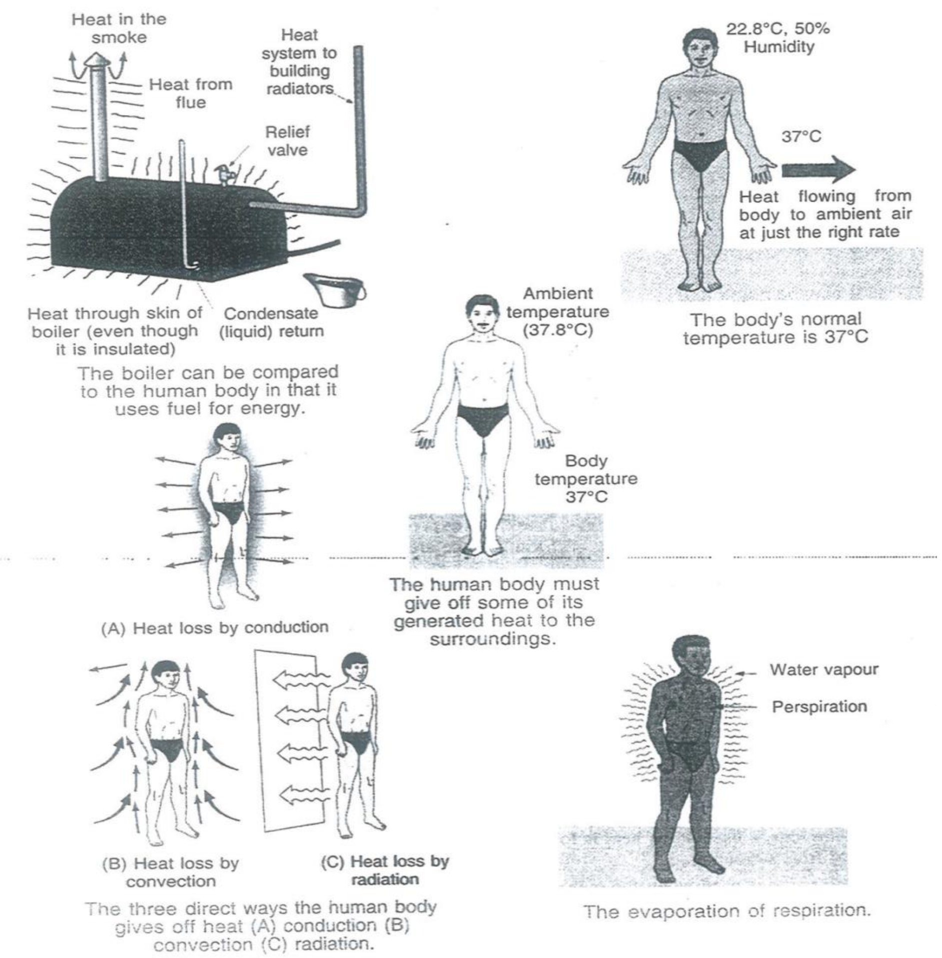
C_c = Factor which accounting for clothing worn.

The value of Q_E becomes zero when Ps = Pc, i.e when the surrounding air temperature is equal to the skin temperature and air is saturated or when it is higher than the erature and the air is nearly saturated.

- ❖ The skin tempplus sing is used when heat is lost to the surrounding and negative sing is used when heat is gained from surroundings.
- ❖ The plus sign is used when the temperature of the body rises and negative sign is used when the temperature of the body falls.

The value of Q_e is never negative as when p_s is less than p_v, the skin will not absorb moisture from the surrounding air as it is in sutures state. The only way for equalizing the pressure difference is by increasing p_s to p v by rise of skin temperature from the sensible heat flow from air skin.

3. The heat loss or gain by radiation (Q_R) from the body to the surroundings depends upon the mean radiant temperature. It is the average surface temperature of the surrounding object when properly weighted, and varies from place to place inside the room. When the mean radiant temperature is lower the day bulb temperature of air in the room, Q_R is positive i.e the body will undergo a radiant heat loss. On the other hand, if the mean radiant temperature is higher than will undergo radiant heat loss. On the other hand, if the mean radiant temperature is higher than the dry bulb temperature of air in the room, Q_R is negative i.e the body will undergo a radiant heat gain.



The boiler can be compared to the human body in that it uses fuel for energy.

The body's normal temperature is 37°C

(A) Heat loss by conduction

The human body must give off some of its generated heat to the surroundings.

(B) Heat loss by convection

(C) Heat loss by radiation

The three direct ways the human body gives off heat (A) conduction (B) convection (C) radiation.

The evaporation of respiration.

4. The heat loss by convection (QC) from the body to the surroundings is given by where

$Q_C = UA\ (t_B - t_s)$

U = Body film coefficient of heat transfer,

A = Body surface area = 1.8 m2 for normal man,

T_b = Temperature of the body, and

Ts = Temperature of the surroundings.

When the temperature of the surroundings (t_s) is higher than the temperature of the body (t_B), then QC will be negative, i.e the heat will be gained by the body. On the other hand if the temperature of the surroundings (t_S) is lower than the temperature of the body (TB), then QC will be the positive, i,e the heat will be lost by the body. Since the body film coefficient of heat transfer increases with the increase in air velocity, therefore higher air velocities will produce uncomforted t when Ts is higher than t_B. The higher air velocities are recommended when t_S is lower than t_B.

5. When Q_E, Q_R and Q_C are high and positive and ($Q_E + Q_R + Q_C$) is greater than (Q_M-W), the heat stored in the body (Qs) will be negative i.e the body temperature falls sown. Thus the sick, weak, old or a fasting man feels colder. On the other a man gets fever when high internal body actives increases Q_M to such extent so that Q_S becomes positive for the given Q_E, Q_R and Q_C.

The heat stored in the body has maximum and minimum limits which when exceed bring death. The usual body temperature, for a normal man (when Qs =0) is 37°C (98.6°F). the temperature of the body temperature of the body when falls below 36.5°C (98°F) and exceeds 40.5°C (105°F) is dangerous. There is some kind of thematic control called vasomotor control mechanism in the human body which maintains the temperature of body at the normal level of 37°C, by regulating the blood supply to the skin. When the temperature of the body falls (i.e the heal stored QS in the body is negative), then the vasomotor control decreases the circulation of blood which decreases conductivity of never cells and other tissues between the skin and the inner body cells. This allows temperature to fall but allows higher inner temperature of body cells beneath. When the temperature of the body rises (i.e the heat stared Qs in the body is positive), then the vasomotor control increases blood circulation which increases conductivity of tissues and hence allows less temperature drop between the skin and inner body cell.

The human body feels comfortable when there is no changes in the body temperature, i.e when the heat stored in the body Qs is zero. Any variation in the body temperature acts as a stress to the brain which ultimately results in either perspiration or shivering.

3.28 Physiological Hazards Resulting from Heat

In summer the temperature of the surrounding is always higher than temperature of the body. Thus the body will gin heat from the surrounding by means of radiation and convection processes. The body can dissipate eat only through evaporation of sweat. When the health loss by evaporation in unable to cope with the heat gain, there will bestrgae of heat in the body and the temperature of body rises. Several physiological hazards exist, the severity of which depends upon the extent and time duration of body temperature rise. Following are some of the physiological hazards which may result due to the rise in body temperature.

1. Heat exhaustion. It is due to the failure of normal blood circulation. The symptoms of heat exhaustion ileuses fatigue, headache, dizziness, vomiting and abnormal mental reaction such as irritability, severe heat exhaustion may close fainting. It does not cause permanent injury to the body and recovery is usually rapid when the person is removed to a cool places.
2. Heat cramp. It results from loss of salt due to an excessive rate of body perspiration. It cases sever pain in the calf and thing muscles. The heat cramp may be largely avoided by using salt tablets.
3. Heat stroke. It is most serious hazarded, when man is exposed to excessive heat and work, the body temperature may rise rapidly to 40.5°C (105°F) or higher. At such elevated temperatures, sweating cease and the man may enter a soma, with death imminent. A person experiencing a hat stroke may have permanent damage to the brain. The heat stroke may be avoided by taking sufficient water at frequent intervals. It has been found that man doing hard work in the sun requires one liter of water per hour.

3.29 Factors Affecting Human Comfort

In designing winter or summer air conditioning system, the designer should be well conversant with a number of factors which physiologically affect human comfort. The important farces are as follows:

1. Effective temperature, 2. Heat production and regulation in human body 3. Heat and moister losses from the human body, 4. Moisture content of air, 5. Quality and quantity of air. 6. Air motion, 7. Hot and cold surfaces and 8. Air stratification

   These factors are discussed, in detail, in the following articles:

Effective Temperature The degree of warmth or cold felt by a human body depends mainly on the following three factors:

1. Dry bulb temperature, 2. Relative humidity and 3. Air velocity.
   In order to evaluate the combined effect of these factors, the effective temperature is employed. It is defined as that index which collates the com binned effects of air temperature, relative humidity and air velocity on the human body. The numerical value of effective temperate is made equal to the temperature of stills (i.e 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or clones as produced under the given conditions.

The practical application of the concept of effective temperature is presented by the comfort chart, as shown in Fig 171. This chart is the result of research made on different kinds of people subjected to wide range of environmental temperature, relative humidity and air movement by the American Society of Heating, Refrigeration and Air conditioning Engineers (ASHRAE). It is appreciable to reasonably still air (5 to 8 m/min air velocity) to situations where the occupants are seated at rest or doing light work and to space whose enclosing surfaces are at a mean temperature equal to the air dry bulb temperature.

In the comfort chart, as shown in Fig.17.1, the dry bulb temperature is taken as abscissa and the wet bulb temperature of ordinates. The relative humidity lines are repotted from the psychometric chart. The statistically prepared graphs corresponding to summer and winter season are also superimposed. These graphs have effective temperature scale as abscissa and % of people feeling comfortable as ordinate.

A close study of the chart reveals that the several combination of wet and dry bulb temperatures with different relative humidity will produces the same effective temperature.

However, all points located on a given effective temperature line do not indicate conditions of equal comfort or discomfort. The extremely high or low relative humidifies may process conditions of discomfort repulses or existent effective temperate. The moist deniable relive humidly rages lies between 30 and 70 per cent. When the relative humidity is much below 30 per cent, the mucous membranes and the skin surface become too dry for comfort and health. On the other hand, if the relative humidity is above 70 per cent, there is a tendency for a clammy or sticky sensation to develop. The curves at the top and bottom, as shown in Fig. 17.1, indicate the parentages of person participating in tests, who found various effective temperatures satisfactory for comfort.

The comfort chart shows the range for both summer and winter condition within which a condition of comfort exists for most people. For summer conditions, the chart indicates that a maximum of 98 percent people felt comfortable for an effective temperature of 21.6°C. For winter conditions, chart indicates that an effective temperature of 20°C was desired by 97.7 percent people. It has been found that comfort; women require 0.5°C higher effect give temperature than men. All men and women above 40 years of age preset 0.5°Chiger effective temperature than the person below 40 years of age.

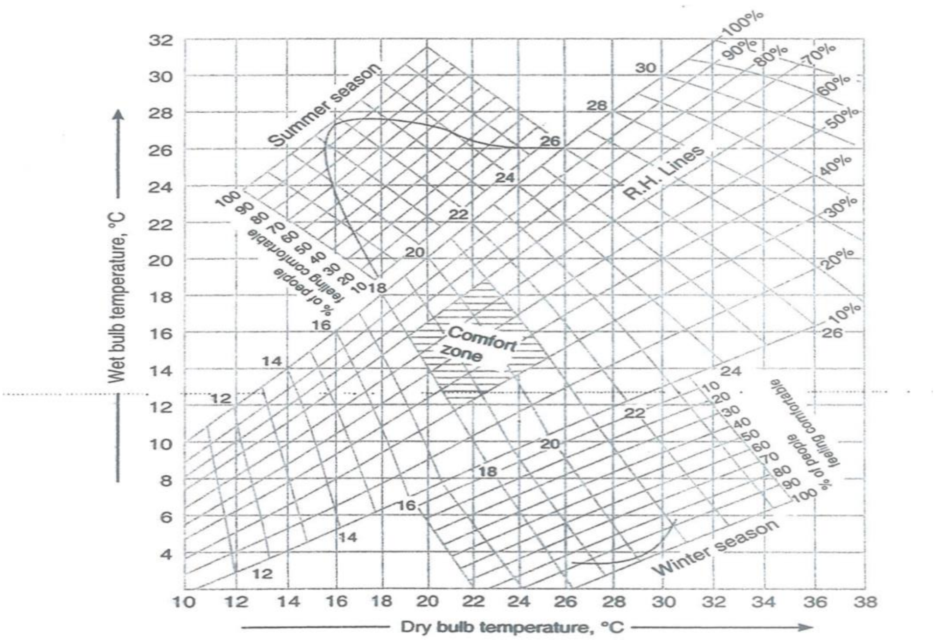


Fig. 17.1. Comfort chart for still air (air velocities from 5 to 8 m/min)

It may be noted that the comfort chart, as shown in \fig.17.1, does not take into account the carnations in comfort conditions when there are wide variations in the mean radiant temperature (MRT). In the range of 26.5°C, a rise of 0.5°C in mean radiant temperature above the room dry bulb temperature raises the effete temperature by 0.5°C. The effect of mean radiant temperature on comfort is less pronounced at high temperatures than at low temperatures.

The comfort conditions for persons at work vary with the rate of work and the amount of clothing worn. In general, the greater the dredge activity, the lower the effective temperature necessary for comfort.

Fig.17.2 shows the variation in effective temperature with different air velocities. We see that for the atmospheric conditions of 24°C dry bulb temperature and16°C wet bulb temperature correspond to about 21°C with nominally still air (velocity 6m/min) and it is about 17°C at an air velocity of 210 m/min. the same effective temperature is observed at higher dry bulb and wet bulb temperatures it higher velocities. The case is reversed after 37.8°C as in that case higher velocities will increases sensible heat flow from air to body and will decrease comfort. The same effective temlperarue means same feeling of warmth, but it does not means same comfort.

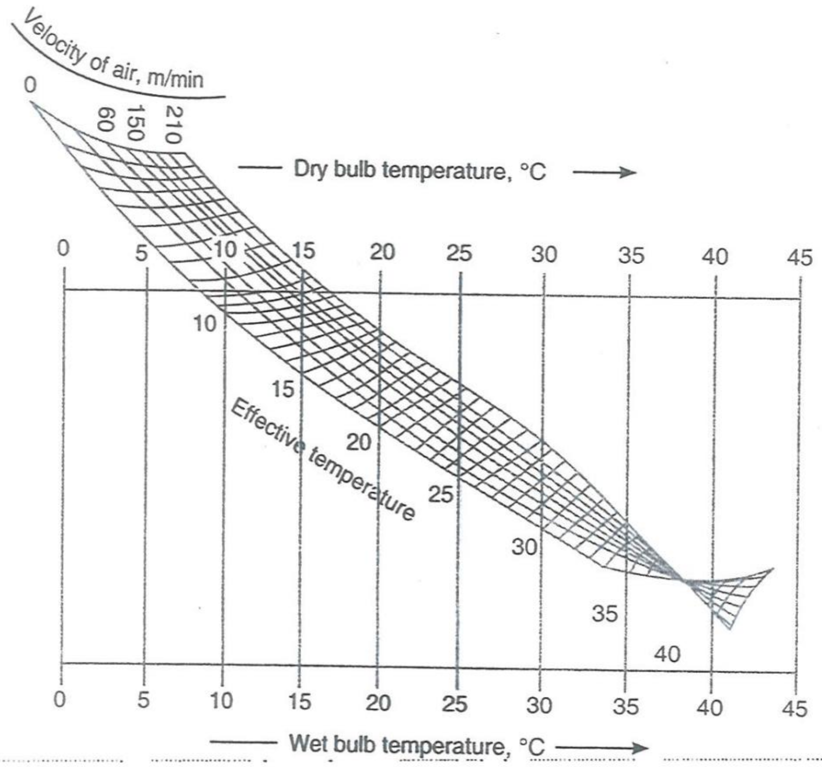


Fig. 17.2. Variation of effective temperature with air velocity.

3.30 Modified Comfort Chart

The comfort chart, as shown in Fig.17.1 has become obsolete now-a-days due to its short comings of over exaggeration of humidity at lower temlperarue and under estimation of humidity at heat to trance level. The modified comfort char according to ASHRAE is shown in Fig.17.3 and it is commonly use these days. This chart was developed on the basis or research done in 1963 by then stirred fro environmental research at Kansas State University. The mean radiant temperature was kept equal to dry bulb temperature and air velocity was less than 0.17 m/s.

17.7 Heat Production and Regulation in Human Body

The human body acts like a heat entice which gets its energy from the combustion of food within the body. The process of combustion (called metabolism) produces het and energy due to the oxidation of redacts in the body by oxygen obtained from in held air. The rate of heat production depends upon the individual’s health, his physical activity and his environment. The rate at which the body produces heat is metabolic rate. The heat production from a normal healthy person when a sleep (called based metabolic rate) is about 60 wtts and it is about ten times more for a person carrying out sustained very hard work.

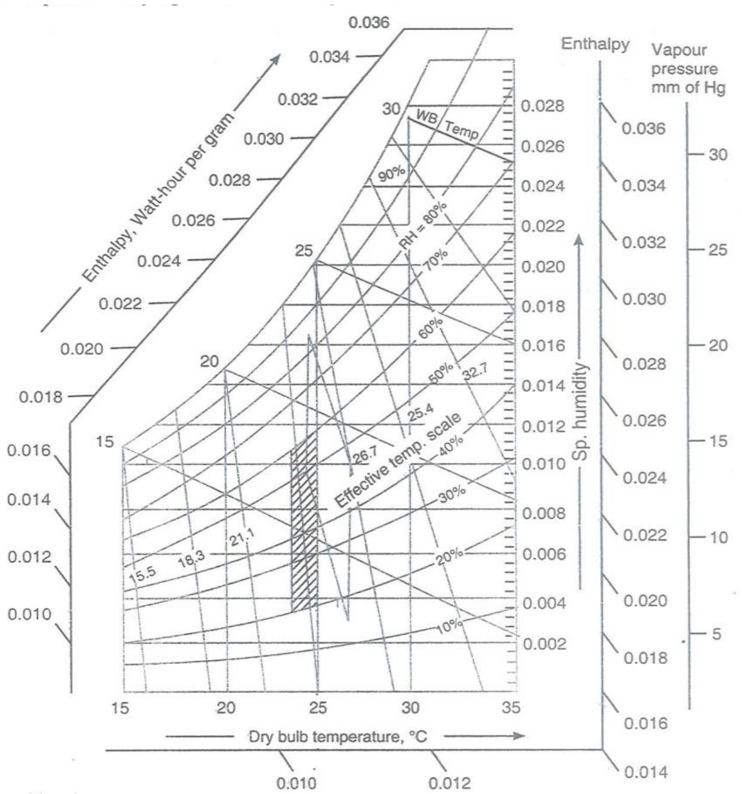


Fig. 17.3. Modified comfort chart.

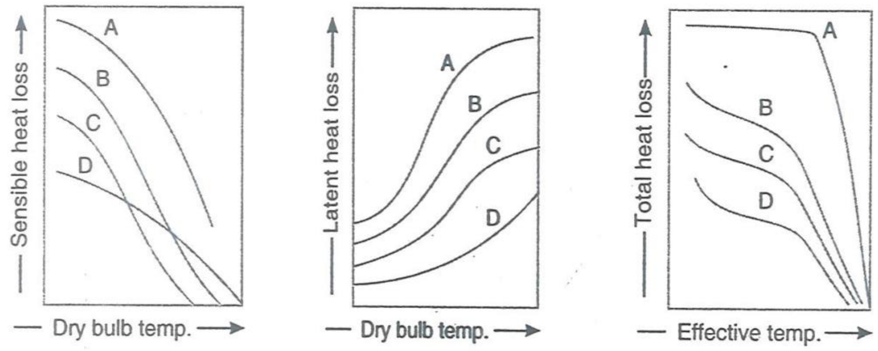
Since the body has a thermal efficiency of 20 per cent, therefore the remaining 80 per cent of the heat must be rejected to the surrounding environment, otherwise accumulation of heat result which causes discomfort. The rate and the manner of rejection of heat is controlled by the automatic regulation system of a human body.

In order to effect the loss of heat from the body to race cold, the body may react to bring more blood to the capillaries in the skin. The heat looses from the skin, now, may take place by radiation, convection and by evaporation. When the process of radiation or convection or both fails process necessary loss of heat, the sweat glands become more active and ore moister is debited on the kin, carrying heat always as it evaporates. It may be noted that when the temperature of surrounding air and objects is below the blood temperature, the heat is removed by rendition and convection. On the other hand, when the temperature of surrounding air is above the blood temperature, the heat is removed by evaporation only. In case the body fails to throw off the requisite amount of heat, the blood temperature rises. This results in the accumulation of heat which will cause discomfort.

The human body attempts to maintain its temperatures when exposed to cold by the with drawl of blood from the outer portions of the skin, by decreased blood circulation and by an increased rate of metabolism.

3.31 Heat and Moisture Losses from the Human Body

This heat is given off from the human body as either sensible or latent heat or both. In order to design any air-conditioning system for spaces which human bodies are to occupy, it is necessary to know the rates at which these two forms of heat are given off under different conditions of air temperature and bodily activity.



(*a*) Relation between sensible heat loss from the human body and dry bulb temperature for still air.

(*b*) Relation between latent heat loss from the human body and dry bulb temperature for still air.

(*c*) Relation between total heat loss from the human body and effective temperature for still air.

Fig. 17.4

Fig.17.4(a) shows the graph between sensible heat loss by radiation and convection for an average man and the dry bulb temperate for different types of acidity. Fig.17.4 (b) shows the graph between the latent heat loss by evaporation for an average man and dry bulb temperature for different types activity.

The total heat loss from the human body under varying effective temperatures is show in Fig.17.4(c). From curve D, which apples to men at rest, we see that from about 19°C to 30°C effective temperature, the heat loss inconstant. At the lower effective temperature, the heat dissipation increases which results in a feeling of coolness. At higher effective temperature, the ability to lose heat rapidly decreases resulting in sever discomfort. The curves A, B, C and D show in Fig.17.4 represents as follows:

Curve A – Men working at the rate of 90 kN-m/h

Curve B – Men working at the rate of 45 kN-m/h

Curve C – Men working at the rate of 22.5 kN-m/h

Curve D -- Men at rest.

3.32 Moisture Content of Air

The dry bulb temperature, relative humidity and air motion are inter-related. The moisture content of outside air during winter is generally low and it is above the average during simmer, because the capacity of the air to carry moisture is dependent upon its dry bulb temperature. This means that in winter, if the cold outside air having a low mist rue content leaks into the conditioned space, it will cause a low relative humidity unless minster is assed to the air by the processes of humidification. In summer, the reveres will take place unless moisture is removed from the inside air by the dehumidification process. Thus, while designing an air-conditioning system, the proper dry bulb temperature for either summer or winter must be selected in accordance with the practical consideration of relative humidity's which are feasible. In general, for winter conditions in the average residence, relative humidity above 35 to 40 per cent are not practical. In summer comfort cooling, the air of he occupied space should not have a relative humidity above 0 per cent. With theses limitations the necessary dry bulb temperature for the air may be depend from the comfort chart.

3.33 Quality and Quantity of Air

The air in an occupied space should, at all times, be free room toxic, unhealthful or disagreeable fumes such as carbon dioxide. It should also be free from dust and odour. In order to obtain these conditions, enough clean outside in must allows be supplied to an occupied space to counteract or adequately dilute the sources of contamination.

The concentration of odour in a room depends upon many factors such as directory and hygienic habits of occupant, type and amount f outdoor air supplied, room volume per occupant and types of odour sources. In general. When here is no smoking in a room, 1m3/min per person of outside air will take care of all the conditions. But when smoking takes place in a room,1.5 m3/min per person of outside air is necessary. In most air-conditioning systems, alarge amount of air is reticulated over and above the required amount of outside air to satisfy the minimum ventilation conditions in regard to odour and purity. For general application, a minimum of 0.3 m3/min of outside air pr person, mixed with 0.6m3/min of recalculated air is good. The recommended and minimum values for the outside air require per person are given in Chapter 19 on cooling load estimation

3.34 Air Motion

The air motion which included the distribution of air is very important to maintain uniform temperate in the conditioned space. No air conditioning system is satisfactory unless the air handled is properly circulated and distributed. Ordinarily, the air velocity in the occupied zone should not exceed 8 to 12m/min. The air velocities in the space above the occupied zone should be very high in order to produce goof distribution of air in the occupied zone, provided that the air in motion does not produce any objectionable noise. The flow of air should be preferably towards the faces of the indicial rather than form the rear in the occupied zone. Also for the proper and perfect distribution of air the air-conditioned space, down flow should be preferred instead of up flow.

The air motion without proper air distribution produces local cooling sensation known as draft.

3.35 Cold and Hot Surfaces

The cold or hot objects in a conditioned space may cause discomfort to the occupants. A single glass of large area when exposed to the outdoor air during winter will produce Discomfort

The atmospheric air contains 0.03% to 0.04% by volume of carbon dioxide and it should not increase 0.6% which is necessary for proper functioning of respiratory system. The carbon dioxide, in excess of 2% dilutes oxygen contents and makes breathing difficult. When the carbon dioxide exceeds 6%, breathing is very difficult and 10% carbon dioxide causes loss of consciousness. A normal man at rest in breathing exhales about 0.015 to 0.018 m3/h of carbon dioxide to the occupants of a room by absorbing heat from them by radiation. On the other hand, a ceiling that is warmer than the room air during summer cusses discomfort. Thus, in the designing of an air conditioning system, the temperature of the surfaces to which the body may be exposed must be given considerable

3.36 Air Stratification

When air is heated, its density decreases and thus it rises to the upper part of the confined space. This results in a considerable variation in the temperatures between the floor and ceiling levels. The movement of the air to produce the temperature gradient from floor to ceiling is termed as air stratification. In order to achieve comfortable conditions in the occupied space, the air conditioning system must be designed to reduce the air stratification to a minimum.

3.37 Factors Affecting Optimum Effective Temperature

The important factors which affect the optimum effective temperature are as follows:

1. Climatic and seasonal differences. It is a known fact that the people living in colder climates feel comfortable at lower effective temperatures than those living in warmer religions. There is a relationship between the optimum indoor effective temperature

and the optimum outdoor temperature, which changes with seasons. We see from the comfort chart (Fig.171.1) that in winter the optimum effective temperature is 19°C whereas in summer this temperature is 22°C.

2. **Clothing:** It is another important factor which affects the optimum effective temperature. It may be noted that the person with light clothing need less optimum temperature than a person with heavy clothing.
3. **Age and sex**. We have already discussed that the women of all ages require high reflective temperature (about 0.5°C) than men, similar is the case with young and old people. The children also need higher effective temperature than adults. Thus, the maternity halls are always kept at an effective temperature of 2 to 3°C higher than the effective temperature used for adults.
4. **Duration of stay.** It has been established that if the stay in a room is shorter (as in the case of persons going to banks), then higher effective temperature is required than that needed for long stay (as in the case of persons working in an office).
5. **Kind of activity.** When the activity of the person is heavy such as people working in a factory, dancing hall, then low effective temperature is need than the people sitting in cinema hall or auditorium**.**
6. **Density of Occupants.** The effect of body radiant heat from person to person particularly in a densely occupied space like auditorium is large enough which rewire alight lower effective temperature.