Lecture -14- to lecture -17-

Advance applications on psychrometric chart

1-Summer air conditioning system

Schematic diagrams of summer air conditioning systems are showed in figures(1-4). The essential air processing components consisting of a filter, a cooling and dehumidifying coil and a fan are arranged in series. These are connected using metal ducts through which air flows. For ventilation purposes, a portion of the air withdrawn from the conditioned space (see figures(2, 3 and 4)) and mixed with the fresh air before inlet to coil.

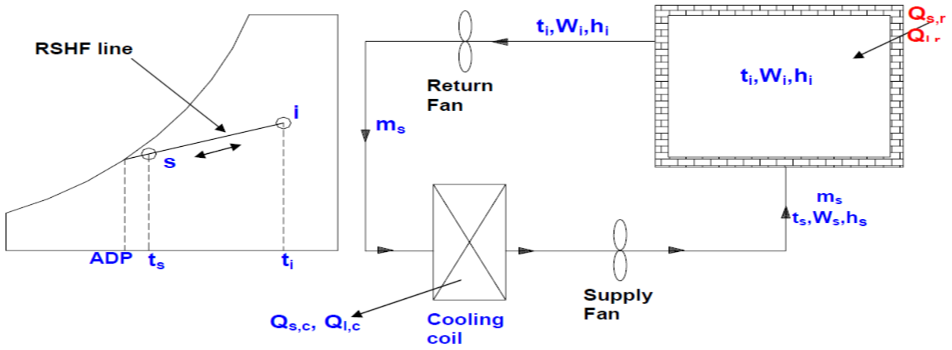


Fig.1: A summer air conditioning system with 100% re-cerculated air

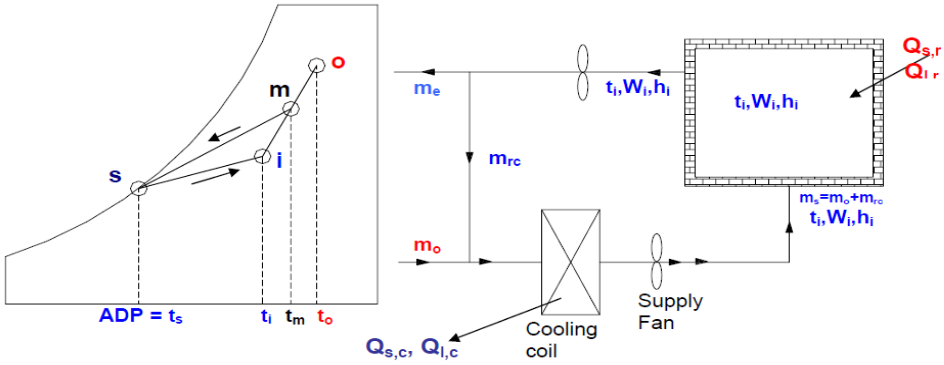


Fig.2: A summer air conditioning system with outdoor air for ventilation without by-pass factors

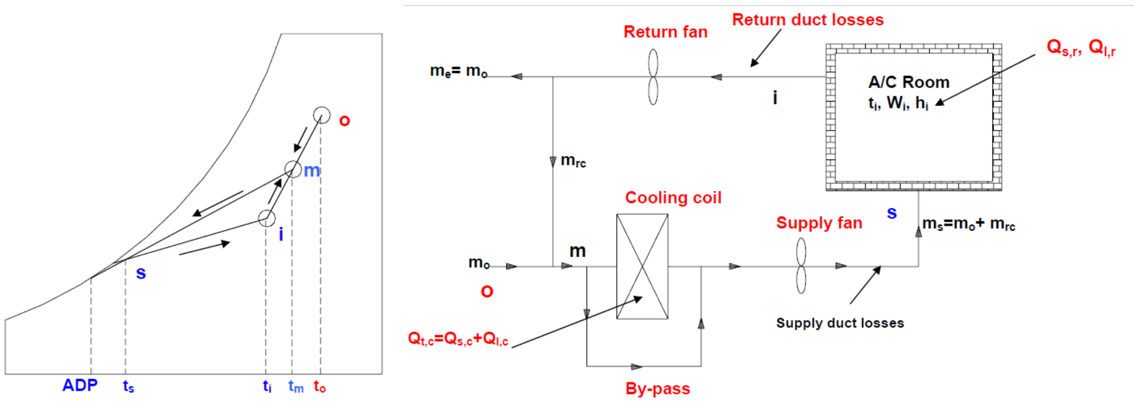


Fig.3: A summer air conditioning system with outdoor air for ventilation with by-pass factors

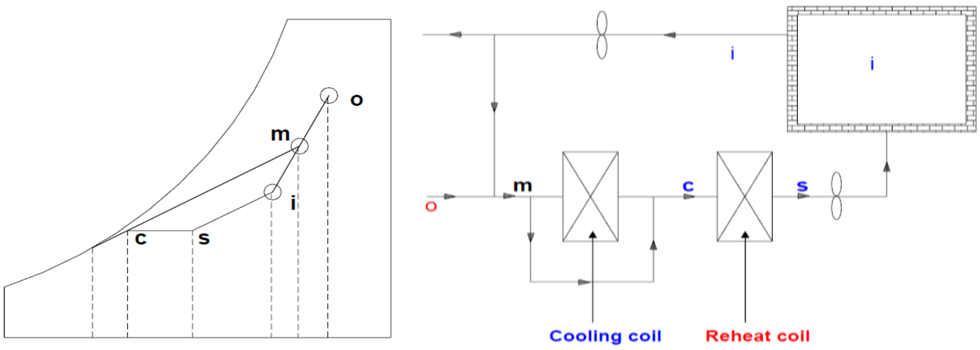


Fig.4: A summer air conditioning system with both reheat coil and by-pass factor for high latent cooling load applications

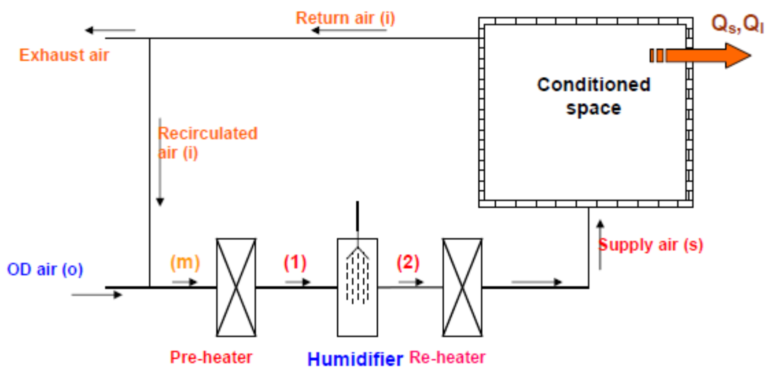
2-Winter air conditioning system

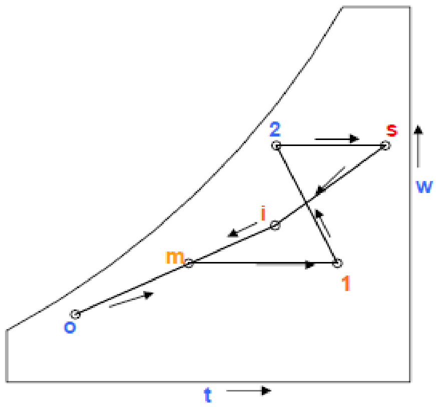
• In winter, outside conditions are cold and dry. As a result, there will be a continuous transfer of sensible heat as well as moisture latent heat from the conditioned space to outside. Hence, in order to maintain comfort conditions (24 C and 60% RH.), air supplied to the conditioned space should be heated and humidification.

• heated and humidification of air can be achieved by different schemes:

a) By using a preheat coil, humidifier and a reheat coil.

b) By using an air washer and a reheat coil.





What is the advantage of preheated in of the winter type of A/C system?

The preheating of air is advantageous as it ensures that the water in the humidifier/ or air washer does not freeze. In addition, by controlling the heat supplied in the preheater, one can control the moisture content in the conditioned space.

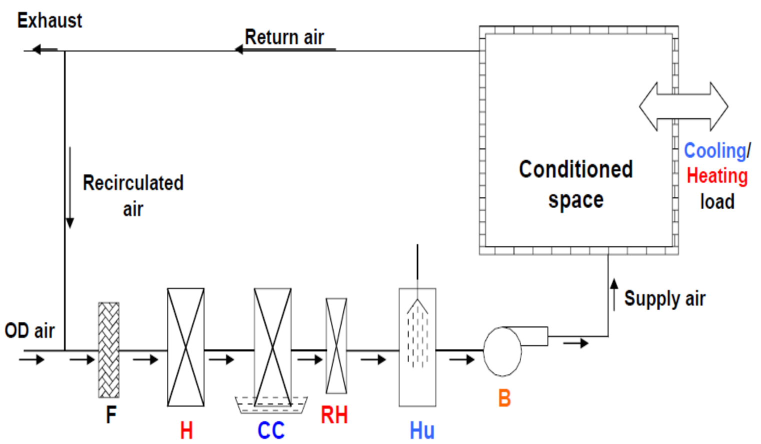
3- Year-round air conditioning system

. In many countries, the summer as well as winter both are very uncomfortable, under such adverse conditions a complete air conditioning system can be used for providing air conditioning throughout the year. The system consists of a filter, a heating coil, a cooling and dehumidifying coil, a reheating coil, a humidifier and a blower. In addition to these, actual system consist of dampers for controlling flow rates of recirculated and outdoor air.

. Filter is introduced for purifying the air

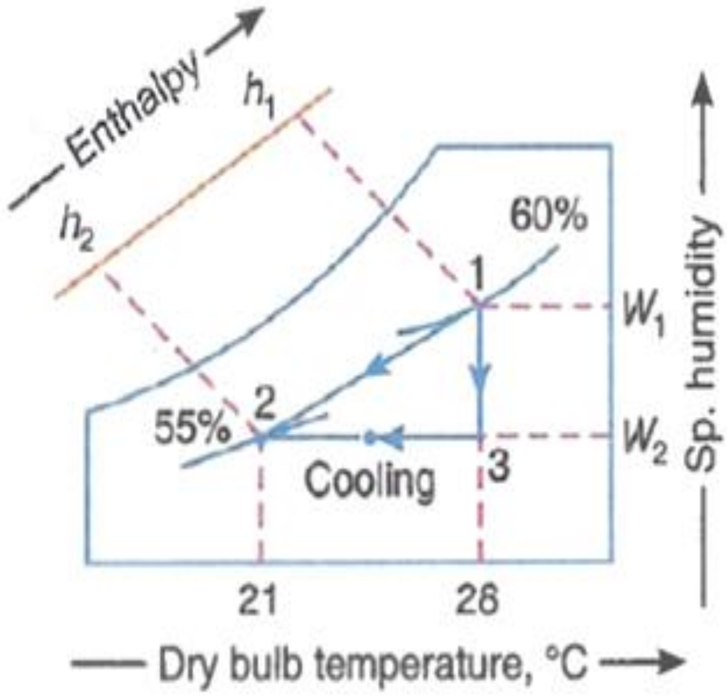
. A blower is introduced for circulating air into the conditioned space.

. In winter, the cooling coil is made inoperative and heating coil and humidifier operate to heat and humidity air whereas in summer, we need cool and dry air for that we require cooling and dehumidification coil.



F: Filter
H:Heating coil
CC: Cooling and dehumidifying coil
RH: Re-heating coil
HU: Humidifier
B: Blower

<u>Example 1:</u> An air conditioning plant is required to supply 60 m^3/min 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.



Mass of water drained = $m_w = m_a(W_1 - W_2)$

$m_a = \dfrac{V}{v_{s2}}$

Capacity of cooling coil = $C.C = m_a(h_1 - h_2)$

<u>From psychrometric chart.</u>

Specific humidity of air at point 1= W_1=0.0142 kg/kg of dry air

Specific humidity of air at point 2= W_2=0.0084 kg/kg of dry air

Specific volume of air at point 2= v_{s2}=0.845 m^3/kg of dry air

Mass of air circulated = $m_a = \frac{V}{v_{s2}}$ = 71 kg/min

Mass of water drained = $m_w = m_a(W_1 - W_2)$= 71 (0.0142 – 0.0084) = 0.412 kg/min
= 0.412× 60= 24.72 kg/h

Enthalpy of air at point 1= h_1 =64.8 kJ/kg of dry air

Enthalpy of air at point 2= h_2 =42.4 kJ/kg of dry air

Capacity of cooling coil = $C.C = m_a(h_1 - h_2)$= 71 (64.8-42.4)= 1590.4 kJ/ min
=1590.4/210 = 7.57 TR

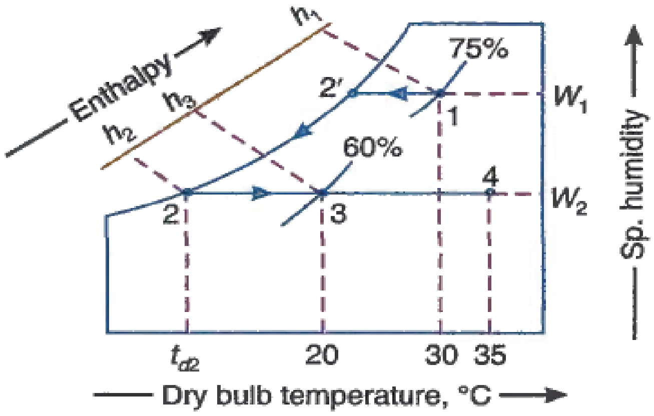
Example 2: Following data refers to an air conditioning system to be designed for an industrial process for hot and wet climate: Outside conditions = 30C DBT and 75% RH, required inside conditions = 20 C DBT and 60% RH. The required condition is to be achieved first by cooling and dehumidifying and then by heating. If 20 m^3 of air 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

4- by-pass factor of the heating coil, if its surface temp. is 35C.



1. *Capacity of the cooling coil in tonnes of refrigeration*

Enthalpy of air at point 1,

$$h_1 = 81.8 \text{ kJ / kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 34.2 \text{ kJ / kg of dry air}$$

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

$$= m_a (h_1 - h_2) = 22.6 (81.8 - 34.2) = 1075.76 \text{ kJ / min}$$

$$= 1075.76/210 = 5.1 \text{ TR} \quad \textbf{Ans.}$$

2. *Capacity of the heating coil in kW*

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

$$h_3 = 42.6 \text{ kJ / kg of dry air}$$

∴ Capacity of the heating coil

$$= m_a (h_3 - h_2) = 22.6 (42.6 - 34.2) = 189.84 \text{ kJ / min}$$

$$= 189.84/60 = 3.16 \text{ kW} \quad \textbf{Ans.}$$

3. *Amount of water removed per hour*

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

$$W_1 = 0.0202 \text{ kg / kg of dry air}$$

and specific humidity of air at point 2,

$$W_2 = 0.0088 \text{ kg / kg of dry air}$$

∴ Amount of water removed per hour

$$= m_a (W_1 - W_2) = 22.6 (0.0202 - 0.0088) = 0.258 \text{ kg / min}$$

$$= 0.258 \times 60 = 15.48 \text{ kg / h} \quad \textbf{Ans.}$$

4. *By-pass factor of the heating coil*

We know that by-pass factor,

$$BPF = \frac{t_{d4} - t_{d3}}{t_{d4} - t_{d2}} = \frac{35 - 20}{35 - 12.2} = 0.658 \quad \textbf{Ans.}$$

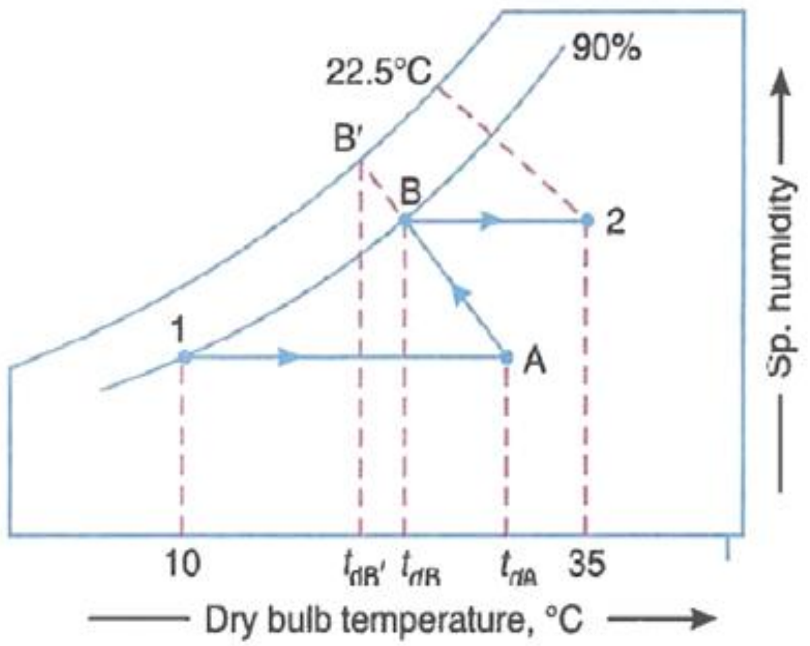
... [From psychrometric chart, $t_{d2} = 12.2°C$]

<u>Example 3</u>: Air at 10 °C DBT and 90% RH is to be brought to 35 °C DBT and 22.5 °C WBT with the help of winter air conditioner. If the humidified air comes out of the humidifier at 90% RH, draw the various processes involved on a psychrometrics chart and find:

1-The temperature to which the air should be preheated,

2-The efficiency of the air-washer.

Solution/ the processes involved preheating of entering air in heating coil and humidification of air in humidifier, and reheating of humidified air in a reheater.



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

$$t_{dA} = 31.2° \text{ C} \quad \textbf{Ans.}$$

2. ***Efficiency of the air-washer***

From the psychrometric chart, we find that

$$t_{dB} = 18.5° \text{ C} ; \quad \text{and} \quad t_{dB'} = 17.5° \text{ C}$$

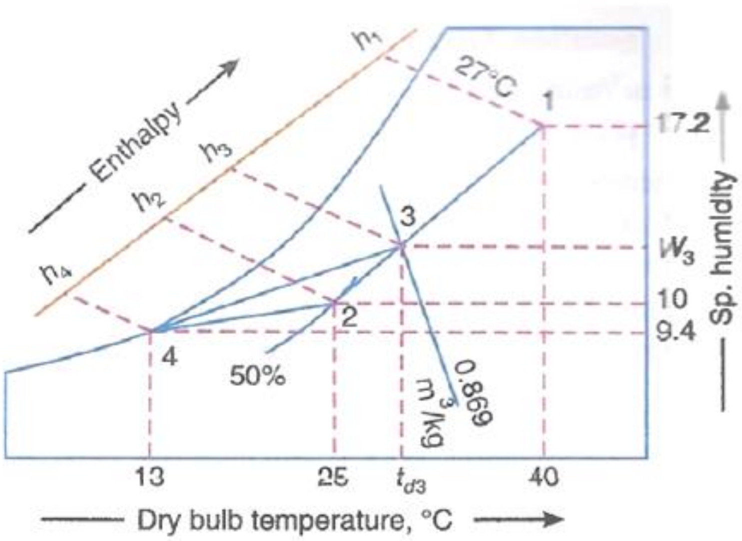
We know that efficiency of the air-washer

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

$$= \frac{31.2 - 18.5}{31.2 - 17.5} = 0.927 \text{ or } 92.7\% \quad \textbf{Ans.}$$

<u>Example 4</u>: An air handling unit in an 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 80% recirculated 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* $\frac{g\ of\ water\ vapour}{kg\ of\ dry\ air}$	*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



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,

$$t_{d3} = 28.3°\text{C}$$

$$\boldsymbol{h_3 = \frac{m_1}{m_T} h_1 + \frac{m_2}{m_T} h_2}$$

$$\boldsymbol{h_3 = 0.2h_1 + 0.8h_2}$$

Total cooling load

We know that mass of air entering the cooling coil,

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

$\therefore$ 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 ($m_{a3} = 5178$ kg / min) comprises 20% of fresh air, therefore mass of fresh air supplied at point 1,

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

and fresh air load $= m_{a1}(h_1 - h_2) = 1035.6\ (85 - 51) = 35\ 210$ kJ / min

$= 35\ 210 / 210 = 168$ TR **Ans.**

$\therefore$ Room heat gain $=$ Total cooling load – Fresh air load

$= 517.8 - 168 = 349.8$ TR **Ans.**

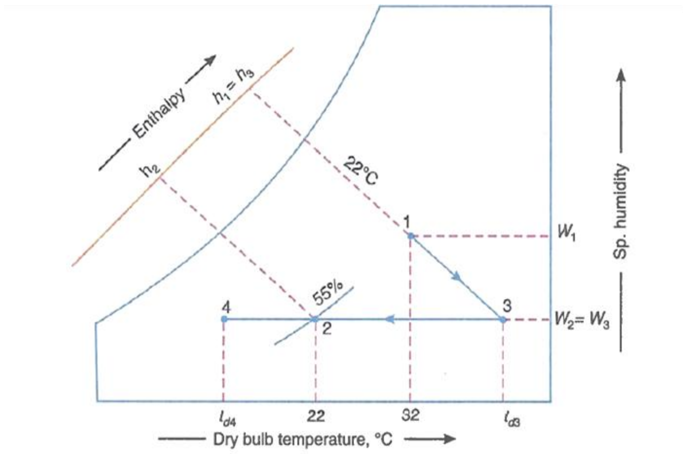
Or $\boldsymbol{Room\ heat\ gain = m_{a3}\ (h_2 - h_4) = 5178 \times (51 - 36.8) = 73528 \frac{kJ}{min} = 349.8TR}$

<u>Example 5:</u> 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 0.5 m^3/min/person. The comfort conditions are achieved first by chemical dehumidification and by cooling coil. Determine:

1-Dry bulb temp. of air at exit of dehumidifier,

2-Capacity of dehumidifier,

3-Capacity and surface temp. of cooling coil, if the by-pass factor is 0.3



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 point 3,

$$t_{d3} = 41°\text{C} \quad \textbf{Ans.}$$

2. Capacity of dehumidifier

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

$$h_1 = h_3 = 64.5 \text{ kJ / kg of dry air}$$

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,

$$v_{s1} = 0.881 \text{ m}^3 \text{ / kg of dry air}$$

We know that mass of air supplied,

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

$\therefore$ Capacity of the dehumidifier

$$= m_a (W_1 - W_3)$$
$$= 34.05\,(0.0123 - 0.0084) = 0.1328 \text{ kg / min}$$
$$= 0.1328 \times 60 = 7.968 \text{ kg / h} \quad \textbf{Ans.}$$

3. Capacity and surface temperature of cooling coil

We know that capacity of the cooling coil

$$= m_a (h_3 - h_2) = 34.05\,(64.5 - 45) = 664 \text{ kJ/min}$$
$$= 664 / 210 = 3.16 \text{ TR} \quad \textbf{Ans.} \qquad ...(\because 1\text{TR} = 210 \text{ kJ/min})$$

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

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

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

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

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

<u>Example 6:</u> An air conditioning plant is to be designed for a small office for winter conditions with following data: Outdoor conditions = 10 °C DBT and 8 C WBT

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

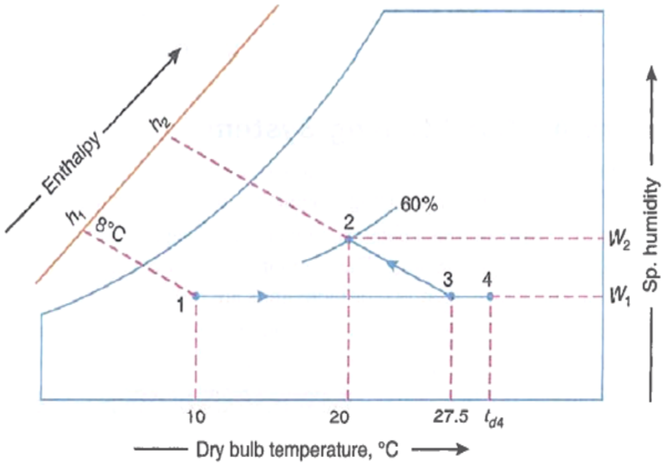
Amount of air circulation = 0.3 m^3/min/person

Seating capacity of the office = 50 persons

The required condition is achieved first by heating and then by adiabatic humidifying. Find:

1-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.



$$v_{s1} = 0.81 \text{ m}^3/\text{kg of dry air}$$

∴ Mass of air supplied per minute,

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

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

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

$$h_1 = 24.8 \text{ kJ / kg of dry air}$$

and enthalpy at point 2, $h_2 = 42.6$ kJ / kg of dry air

We know that heating capacity of the coil

$$= m_a(h_2 - h_1) = 18.52\ (42.6 - 24.8) = 329.66 \text{ kJ/min}$$

$$= 329.66 / 60 = 5.5 \text{ kW } \textbf{Ans.}$$

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

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

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

or $0.32\ (t_{d4} - 10) = t_{d4} - 27.5$ or $0.32\ t_{d4} - 3.2 = t_{d4} - 27.5$

∴ $t_{d4} = 24.3 / 0.68 = 35.7°\text{C}$ **Ans.**

2. *Capacity of the humidifier*

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

$$W_1 = 0.0058 \text{ kg / kg of dry air}$$

and specific humidity at point 2,

$$W_2 = 0.0088 \text{ kg / kg of dry air}$$

We know that capacity of the humidifier,

$$= m_a(W_2 - W_1) = 18.52\ (0.0088 - 0.0058) = 0.055 \text{ kg / min}$$

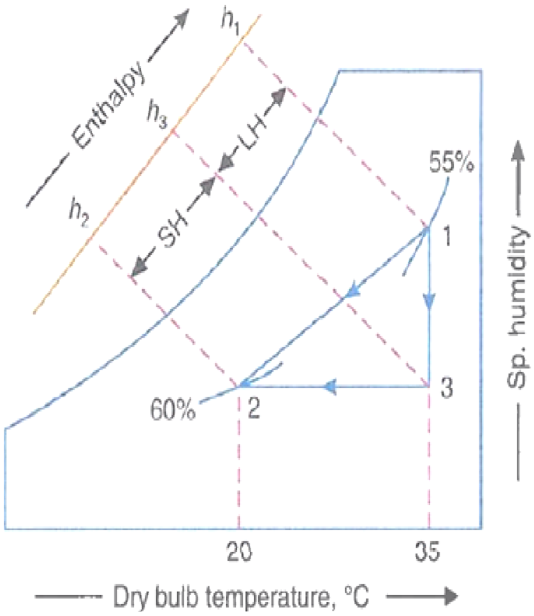
$$= 0.055 \times 60 = 3.3 \text{ kg / h } \textbf{Ans.}$$

<u>Example 7</u>: The amount of air supplied to an air conditioned hall is 300 m^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.

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

$\therefore$ Mass of air supplied,

$$m_a = \frac{v_1}{v_{s1}} = \frac{300}{0.9} = 333.3 \text{ kg / min}$$



Sensible heat removed from the air

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

$$h_1 = 85.8 \text{ kJ/kg of dry air}$$

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 fron the air,

$$SH = m_a (h_3 - h_2)$$
$$= 333.3 (57.4 - 42.2) = 5066.2 \text{ kJ/min} \textbf{ Ans.}$$

Latent heat removed from the air

We know that latent heat removed from the air,

$$LH = m_a (h_1 - h_3)$$
$$= 333.3 (85.8 - 57.4) = 9465.7 \text{ kJ/min} \textbf{ Ans.}$$

Sensible heat factor for the system

We know that sensible heat factor for the system,

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

Room Sensible Heat Factor:

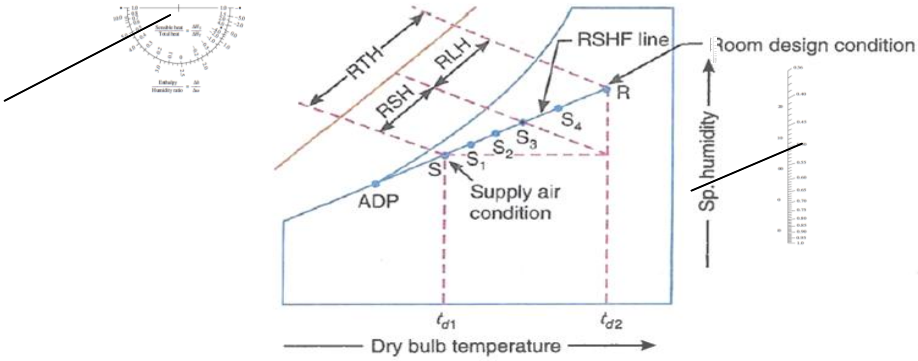
It is defined as the ratio of the room sensible heat to the room total heat. The room sensible heat factor,

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

Where:

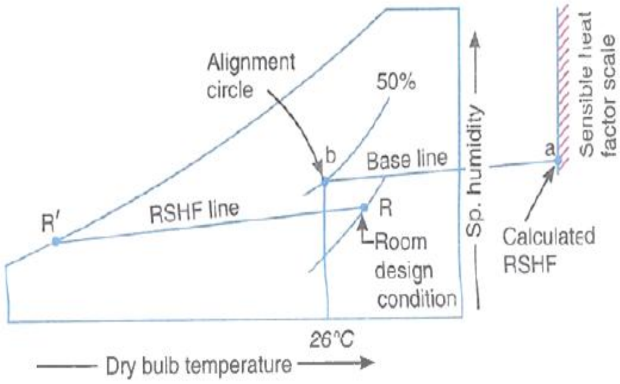
$$RSH = Room\ sensible\ heat$$
$$RLH = Room\ latent\ heat$$
$RTH = Room\ total\ heat$ = Room load

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 S_4, it is maximum of all the four points.



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. 18.15. The point a represents the calculated value of $RSHF$.

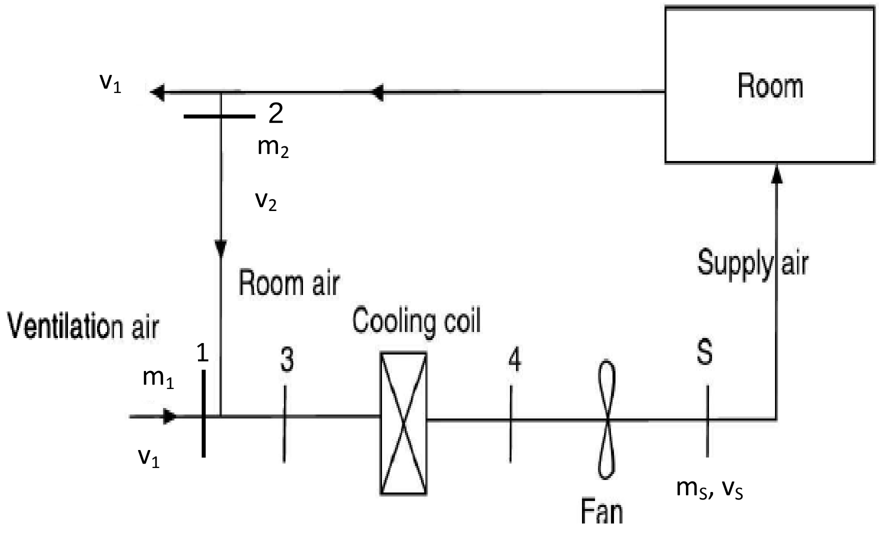


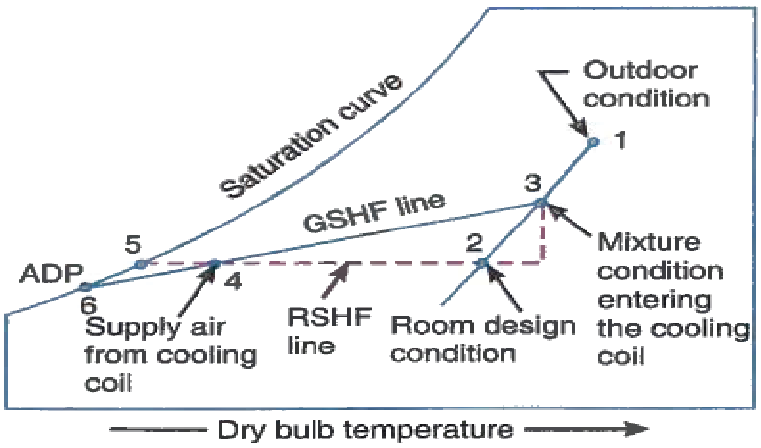
2. Join point a with the *alignment circle or the reference point b. The line ab is called base line.
3. Mark point R on the psychrometric chart to represent the room design conditions.
4. Through point R draw a line RR' parallel to the base line ab. This line is the required room sensible heat factor 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)*.

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, the grand sensible heat factors:





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

where TSH = Total sensible heat = $RSH + OASH$

TLH = Total latent heat = $RLH + OALH$

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

$= RSH + RLH + (OASH + OALH)$ = **Air conditioning Equipment load**

Let v_1 = Volume of outside air or ventilation in m³/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_1 = 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,

$$OASH = 0.020\,44\ v_1 (t_{d1} - t_{d2})\ \text{kW} \qquad \ldots \text{(Refer Art. 16.10)}$$

Outside air latent heat,

$$OALH = 50\ v_1(W_1 - W_2)\ \text{kW} \quad \ldots \text{(Refer Art. 16.13)}$$

and outside air total heat,

$$OATH = OASH + OALH$$ = **Ventilation load**

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

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

Generally, the air supplied to the air conditioning plant is a mixture of fresh air (or outside air or ventilation) and the recirculated air having the properties of room air. On the psychrometric chart, as shown in Fig. 18.16, 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. 18.16. This line, when produced up to the saturation curve, gives apparatus dew point (*ADP*).

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. 18.16, is the intersection of *GSHF* line and *RSHF* line. This point gives the ideal conditions for supply air to the room.

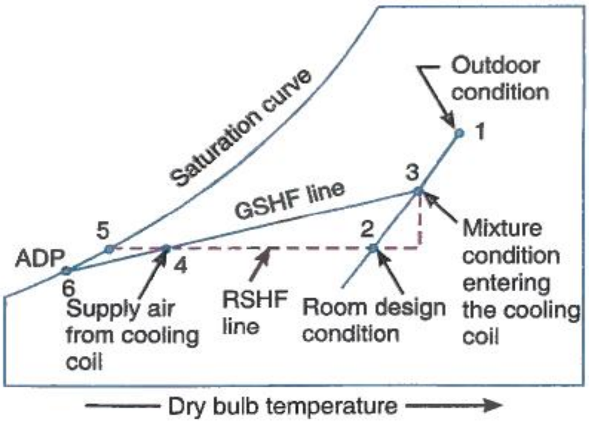


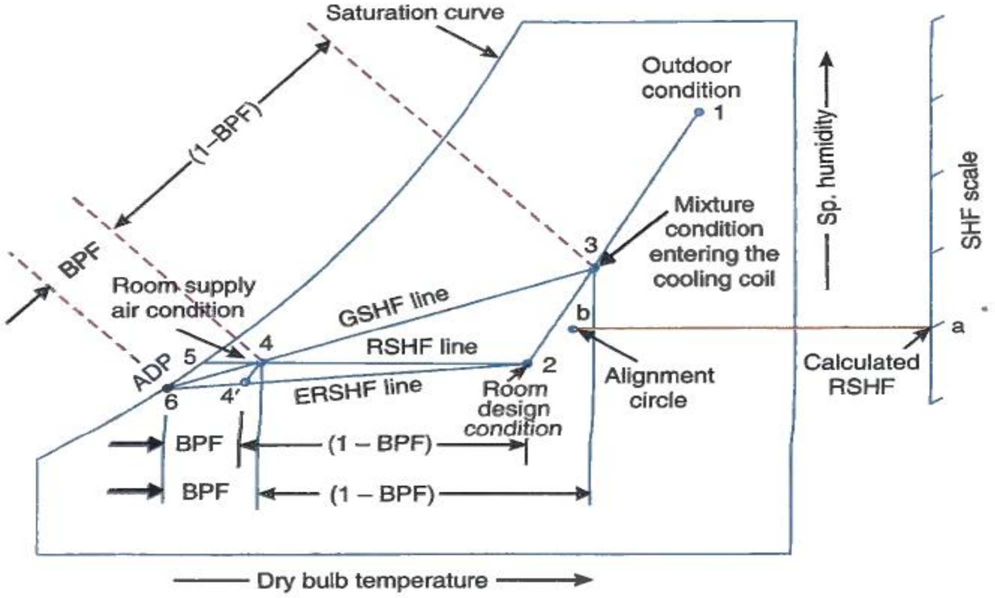
Fig. 18.16. Grand sensible heat factor.

Effective Room Sensible Heat Factor

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

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

where
$$\begin{aligned} ERSH &= \text{Effective room sensible heat} = RSH + OASH \times BPF \\ &= RSH + 0.020\,44\, v_1 (t_{d1} - t_{d2})\, BPF \\ ERLH &= \text{Effective room latent heat} = RLH + OALH \times BPF \\ &= RLH + 50\, v_1 (W_1 - W_2)\, BPF \\ ERTH &= \text{Effective room total heat} = ERSH + ERLH \\ BPF &= \text{By-pass factor} \end{aligned}$$



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 :

$$\begin{aligned} ERSH &= 0.020\,44\, v_d (t_{d2} - ADP)(1 - BPF)\ \text{kW} \\ ERLH &= 50\, v_d (W_2 - W_{ADP})(1 - BPF)\ \text{kW} \\ ERTH &= 0.02\, v_d (h_2 - h_{ADP})(1 - BPF)\ \text{kW} \end{aligned}$$

and

where v_d = Volume of dehumidified air to room or space in m³/min,

ADP = Apparatus dew point in °C,

W_{ADP} = Specific humidity at apparatus dew point in kg / 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

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

h_2 = Enthalpy of air at room condition, and

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

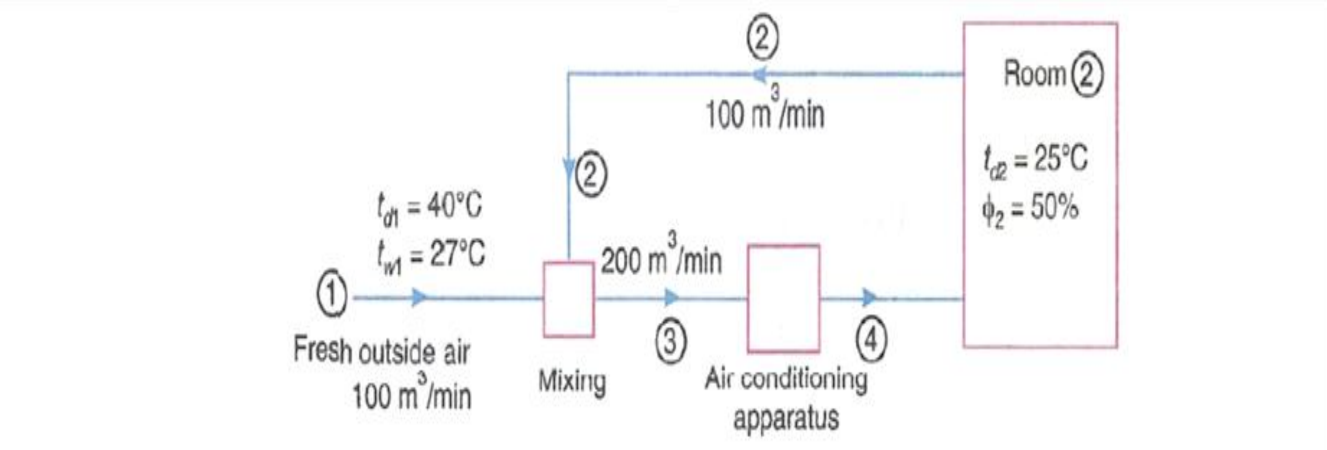
Example 8: In air conditioning system, the inside and outside conditions are dry bulb temperature 25 C, relative humidity 50% and dry bulb temperature 40C, wet bulb temperature 27C respectively. The room sensible heat factor is 0.8. 50% of the room air is rejected to atmosphere and an equal quantity of fresh air added before air enters the air conditioning apparatus. If the fresh air added is 100 m^3/min, determine:

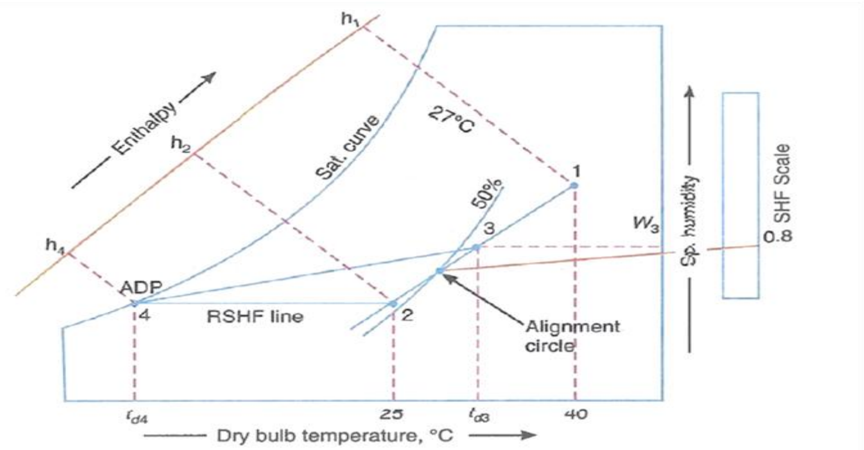
1-Room sensible and latent heat load

2-Sensible and latent heat load due to fresh air

3-Apparatus dew point

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





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

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,

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

$\therefore$ Room sensible heat load,

$$RSH = m_a\, c_{pm}\,(t_{d2} - t_{d4})$$
$$= 240 \times 1.022\,(25 - 11.8) = 3238 \text{ kJ / min}$$
$$= 3238/60 = 53.96 \text{ kJ/s or kW} \quad \textbf{Ans.}$$

... [$\because$ From psychrometric chart, $t_{d4} = 11.8$°C]

and room total heat load,

$$RTH = m_a\,(h_2 - h_4) = 240\,(50 - 33) = 4080 \text{ kJ/min}$$
$$= 4080/60 = 68 \text{ kJ/s or kW}$$

$\therefore$ Room latent heat load,

$$RLH = RTH - RSH$$
$$= 68 - 53.96 = 14.04 \text{ kW} \quad \textbf{Ans.}$$

2. Sensible and latent heat load due to fresh air

We know that mass of fresh air supplied,

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

$\therefore$ Sensible heat load due to fresh air

$$= m_F\, c_{pm}\, (t_{d1} - t_{d2})$$
$$= 120 \times 1.022\, (40 - 25) = 1840 \text{ kJ/min}$$
$$= 1840 / 60 = 30.67 \text{ kJ/s or kW} \quad \textbf{Ans.}$$

and total heat load due to fresh air

$$= m_F\,(h_1 - h_2) = 120\,(85.2 - 50) = 4224 \text{ kJ/min}$$
$$= 4224 / 60 = 70.4 \text{ kJ/s or kW}$$

$\therefore$ Latent heat load due to fresh air

$$= \text{Total heat load} - \text{Sensible heat load}$$
$$= 70.4 - 30.67 = 39.73 \text{ kW} \quad \textbf{Ans.}$$

3. Apparatus dew point

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

$$t_{d4} = 11.8°\text{C} \quad \textbf{Ans.}$$

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. 18.20. From the psychrometric chart, we find that humidity ratio corresponding to point 3,

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

and dry bulb temperature corresponding to point 3,

$$t_{d3} = 32.5°\text{C} \quad \textbf{Ans.}$$

<u>Example 9:</u> An air conditioned auditorium is maintained at 27 C DBT and 60% RH. The ambient conditions are 40 C DBT and 30 C WBT. The total sensible heat load is 100000 kJ/h and the total latent heat load is 40000 kJ/h. 60% of return air is recirculated and mixed with 40% of make-up after the cooling coil. The condition of air leaving the cooling coil is at 18 C. Calculate:

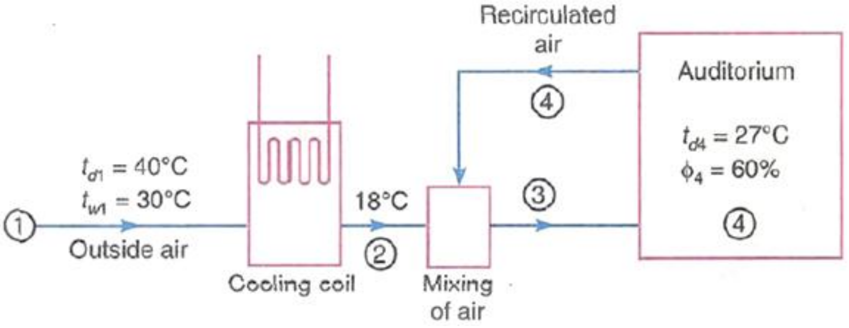
1-Room sensible heat factor
2-The condition of air entering the auditorium
3-The amount of make-up air
4-Apparatus dew point.
5- By-pass factor of the cooling coil.

We know that room sensible heat factor,

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

2. Condition of air entering the auditorium

The line diagram for processes involved in the air conditioning of an auditorium is shown in Fig. 18.25. These processes are shown on the psychrometric chart as discussed below :



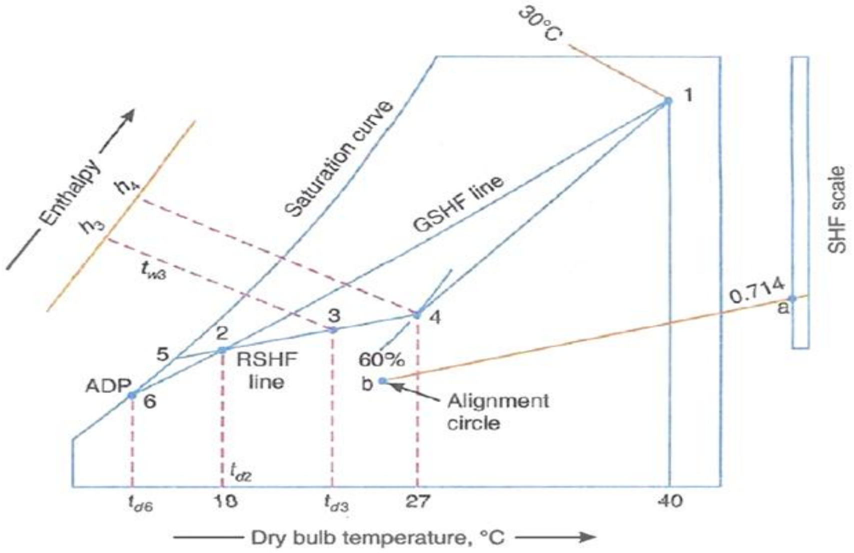


Fig. 18.26

We know that mass of supply air to the auditorium,

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, ϕ_3 = 72% **Ans.**

$$t_3 = \frac{m_2}{m_T}.t_2 + \frac{m_4}{m_T}.t_4$$

$$t_3 = 0.4.t_2 + 0.6.\ t_4$$

3. *Amount of make-up air*

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

$$h_4 = 61 \text{ kJ/kg of dry air}$$

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

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

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

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

$$= 0.4 \times 28\,000 = 11\,200 \text{ kg / h} \textbf{ Ans.}$$

4. *Apparatus dew point*

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

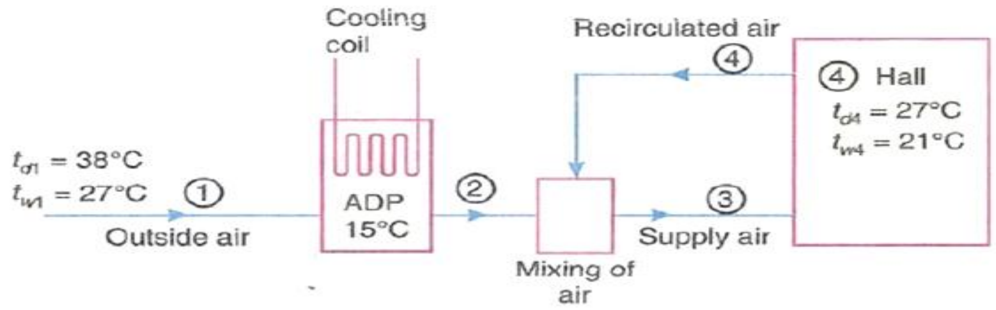
$$ADP = t_{d6} = 13° \text{ C} \textbf{ Ans.}$$

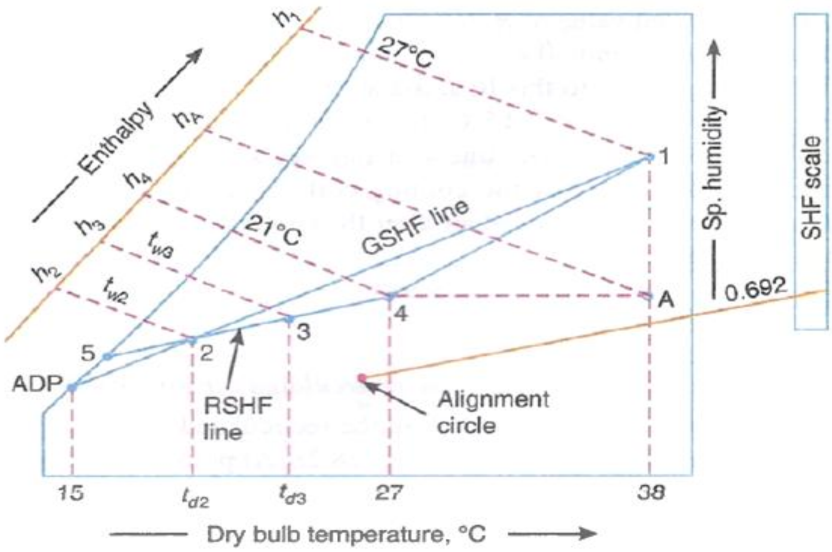
5. *By-pass factor of the cooling coil*

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

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

<u>Example10</u>: An air conditioned hall is to be maintained at 27 ˚C DBT and 21 ˚C WBT. 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 DBT and 27 C WBT is 25 m^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 quality of recirculated air from the hall is 60%. This quality is mixed with the conditioned air after the cooling coil. Determine 1- Condition of air after the coil and before the recirculated air mixes with it. 2- Condition of air entering the hall, i.e. after mixing with recirculated air, 3- Mass of fresh air entering the cooler, 4- By-pass factor of the cooling coil, 5-Refrigerating load on the cooling coil.





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,

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

$$= 336.23/60 = 5.6 \text{ kW}$$

$\therefore$ 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,

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

1. *Condition of air after the coil and before the recirculated air mixes with it*

The condition of air after the coil and before the recirculated air mixes with it is shown by point 2 on the psychrometric chart, as shown in Fig. 18.28. At point 2, we find that

Dry bulb temperature, $t_{d2} = 19°\text{C}$ **Ans.**

Wet bulb temperature, $t_{w2} = 17.5°\text{C}$ **Ans.**

2. *Condition of air entering the hall, i.e. after mixing with recirculated air*

The condition of air entering the hall, *i.e.* after mixing with recirculated air, is shown by point 3 on the psychrometric chart, as shown in Fig. 18.28. At point 3. we find that

Dry bulb temperature, $t_{d3} = 24°\text{C}$ **Ans.**

Wet bulb temperature, $t_{w3} = 19.8°\text{C}$ **Ans.**

$t_3 = \frac{m_2}{m_T}.t_2 + \frac{m_4}{m_T}.t_4$

$t_3 = 0.4.t_2 + 0.6.\ t_4$

3. *Mass of fresh air entering the cooler*

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

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

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

... (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,

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

5. *Refrigerating load on the cooling coil*

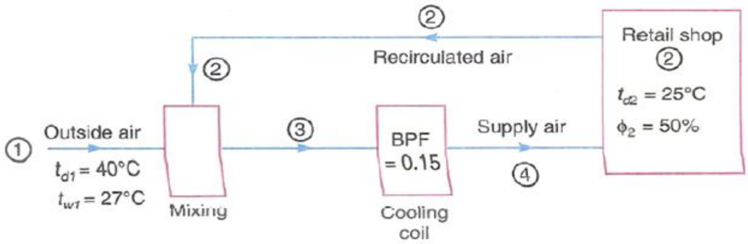
We know that the refrigerating load on the cooling coil

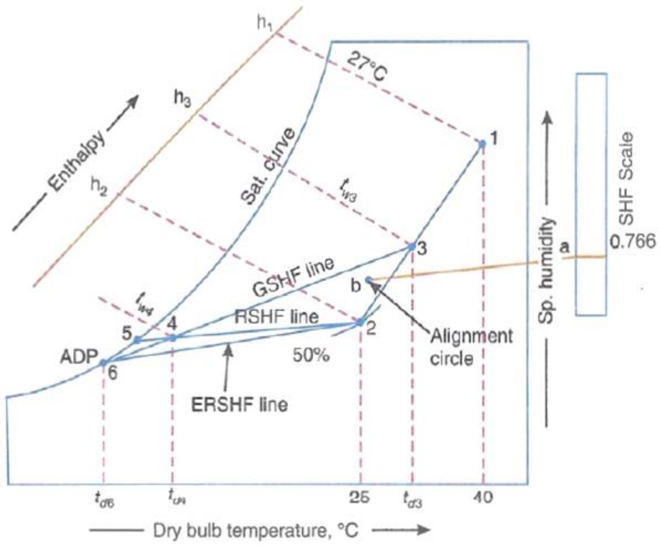
$$= m_F(h_1 - h_2) = 375(85 - 49) = 13\,500 \text{ kJ/min}$$

$$= 13\,500/210 = 64.3 \text{ TR} \quad \textbf{Ans.}$$

Example 11: A retail shop located in a city at 30 N latitude has the following loads: Room sensible heat = 58.15 kW, Room latent heat =14.54 kW. The summer outside and inside design conditions are: 40 C DBT, 27C WBT, and 25C DBT, 50% RH, respectively. 70 m^3/min of ventilation air is used. Determine the following, if the by-pass factor of the cooling coil is 0.15.
1- Ventilation load, 2- Grand total heat, 3- Effective room sensible heat factor, 4-Apparatus dew point, 5- Dehumidified air quantity 6- Condition of air entering and leaving the apparatus.

Solution. Given : $RSH = 58.15$ kW ; $RLH = 14.54$ kW ; $t_{d1} = 40°C$; $t_{w1} = 27°C$; $t_{d2} = 25°C$; $\phi_2 = 50\%$; $v_1 = 70$ m³/min ; $BPF = 0.15$





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

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

and enthalpy of air at point 2,

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

1. *Ventilation load*

We know that outside air sensible heat,

$$OASH = 0.02044\, v_1(t_{d1} - t_{d2})$$
$$= 0.02044 \times 70\,(40 - 25) = 21.46 \text{ kW}$$

and outside air total heat,

$$OATH = 0.02\, v_1\,(h_1 - h_2) = 0.02 \times 70\,(85.2 - 50) = 49.28 \text{ kW}$$

$\therefore$ Ventilation load $= OATH = 49.28$ kW **Ans.**

2. *Grand total heat*

We know that outside air latent heat,

$$OALH = OATH - OASH = 49.28 - 21.46 = 27.82 \text{ kW}$$

Total sensible heat,

$$TSH = RSH + OASH = 58.15 + 21.46 = 79.61 \text{ kW}$$

Total latent heat,

$$TLH = RLH + OALH = 14.54 + 27.82 = 42.36 \text{ kW}$$

We know that grand total heat,

$$GTH = TSH + TLH = 79.61 + 42.36 = 121.97 \text{ kW}$$ **Ans.**

3. *Effective room sensible heat factor*

We know that effective room sensible heat,

$$ERSH = RSH + OASH \times BPF$$
$$= 58.15 + 21.46 \times 0.15 = 61.37 \text{ kW}$$

and effective room latent heat,

$$ERLH = RLH + OALH \times BPF$$
$$= 14.54 + 27.82 \times 0.15 = 18.71 \text{ kW}$$

$\therefore$ Effective room sensible heat factor,

$$ERSHF = \frac{ERSH}{ERSH + ERLH} = \frac{61.37}{61.37 + 18.71} = 0.766 \text{ Ans.}$$

4. *Apparatus dew point*

Mark the calculated value of $ERSHF = 0.766$ 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 parallel to this line ab to intersect the saturation curve at point 6. From the psychrometric chart, we find that apparatus dew point,

$$ADP = t_{d6} = 11°\text{C Ans.}$$

5. *Dehumidified air quantity*

Let v_d = Volume of dehumidified air to room in m³/min.

We know that effective room sensible heat ($ERSH$),

$$61.37 = 0.02044\, v_d\, (t_{d2} - ADP)\, (1 - BPF)$$
$$= 0.02044\, v_d\, (25 - 11)\, (1 - 0.15) = 0.243\, v_d$$

$$\therefore \quad v_d = 61.37/0.243 = 253 \text{ m}^3/\text{min Ans.}$$

6. *Condition of air entering and leaving the apparatus*

We know that volume of recirculated air

$$= v_d - v_1 = 253 - 70 = 183 \text{ m}^3/\text{min}$$

Thus 183 m³/min of recirculated air is mixed with 70 m³/min of ventilation air. The mixing condition is shown at point 3, such that

$$\text{Length 2-3} = \text{Length 1-2} \times \frac{70}{253}$$

$$\boxed{\text{or } t_3 = \frac{m_1}{m_T} t_1 + \frac{m_2}{m_T} t_2 = \frac{70}{253} \times 40 + \frac{183}{253} \times 25 = 29}$$

From the psychrometric chart, we find that dry bulb temperature and wet bulb temperature of air entering the apparatus at point 3,

$$^*t_{d3} = 29°\text{C, and } t_{w3} = 20.7 \text{ °C Ans.}$$

Through point 3, draw a line 3-6 (known as GSHF line) and mark point 4 on this line such that

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

$$\boxed{BPF = \frac{t_{ADP} - t_4}{t_{ADP} - t_3}}$$

From the psychrometric chart, we find that dry bulb temperature and wet bulb temperature of air leaving the apparatus at point 4,

$$t_{d4} = 13.7° \text{ C, and } t_{w4} = 12.7 \text{ °C Ans.}$$

Notes : 1. The dry bulb temperature at point 4 may also be obtained as follows :

Or

$$RSH = 0.02044\, v_d\, (t_{d2} - t_{d4})$$
$$58.15 = 0.02044 \times 253\, (25 - t_{d4})$$
$$\therefore \quad t_{d4} = 13.76° \text{ C}$$

Example 12: **The following data refer to summer air conditioning of building:**

Outdoor conditions = 43 °C DBT and 27 °C WBT

Desired inside conditions = 25 °C DBT and 50% RH

Room sensible heat gain = 84000 kJ/h

Room latent heat gain = 21000 kJ/h

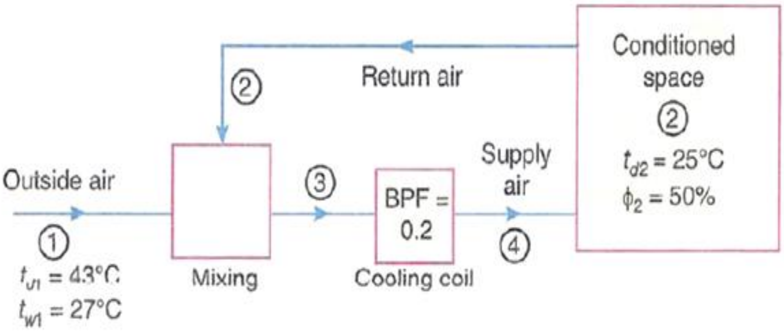
By-pass factor of the cooling coil used = 0.2

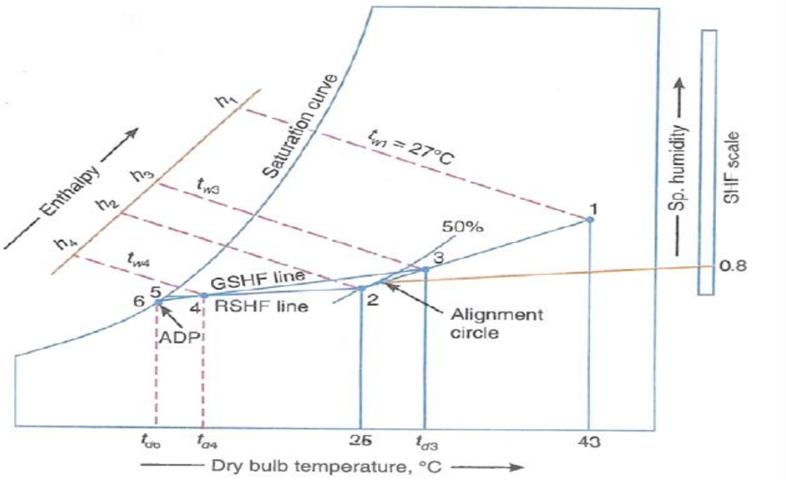
The return air from the room is mixed with the outside air before entry to cooling coil in the ratio of 4: 1 by mass. Determine:

A-apparatus dew point of cooling coil, B- Entry and exit conditions of air for cooling coil, C- Fresh air mass flow rate, and D- Refrigeration load on the cooling coil

Solution. Given : $t_{d1} = 43°C$; $t_{w1} = 27°C$; $t_{d2} = 25°C$; $\phi_2 = 50\%$; $RSH = 84\ 000$ kJ / h ; $RLH = 21\ 000$ kJ / h ; $BPF = 0.2$

The flow diagram for the conditioned space is shown in Fig. 18.33 and it is represented on the psychrometric chart as discussed below :





First of all, mark the outside condition of air at 43°C dry bulb temperature and 27°C wet bulb temperature on the psychrometric chart as point 1, as shown in Fig. 18.34. Now mark the inside conditions of air at 25°C dry bulb temperature and 50% relative humidity as point 2. We know that room sensible heat factor,

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{84\ 000}{84\ 000 + 21\ 000} = 0.8$$

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*). From point 2, draw a line 2-5 parallel to this line. This line 2-5 is called *RSHF* line. Since the return air from the conditioned space is mixed with outside air before entry to the cooling coil in the ratio of 4 : 1, therefore the condition of air entering the cooling coil after mixing process is marked on the line 1-2 by point 3, such that

$$\text{Length 2-3} = \frac{\text{Length 1-2}}{5}$$

$$t_3 = \frac{m_1}{m_T} t_1 + \frac{m_2}{m_T} t_2 = \frac{1}{5} t_1 + \frac{4}{5} t_2$$

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

$$\frac{\text{Length 4-6}}{\text{Length 3-6}} = BPF = 0.2$$

$$\boxed{BPF = \frac{t_6 - t_4}{t_6 - t_3} = 0.2}$$

(a) Apparatus dew point

The point 6 on the psychrometric chart is the apparatus dew point. By reading the value from the chart, we find that

Apparatus dew point (*ADP*)

$$= t_{d6} = 11°\text{C} \textbf{ Ans.}$$

Note : The apparatus dew point (*ADP* or t_{d6}) may be obtained by using any one of the following relations :

$$BPF = \frac{t_{d4} - t_{d6}}{t_{d3} - t_{d6}} = \frac{h_4 - h_6}{h_3 - h_6} = \frac{W_4 - W_6}{W_3 - W_6}$$

The value of t_{d3} as read from the psychrometric chart is 28.8° C. Using first relation, we have

$$0.2 = \frac{t_{d4} - t_{d6}}{28.8 - t_{d6}}$$

By trial and error, we find that

$$t_{d4} = 14.56°\text{ C and } t_{d6} = 11°\text{ C}$$

(b) Entry and exit conditions of air for cooling coil

The point 3 and point 4 represent the entry and exit condition of air for cooling coil as shown in Fig. 18.34. From the psychrometric chart, we find that

Dry bulb temperature of entering air,

$$t_{d3} = 28.8°\text{ C} \textbf{ Ans.}$$

Wet bulb temperature of entering air,

$$t_{w3} = 19.9°\text{ C} \textbf{ Ans.}$$

Dry bulb temperature of exit air,

$$t_{d4} = 14.5°\text{ C} \textbf{ Ans.}$$

and wet bulb temperature of exit air,

$$t_{w4} = 13°\text{ C} \textbf{ Ans.}$$

(c) ***Fresh air mass flow rate***

From the psychrometric chart, we find that enthalpy of air at point 2 ,

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

and enthalpy of air at point 4,

$$h_4 = 36.8\text{ kJ/kg of dry air}$$

We know that mass of dehumidified air or the total mass of air flowing,

$$m_a = \frac{\text{Room total heat}}{h_2 - h_4} = \frac{RSH + RLH}{h_2 - h_4}$$

$$= \frac{84\,000 + 21\,000}{50 - 36.8} = 7955\text{ kg / h}$$

Since this mass of air contains return air and fresh air in the ratio 4 : 1, therefore fresh air mass flow rate,

$$m_F = 7955 \times \frac{1}{5} = 1591\text{ kg / h} \textbf{ Ans.}$$

OBJECTIVE TYPE QUESTIONS

1. In summer air conditioning, the air is
   (*a*) cooled and humidified (*b*) cooled and dehumidified
   (*c*) heated and humidified (*d*) heated and dehumidified

2. In winter air conditioning, the air is
   (*a*) cooled and humidified (*b*) cooled and dehumidified
   (*c*) heated and humidified (*d*) heated and dehumidified

3. For summer air conditioning, the relative humidity should not be less than
   (*a*) 40% (*b*) 60% (*c*) 75% (*d*) 90%

4. For winter air conditioning, the relative humidity should not be more than
   (*a*) 40% (*b*) 60% (*c*) 75% (*d*) 90%

5. The sensible heat factor for auditorium or cinema hall is generally kept as
   (*a*) 0.6 (*b*) 0.7 (*c*) 0.8 (*d*) 0.9

6. The conditioned air supplied to the room must have the capacity to take up
   (*a*) room sensible heat load only (*b*) room latent heat load only
   (*c*) both room sensible heat and latent heat loads

7. The alignment circle is marked on the psychrometric chart at
   (*a*) 20°C *DBT* and 50% *RH* (*b*) 26°C *DBT* and 50% *RH*
   (*c*) 20°C *DBT* and 60% *RH* (*d*) 26°C *DBT* and 60% *RH*

8. The supply air state of cooling coil with a by-pass factor (*BPF*) lies at
   (*a*) intersection of *RSHF* line with saturation curve
   (*b*) intersection of *GSHF* line with saturation curve
   (*c*) point dividing *RSHF* line in proportions of *BPF* and (1-*BPF*)
   (*d*) intersection of *RSHF* line and *GSHF* line

9. The effective room sensible heat factor (*ERSHF*) is given by
   (*a*) $\dfrac{RSH}{RLH}$ (*b*) $\dfrac{RSH + OASH}{RLH + OALH}$
   (*c*) $\dfrac{RSH + OASH \times BPF}{(RSH + RLH) + (OASH + OALH)\ BPF}$ (*d*) none of these

   where RSH = Room sensible heat,
   RLH = Room latent heat,
   $OASH$ = Outside air sensible heat,
   $OALH$ = Outside air latent heat,
   BPF = By-pass factor.

10. In Fig. 18.37, line 2-6 is the effective room sensible heat factor (*ERSHF*) line, the line 2-5 is the room sensible heat factor (*RSHF*) line and the line 3-6 is the grand sensible heat factor (*GSHF*) line. Which of the following statements is correct ?

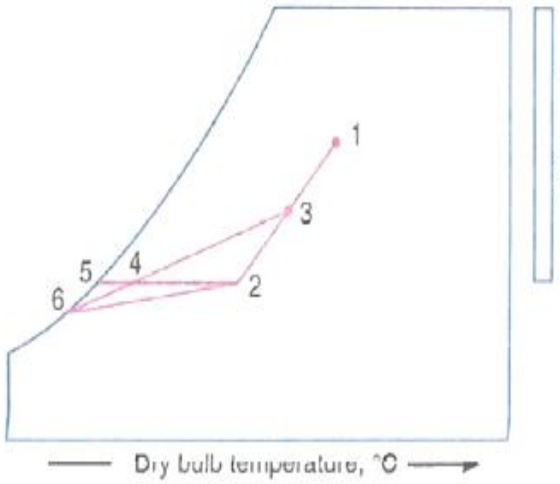


Fig. 18.37

(*a*) *GSHF* is greater than *RSHF* and *ERSHF* both

(*b*) *GSHF* is less than *RSHF* and *ERSHF* both

(*c*) *GSHF* is greater than *RSHF* but less than *ERSHF*

(*d*) *GSHF* is less than *RSHF* but greater than *ERSHF*

ANSWERS

1. (*b*)	2. (*c*)	3. (*b*)	4. (*a*)	5. (*b*)
6. (*c*)	7. (*b*)	8. (*d*)	9. (*c*)	10. (*d*)