



## Air Conditioning

### Syllabus

1. Properties of air and water
2. Psychrometric of air conditioning processes
3. Cooling load calculations
4. Calculation of heating load
5. Design of air distribution systems
6. Types of air distribution systems
7. Pipe sizing

### References

1. **Air conditioning engineering, fifth edition, by W.P. Jones**
2. **Principle of heating, ventilation and air conditioning with worked examples, by Nihal E Wijesundera**
3. **Refrigeration and air conditioning, by W.F. Stoecker**
4. **هندسة تكييف الهواء والتثليج, تأليف د. خالد أحمد الجودي**

### 1.1 Air Conditioning Concept

In the hot weather regions especially in the south of Iraq, the air conditioning for the majority of peoples maybe just (air cooling) because of the long summer months throughout the year. Air conditioning means the control of an atmospheric environment for the comfort of humans. The controlling of environments includes the following main parameters:

1. Air temperature
2. Relative humidity
3. Air motion
4. Contaminants (pollutants)



The air temperature and relative humidity are the most important in air conditioning and so they will be studied intensively here.

## 1.2 Properties of Moist Air

Atmospheric air is a mixture of many gases plus water vapor and pollutants. It consists of five main gases as shown in the table below:

Gas	Molecular weight	Volume fraction
<u>Nitrogen</u>	28.02	78.03
<u>Oxygen</u>	32	20.99
Argon	39.91	0.94
<u>Carbon dioxide</u>	44	<u>0.03</u>
Hydrogen	2.02	0.01

The mixture of these gases is called *dry air*. the molecular weight of the dry air can be calculated as follows:

$$M = 28.02 \times 0.7803 + 32 \times 0.2099 + 39.91 \times 0.0094 + 44 \times 0.0003 + 2.02 \times 0.0001 = 28.97$$

Most air involves a mixture of dry air and water vapor. The amount of water vapor may vary from zero to a maximum depending on the temperature and pressure of the mixture.

The molecular weight of water vapor  $H_2O$  can be estimated as:

$$M = 2 \times 1.01 + 1 \times 16 = 18.02 \text{ kg/kmol}$$

## 1.3 Fundamentals Parameters

Moist air up to about three atmospheres pressure obeys the perfect gas law with sufficient accuracy for most engineering calculations. The Dalton law for a mixture of perfect gases states that the mixture pressure is equal to the sum of the partial pressures of the constituents:



$$P = P_1 + P_2 + P_3 + \dots$$

The pressure of moist air is:

$$P = P_{N_2} + P_{O_2} + P_{Ar} + P_{CO_2} + P_{H_2} + P_V$$

The total pressure of moist air is the sum of the partial pressures of the dry air and the water vapor:

$$P = P_a + P_v$$

$P$  : Total pressure of moist air

$P_a$ : Pressure of dry air

$P_v$ : pressure of vapor

### Example 1.1:

Saturated air at 1 atmospheric pressure and temperature of 26°C. Calculate the pressure of the vapor and dry air.

### Solution:

From the saturated steam table, the saturation pressure corresponding to the temperature 26°C is equal to 3.36kPa.

$$P_v = 3.36 \text{ kPa}$$

$$P = P_a + P_v$$

$$P_a = P - P_v$$

*The pressure of the dry air is:*



$$P_a = 101.325 - 3.36 = 97.965 \text{ kPa}$$

**Dry bulb temperature (DBT):** It is the air temperature recorded or measured by a thermometer. Its denoted by  $T$  or  $T_{db}$  or **DBT**.

**Wet-bulb temperature (WBT):** It is the lowest temperature of air recorded by a thermometer when its bulb is covered with wet wick and exposed to an air stream. It's denoted by  $T_w$  or  $T_{wb}$  or **WBT**.

**Dew point temperature (DPT):** It is the *temperature* at which air is saturated with water vapor. It is denoted by  $T_{dp}$  or **DPT**.

Dew point temperature can be calculated from the approximate equation as follows:

$$T_{dp} = \frac{4030(T_{db} + 235)}{4030 - (T_{db} + 235) \ln \phi} - 235$$

$T_{dp}$  : dew point temperature °C

$T_{db}$  : dry-bulb temperature °C

$\phi$  : relative humidity (in fraction)

## 1.4 Pressure of Water Vapor in Moist Air

The vapor pressure of saturated air can be found directly from the steam table at a given temperature or can be estimated approximately as follows:

$$P_{sat} = Epx \left[ 25.317 - \frac{5144}{T_{sat}} \right]$$

Where;



$P_{sat}$  in Pa  
 $T_{sat}$  in K

In the case of unsaturated air, the vapor pressure can be calculated from the following relation:

$$P_v = P_{sw} - P_B A (T_d - T_w)$$

$P_v$  Vapor partial pressure

$P_{sw}$  Saturated vapor pressure at wet bulb temperature

$P_B$  Barometric pressure

$A$  Constant which is given as:

$$A = 6.66 \times 10^{-4} \text{ } ^\circ\text{C}^{-1} \quad T_w \geq 0 \text{ } ^\circ\text{C}$$

$$A = 5.94 \times 10^{-4} \text{ } ^\circ\text{C}^{-1} \quad T_w < 0 \text{ } ^\circ\text{C}$$

$T_d$  Dry bulb temperature  $^\circ\text{C}$

$T_w$  Wet bulb temperature  $^\circ\text{C}$

### Example 1.2:

Calculate the partial pressure of water vapor and dew point of moist air with 20  $^\circ\text{C}$  dry bulb temperature and 15  $^\circ\text{C}$  wet bulb temperature under barometric pressure of 95kPa.

### Solution:

From the steam table, the saturation pressure at WBT is equal to 1.707 kPa, so the partial pressure of the vapor is:

$$\begin{aligned} P_v &= 1.707 - 95 \times 6.66 \times 10^{-4} (20 - 15) \\ &= 1.391 \text{ kPa} \end{aligned}$$



From the steam table dew point temperature can be evaluated which is equal to the saturation temperature corresponding to the pressure 1.391;

$$P=1.227 \text{ kPa} \quad \text{at} \quad T=10 \text{ }^{\circ}\text{C}$$

$$P=1.4014 \text{ kPa} \quad \text{at} \quad T=12 \text{ }^{\circ}\text{C}$$

Using interpolation:

$$T_{dp} = 12 - \frac{12-10}{1.4014-1.227} (1.4014 - 1.391) = 11.88 \text{ }^{\circ}\text{C}$$

Or by using the equation:

$$P_{sat} = Epx \left[ 25.317 - \frac{5144}{T_{sat}} \right]$$

$$1391 = Epx \left[ 25.317 - \frac{5144}{T_{sat}} \right]$$

$$\ln(1391) = \left[ 25.317 - \frac{5144}{T_{sat}} \right]$$

$$T_{sat} = [25.317 - \ln(1391)]^{-1} \times 5144 = 284.53\text{K} = 11.53 \text{ }^{\circ}\text{C}$$



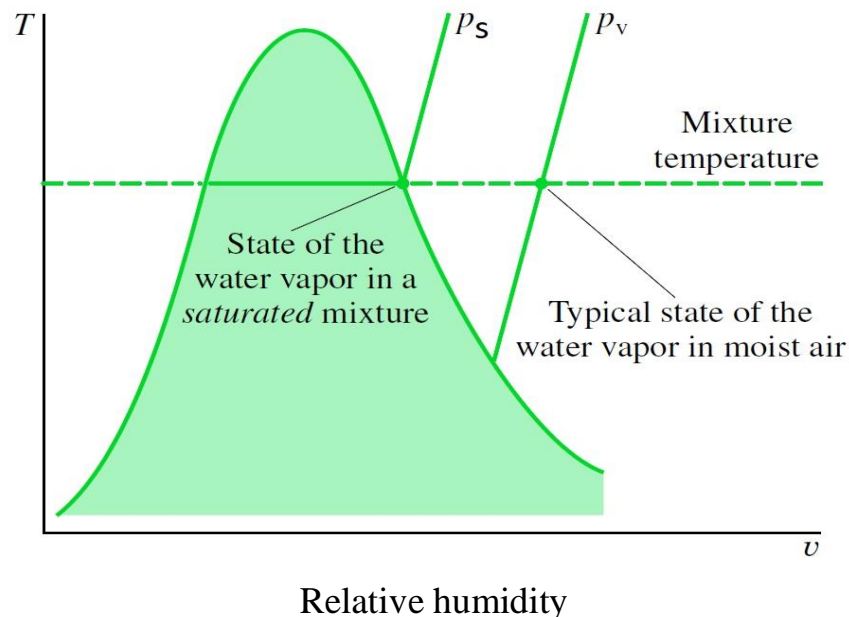
## 2.1 Relative humidity (RH or $\phi$ )

It is the ratio of the mass of water vapor in a given volume of air at any temperature and pressure to the maximum amount of mass of water vapor that the same volume of air can hold at the same temperature conditions.

The air contains the maximum amount of water vapor at the saturation conditions.

The relative humidity can be defined as the ratio of the partial pressure of the vapor  $P_v$  in the mixture to the saturation pressure  $P_s$ :

$$RH = \frac{P_v}{P_s}$$





The relative humidity can also be defined in term of density or mole fraction of the moist air as follows:

$$RH = \frac{\rho_v}{\rho_s} = \frac{v_s}{v_v} = \frac{x_v}{x_s}$$

$\rho_v$ : density of moist air

$\rho_s$  : density of moist air at saturation conditions

$x_v$  : mole fraction of moist air

$x_s$  : mole fraction of moist air at saturation conditions

$v_v$  : specific volume of moist air

$v_s$  : specific volume of moist air at saturation condition

### Example2.1:

Moist air at 20 °C(DBT) and 15 °C (WBT) and atmospheric pressure 95kPa. Calculate the vapor density and relative humidity of the moist air.

### Solution:

The partial pressure of the water vapor in the moist air is calculated from the equation:

$$P_v = P_{sw} - P_B A (T_d - T_w)$$

$$P_{sw} = 1.707 \text{ kPa} \quad \text{at } 15 \text{ }^\circ\text{C (WBT)}$$

$$\begin{aligned} P_v &= 1.707 - 95 \times 6.66 \times 10^{-4} (20 - 15) \\ &= 1.391 \text{ kPa} \end{aligned}$$





From ideal gas law the vapor density can be estimated as follows:

$$\rho_v = \frac{P_v}{RT_v} = \frac{1.391}{0.4615 (20+273)} = 0.0103 \text{ kg/m}^3$$

To calculate relative humidity, the saturation pressure 20 °C(DBT) can be obtained from the steam table:

$$P_s = 2.337 \text{ kPa}$$

$$RH = \frac{P_v}{P_s} = \frac{1.391}{2.337} = 0.595 = 59.5\%$$

## 2.2 Moisture content(humidity ratio)

It is the ratio of the mass of water vapor to the mass of dry air contained in the sample air. It is denoted by  $w$ .

$$w = \frac{\text{mass of water vapor in air}}{\text{mass of dry air}} \text{ kg/kg of dry air}$$

$$w = \frac{m_v}{m_a}$$

The ideal gas equation can be applied to dry air and water vapor.

$$P_a V_a = m_a R_a T$$

$$P_v V_v = m_v R_v T$$



The mixture of dry air and water vapor at the same temperature and occupying the same volume

$$\frac{P_v}{P_a} = \frac{m_v R_v}{m_a R_a}$$

$$W = \frac{m_v}{m_a} = \frac{P_v R_a}{P_a R_v}$$

$$R_a = 0.287 \text{ kJ/Kg K (dry air)}, R_v = 0.4615 \text{ kJ/Kg K (vapor)}$$

$$W = \frac{0.287 P_v}{0.4615 P_a}$$

$$W = 0.622 \frac{P_v}{P_a}$$

$$W = 0.622 \frac{P_v}{P_B - P_v} \quad \text{kg of water vapor/kg of dry air}$$

### Example 2.2:

Moist air at 20 °C(DBT) and 15 °C (WBT) and atmospheric pressure 95kPa. Calculate the moisture content in the air.

### Solution:

$$P_v = P_{sw} - P_B A (T_d - T_w)$$

$$P_{sw} = 1.707 \text{ kPa at } 15 \text{ °C (WBT)}$$

$$\begin{aligned} P_v &= 1.707 - 95 \times 6.66 \times 10^{-4} (20 - 15) \\ &= 1.391 \text{ kPa} \end{aligned}$$



The moisture content is:

$$w = 0.622 \frac{P_v}{P_B - P_v} = 0.622 \frac{1.391}{95 - 1.391} = 0.00924 \frac{kg}{kgda}$$

### Example 2.3:

Calculate the moisture content of air at 60 percent relative humidity when the temperature is 30°C. The barometric pressure is the standard value of 101.325 kPa.

**Solution:**

$$w = 0.622 \frac{P_v}{P_B - P_v} = 0.622 \frac{RH \times P_s}{P_B - RH \times P_s}$$

The pressure of saturated air at 30°C is 4.241 kPa

$$w = 0.622 \frac{0.6 \times 4.241}{101.325 - 0.6 \times 4.241} = 0.0160 \text{ kg/kgda}$$

## 2.3 Specific volume of moist air

It is the volume of 1kg of dry air and water vapor mixture. In this mixture, each constituent occupies the same volume and the same temperature, but each exerts its partial pressure.

### Example 2.4:

Calculate the specific volume of air at 20 °C (DBT) and a moisture content of 0.00734 kg/kg of dry air at a Prometric pressure of 95 kPa.



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### Solution:

the water vapor occupies the same volume and temperature as the moist air. The volume of the vapor can be calculated using the gas equation:

$$P_v V_v = m_v R_v T$$

$$V_v = \frac{m_v R_v T}{P_v}$$

The partial pressure of the vapor in the moist air can be estimated as:

$$w = 0.622 \frac{P_v}{P_B - P_v}$$
$$0.00734 = 0.622 \frac{P_v}{95 - P_v}$$
$$P_v = 1.1080 \text{ kPa}$$

$$V_v = \frac{0.00734 \times 0.461 \times (20 + 273)}{1.1080} = 0.895 \text{ m}^3$$

Or by using the gas equation for dry air:

$$V_a = \frac{m_a R_a T}{P_a} = \frac{1 \times 0.287 \times (20 + 273)}{95 - 1.1080} = 0.896 \text{ m}^3$$

## 2.4 Enthalpy

Heat is supplied to gas in two ways: at constant volume or constant pressure. In a constant volume process, no work will be done by the gas, and the internal energy of the gas increases. If a heat exchange occurs at constant pressure, as well as a change in internal energy taking place, work may be done. This leads to the definition of enthalpy H:



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$$H=U+PV \quad , \quad J$$

Or specific enthalpy for a unit mass of the working fluid:

$$h=u+pv \quad , \quad J/kg$$

The enthalpy of moist air is the summation of the enthalpy of 1 kg of dry air and enthalpy of water vapor in the moist air.

$$h = h_a + w h_v$$

$h$	enthalpy of moist air	kJ/kg
$h_a$	enthalpy of dry air	kJ/kg
$h_v$	enthalpy of water vapor	kJ/kg
$w$	moisture content	kg/kg of dry air

The enthalpy of dry air  $h_a$  over the range  $0\text{ }^\circ\text{C}$  to  $60\text{ }^\circ\text{C}$  is given as:

$$h_a = 1.007 T - 0.026 \quad \text{kJ/kg}$$

For the lower temperature, down to  $-10\text{ }^\circ\text{C}$ , the approximate equation is:

$$h_a = 1.005 T$$

And the enthalpy of the water vapor is:

$$h_v = 2501 + 1.84 T \quad \text{kJ/kg}$$

The enthalpy of moist air at barometric pressure (101.325 kPa) becomes:

$$h = 1.007 T - 0.026 + w( 2501 + 1.84 T) \quad \text{kJ/kg}$$



$T$  : temperature of moist air °C

### Example 2.5:

Calculate the enthalpy of moist air at a dry bulb temperature of 24 °C and relative humidity of 40%. The barometric pressure is 101.325kPa.

### Solution:

$$w = 0.622 \frac{P_v}{P_B - P_v} = 0.622 \frac{RH \times P_s}{P_B - RH \times P_s}$$

From the steam table, the saturation pressure corresponding to 24 °C is 2.982kPa or can use the following equation:

$$P_{sat} = \text{Exp} \left[ 25.317 - \frac{5144}{T_{sat}} \right] = \text{exp} \left[ 25.317 - \frac{5144}{(24 + 273)} \right] = 2972.43 \text{ Pa}$$
$$= 2.972 \text{ kPa}$$

$$w = 0.622 \frac{RH \times P_s}{P_B - RH \times P_s} = 0.622 \frac{0.4 \times 2.982}{101.325 - 0.4 \times 2.982}$$
$$= 0.0074 \text{ kg/kgda}$$

$$h = 1.007 T - 0.026 + w(2501 + 1.84 T)$$

$$h = 1.007 \times 24 - 0.026 + 0.0074(2501 + 1.84 \times 24)$$

$$h = 43.00 \text{ kJ/kg}$$



### 3.1 Degree of saturation( $\mu$ )

It is the ratio of moisture content at moist air to the moisture content of saturation air at the same temperature.

$$\mu = \frac{w}{w_s}$$

$$\mu = \frac{w}{w_s} = \frac{0.622 \frac{P_v}{P_B - P_v}}{0.622 \frac{P_s}{P_B - P_s}} = \frac{P_v}{P_s} \frac{P_B - P_s}{P_B - P_v}$$

$$\mu = RH \frac{P_B - P_s}{P_B - P_v}$$

For **humid air**, the numerical value of the degree of saturation  $\mu$  very closed to that of relative humidity **RH** . In the case of dry or saturation air, both  $\mu$  and **RH** are equal.

#### **Example 3.1:**

Calculate the degree of saturation and relative humidity of moist air at a dry bulb temperature of 40 °C and a wet-bulb temperature of 20 °C. The barometric pressure is 101.325kPa.

#### **Solution:**

The saturation pressures corresponding to 40 °C and 20 °C are 7.375 kPa and 2.337 kPa respectively.

The water vapor pressure is calculated as follows:

$$P_v = P_{sw} - P_B A (T_d - T_w)$$



$$P_v = 2.337 - 101.325 \times 6.66 \times 10^{-4} (40 - 20) = 0.987 \text{ kPa}$$

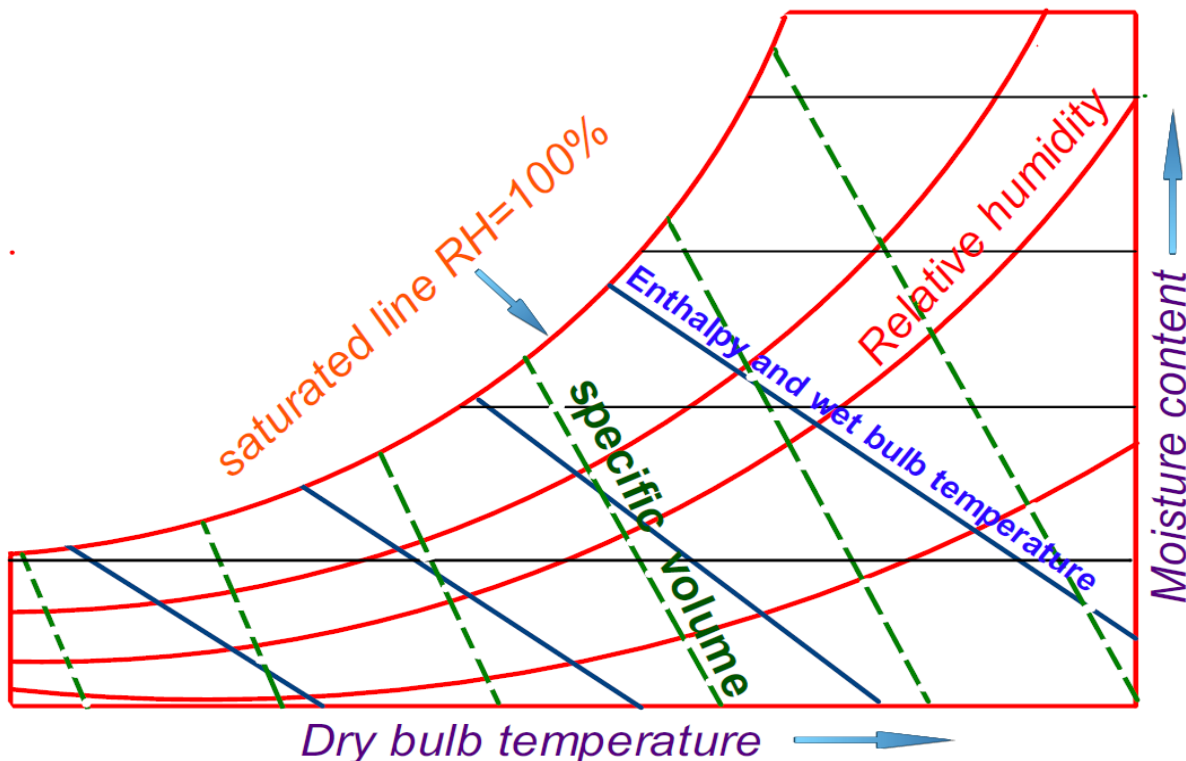
$$RH = \frac{P_v}{P_s} = \frac{0.987}{7.375} = 0.134 = 13.4\%$$

$$\text{Degree of saturation } \mu = RH \frac{P_B - P_s}{P_B - P_v}$$

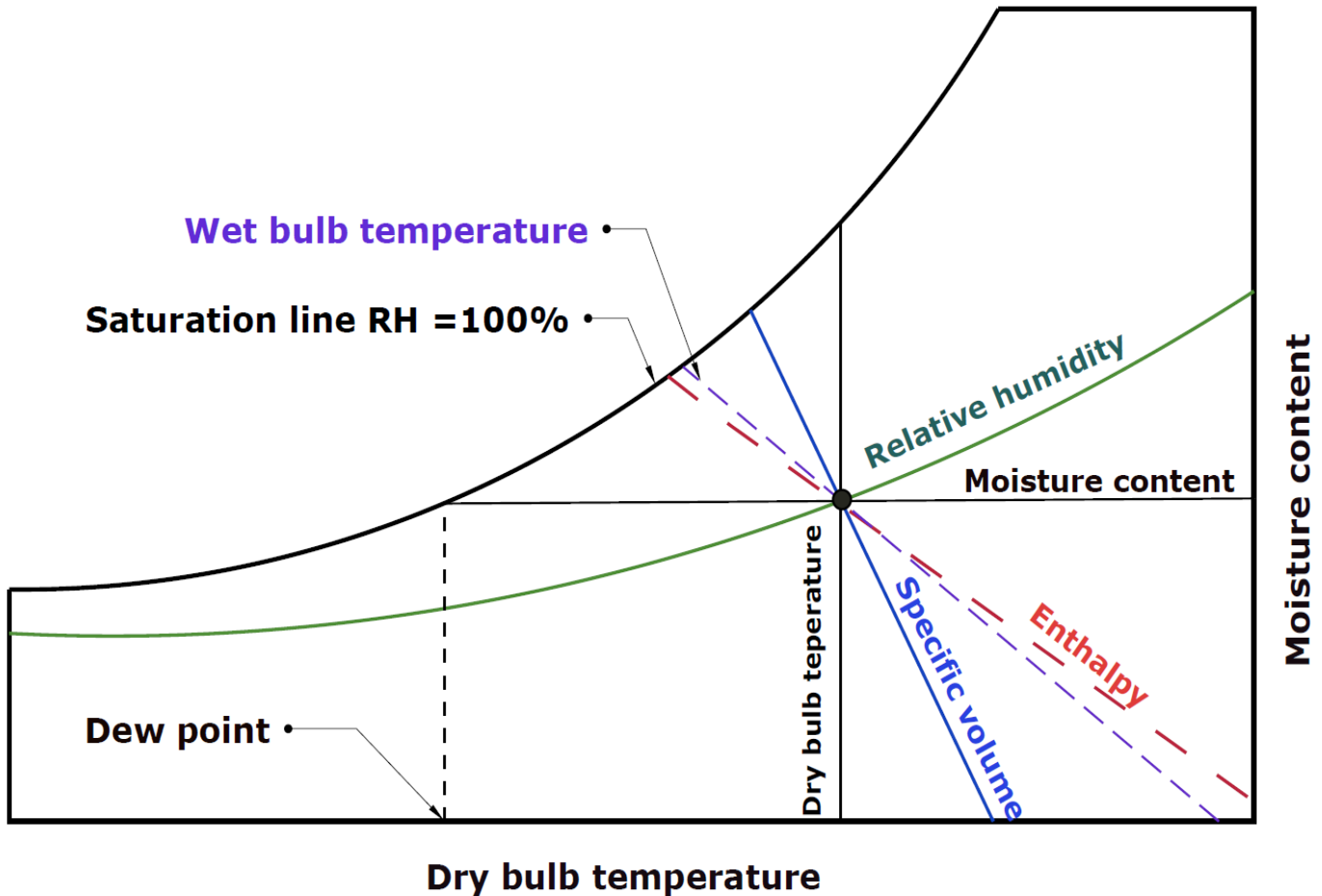
$$\mu = 0.134 \frac{101.325 - 7.375}{101.325 - 0.987} = 0.1255 = 12.55\%$$

### 3.2 Psychrometric chart

Psychrometric chart is a graphical representation of the thermodynamics of humid air like moisture content, enthalpy, specific volume, relative humidity, dew point temperature, and wet bulb temperature. These properties vary with atmospheric pressure and altitude. The psychrometric chart for standard atmospheric pressure 101.325kPa at sea level will be considered here.

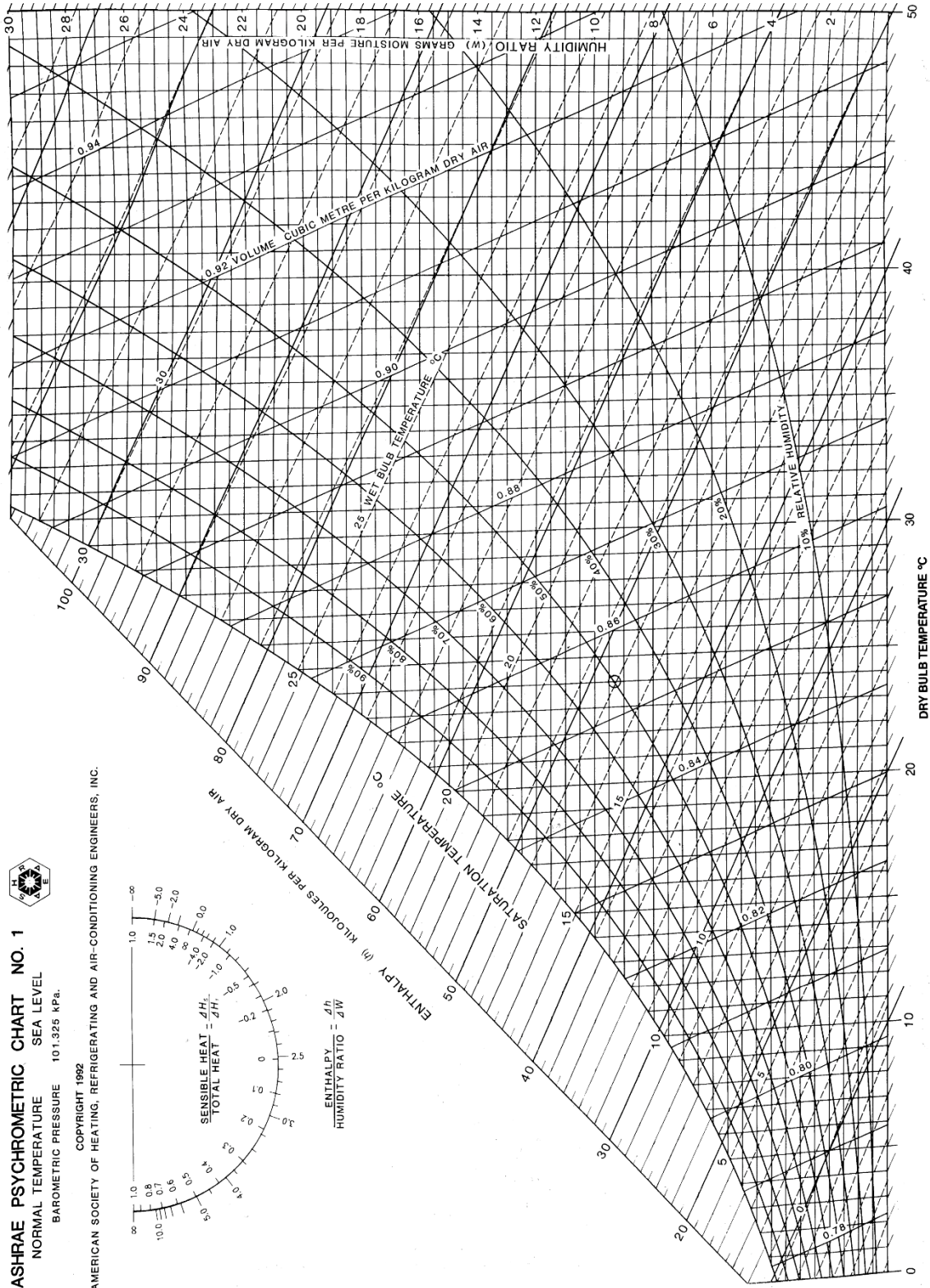






In constructing the psychrometric chart, two properties of moist air are taken as basic coordinates. The two common choices for this purpose are moisture content and dry-bulb temperature (*W-DBT*) or any two properties like DBT and RH or DBT and WBT...etc.

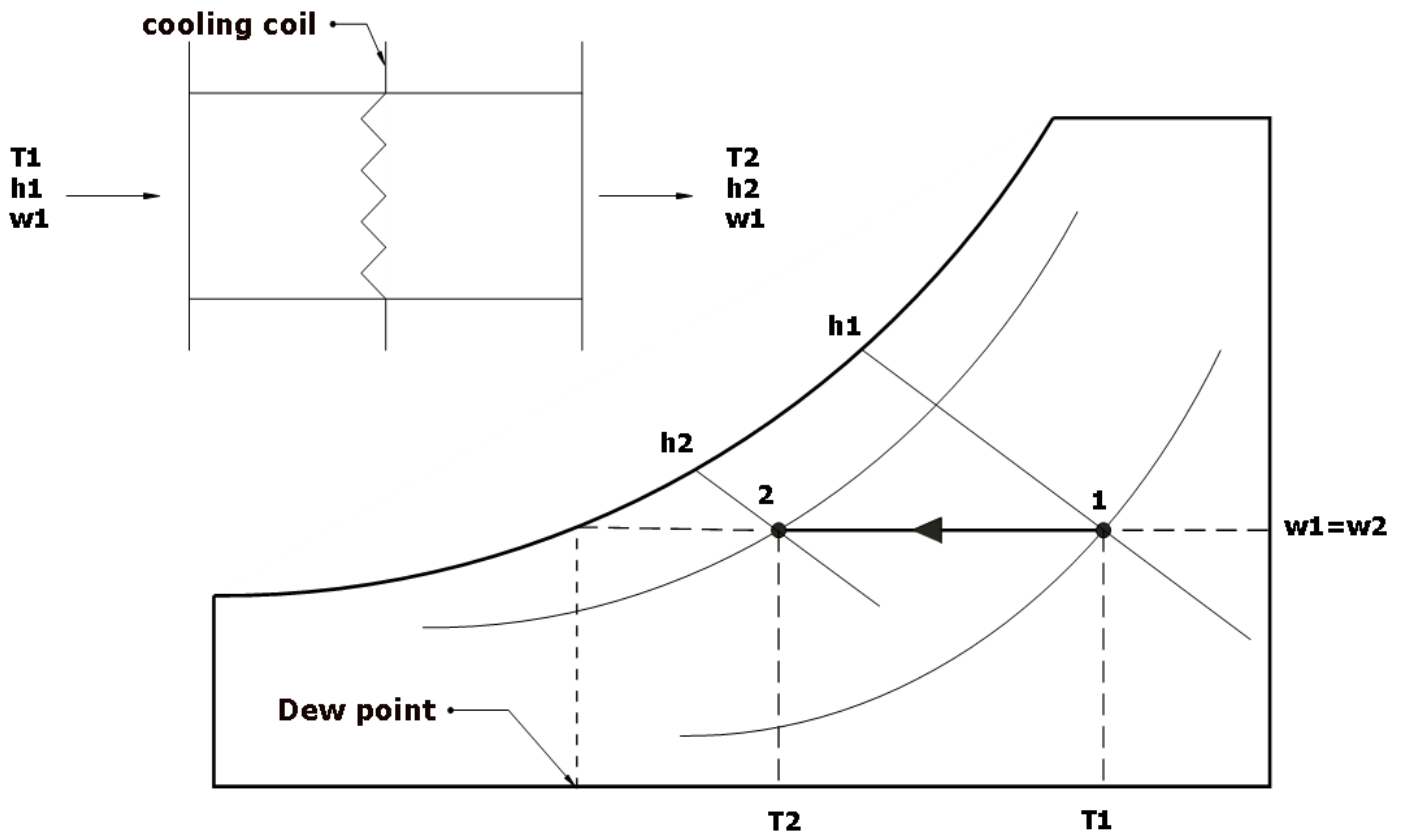
The above figure explained a state point and all the variables on the psychrometric chart.



### 3.3 Air conditioning processes

#### 1. Sensible cooling

Sensible cooling of humid air can be done by passing it over a cooling coil whose surface temperature is kept below the dry bulb temperature of entering air and above the dew point temperature. No moisture is added or removed and dew point temperature remains constant throughout the sensible cooling process.





The sensible cooling during the process is given by:

$$Q = \dot{m}(h_1 - h_2)$$

Or

$$Q = \dot{m} C_p (T_1 - T_2)$$

$C_p$  for moist air is given as:

$$C_p = C_{pa} + wC_{pv}$$

$C_{pa}$  : specific heat for dry air

$C_{pv}$  : specific heat for water vapor

If we take an approximate value for  $C_{pa} = 1.013 \text{ kJ/kgK}$  and  $C_{pv} = 1.89 \frac{\text{kJ}}{\text{kg}} \text{K}$ , the specific heat of moisture air becomes 1.013-1.026 kJ/kg K depending on the moist air conditions. If accuracy not very important, an approximate value of  $C_p$  can be taken as 1.02 kJ/kg K. The change in enthalpy ( $\Delta h$ ) is preferred for all air conditioning processes.

### Example 3.2:

Humid air at 40 °C (DBT) and 20 °C(WBT) flow over a cooling coil at a rate of 5000 m<sup>3</sup>/hr. It sensible cooled to 26 °C. Determine:

- Final WBT of air
- The total heat transferred

### Solution:



The point 1 is located on the chart at the intersection of 40 °C (DBT) and 20 °C(WBT) lines. The value of enthalpy  $h_1 = 57.5$  kJ/kg and  $h_2 = 41.5$  kJ/kg. specific volume of air  $v_1 = 0.896$  m<sup>3</sup>/kg and WBT at outlet = 15.5 °C.

The total heat transfer rate is:

$$Q = \dot{m}(h_1 - h_2) = \frac{\dot{V}}{v} (h_1 - h_2) = \frac{5000}{0.897} (57.5 - 41.5) = 24.774 \text{ kW}$$

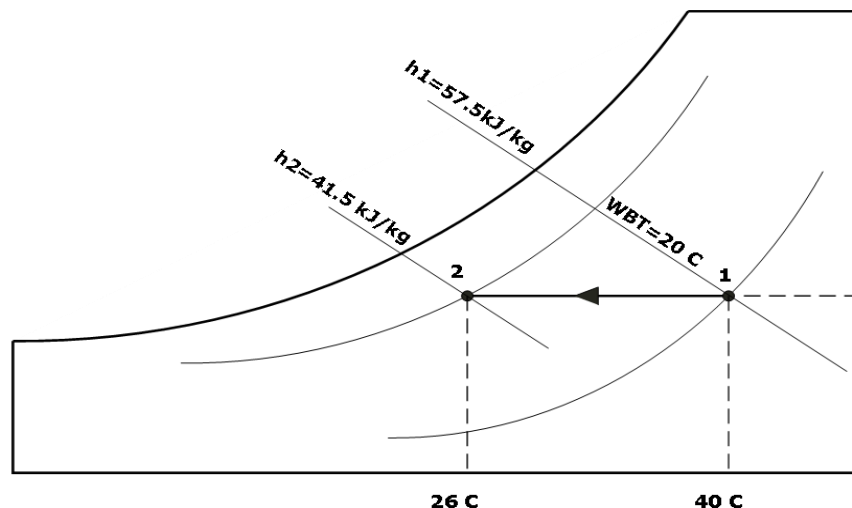
Or can be calculated approximately as:

$$Q = \dot{m} C_p (T_1 - T_2) = \frac{5000}{0.897} \times 1.02 \times (40 - 26) = 22.11 \text{ kW}$$

The error percentage between the accurate and approximated value is:

$$ERR = \frac{24.774 - 22.11}{24.774} = 0.108 = 10.8\%$$

The error percentage is somewhat large. So it is highly recommended to use  $Q = \dot{m}(h_1 - h_2)$  instead of  $Q = \dot{m} C_p (T_1 - T_2)$  to calculate accurate heat transfer rate for all the psychrometric processes.

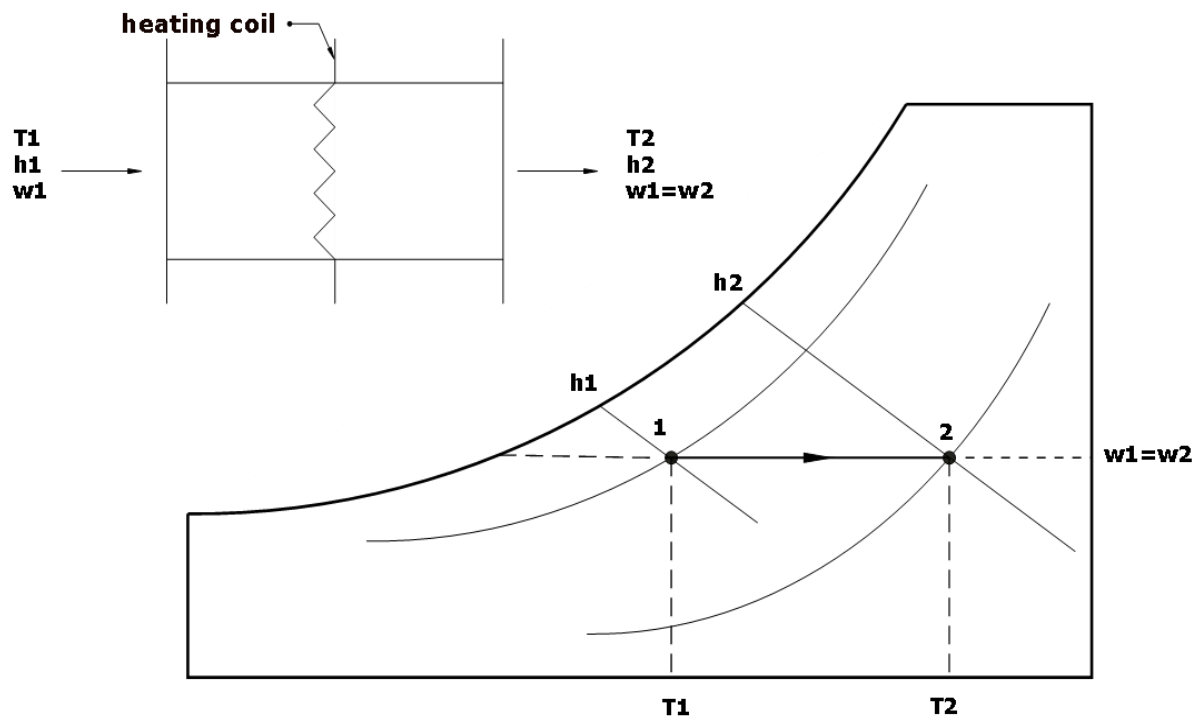


## 4.1.Sensible heating

Sensible heat will be added to the humid air when passing it over a hot dry surface. The hot surface temperature is above the dry bulb temperature of the air. The moisture content remains constant during sensible heating (see the figure below).

The heat added to the air can be estimated as follows:

$$Q = \dot{m}(h_1 - h_2)$$



**Sensible heating**

Or can be calculated approximately as:

$$Q = \dot{m} C_p (T_2 - T_1)$$



### Example 4.1:

Saturated air at 10 °C flows over a heating coil at a rate of 50 m<sup>3</sup>/min. air leaves the coil at 40 °C. sketch the process on a psychrometric chart and determine the following:

- The wet-bulb temperature of the air
- The sensible heat transferred
- The total heat transferred

### Solution:

Point 1 is located on the saturation curve (RH=100%). The properties of saturated air can be calculated from the psychrometric chart as follows:

$$h_1 = 29.7 \frac{kJ}{kg}, w_1 = w_2 = 0.0077 \frac{kg}{kgda}, v_1 = 0.812 m^3/kg$$

The enthalpy at point 2 is  $h_2 = 60.2 \frac{kJ}{kg}$

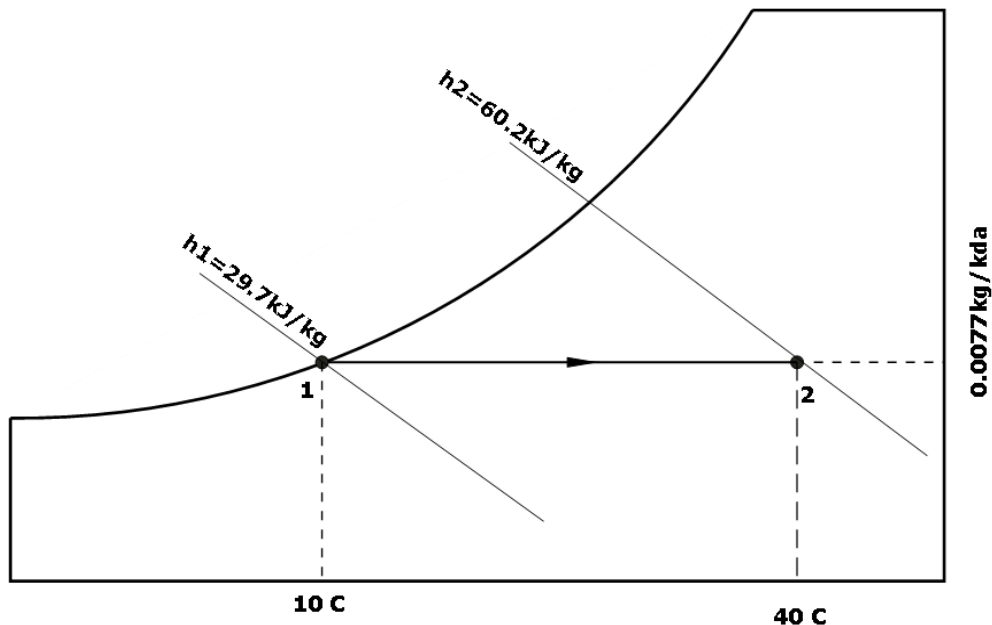
a)  $WBT_1 = DBT_1 = 10 \text{ }^\circ\text{C}$  (Saturated state which means RH=100%)

$$WBT_2 = 20.2^\circ\text{C}$$

b) sensible heat transfer rate  $Q = \dot{m}(h_1 - h_2)$

$$Q = \frac{50}{60 \times 0.812} (60.2 - 29.7) = 31.301 \text{ kW}$$

c) The total heat is equal to the sensible heat as no change in the moisture content during the sensible heating.



## 4.2 Cooling and Dehumidification of Humid Air

Here the air is to be cooled and during cooling, water vapor is to be separated from the air. Moisture separation (condensation) will occur only when the moist air is cooled to a temperature below its dew point temperature. The effective surface temperature of the cooling coil kept below the initial dew point temperature of the air is called **apparatus dew point ADP or  $T_c$** . The cooling and dehumidification process is shown in the figure.

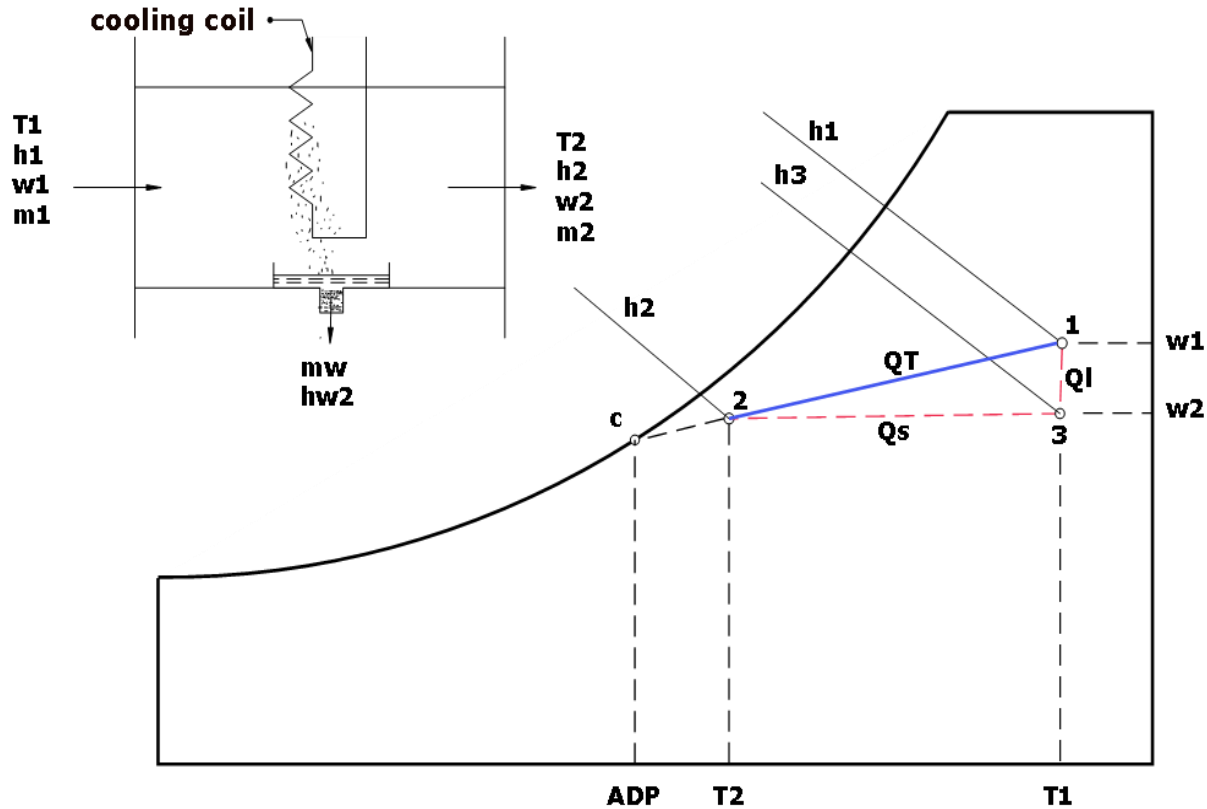
The energy and mass balance equations of the cooling and dehumidification process are:

$$m_1 h_1 = m_2 h_2 + Q_{1-2} + m_w h_{w2}$$

$h_{w2}$  is the enthalpy of saturated water at the surface temperature of the cooling coil.

$$m_2 = m_1 - m_w$$





$m_w$  is very small in comparison with  $m_1$ , therefore  $m_2 = m_1 = m$  is considered approximately correct.

$$m h_1 = m h_2 + Q_{1-2} + m_w h_{w2}$$

$$m w_1 = m w_2 + m_w$$

$$m_w = m(w_1 - w_2)$$



$$Q_{1-2} = m(h_1 - h_2) - (w_1 - w_2)h_{w2}$$

The term  $(w_1 - w_2)h_{w2}$  is small in comparison with the term  $(m(h_1 - h_2))$ , so it can be neglected and the coil cooling load can be written as:

$$Q_{1-2} = m(h_1 - h_2)$$

The cooling and dehumidification process involves sensible and latent heat transfer processes. The total heat transfer throughout the cooling and dehumidification process can be given as:

$$Q_T = Q_S + Q_L$$

$$Q_T = m(h_1 - h_2)$$

$$Q_S = m(h_3 - h_2) \approx m C_p(T_3 - T_2)$$

$$Q_L = m(h_1 - h_3) = m(w_1 - w_2)h_{fg}$$

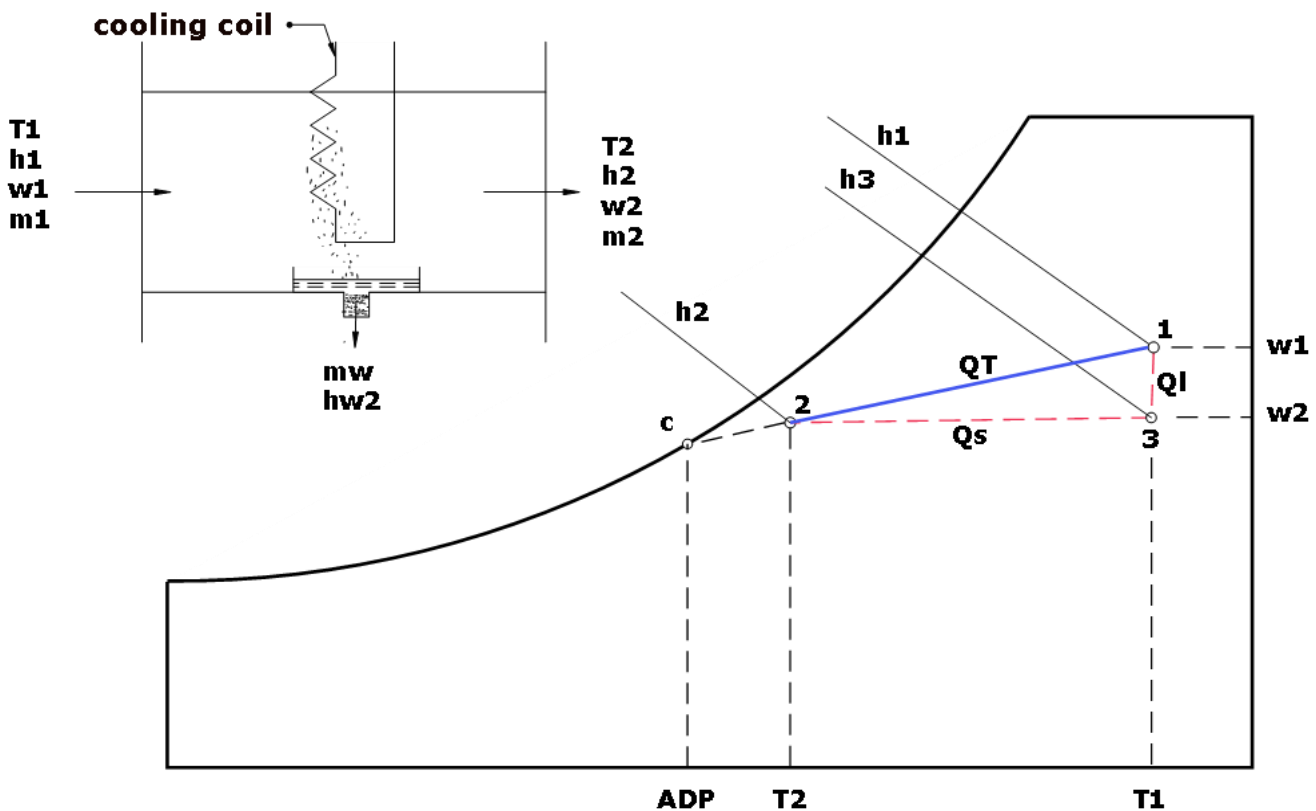
### 4.3 Bypass Factor of the Coil

If all air following over the coil comes into contact with the cooling or heating surface of the coil for a long time, then the air temperature leaving the coil and the coil surface temperature will be the same. Practically air does not remain in contact with the coil surface for a long time and also some part of the air does not contact

the coil surface. That is means some air will bypass the coil. Therefore, the temperature of the outlet air is higher than the cooled surface (cooling process) or lower than the hot surface (heating process). The bypass factor can be written as:

$$BF = \frac{\text{Temperature, enthalpy, moisture content differences that is not achieved}}{\text{Temperature, enthalpy, moisture content differences that could be achieved}}$$

$$BF = \frac{h_2 - h_c}{h_1 - h_c}$$





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In the case of humidification or dehumidification processes the bypass factor can be given as follows:

$$BF = \frac{w_2 - w_c}{w_1 - w_c}$$

Also, the bypass factor can be expressed in term of temperature as:

$$BF = \frac{T_2 - T_c}{T_1 - T_c}$$

The contact factor **CF** is equal:

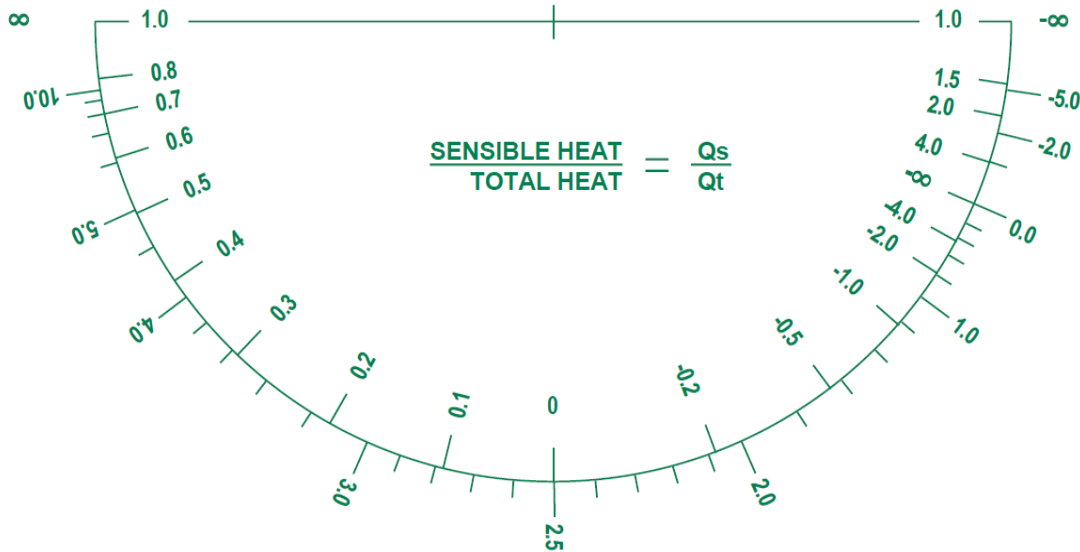
$$CF = 1 - BF$$

$$CF = \frac{h_1 - h_2}{h_1 - h_c} = \frac{w_1 - w_2}{w_1 - w_c} = \frac{T_1 - T_2}{T_1 - T_c}$$



## 5.1 Sensible heat ratio SHR

It is defined as the sensible heating or cooling load divided by the total heating or cooling load.



$$SHR = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{Q_s}{Q_s + Q_L}$$

**SHR** is **1.0** in case of no latent heat transfer and **SHR** equal to **0** for no sensible heat transfer.

$$SHR = \left. \begin{array}{l} 0 \\ 0 < SHR < 1 \\ 1 \end{array} \right\} \begin{array}{l} \text{no sensible heat} \\ \text{sensible + latent heat} \\ \text{no latent heat} \end{array}$$



**Example 5.1:**

Humid air at 32 °C DBT and 50% RH enters a cooling coil at 2.778 m<sup>3</sup>/s. If the air leaving the coil should be at 20 °C DBT and 18°C WBT, determine:

- The surface temperature of the coil
- Bypass factor of the coil
- Sensible heat ratio of the coil
- Total heat removed from the air
- Mass of water vapor condensed on the coil.

**Solution:**

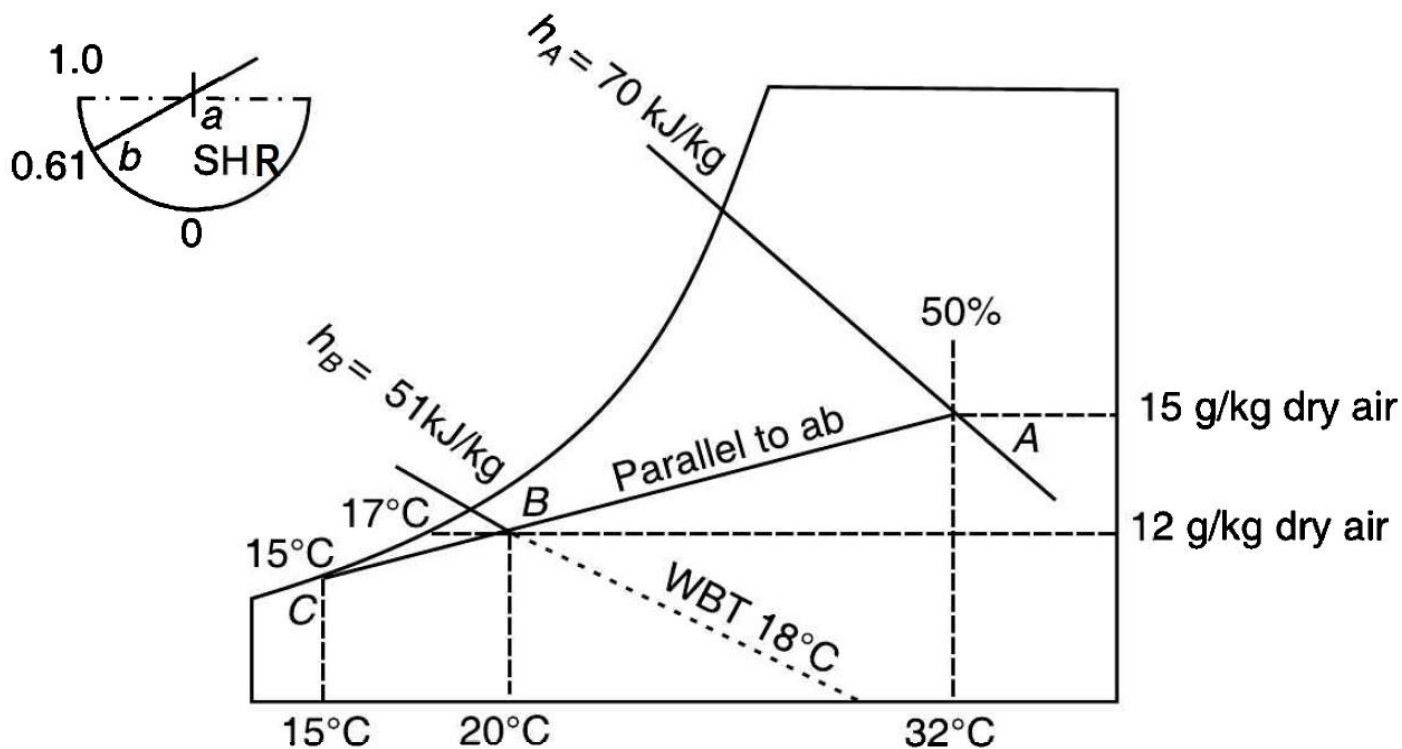
**a)**

Extend the line **AB** to meet the 100% RH curve at point **C** which indicates the coil surface temperature of 15 °C. (See the figure)

**b)** the coil bypass factor is:

$$BF = \frac{T_B - T_c}{T_A - T_c} = \frac{20 - 15}{32 - 15} = 0.29$$

**c)** To get the sensible heat ratio of the cooling coil, draw a line parallel to the process line **AB** through the center of the **protractor** give on the chart. Then read the **SHR** scale on the **protractor**, which is **0.61** in this case.



d)

The specific volume of the moist air at the inlet conditions (32 °C DBT and 50% RH) is 0.885 m<sup>3</sup>/kg, therefore the total heat removed from the air is:

$$Q_T = m(h_A - h_B) = \frac{V}{v}(h_A - h_B) = \frac{2.778}{0.885} (70 - 51) = 59.662 \text{ kW}$$

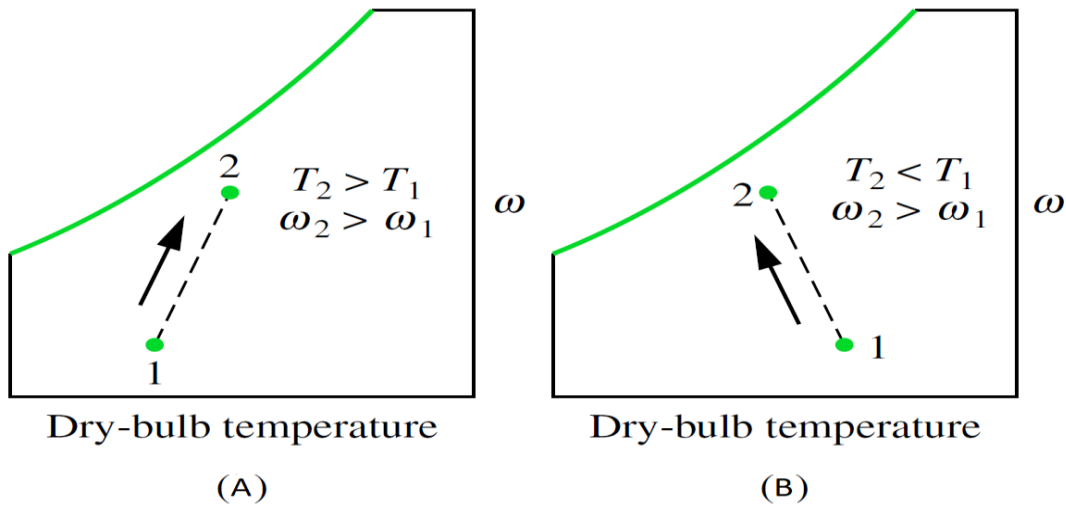
e)

mass of water vapor condensed :

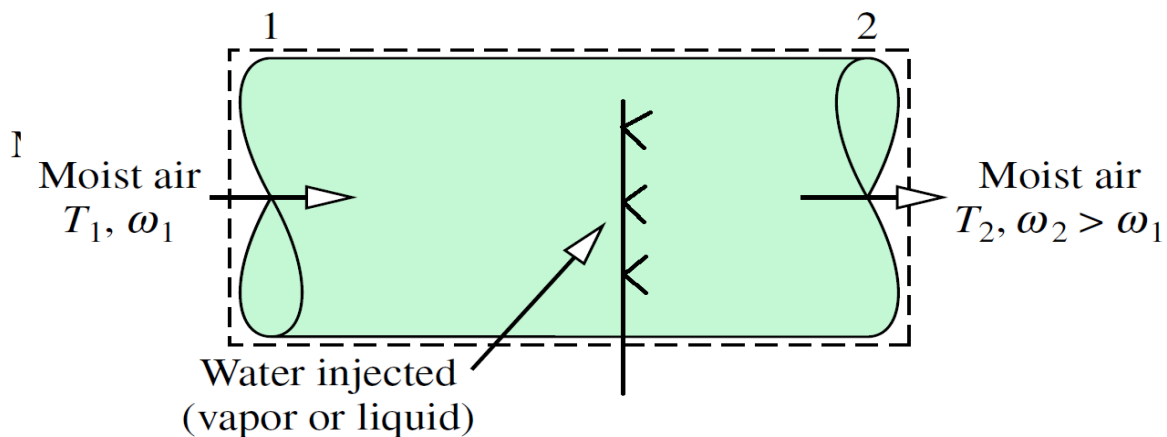
$$m_w = m(w_A - w_B) = \frac{2.778}{0.885} (0.015 - 0.012) = 0.0094 \text{ kg/s}$$

## 5.2 Humidification

Some time needs to increase the moisture content in the supplied air to the conditioned space. One way to accomplish this is to inject steam or water liquid into the air. The temperature of the air leaving the humidifier depends on the conditions of the water introduced. When relatively high-temperature steam is injected, both moisture content and dry bulb temperature would be increased. If water liquid was injected, the moist air may exit the humidifier with a lower temperature than the inlet as shown in the figures.



(A) Steam injected. (B) Liquid injected.



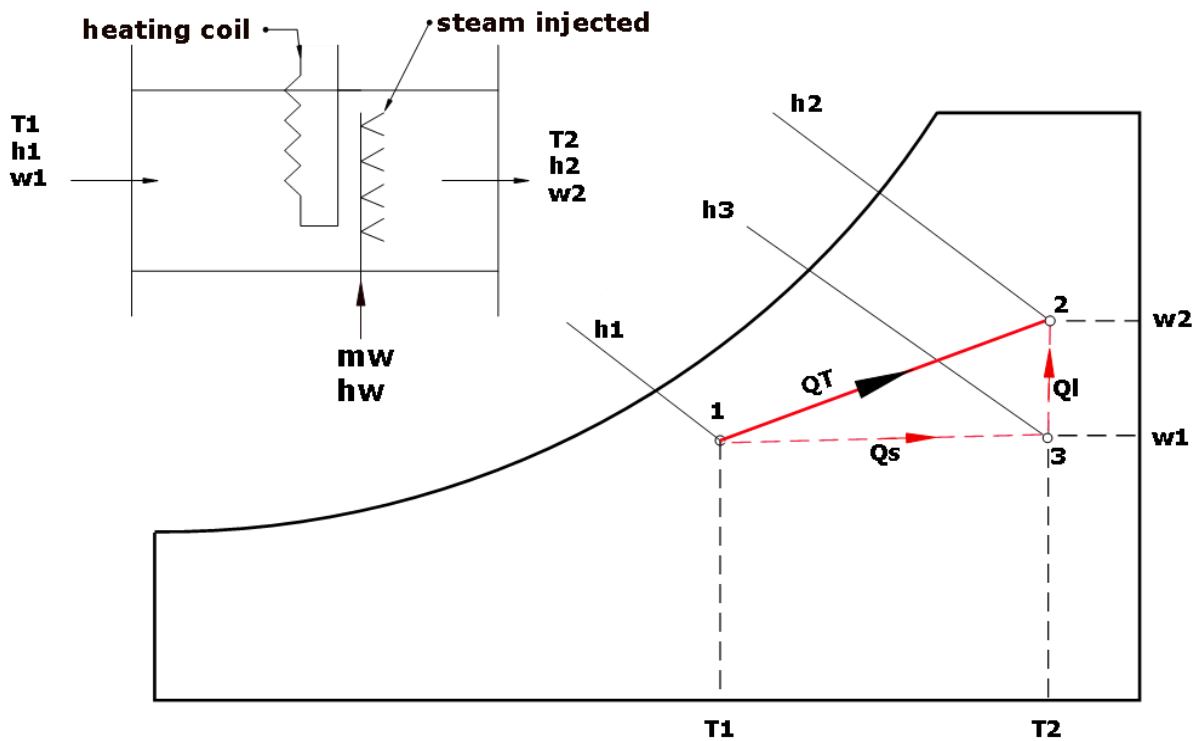


### 5.3 Heating and Humidification

During the heating process in Winter, the relative humidity of the warm air decreases, and then an indoor thermal discomfort happens. Water vapor is added to the air stream through steam nozzles to increase the relative humidity for comfort and health.

The mass of steam added during the humidification process can be given as follows:

$$m_w = m(w_2 - w_1)$$



**Heating and Humidification process**



The energy balance through the heating and humidification process shown in the figure is:

$$m h_1 + m_w h_w + Q = m h_2$$

The heat transfer during the process is written as:

$$Q = m(h_2 - h_1) - m_w h_w$$

From the mass balance, the mass of injected water vapor is :

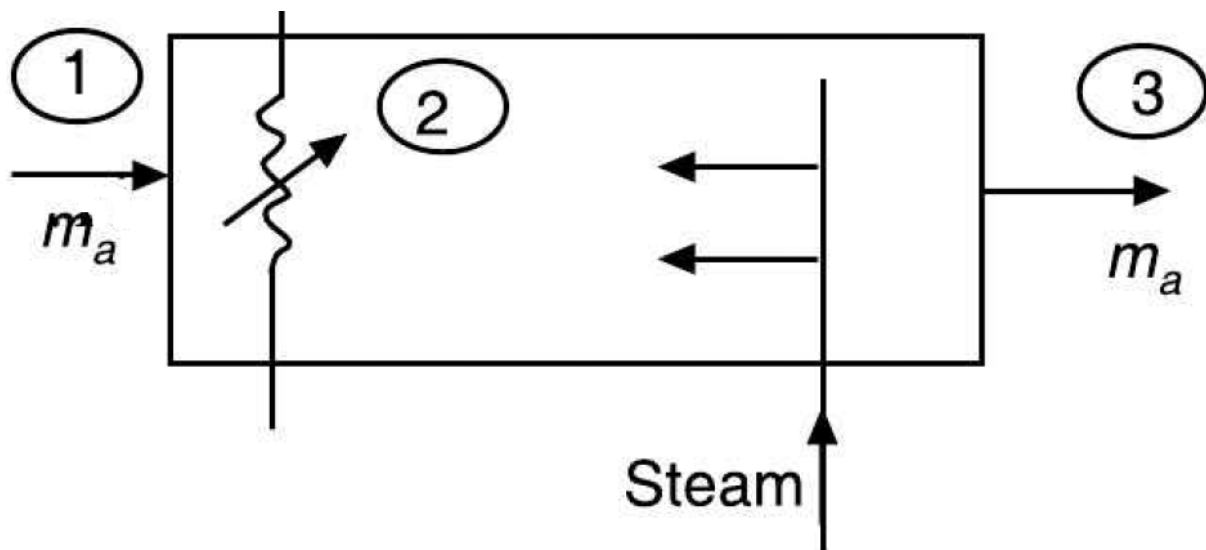
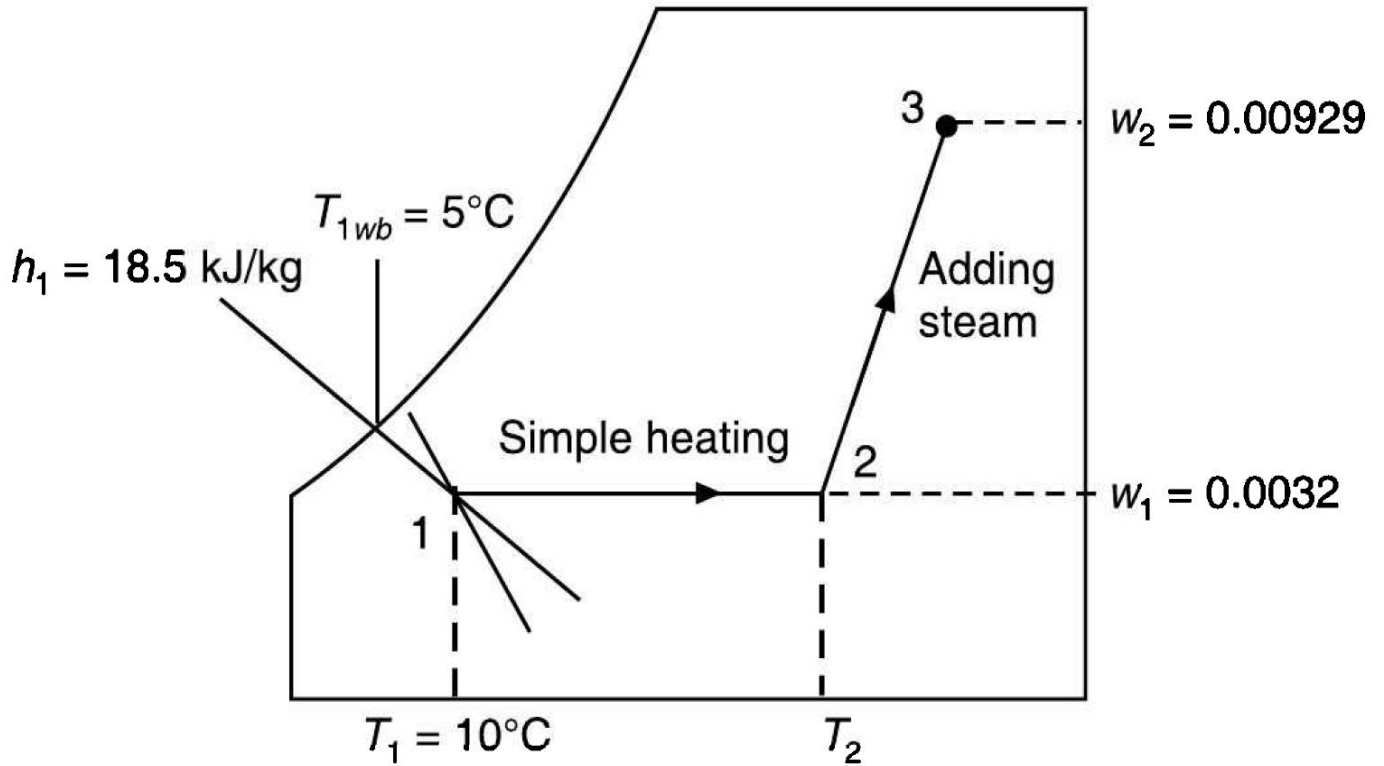
$$m_w = m(w_2 - w_1)$$

### Example 5.2

Air enters a chamber at 10 °C DBT and 5 °C WBT at a rate of 100 m<sup>3</sup>/min. the barometer reads a pressure of 101.325kPa. while passing through the chamber, the air absorbs sensible heat at the rate of 40 kW and picks up 45 kg/hr. of saturated steam at 105 °C. Determine the dry and wet bulb temperatures (DBT and WBT) of the air leaving the chamber.

### Solution

As shown in the figure, the processes is a simple sensible heating





followed by humidification of air by the addition of steam.

The specific volume of the entering air is **0.813 m<sup>3</sup>/kg**

The mass flow rate of air is:

$$m = \frac{100 \times 60}{0.813} = 7380 \text{ kg/hr.}$$

By mass balance of moisture,

$$m_w = m(w_3 - w_1) ; \text{ note that } w_1 = w_2$$

$$45 = m(w_3 - w_1)$$

From the psychrometric chart  $w_1 = 0.0032 \text{ kg/kgda}$

$$w_3 = w_1 + \frac{45}{m_a} = 0.0032 + \frac{45}{7380} = 0.00929 \text{ kg/kgda}$$

By the energy balance,

$$Q = m(h_3 - h_1) - m_w h_w$$

$h_w = 2683.6 \text{ kJ/kg}$ , enthalpy of saturated vapor at 105 °C.

$$40 = \frac{7380}{3600} (h_3 - 18.5) - \frac{45}{3600} \times 2683.6$$

$$h_3 = 54.376 \text{ kJ/kg}$$

From the psychrometric chart at point 3  $h_3 = 54.376 \text{ kJ/kg}$  and  $w_3 = 0.00929 \text{ kg/kgda}$ ,

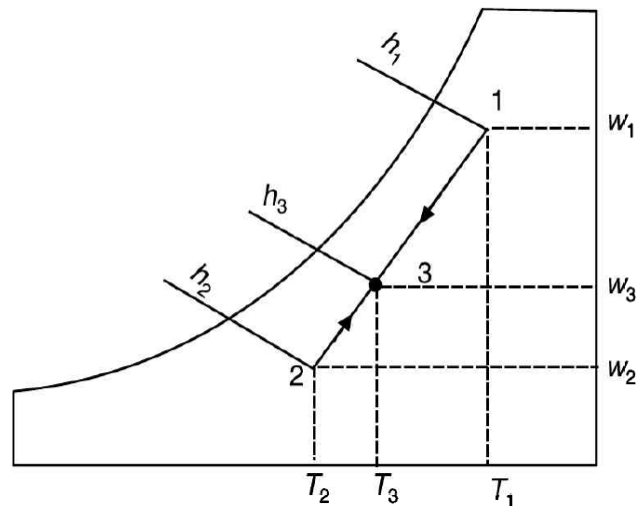
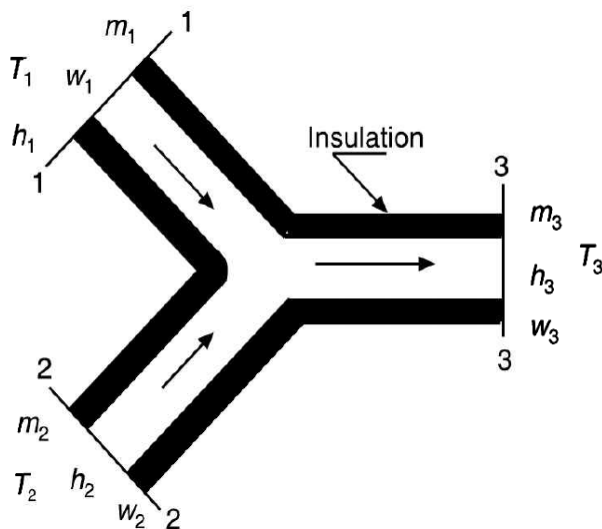
$$T_{db} = 26.5 \text{ °C}$$

$$T_{wb} = 18.1 \text{ °C}$$

## 6.1 Adiabatic Mixing

Mixing of two moist air streams is a common process in air conditioning. For design purposes, it is usually assumed that the mixing process is adiabatic under steady-state conditions.

In the figure below we consider two inlet ports 1 and 2 and an exit port 3. The heat balance of the system gives:



$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$$

The mass balance of air gives:

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

For the mass balance of water vapor in the moist air:

$$\dot{m}_1 w_1 + \dot{m}_2 w_2 = \dot{m}_3 w_3$$



if the heat capacities of the moist air stream 1,2, and 3 are equal, the following approximate relation can be given as:

$$\dot{m}_1 T_1 + \dot{m}_2 T_2 = \dot{m}_3 T_3$$

Or

$$\dot{m}_1 T_1 + \dot{m}_2 T_2 = (\dot{m}_1 + \dot{m}_2) T_3$$

$$T_3 = \frac{\dot{m}_1 T_1 + \dot{m}_2 T_2}{\dot{m}_1 + \dot{m}_2}$$

Similarly:

$$h_3 = \frac{\dot{m}_1 h_1 + \dot{m}_2 h_2}{\dot{m}_1 + \dot{m}_2}$$

And,

$$w_3 = \frac{\dot{m}_1 w_1 + \dot{m}_2 w_2}{\dot{m}_1 + \dot{m}_2}$$

### Example 6.1:

2 kg/s of moist air at 45°C (DBT) and 30°C (WBT) is mixed with 1 kg/s of moist air at 25°C and relative humidity of 30% in an adiabatic mixing chamber. Determine the enthalpy, dry bulb temperature, and moisture content of the moist air at the mixing point.



**Solution:**

From psychrometric chart at 45°C (DBT) and 30°C (WBT) :

$$h_1 = 98.32 \frac{kJ}{kg}$$

$$w_1 = 0.02055 kg/kg da$$

At 25°C and relative humidity of 30% :

$$h_2 = 40.13 \frac{kJ}{kg}$$

$$w_2 = 0.00589 kg/kg da$$

$$h_3 = \frac{\dot{m} h_1 + \dot{m}_2 h_2}{\dot{m}_1 + \dot{m}_2} = \frac{2 \times 98.32 + 1 \times 40.13}{(2+1)} = 78.92 kJ/kg$$

$$w_3 = \frac{\dot{m} w_1 + \dot{m}_2 w_2}{\dot{m}_1 + \dot{m}_2} = \frac{2 \times 0.02055 + 1 \times 0.00589}{2+1} = 0.01566 kg/kg da$$

$$h_3 = 1.007 T_3 - 0.026 + w(2501 + 1.84 T_3)$$

$$78.92 = 1.007 T_3 - 0.026 + 0.01566 (2501 + 1.84 T_3)$$

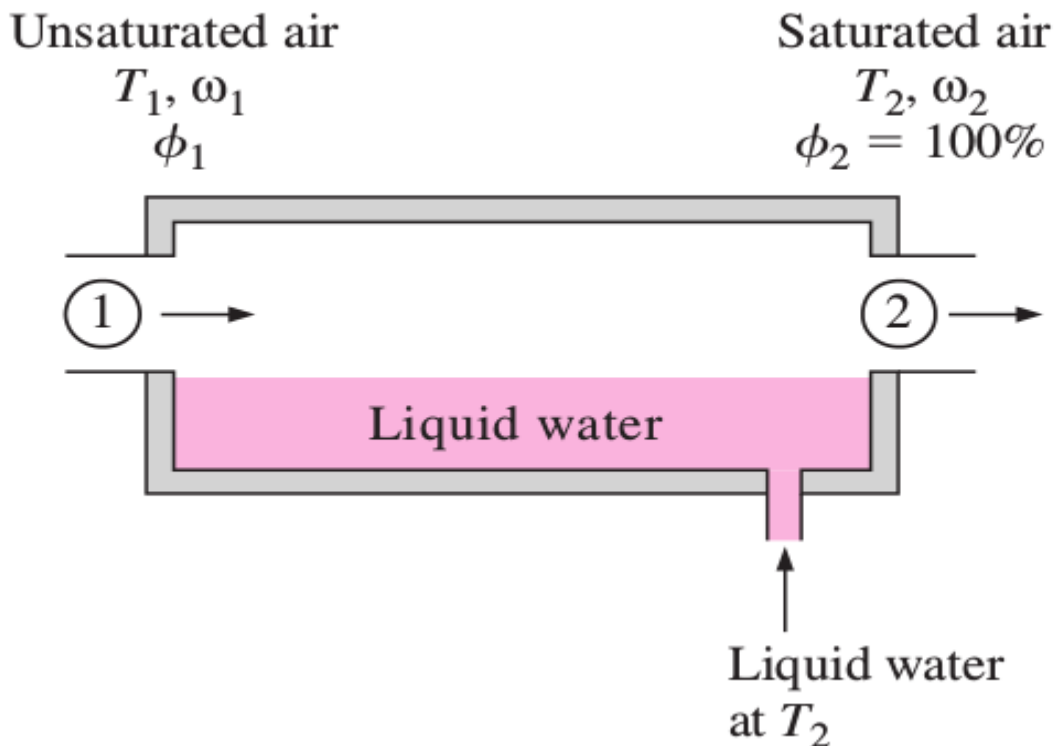
$$T_3 = 38.4^\circ C$$

or can be calculated approximately by assuming that the specific heat of inlet and outlet air is the same:

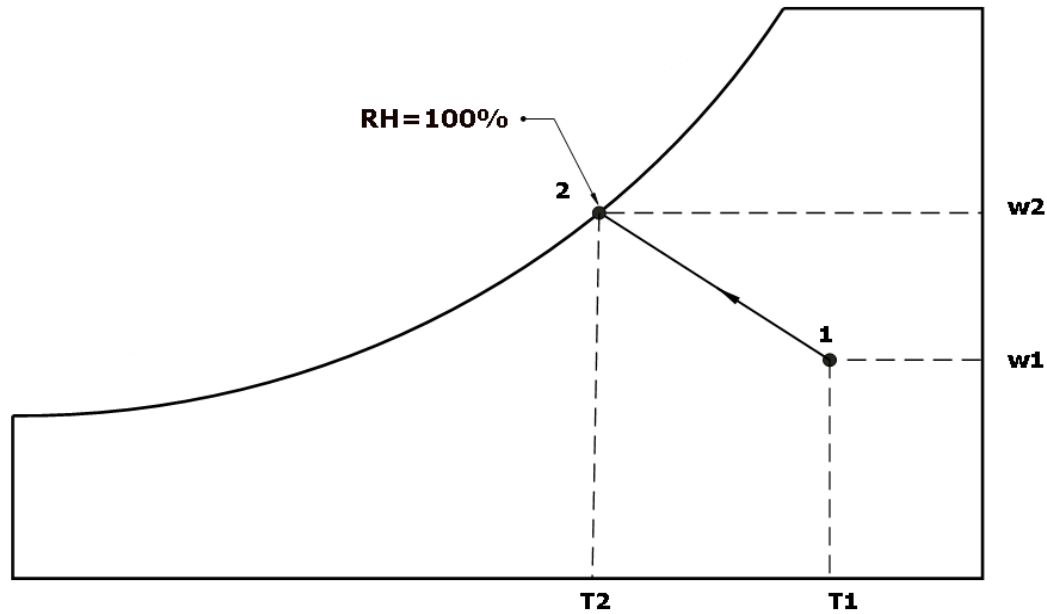
$$T_3 = \frac{\dot{m} T_1 + \dot{m}_2 T_2}{\dot{m}_1 + \dot{m}_2} = \frac{2 \times 45 + 1 \times 25}{2 + 1} = 38.33^\circ\text{C}$$

## 6.2 Adiabatic saturation

The adiabatic process is a humidification process in which no heat enters or leaves the system under consideration. Air is flowing through a perfectly insulated duct with an open water tank in the bottom of it. If the duct is infinitely long, the air at the outlet will be 100% saturated.







the heat balance equation is:

$$h_1 + (w_2 - w_1)h_f = h_2$$

Where  $h_f$  is the enthalpy of the injected water at  $T_2$ .

The term  $(w_2 - w_1)h_f$  small in comparison with  $h_1$  and  $h_2$ , therefore the above equation becomes:

$$h_1 \approx h_2$$

That means the enthalpy during the adiabatic saturation process approximately *remains constant* as well as it is of *constant wet-bulb temperature*.



Also when  $h_1 \approx h_2$ , the latent heat added to the air approximately equal to the sensible heat removed from the air:

$$C_p (T_1 - T_2) = (w_2 - w_1)h_{fg}$$

Or

$$T_2 = T_1 - \frac{(w_2 - w_1)h_{fg}}{C_p}$$

Where  $C_p$  is the specific heat of moist air :

$$C_p = C_{pa} + w C_{pv}$$

$h_{fg}$  : latent heat of evaporation at water temperature.

### Example 6.2:

Air at 30 °C (DBT) and 17 °C (WBT) and a barometric pressure of 1.050 bar enters an equipment where it undergoes an adiabatic saturation process. The air leaving with a moisture content of 5 g/kg higher than that it was while entering. Calculate the following:

- The moisture content of the air entering the equipment.
- Dry bulb temperature, enthalpy, and relative humidity of the air leaving the equipment.

### Solution:

a)

The vapor pressure at the inlet point can be calculated as follows:

$$P_v = P_{sw} - P_B A (T_d - T_w)$$

$$P_{sw} = 1.936kPa \text{ (saturation pressure at 17°C (WBT))}$$



$$P_{v1} = 1.936 - 105 \times 6.66 \times 10^{-4} (30 - 17) = 1.027 \text{ kPa}$$

$$w_1 = 0.622 \frac{P_{v1}}{P_B - P_{v1}} = 0.622 \frac{1.027}{105 - 1.027} = 0.00614 \frac{\text{kg}}{\text{kg}} da$$

b)

$$w_2 = w_1 + 0.005 = 0.00614 + 0.005 = 0.01114 \frac{\text{kg}}{\text{kg}} da$$

$$w_2 = 0.622 \frac{P_{v2}}{P_B - P_{v2}}$$

$$0.01114 = 0.622 \frac{P_{v2}}{105 - P_{v2}}$$

$$1.17 - 0.01114 P_{v2} = 0.622 P_{v2}$$

$$P_{v2} = 1.848 \text{ kPa}$$

$$P_{v2} = P_{sw} - P_B A (T_{d2} - T_w)$$

$$1.848 = 1.936 - 105 \times 6.66 \times 10^{-4} (T_{d2} - 17)$$

$$T_{d2} = 18.258 \text{ }^\circ\text{C}$$

$$h_2 = 1.007 T_{d2} - 0.026 + w_2 (2501 + 1.84 T_{d2})$$

$$h_2 = 1.007 \times 18.258 - 0.026 + 0.01114 (2501 + 1.84 \times 18.258)$$

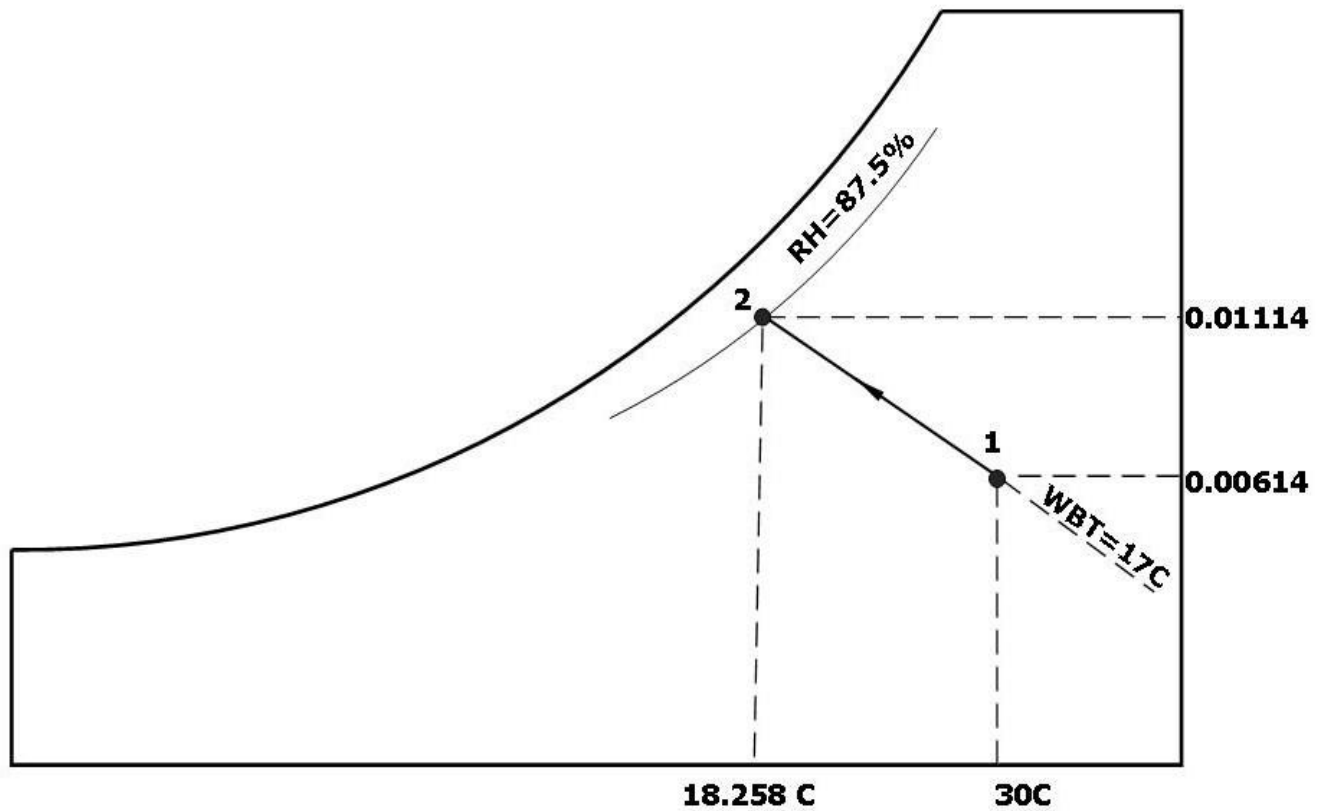
$$h_2 = 46.595 \text{ kJ/kg}$$

$$P_{sat2} = \text{Exp} \left[ 25.317 - \frac{5144}{T_{sat2}} \right]$$



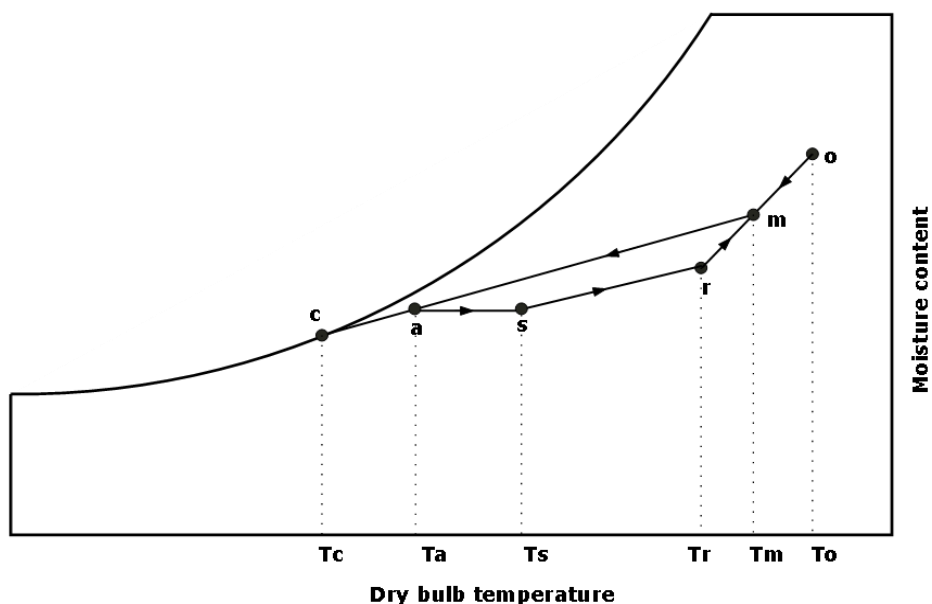
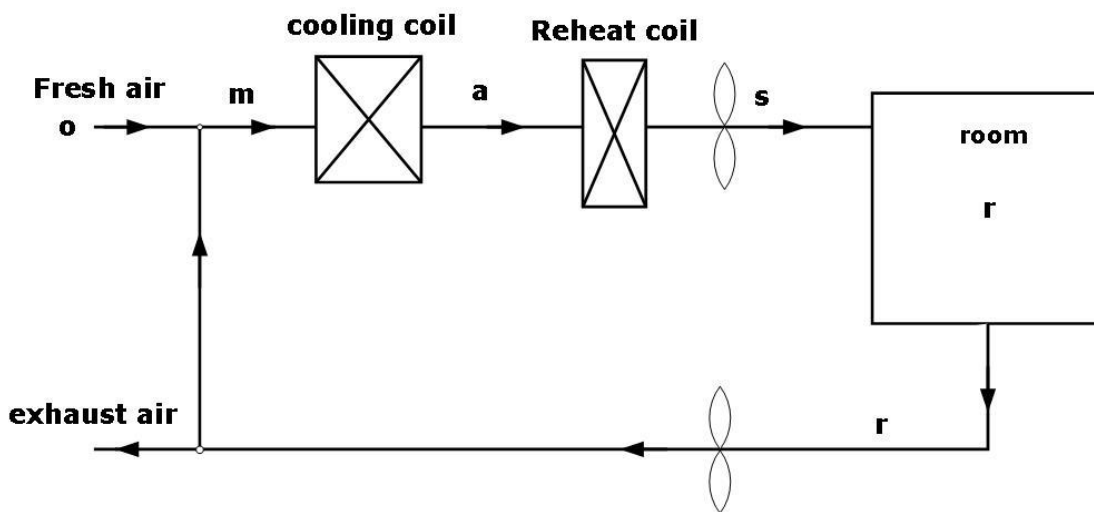
$$P_{sat2} = Epx \left[ 25.317 - \frac{5144}{18.258+273} \right] = 2.113 \text{ kPa}$$

$$RH_2 = \frac{P_{v2}}{P_{sat2}} = \frac{1.848}{2.113} = 0.875 = 87.5\%$$



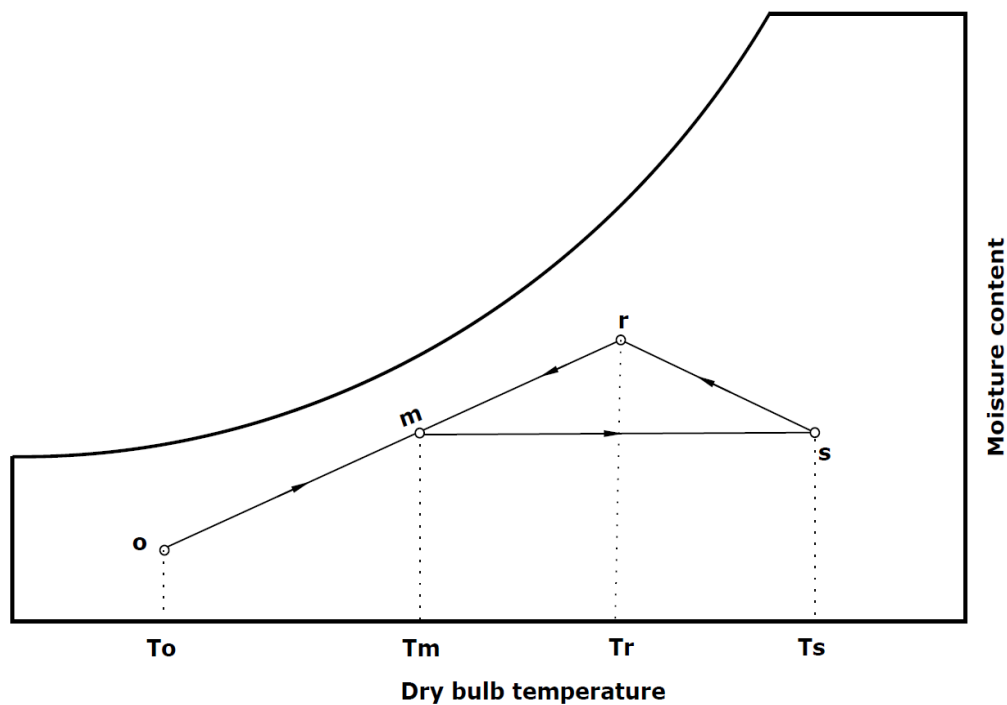
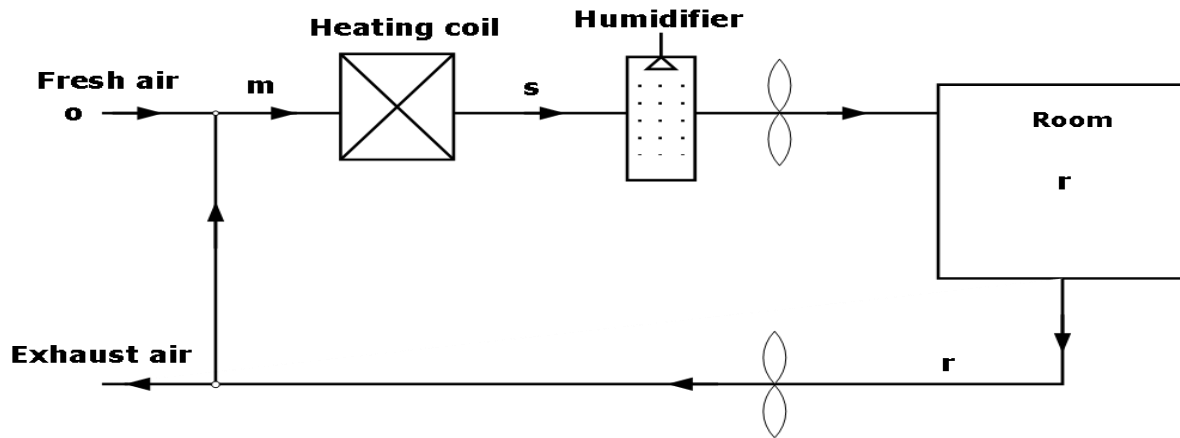
## 7.1 Summer air conditioning systems

The figure below shows a simple summer air conditioning system. In this system outdoor air denoted by (o) mixes with return air from the conditioned space (room r) at point m. The air mixture is then cooled and dehumidified through the cooling coil at point a. In case of high relative humidity, it is preferred to sensibly heat the air to point s before supplying it to the conditioned space.



## 7.2 Winter air conditioning systems

In a typical winter air conditioning system shown in the figure below, fresh air mixes with return air from the room at point  $m$  then heated sensibly by the heating coil. finally, the air mixture was humidified in the humidifier and supplied to the room.





### Example 7.1:

An air-conditioned space is maintained at 25 °C (DBT) and 50% relative humidity. 1m<sup>3</sup>/s of outdoor air at 45 °C (DBT) and 30 °C (WBT) was mixed with 2m<sup>3</sup>/s of return air from the space. The air mixture then cooled by cooling coil to 10 °C (DBT) and saturated state. Determine:

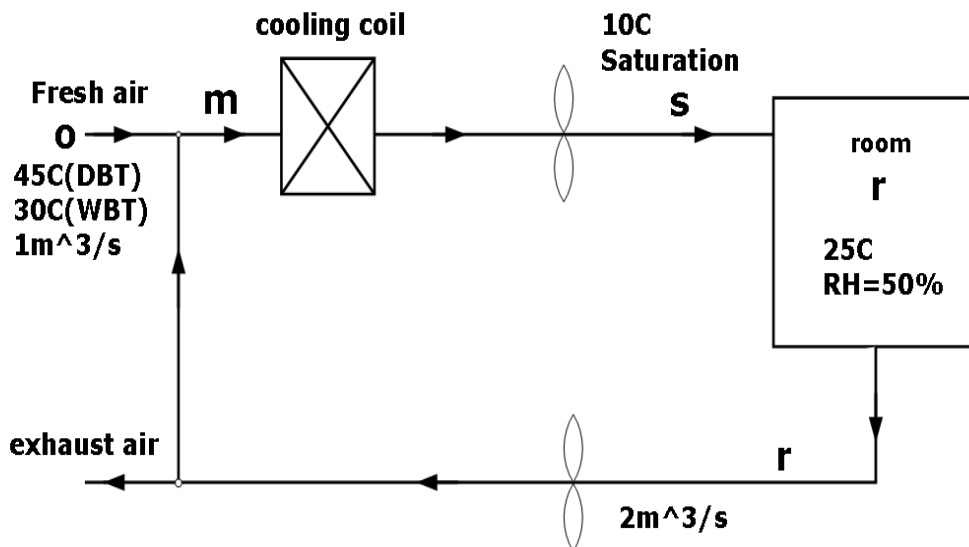
- the mass flow rate of the supplied air to space.
- moisture content of the air after the mixing point.
- the cooling capacity of the coil.
- mass of condensate on the cooling coil.

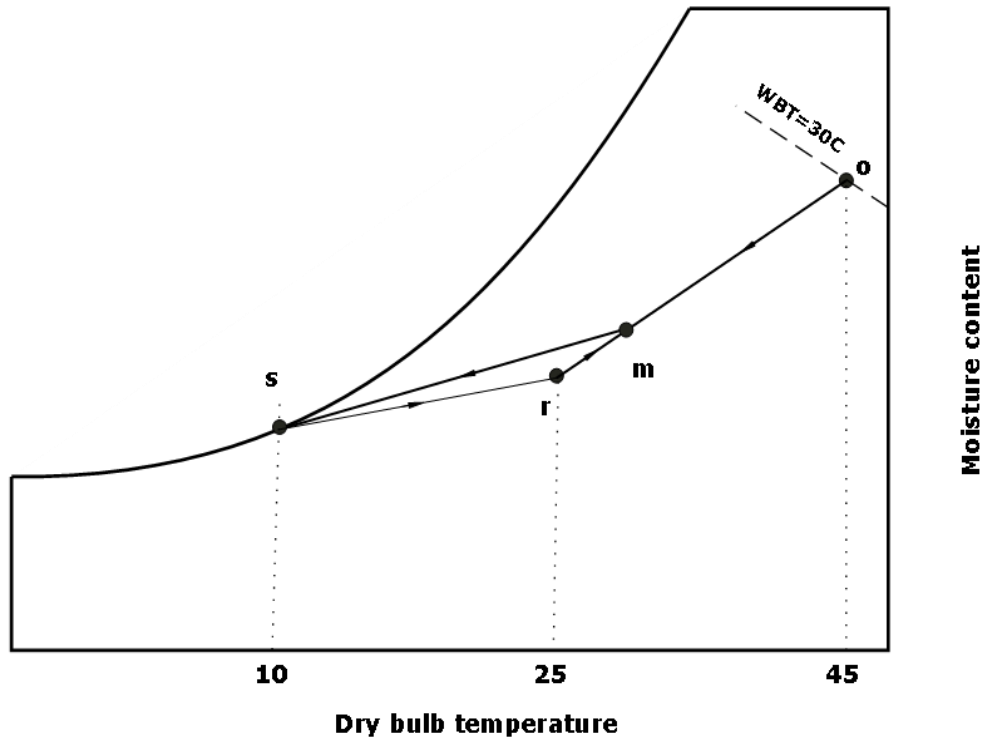
### Solution:

a)

For the supplied air at 10 °C and saturated air of point s (RH=100%),  
 $v=0.8112\text{m}^3/\text{kg}$

$$\dot{m} = \frac{\dot{V}}{v} = \frac{3}{0.8112} = 3.698 \text{ kg/s}$$





b)

$$\dot{m}_r = \frac{\dot{V}_r}{v_r} = \frac{2}{0.8572} = 2.333 \text{ kg/s} ; (25 \text{ }^\circ\text{C, RH=50\%})$$

$$w_r = 0.00988 \text{ kg/kg da}$$

$$\dot{m}_o = \frac{\dot{V}_o}{v_o} = \frac{1}{0.9301} = 1.075 \text{ kg/s} ; 45 \text{ }^\circ\text{C (DBT) and } 30 \text{ }^\circ\text{C (WBT)}$$

$$w_o = 0.02055 \text{ kg/kg da}$$

$$\dot{m}_r w_r + \dot{m}_o w_o = \dot{m}_m w_m$$

$$2.333 \times 0.00988 + 1.075 \times 0.02055 = (2.333 + 1.075)w_m$$





$$w_m = 0.0132 \text{ kg/kg da}$$

c)

$$\dot{m}_o h_o + \dot{m}_r h_r = \dot{m}_m h_m$$

$$1.075 \times 92 + 2.333 \times 50.29 = (1.075 + 2.333)h_m$$

$$h_m = 58.4712 \text{ kJ/kg}$$

$$Q = \dot{m}(h_m - h_s) = 3.698(58.4712 - 29.27) = 107.986 \text{ kW}$$

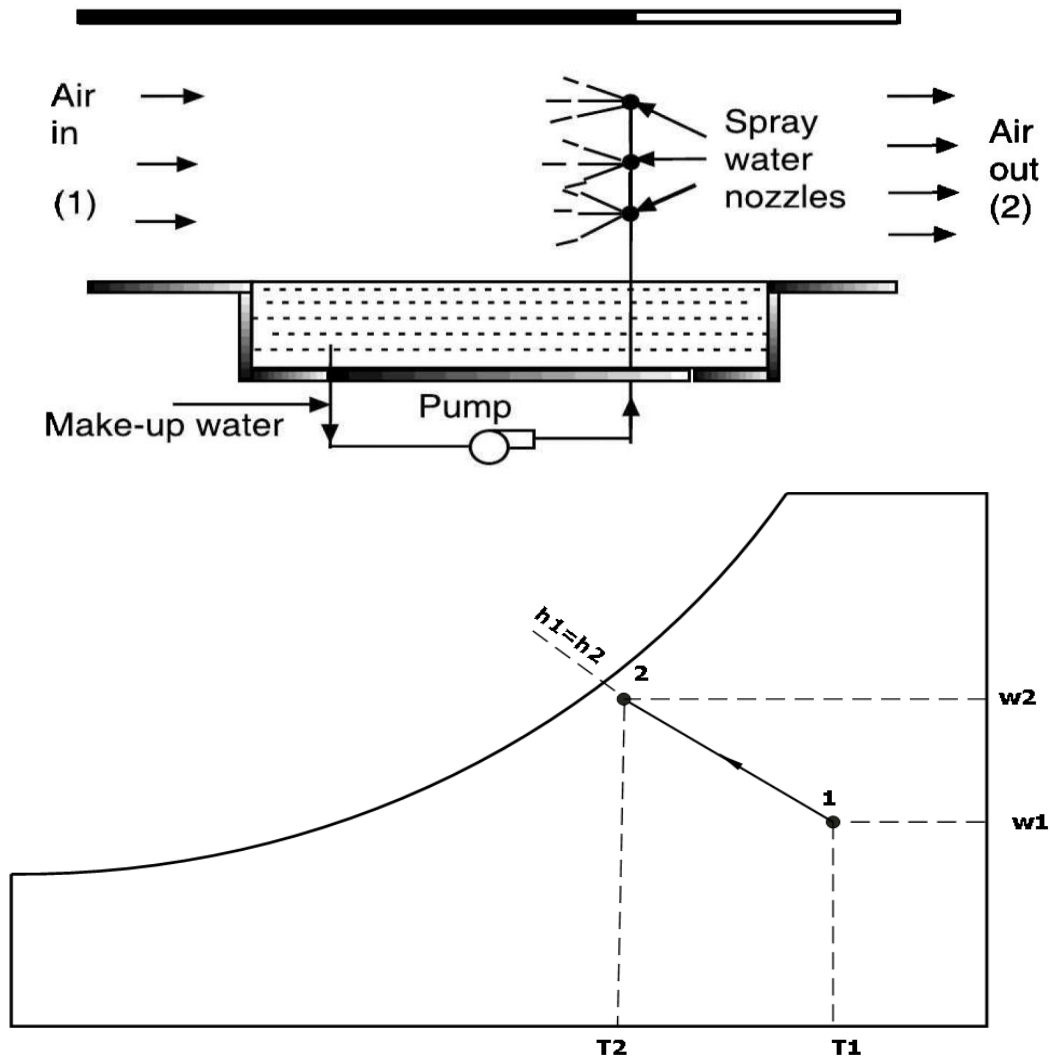
d)

$$\begin{aligned} m_c &= \dot{m}(w_m - w_s) = 3.698(0.0132 - 0.00763) \\ &= 0.02059 \text{ kg/s} \end{aligned}$$

### 7.3 Evaporative cooling

In dry weather area, the difference between dry bulb and wet bulb temperatures is high enough to use water as cooling media. Let's consider warm air blowing over the water surface as shown in the figure. The air contacts the water surface and dissipates heat to the water by conduction. Water particles separate from the water surface, evaporate, and mix with the air. For the evaporation of water latent heat is to be supplied which comes from both air and water. Therefore air and water get cooled. This process of cooling is known as *Evaporative cooling*. In the Arab countries sometimes called *Desert air conditioner*.

During the evaporative cooling, the moisture content increases, and the *enthalpy and wet bulb temperature* remain constant.



The effectiveness of the evaporative cooling can be given as follows:

$$\epsilon = \frac{T_1 - T_2}{T_1 - T_{wb}}$$



It is clear that the effectiveness of the evaporative cooling reaches the maximum value (100%) when the outlet temperature  $T_2$  is equal to the wet-bulb temperature of the moist air.

when  $h_1 = h_2$ , the latent heat added to the air equal to the sensible heat removed from the air:

$$C_p (T_1 - T_2) = (w_2 - w_1)h_{fg}$$

Or

$$T_2 = T_1 - \frac{(w_2 - w_1)h_{fg}}{C_p}$$

### Example 7.2

2500 m<sup>3</sup>/hr air enters evaporative air cooler at 40 °C (DBT) and 20 °C (WBT). If the moisture content of the outlet increase by 90% more than that of the inlet. calculate the effectiveness and the cooling capacity of the evaporative air cooler.

#### Solution:

From the psychrometric chart at 40 °C (DBT) and 20 °C, the moisture content is:  $w_1 = 0.00619 \text{ kg/kg da}$

$$w_2 = w_1 + 0.9w_1 = 1.9 \times 0.00619 = 0.01176 \text{ kg/kg da}$$

$$T_2 = T_1 - \frac{(w_2 - w_1)h_{fg}}{C_p} = 40 - \frac{(0.01176 - 0.00619) \times 2500}{1.02} = 26.35 \text{ °C}$$

$$\varepsilon = \frac{T_1 - T_2}{T_1 - T_{wb}} = \frac{40 - 26.35}{40 - 20} = 0.683$$



The outlet temperature  $T_2$  can be also calculated as follows:

$$h_1 = h_2$$

$$h_1 = 1.007 T_1 - 0.026 + w_1(2501 + 1.84 T_1)$$

$$\begin{aligned} h_1 &= 1.007 \times 40 - 0.026 + 0.00619(2501 + 1.84 \times 40) \\ &= 56.191 \text{ kJ/kg} \end{aligned}$$

$$h_2 = 1.007 T_2 - 0.026 + w_2(2501 + 1.84 T_2)$$

$$56.191 = 1.007 T_2 - 0.026 + 0.01176(2501 + 1.84 T_2)$$

$$T_2 = 26.06 \text{ }^\circ\text{C}$$

$$v_1 = 0.8951 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{V}}{v_1} = \frac{2500}{3600 \times 0.8951} = 0.776 \text{ kg/s}$$

$$\begin{aligned} \text{cooling capacity} &= \dot{m}(w_2 - w_1)h_{fg} \\ &= 0.776((0.01176 - 0.00619) * 2500) \\ &= 10.806 \text{ kW} \end{aligned}$$

Or

$$\begin{aligned} \text{cooling capacity} &= \dot{m} C_p(T_2 - T_1) \\ \text{cooling capacity} &= 0.776 \times 1.02 \times (40 - 26.35) \\ &= 10.804 \text{ kW} \end{aligned}$$

## **8. Cooling Load Calculations**

The human body continuously generates heat at a rate varying between 100W and 500 W depending upon the health and activities of the individuals. Body temperature must be maintained with a narrow temperature range to avoid discomfort. Therefore, heat must be dissipated in a controlled manner. The comfort conditions are not the same for men, women, children, and old people. A large number of human experiments have been carried out to understand the comfort conditions. The experiments show that the majority of people feel comfortable in summer at:

1. Dry bulb temperature =  $25 \pm 1$  °C
2. Relative humidity =  $50 \pm 5\%$
3. Air velocity = 15 to 20 m/min

For the winter season the comfort conditions are:

1. Dry bulb temperature =  $22 \pm 1$  °C
2. Relative humidity = 50 %
3. Air velocity = 9 to 12 m/min

### **8.1 Thermal exchanges of the body with the environment**

The rate at which the body generates heat is called the *metabolic rate*. The human body is considered a heat engine with 20% thermal efficiency. This means that if the body generates heat of 100W due to digestion of food, heat at a rate of 80W must be dissipated to the surroundings. The temperature of a healthy man is 36.9 °C.



The heat exchange between the human body and the surroundings is through three modes: radiation, convection, and evaporation of moisture. The total heat exchanges can be given as:

$$Q = Q_r + Q_c + Q_e$$

$Q_r$ : radiation heat rate, (W)

$Q_c$ : convection heat rate, (W)

$Q_e$ : evaporation heat rate, (W)

The human feels comfortable only when the rate of heat generation due to metabolic rate is equal to the heat loss to the surroundings. Hence, one can apply the first law of thermodynamics to a body as:

$$Q_m - W = Q_e \pm Q_r \pm Q_c \pm Q_s$$

$Q_m$  : Metabolic heat produced within the body

$W$  : Useful rate of working

$Q_e$  : Heat lost by evaporation

$Q_r$  : Heat lost or gained by radiation

$Q_c$  : Heat lost or gained by convection

$Q_s$  : Heat stored in the body



**Convective heat loss**  $Q_c = U A (T_b - T_a)$

if the surrounding temperature is higher than the human body temperature, the convective heat  $Q_c$  is negative (heat gain).

$U$  : overall heat transfer coefficient

$T_b$  : human body temperature, °C

$T_a$  : air temperature, °C

$A$  : the surface area of the human body which can be estimated from the following relation:

$$A = 0.202 m^{0.425} h^{0.725} , (m^2)$$

Where;

$m$  : mass of the human; kg

$h$  : height of the human; m

**Radiation heat loss**  $Q_r = \sigma A (T_b^4 - T_a^4)$

$\sigma$  : Stefan -Boltzmann constant,  $\sigma = 5.67 \times 10^{-8} \frac{W}{m^2 K^4}$

The temperature here should be in  $K$

If  $T_a > T_b$  , the radiation heat transfer  $Q_r$  becomes negative (heat gain).



### Evaporative heat loss $Q_e = K_d A (P_s - P_v) h_{fg} K_c$

$K_d$  : diffusion coefficient in kg of water evaporated per unit area and pressure difference per hour.

$P_s$  : saturation pressure of water vapor corresponding to the skin temperature.

$P_s$  : vapor pressure in the air

$h_{fg}$  : latent heat of evaporation

$K_c$  : a factor for the type of clothing

According to *Belding* and *Hatch*, the convective, radiative, and evaporative heat transfer rates from the naked body of an average adult,  $Q_c$ ,  $Q_r$ , and  $Q_e$ , respectively, are given by:

$$Q_c = 14.8 \sqrt{V} (T_b - T_a)$$

$$Q_r = 11.603 (T_b - T_s)$$

$$Q_e = 181.76 V^{0.4} (P_{s,b} - P_v)$$

$V$  : air velocity m/s

$T_b$  : human body temperature

$T_s$  : the temperature of the surrounding surfaces

$T_a$  : Air temperature

$P_{s,b}$  : saturated vapor pressure at the surface temperature of the body; *kPa*

$P_v$  : partial pressure of water vapor in air *kPa*





### 8.3 Metabolic rate

The metabolic rate depends on the activity. It is normally measured in the unit “**met**”. A met is defined as the metabolic rate per unit area of a sedentary person and is found to be equal to about **58.2 W/m<sup>2</sup>**. This is also known as “basal metabolic rate”. The following table shows typical metabolic rates for different activities:

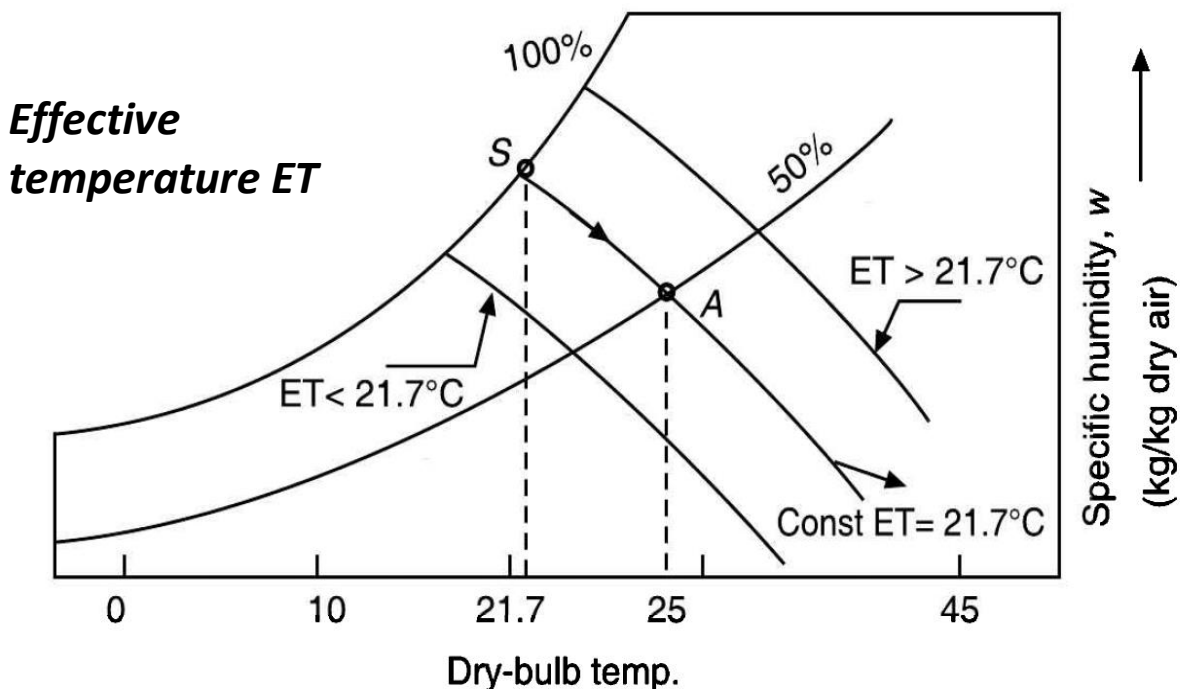
**Typical metabolic rates**

<b>Activity</b>	<b>Specifications</b>	<b>Metabolic rate (met)</b>	<b>Metabolic rate (W/m<sup>2</sup>)</b>
<b>Resting</b>	<b>Sleeping</b>	<b>0.7</b>	<b>41</b>
	<b>Reclining</b>	<b>0.8</b>	<b>47</b>
	<b>Seated, Quite</b>	<b>1</b>	<b>58</b>
	<b>Standing, relaxed</b>	<b>1.2</b>	<b>70</b>
<b>Walking</b>	<b>0.89 m/s</b>	<b>2</b>	<b>116</b>
	<b>1.79 m/s</b>	<b>3.8</b>	<b>221</b>
<b>Office activity</b>	<b>Typing</b>	<b>1.1</b>	<b>64</b>
<b>Driving</b>	<b>Car</b>	<b>1-2</b>	<b>58-116</b>
	<b>Heavy vehicle</b>	<b>3.2</b>	<b>186</b>
<b>Domestic activities</b>	<b>Cooking</b>	<b>1.6-2</b>	<b>93-116</b>
	<b>Washing dishes</b>	<b>1.6</b>	<b>93</b>
	<b>House cleaning</b>	<b>2-3.4</b>	<b>116-198</b>
<b>Dancing</b>	<b>-</b>	<b>2.4-4.4</b>	<b>140-256</b>
<b>Teaching</b>	<b>-</b>	<b>1.6</b>	<b>93</b>
<b>Games and sports</b>	<b>Tennis, singles</b>	<b>3.6-4</b>	<b>210-233</b>
	<b>Gymnastics</b>	<b>4</b>	<b>233</b>
	<b>Basketball</b>	<b>5-7.6</b>	<b>291-442</b>
	<b>Wrestling</b>	<b>7-8.7</b>	<b>407-506</b>

## 8.2 Effective temperature

People feel comfortable only when dry bulb temperature, relative humidity, and air motion are within certain limits. Effective temperature  $ET$  is an index of the measure of comfort. The effective temperature is the temperature of saturated air at which people feel comfortable as experienced in the actual unsaturated environment.

The effective temperature is a measure of feeling warmth or cold by the human body in presence of air temperature, relative humidity, and air motion. To construct a constant  $ET$  line on the psychrometric chart, mark point  $A$  on the intersection of 50% RH and 25 °C DBT (see the figure). Draw a vertical line from the x-axis corresponding to 21.7 °C intersecting the saturation curve at point  $S$  (RH=100%). Join the points  $S$  and  $A$  indicating constant  $ET$  line. This is an approximate method of obtaining a rough idea about effective temperature  $ET$ .



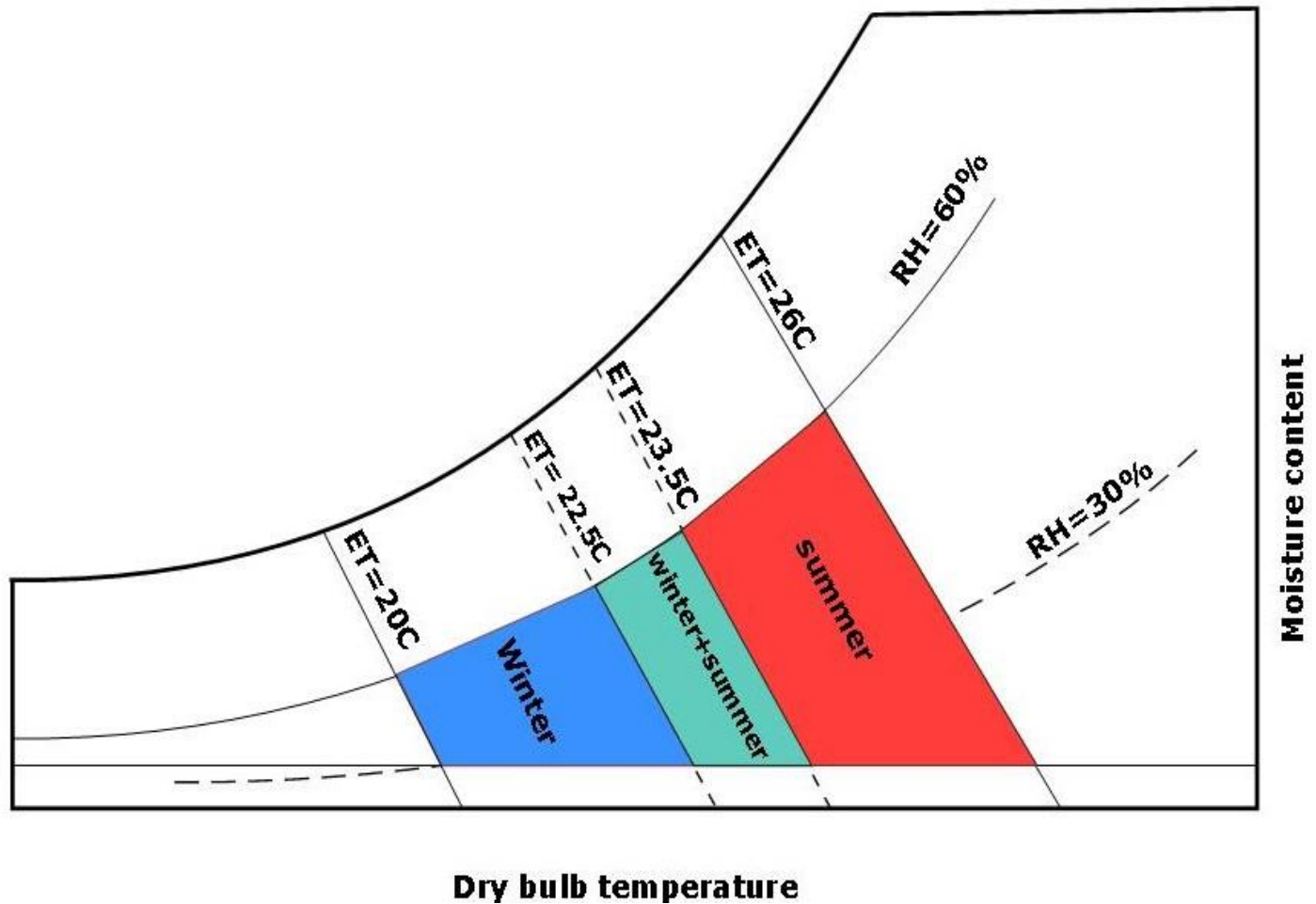


The following table shows the effective temperature in the summer and winter seasons.

**Table: Effective temperature in summer and winter seasons.**

Summer		Winter	
ET °C	RH %	ET °C	RH %
26.7	25	25.6	10
26.1	30	24.4	20
25.6	40	23.9	30
25	50	23.3	40
23.9	60	22.8	50
23.3	70	22.2	60
-	-	21.7	70
-	-	21.1	80

The comfortable zone can be plotted on a psychrometric chart in term of effective temperature *ET* as shown in the figure below:



**Comfort chart for a sedentary person (activity  $\approx$  1.2 met)**



**Table: Heat gain from occupants**

Activity	Location	Total Heat, W		Heat Gain, W	
		Adult male	Adjusted M/F	Sensible	Latent
Seated theater	Theater	115	105	70	35
Seated, very light work	Office, hotels, apartments	130	115	70	45
Moderately active office work	Office, hotels, apartments	140	130	75	55
Standing, light work, walking	Department store; retail store	160	130	75	55
Standing, walking,	Drug store, bank	160	145	75	70
Sedentary work	Restaurant	145	160	80	80
Light benchwork	Factory	235	220	80	140
Moderate dancing	Dance hall	265	250	90	160
Walking 4.8 km/h; light machine work	Factory	295	295	110	185
Bowling	Bowling alley	440	425	170	225
Heavy work	Factory	440	425	170	255
Heavy machine work; lifting	Factory	470	470	185	285
Athletics	Gymnasium	585	525	210	315



## **9.1 Selection of outdoor design conditions**

Outside design conditions are important parameters for the energy efficiency of buildings. The result of incorrect selection of outdoor design conditions can be caused discomfort and energy consumption. The selection of the outdoor design conditions may only prevail for a short time, and most of the time the actual external conditions will be different from the design values the outside design conditions depends on the following factors:

1. Type of the structure, (heavy construction, medium, or light)
2. Thermal insulation of the building walls and roofs.
3. Percentage of glass area.
4. Type of usage
5. Nature of occupancy
6. Daily range of temperature.

## **9.2 Outdoor design conditions for summer and winter:**

ASHRAE handbook-Fundamentals 2017 lists outdoor design conditions for major locations around the world for 0.4%, 2%, 5%, or 10 % of total hours in a year. Choosing of 0.4 % value or 2 % value or 5 % or 10% value depends on the specified requirements and experience of the HVAC designer. The following table shows the outdoor design condition for four locations only in Iraq that are listed in ASHRAE handbook-Fundamentals 2017.



Table: Outdoor design condition in summer for some locations in Iraq

City	0.4% Value		2% Value		5% Value		10% Value		Daily Range
	DBT	WBT	DBT	WBT	DBT	WBT	DBT	WBT	
Basrah	50.2	21.8	48.9	21.6	47.4	21.6	46.1	21.2	17.7
Baghdad	48.2	22.1	46.9	21.7	45.5	21.3	44.1	21	15.2
Mosul	47.1	21.3	45.8	21.2	44.6	20.7	43.2	20.3	15.5
Kirkuk	47.2	21	45.8	21	44.5	20.7	43.2	20.1	13.3

Table: Outdoor design condition in winter for some locations in Iraq

City	DBT °C Dry bulb temperature		DPT °C Dew Point temperature	
	99.6%	99%	99.6%	99%
Basrah	3	4.8	-6.9	-4.9
Baghdad	1.4	3	-8	-6.1
Mosul	-0.9	0.3	-9.9	-7.1

If no special requirements, the winter outside design conditions in the south of Iraq is roughly selected as **0 °C (DBT)** and **RH=85%**.

### **9.3 Indoor design conditions for summer and Winter:**

According to ASHRAE comfortable zone, the following indoor design conditions are recommended:

Inside design conditions for Summer:

$T_{db}$  between 22.5 to 26.0°C at RH of 60%

$T_{db}$  between 23.5 to 27.0°C at DPT of 2°C

Inside design conditions for Winter:

$T_{db}$  between 20.0 to 23.5°C at RH of 60%

$T_{db}$  between 20.5 to 24.5°C at DPT of 2°C

In the absence of any special requirements, the indoor design conditions in most practical applications in Iraq is roughly selected as 25°C and RH=50%.

#### 9.4 Steady-state heat conduction in buildings

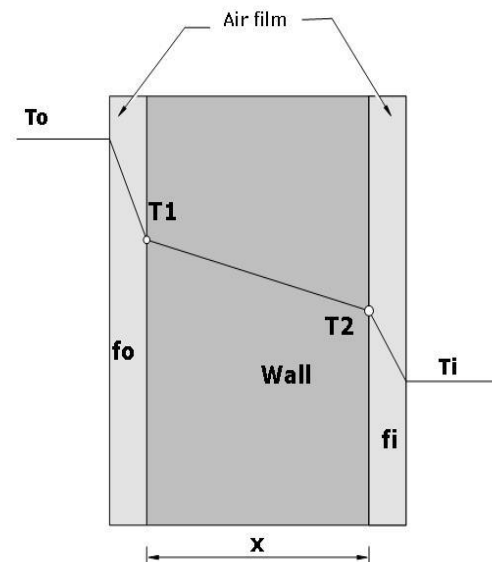
The heat gain or loss through building structures like walls, roof, windows, and doors are the major part of the sensible cooling or heating load of the buildings.

Consider the wall of single homogenous material as shown in the figure. If one surface of the wall is exposed to higher air temperature, then heat transfers through the wall to the surface of the lower temperature. The heat transfer can be given as follows:

$$Q = U A (T_o - T_i), W$$

Where  $U$  is the overall heat transfer coefficient

$A$  : area normal to the direction of heat flow





The overall heat transfer coefficient  $U$  can be estimated as follows:

$$U = \frac{1}{R_t}, \text{ (W/ m}^2 \text{ }^\circ\text{C)}$$

Where,  $R_t$  is the total heat resistance of the wall which is equal:

$$R_t = R_{cov,o} + R_{cond} + R_{cov,i}$$

$$R_t = \frac{1}{f_o} + \frac{x}{k} + \frac{1}{f_i}, \text{ (m}^2 \text{ }^\circ\text{C/W)}$$

$f_o$  : outside heat transfer coefficient

$f_i$  : inside heat transfer coefficient

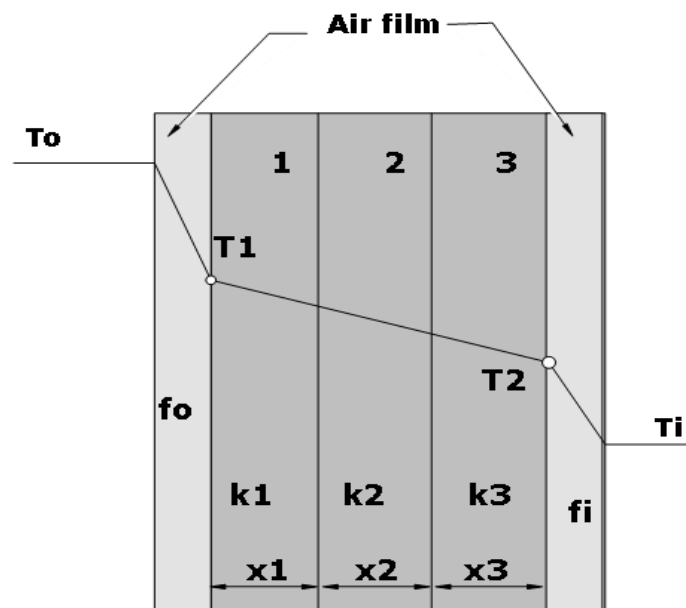
$k$  : thermal conductivity of the wall

$x$  : wall thickness

For multilayer wall of different materials as shown in the figure, the total heat resistance of the wall is:

$$R_t = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_i}$$

And the overall heat transfer coefficient is:

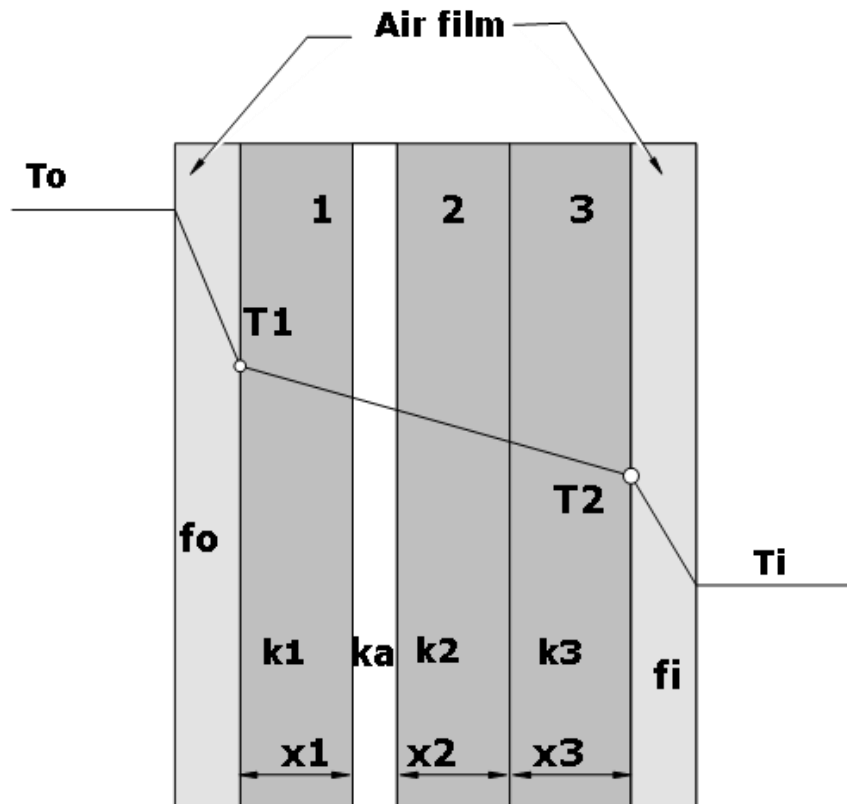


$$U = \frac{1}{R_t}$$

And,

$$Q = U A (T_o - T_i), W$$

When an air gap is provided between the materials as shown in the figure below, then the total heat resistance is given by:



$$R_t = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{1}{k_a} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_i}$$



The heat transfer coefficient of the outside and inside air film is given as:

$$f_i = 9.37 \text{ W/m}^2 \text{ }^\circ\text{C}, \text{ no air motion}$$

$$f_o = 34.1 \text{ W/m}^2 \text{ }^\circ\text{C}, \text{ Winter with wind velocity 24km/hr.}$$

$$f_o = 22.7 \text{ W/m}^2 \text{ }^\circ\text{C}, \text{ Summer with wind velocity 12km/hr.}$$

### Example 9.1:

A building was maintained at  $24^\circ\text{C}$  (DBT) and 50% relative humidity. The outdoor conditions are  $45^\circ\text{C}$  (DBT) and 25% relative humidity. The inner and outer heat transfer coefficients are  $9.37 \text{ W/m}^2 \cdot \text{K}$  and  $22.7 \text{ W/m}^2 \cdot \text{K}$ , respectively. Calculate the heat gain and the inner surface temperature of the building wall (see the figure).

### Solution:

the heat resistance of the wall is:

$$R_t = \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_i}$$

$$R_t = \frac{1}{22.7} + \frac{0.24}{0.6} + \frac{0.05}{0.04} + \frac{0.15}{0.8} + \frac{1}{9.37} = 1.988 \text{ m}^2 \text{ }^\circ\text{C/W}$$



$$U = \frac{1}{R_t} = \frac{1}{1.988} = 0.503 \text{ W/m}^2 \text{ }^\circ\text{C}$$

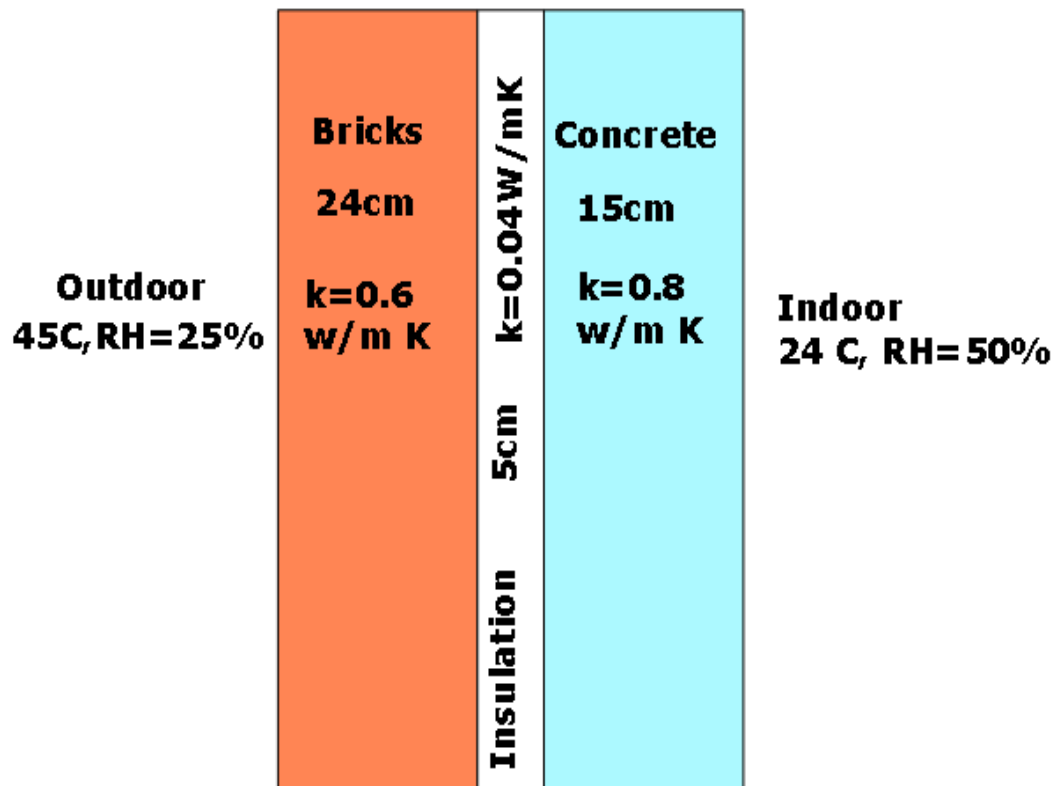
$$Q = U (T_o - T_i) = 0.503(45 - 24) = 10.563 \text{ W/m}^2$$

The inner surface temperature can be evaluated by:

$$Q = f_i (T_{si} - T_i)$$

$$10.563 = 9.37 (T_{si} - 24)$$

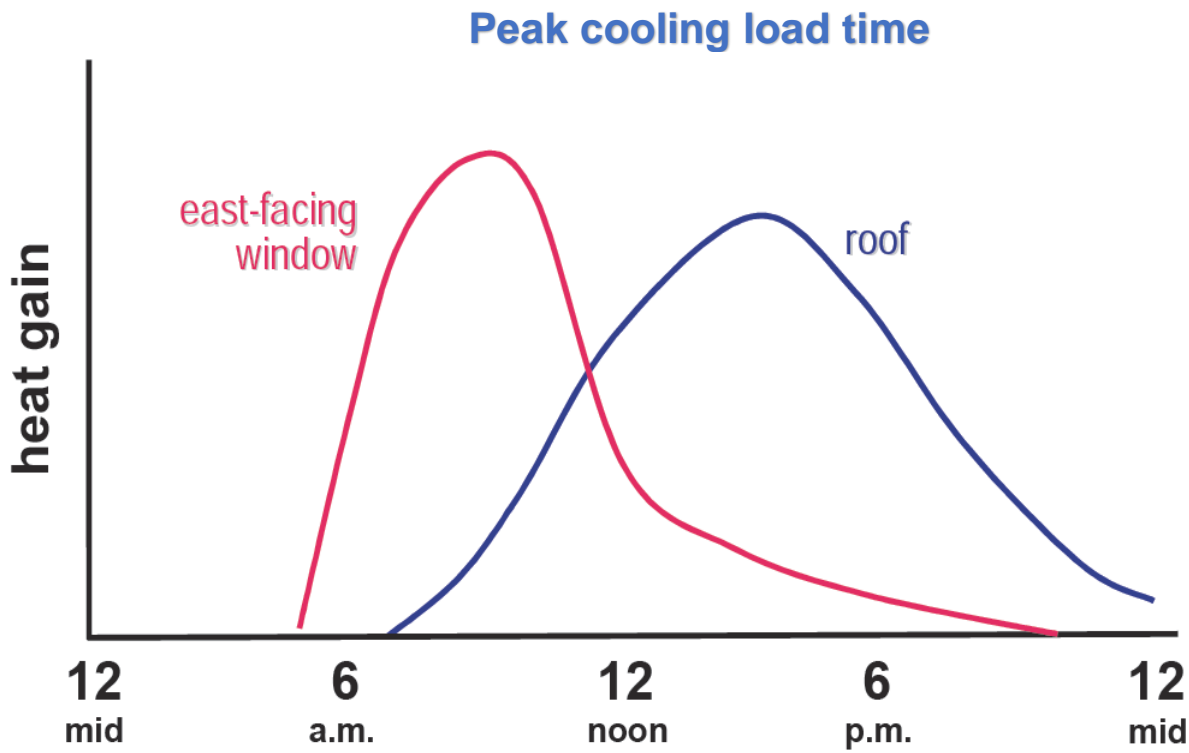
$$T_{si} = 25.127 \text{ }^\circ\text{C}$$



## **10.1 Unsteady-state heat conduction in buildings**

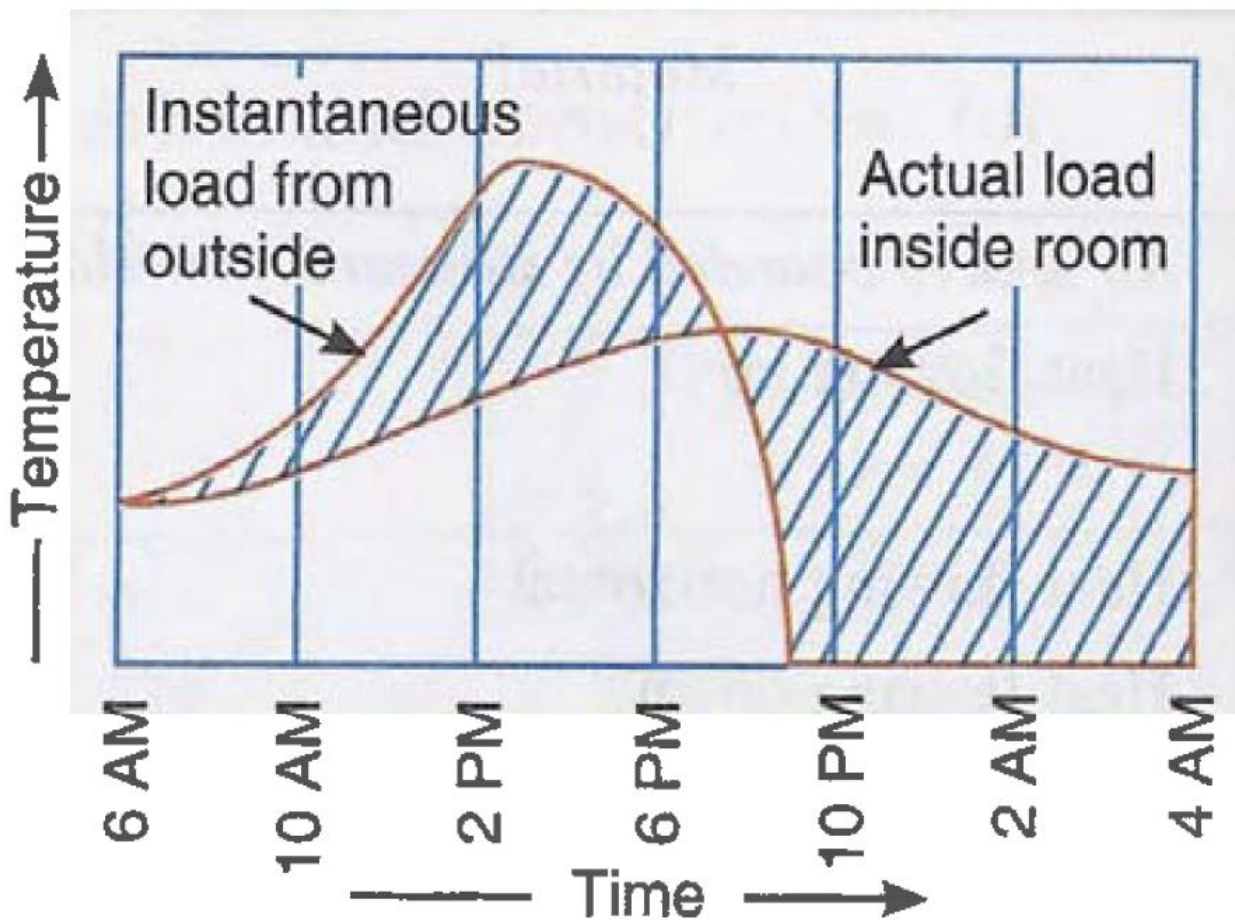
One of the more difficult aspects of estimating the maximum cooling load for space is determining the time at which this maximum load will occur. This is because:

- 1-Heat storage in the building's structures.
- 2-Variation of the solar radiation and ambient temperature with time.
- 3-The individual components of building like walls, roof, windows, and other components not reach a peak heat gain at the same times, for example, the heat gain through the roof will be highest in the late afternoon, when it is warm outside and the sun has been shining on it all day. Conversely, the heat gains due to the sun shining through an east-facing window will be highest in the early morning when the sun is rising in the east and shining directly into the window.



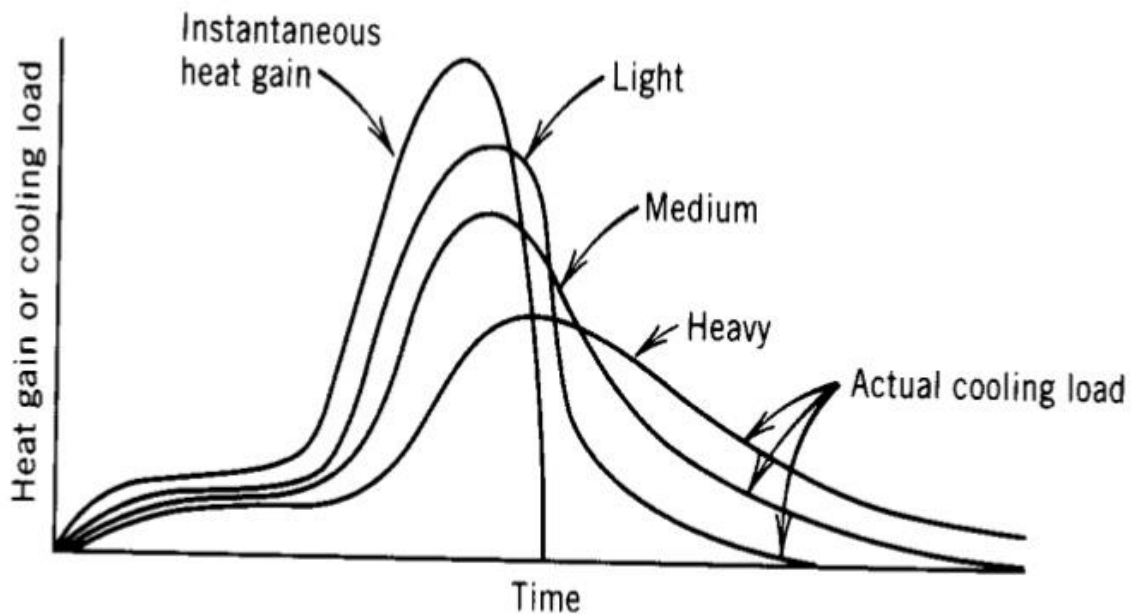
## 10.2 Effect of heat storage in the building's structures

The transition of heat through the walls and roof exposed to the outdoor is not steady due to variation in the ambient temperature and solar radiation intensity over 24 hours. The temperature of walls and roof rises with the rise in outside air temperature and solar radiation and the heat is stored in the wall and roof and other components of the building. The stored heat in the wall or roof is given off to the room when the outside air temperature falls. Therefore, *the instantaneous heat gain from outside is not equal to instantaneous heat gain inside the room*. The heat stored in the wall is given off to later in the evening. (see the figure)



The shaded area above the actual load shows the heat stored and below the actual load shows the heat released by the walls and other structures. The two areas are equal. The heat stored in walls and other structures depends on the thermal mass of the building which equals multiplying the mass of the structure by its specific heat capacity. The increase of thermal mass increases time lag and increasing thermal energy storage capability as shown in the figure below for three types of structure (light, medium, and heavy). The heavy structure (higher thermal mass) has a higher time lag and higher heat storage in comparison with a medium and light structure.

### Heat Gain $\neq$ Cooling Load Thermal storage and Construction Type





A convenient of taking into account this lagging effect of storage and the solar radiation is to use *cooling load temperature difference CLTD*. The heat gain through the external walls and roofs can be given as:

$$Q = U \times A \times CLTD$$

### 10.3 Sol-Air temperature

The sol-air temperature is a hypothetical temperature used to calculate the heat received by the outside surface of a building wall by the combined effect of convection and solar radiation.

The total heat received by the outside surface of the wall by convection and radiation is given by:

$$\begin{aligned}q_{os} &= q_c + q_r = f_o(T_o - T_{os}) + \alpha I \\q_{os} &= q_c + q_r = f_o T_o - f_o T_{os} + \alpha I \\&= f_o \left( T_o + \frac{\alpha I}{f_o} \right) - f_o T_{os} \\&= f_o T_e - f_o T_{os} \\q_{os} &= f_o (T_e - T_{os})\end{aligned}$$

Where;

$T_o$  : outside air temperature

$T_{os}$  : outside surface temperature

$T_e$  : sol-air temperature  $T_e = T_o + \frac{\alpha I}{f_o}$



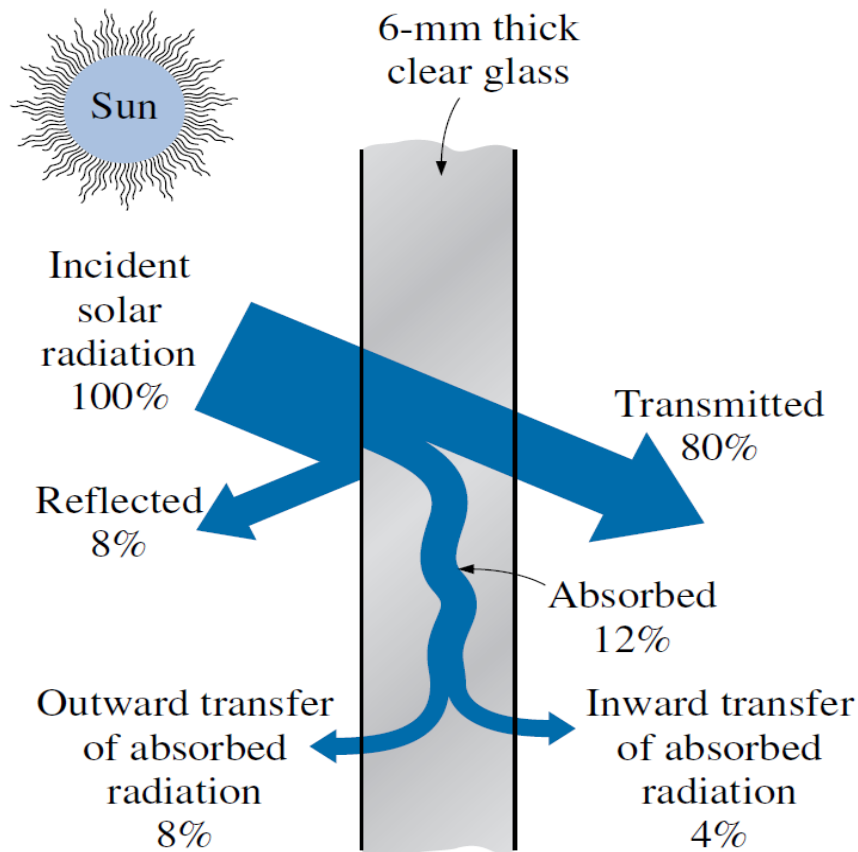
$I$  : total solar radiation intensity ( $\text{W}/\text{m}^2$ )

$\alpha$  : absorptivity of the surface

$f_o$  : outside heat transfer coefficient

### 10.4 Heat gain through the glass area

When solar radiation strikes a glass surface, part of it (about 8 percent for uncoated clear glass) is reflected back to outdoors, part of it (5 to 50 percent, depending on composition and thickness) is absorbed within the glass, and the remainder is transmitted indoors, as shown in Figure below.





the solar heat gain through that window is determined from:

$$Q = A \times SC \times SHG_{max} \times CLF, W$$

$SC$  is shading coefficient is given by:

$$SC = \frac{\text{solar heat gain of product}}{\text{solar heat gain of reference glazing}}$$

$CLF$  is the cooling load factor which refers to heat gains to building material due to solar radiation is not converted to cooling instantly but there is some time lag between heat gain and cooling.

The transmission heat gains through the glass area that results from the temperature difference between the outside and inside air temperature can be given as follows:

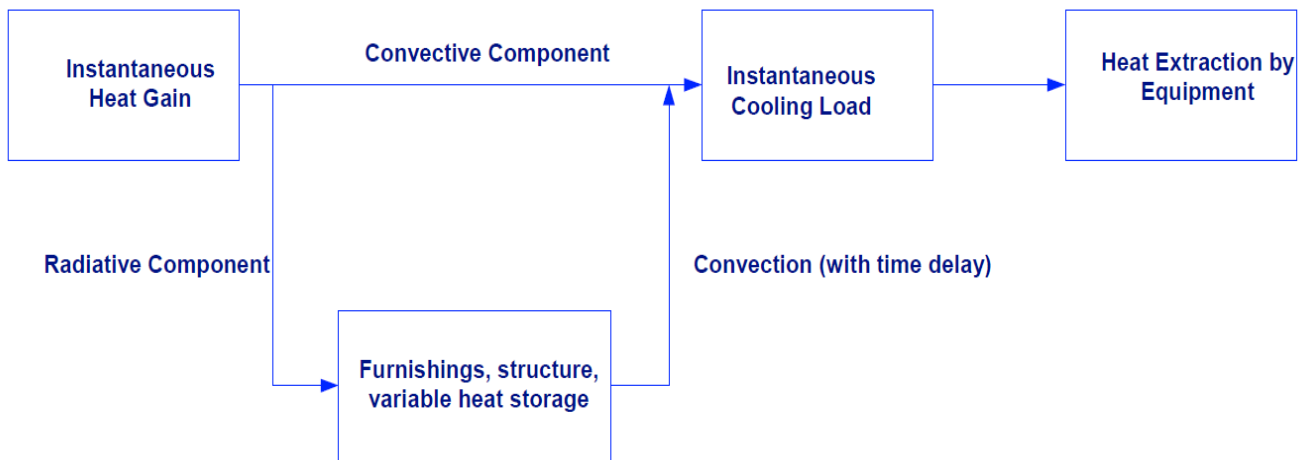
$$Q = A \times U \times CLTD$$

The total heat gain through the glass is:

$$Q = A \times (SC \times SHG_{max} \times CLF + U \times CLTD)$$

## 10.5 Cooling load estimation

The **cooling load** is an hourly rate at which heat must be removed from a building to hold the indoor air temperature at the design value. Theoretically, it may seem logical to address that the space heat gain is equivalent to space cooling load but in practice “**Heat gain  $\neq$  cooling load.**” as shown in the following diagram.



**Schematic Relation of Heat Gain to Cooling Load**

The primary explanation for this difference is the time lag or thermal storage effects of the building elements. Heat gains that enter a building are absorbed/stored by surfaces enclosing the space (walls, floors, and other interior elements). These elements radiate into space even after the heat gain sources are no longer present.



## 10.6 Classification of Cooling Loads

There are two types of cooling load; (a) the sensible load (heat gain) and (b) the latent load (water vapor gain).

### **Sensible Loads**

Sensible heat gain is the direct addition of heat to space, which shall increase space temperature. The factors influencing sensible cooling load:

- 1) Solar heat gain through the building envelope (exterior walls, glazing, skylights, roof, floors over crawl space)
- 2) Partitions (that separate spaces of different temperatures)
- 3) Ventilation air and air infiltration through cracks in the building, doors, and windows
- 4) People in the building
- 5) Equipment and appliances in the space.

### **Latent Loads**

A latent heat gain is heat contained in water vapor. Latent heat does not cause a temperature rise, but it constitutes a load on the cooling equipment. The latent load is the heat that must be removed to condense the moisture out of the air.

The sources of latent heat gain are:

- 1) People (breathing)
- 2) Cooking equipment
- 3) Housekeeping, floor washing, etc.
- 4) Appliances or machinery that evaporates water
- 5) Ventilation air and air infiltration through cracks in the building, doors, and windows

Cooling Loads Classified by Inside-Outside Environment as follows:



### a) External cooling load

1. Sensible loads through opaque envelope assemblies (roofs, walls, floors)
2. Sensible loads through transparent or translucent envelope assemblies (skylights, windows, glazed openings)
3. Sensible loads through ventilation and infiltration (air leakage)
4. Latent loads through ventilation and infiltration.

#### 1) Roofs, External Walls & Conduction through Glass

The equation used for sensible loads from the walls, roof, partitions, and the conduction through glass is:

$$Q = A \times U \times CLTD_c$$

for walls and roofs  $CLTD_c$  is:

$$CLTD_c = (CLTD + LM) \times K + (25.5 - T_R) + (T_o - 29.4)$$

for glass area  $CLTD_c$  is:

$$CLTD_c = CLTD + (25.5 - T_R) + (T_o - 29.4) ; \text{ (table 10)}$$

Where;

$LM$  : latitude-month correction (Table 9)

$K$  : color adjustment factor ( $K=1$  for a dark color, and  $K=0.5$  for light color)

$T_o$  : Average outside temperature on design day

$$T_o = \frac{T_{a.max} + T_{a.min}}{2} = T_{a.max} - \frac{1}{2}(\text{Temperaure daily range})$$



$T_R$  : indoor design temperature

## 2) Solar Load through Glass

$$Q = A \times SC \times SHG_{max} \times CLF$$

Where;

$SC$  : shading coefficient

$CLF$  : cooling load factor; (tables 13, 14)

$SHG_{max}$  : maximum solar heat gain; (table 11)

## 3) Partitions, Ceilings, and Floors

$$Q = A \times U \times (T_{aj} - T_R)$$

$T_{aj}$  : Temperature of adjacent space; in summer  $T_{aj}$  is given by:

$$T_{aj} = T_R + \frac{2}{3}(T_o - T_R)$$



## 4) Ventilation & Infiltration Air

Ventilation air is the amount of outdoor air required to maintain indoor air quality for the occupants

$$Q_s = 1.22 \dot{V} (T_o - T_i)$$

$$Q_L = 2940 \dot{V} (w_o - w_i)$$

$$Q = 1.2 \dot{V} (h_o - h_i)$$

$\dot{V}$  = Ventilation airflow rate.

$T_o$  = Outside dry bulb temperature of the air.

$T_i$  = inside dry bulb temperature of the air.

$w_o, w_i$  = Outside and inside moisture content.

$h_o, h_i$  = Outside and inside enthalpies.

### b. Internal Loads

Internal cooling loads consist of the following:

- 1) Sensible & latent loads due to people
- 2) Sensible loads due to lighting
- 3) Sensible loads due to power loads and motors (elevators, pumps, fans & other machinery)
- 4) Sensible & latent loads due to appliances

The equations used in estimating internal loads are:

#### 1) People

$$Q_s = N \times Q_{sp} \times CLF$$



$$Q_L = N \times Q_{Lp}$$

$N$  = number of people in space.

$Q_{sp}$ ,  $Q_{Lp}$  = Sensible and Latent heat gain from occupancy.

$CLF$  = Cooling Load Factor, by the hours of occupancy.

## 2) Lights

The lights result in sensible heat gain.

$$Q = 1.2 \times W \times CLF$$

For practical applications  $CLF = 1$  and  $Q = W$

In case of no data available about the lighting plan of the conditioned space, the lighting intensity is considered as  $25 \text{ W/m}^2$ .

## 3) Power Loads & Motors

Three different equations are used under different scenarios:

**A.** Heat gain of power-driven equipment and motor when both are located inside the space to be conditioned

$$Q = \frac{P}{\eta}$$

$P$  = power rating from electrical power plans or manufacturer's data

$\eta$  = Equipment motor efficiency, as a decimal fraction.

**B.** Heat gain of when driven equipment is located inside the space to be conditioned space and the motor is outside space or air stream



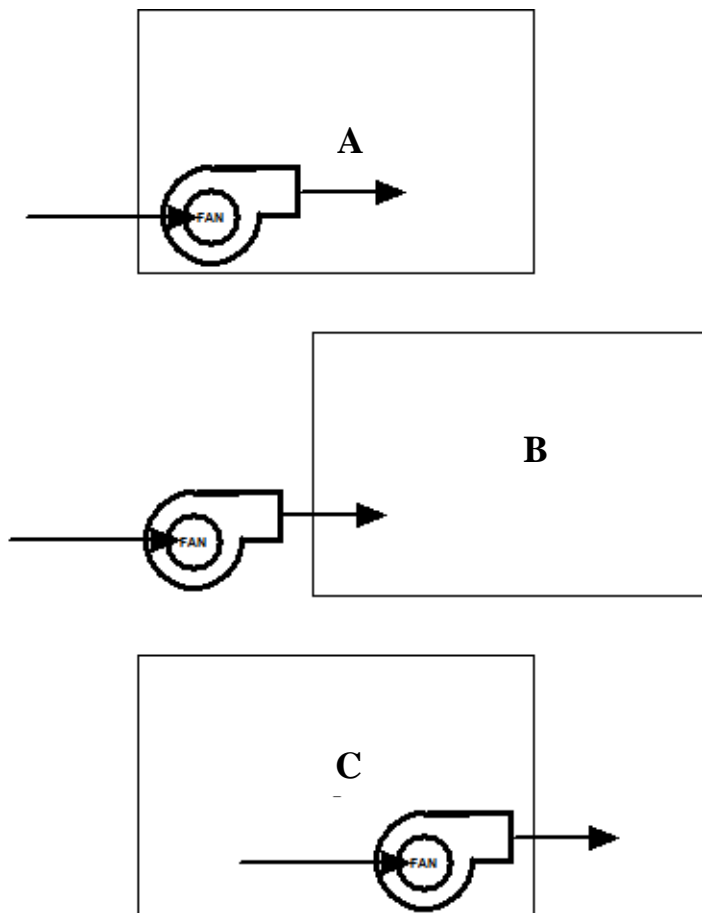
$$Q = P$$

**C.** Heat gain of when driven equipment is located outside the space to be conditioned space and the motor is inside space or air stream.

$$Q = P \times (1 - \eta) / \eta$$

P = Horsepower rating from electrical power plans or manufacturer's data

The figure below explaining the three cases.





## 4) Appliances

$$Q_s = W \times F_u \times CLF$$

$W$  = Installed rating of appliances in watts.

$CLF$  = Cooling Load Factor, by an hour of occupancy.

$F_u$  = Usage factor

$CLF = 1$ , if the operation is 24 hours. Here  $CLF$  value is taken as **1.0** for all applications.

Usage factor  $F_u$  according to ASHRAE fundamentals is 0.06-0.83, it can be taken as **0.4** (average value) for all appliances.

The latent heat due to appliances is:

$$Q_L = W \times \text{Latent heat fraction}$$

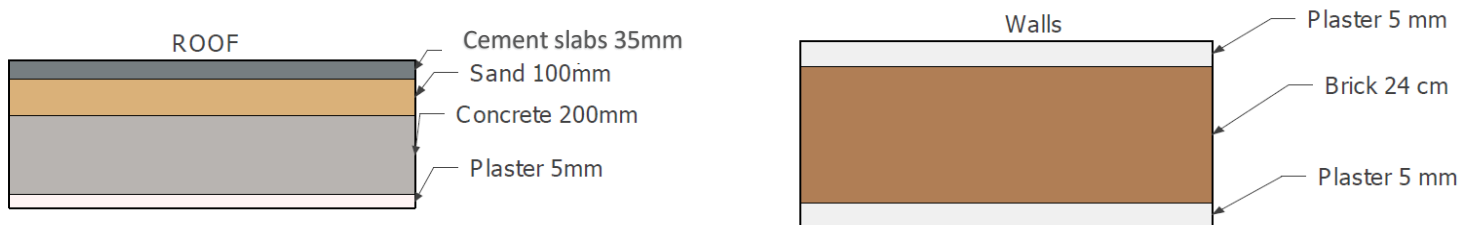
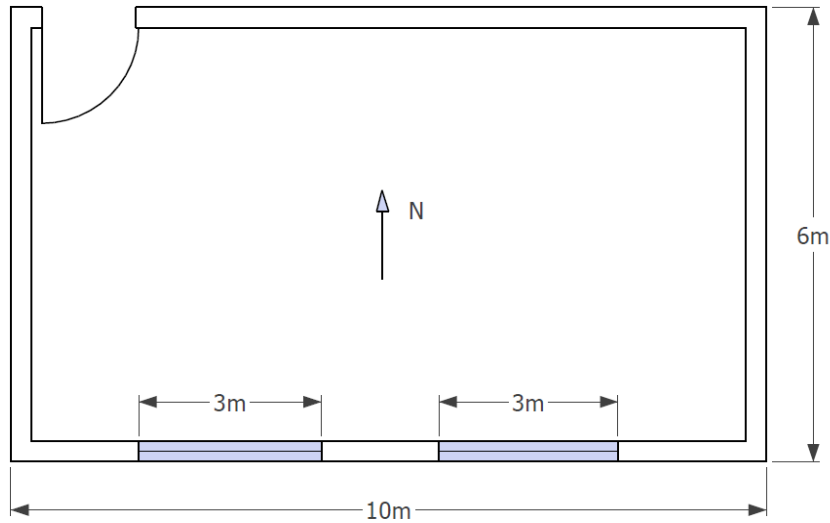
### c. Heat Gain from HVAC System

Supply and/or return fans that circulate or supply air to space add heat to space or system depending on the location relative to the conditioned space. This heat gain is normally estimated as a percentage of the space sensible cooling load (usually 1% to 5%) added to the coil cooling load.

#### Example 11.1

A classroom size (10 × 6 × 4)m lies at latitude 32° N. 40 students attending at 8 am and leaving at 4 pm. The outside conditions are 50 °C maximum dry bulb temperature, 32 °C minimum (DBT), and 25% relative humidity. The inside design conditions are 25 °C (DBT) and 50% relative humidity. The classroom has two windows of (3 × 2)m and a door of (2 × 2.5)m. Neglect heat gains from the door, calculate the cooling load

of the classroom at 5 pm in July. The thermal conductivity of the structure's components is listed in the table below.



Material	Thickness mm	Thermal conductivity W/m.K
Concrete	200	0.9
Brick	240	0.7
Sand	100	2
Cement slabs	35	0.71
Plaster	5	0.75
Glass	6	1.5



## Solution

### A-Walls

$$R_t = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} = \frac{1}{9.37} + \frac{0.005}{0.75} + \frac{0.24}{0.7} + \frac{0.005}{0.75} + \frac{1}{22.7}$$

$$R_t = 0.506 \text{ m}^2 \cdot \text{°C/W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.506} = 1.976 \text{ W/m}^2 \text{ °C}$$

#### 1.South wall

*At 5 pm(17.00), wall group B; Table 7*

$$CLTD = 10 \text{ °C}$$

$$LM = -1.6 \text{ °C, Table 9}$$

$$CLTD_c = CLTD + LM + (25.5 - T_R) + (T_o - 29.4)$$

$$T_o = \frac{T_{max} + T_{min}}{2} = \frac{50 + 32}{2} = 41 \text{ °C}$$

$$CLTD_c = 10 - 1.6 + (25.5 - 25) + (41 - 29.4)$$

$$CLTD_c = 20.5 \text{ °C}$$

$$Q = U \times A \times CLTD_c$$

$$Q = 1.976 \times [(10 \times 4) - 2(3 \times 2)] \times 20.5$$

$$\mathbf{Q = 1134.224W}$$



## 2. West wall

*At 5 pm(17.00), wall group B; Table 7*

$$CLTD = 9 \text{ } ^\circ\text{C}$$

$$LM = 0 \text{ } ^\circ\text{C}, \text{ Table 9}$$

$$CLTD_c = CLTD + LM + (25.5 - T_R) + (T_o - 29.4)$$

$$CLTD_c = 9 + 0 + (25.5 - 25) + (41 - 29.4) = 21.4 \text{ } ^\circ\text{C}$$

$$Q = U \times A \times CLTD_c$$

$$Q = 1.976 \times (6 \times 4) \times 21.4$$

$$\mathbf{Q = 1014.874 \text{ W}}$$

## 3. East wall

*At 5 pm(17.00), wall group B; Table 7*

$$CLTD = 14 \text{ } ^\circ\text{C}$$

$$LM = 0 \text{ } ^\circ\text{C}, \text{ Table 9}$$

$$CLTD_c = CLTD + LM + (25.5 - T_R) + (T_o - 29.4)$$

$$CLTD_c = 14 + 0 + (25.5 - 25) + (41 - 29.4) = 26.1 \text{ } ^\circ\text{C}$$

$$Q = U \times A \times CLTD_c$$

$$Q = 1.976 \times (6 \times 4) \times 26.1$$

$$\mathbf{Q = 1237.77 \text{ W}}$$



#### 4. North wall

At 5 pm(17.00), wall group B; Table 7

$$CLTD = 6 \text{ } ^\circ\text{C}$$

$$LM = 0.5 \text{ } ^\circ\text{C}, \text{ Table 9}$$

$$CLTD_c = CLTD + LM + (25.5 - T_R) + (T_o - 29.4)$$

$$CLTD_c = 6 + 0.5 + (25.5 - 25) + (41 - 29.4) = 18.6 \text{ } ^\circ\text{C}$$

$$Q = U \times A \times CLTD_c$$

$$Q = 1.976 \times [(10 \times 4) - (2 \times 2.5)] \times 18.6$$

$$Q = 1286.376 \text{ W}$$

#### B-Roof

At 5 pm(17.00), roof No.8; Table 5

$$R_t = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{1}{f_o}$$

$$R_t = \frac{1}{9.37} + \frac{0.005}{0.75} + \frac{0.2}{0.9} + \frac{0.1}{2} + \frac{0.035}{0.71} + \frac{1}{22.7} = 0.478 \text{ m}^2 \cdot ^\circ\text{C}/\text{W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.478} = 2.092 \text{ W/m}^2 \text{ } ^\circ\text{C}$$

$$CLTD = 25 \text{ } ^\circ\text{C}$$

$$LM = 0.5 \text{ } ^\circ\text{C}; \text{ Table 9}$$

$$CLTD_c = CLTD + LM + (25.5 - T_R) + (T_o - 29.4)$$

$$CLTD_c = 25 + 0.5 + (25.5 - 25) + (41 - 29.4) = 37.6 \text{ } ^\circ\text{C}$$

$$Q = U \times A \times CLTD_c = 2.092 \times 10 \times 6 \times 37.6$$

$$Q = 4719.552 \text{ W}$$



### C.Windows

$$R_t = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} = \frac{1}{9.27} + \frac{0.006}{1.5} + \frac{1}{22.7} = 0.154 \text{ m}^2 \cdot \text{°C/W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.154} = 6.494 \text{ W/m}^2 \text{ °C}$$

$$CLTD = 7 \text{ °C ; Table 10}$$

$$CLTD_c = CLTD + (25.5 - T_R) + (T_o - 29.4)$$

$$CLTD_c = 7 + (25.5 - T_R) + (T_o - 29.4) = 19.1 \text{ °C}$$

$$Q_c = U \times A \times CLTD_c = 6.494 \times 2(3 \times 2) \times 19.1 = \mathbf{1488.425 \text{ W}}$$

Radiant heat gain;

$$Q_{SR} = A \times SC \times SHG_{max} \times CLF$$

$$SHG_{max} = 227 \text{ W , Table 11}$$

$$SC = 0.95 , \text{Table 26}$$

$$CLF = 0.41 , \text{Table 13}$$

$$Q_{SR} = 12 \times 0.95 \times 227 \times 0.41 = \mathbf{1061 \text{ W}}$$

$$\mathbf{Q = 1488.425 + 1061 = 2549.425 \text{ W}}$$

### D.Infiltration

$$ACH = 1.5 , \text{Table 27}$$

$$\dot{V}_{inf} = \text{space volume} \times \frac{ACH}{3600} = 10 \times 6 \times 4 \frac{1.5}{3600} = 0.1 \text{ m}^3/\text{s}$$

$$Q_{inf} = 1.2 \times \dot{V}_{inf} \times (h_o - h_i)$$

$$h_o = 100.94 \text{ kJ/kg ; [50°C(DBT), 25% RH]}$$



$$h_i = 50.29 \text{ kJ/kg} ; [25^\circ\text{C(DBT), 50\% RH}]$$

$$Q_{inf} = 1.2 \times 0.1 \times (100.94 - 50.29) = \mathbf{6.078 \text{ kW}}$$

$$\mathbf{Q_{inf} = 6078 \text{ W}}$$

## E. Ventilation

Ventilation requirements according to the activity in the space is 5 (L/s)/person.

$$\dot{V}_{vent} = 40 \times 5 = 200 \text{ L/s} = 0.2 \text{ m}^3/\text{s}, \text{ Table 28}$$

$$Q_{vent} = 1.2 \times \dot{V}_{vent} \times (h_o - h_i)$$

The infiltration rate is 0.1 m<sup>3</sup>/s and it covers 50% of the fresh air required for ventilation. Therefore only 0.1 m<sup>3</sup>/s of fresh air needs and the total heat gain due to the ventilation air becomes:

$$Q_{vent} = 1.2 \times 0.1 \times (100.94 - 50.29) = \mathbf{6.078 \text{ kW}}$$

$$\mathbf{Q_{vent} = 6078 \text{ W}}$$

## F. Peoples

Sensible and latent heat gain per person are:

**Table 18**

$$Q_s/p = 60 \text{ W}$$

$$Q_L/p = 40 \text{ W}$$

Total sensible heat gain  $Q_s$  is:

$$Q_s = N \times Q_s/p \times CLF$$





The cooling load factor of 9 hours after entry into the space  $CLF$  is:

$$CLF = 0.38$$

$$Q_s = 40 \times 60 \times 0.38 = \mathbf{912\ W}$$

$$Q_L = 0$$

Total heat gain from peoples

$$Q = Q_s + Q_L = 912 + 0$$

$$\mathbf{Q = 912\ W}$$

## G.Lighting

In case of no information about the lighting plan, the default value can be taken as  $25\ \text{W/m}^2$

$$Q = \text{lighting intensity} \times A \times CLF$$

*In all calculations related to lighting heat gain, CLF is considered equal to 1.*

$$Q = \text{lighting intensity} \times A \times CLF = 25 \times 10 \times 6 \times 1 = 1500\ \text{W} \quad \mathbf{Q =}$$

$\mathbf{1500\ W}$  The total cooling load of the classroom is:

$$Q_t = 1134.224 + 1014.874 + 1237.77 + 1286.376 + 4719.552 + 2549.425 + 6078 + 912 + 1500$$

$$\mathbf{Q_t = 20432.221\ W = 5.81\ TR \approx 6\ TR}$$

coil cooling load capacity = space cooling load + ventilation cooling load =  $20432.221 + 6078 = 26510.221\ \text{W} = \mathbf{7.54\ TR}$



**Table. Summary of the cooling load calculations**

Components	U ,W/m <sup>2</sup> °C	A m <sup>2</sup>	CLTDc °C	SHG <sub>max</sub> W/m <sup>2</sup>	CLF	Q <sub>t</sub> , W
South wall	1.976	28	20.5	-	-	1134.224
West wall	1.976	24	21.4	-	-	1014.874
East wall	1.976	24	26.1	-	-	1237.77
North wall	1.976	35	18.6	-	-	1286.376
Roof	2.092	60	37.6	-	-	4719.552
Windows	6.494	12	19.1	227	0.41	2549.425
Infiltration	-		-	-	-	6078
Ventilation	-		-	-	-	6078
Peoples	-		-	-	0.38	912
Lighting	-	60	-	-	1	1500
<b>Total</b>	<b>-</b>		<b>-</b>	<b>-</b>	<b>-</b>	<b>26510. 221</b>



## **12.1 Heating load Calculations**

Heating load calculations are carried out to estimate the total heat losses from the building in winter. The heat load calculations are executed assuming steady-state conditions (no solar radiation and steady outdoor conditions) and neglecting internal heat sources.

The heating load includes:

- (a) Conduction heat loss to adjoining spaces through the ceiling, interior partition walls, and floor.
- (b) Heat loss due to cold air infiltrating into space from outdoors through doors, windows, and small cracks in the building envelope.
- (c) Heat loss due to ventilation purposes.

### **(a) Conduction heat loss**

When calculating heating loss by conduction through the roof, the exterior walls, and the windows, no credit is given for the effect of the sun shining on the outside surfaces. With this assumption, the amount of heat transferred through the surface is a direct result of the temperature difference between the outdoor and indoor surfaces ( *$\Delta T$  is used instead of CLTD*).

The conductive heat loss through a roof, an exterior wall, or a window can be given as follows:

$$Q = U \times A \times \Delta T$$

Where;

$Q$  = heat loss by conduction [W]

$U$  = overall heat-transfer coefficient of the surface [W/m<sup>2</sup> °C]

$A$  = area of the surface, [m<sup>2</sup>]



$\Delta T = (T_i - T_o)$  = desired indoor dry-bulb temperature ( $T_i$ ) minus the design outdoor dry-bulb temperature ( $T_o$ ).

### (b) Heat loss due to infiltration

During the winter season, air may leak in from outdoors through doors, windows, and small cracks in the building envelope. It contributes to the sensible heat loss of the space because the outdoor air is typically colder than the indoor air.

The outdoor air during the winter season is generally drier than the indoor air. If the building requires humidification, then the latent heat load is considered. The heat loss due to infiltration air is given by:

$$Q_s = 1.22 \dot{V}_{inf} (T_i - T_o)$$

$$Q_L = 2940 \dot{V}_{inf} (w_i - w_o)$$

$$Q = 1.2 \dot{V}_{inf} (h_i - h_o)$$

### (c) Heat loss due to ventilation purposes.

Heat loss due to ventilation is similar to that of infiltration except that the ventilation loss is added to heating coil load while the infiltration loss is added to space heating load.

$$Q_s = 1.22 \dot{V}_{vent} (T_i - T_o)$$

$$Q_L = 2940 \dot{V}_{vent} (w_i - w_o)$$

$$Q = 1.2 \dot{V}_{vent} (h_i - h_o)$$

## 12.2 Inside wall surface temperature

The inside surface temperatures of the walls, roofs are very important in the Winter season to prevent the risk of vapor condensation on the interior wall and roof surfaces. The surface condensation occurs when the surface temperature falling below the dew point temperature of the inside air.

The heat loss from the wall is:

$$Q = \frac{1}{R_t} \times A \times (T_i - T_o)$$

Also, the heat loss through the wall is given as:

$$Q = \frac{1}{R_{fi}} \times A \times (T_i - T_s)$$

From the two above equations, we can write:

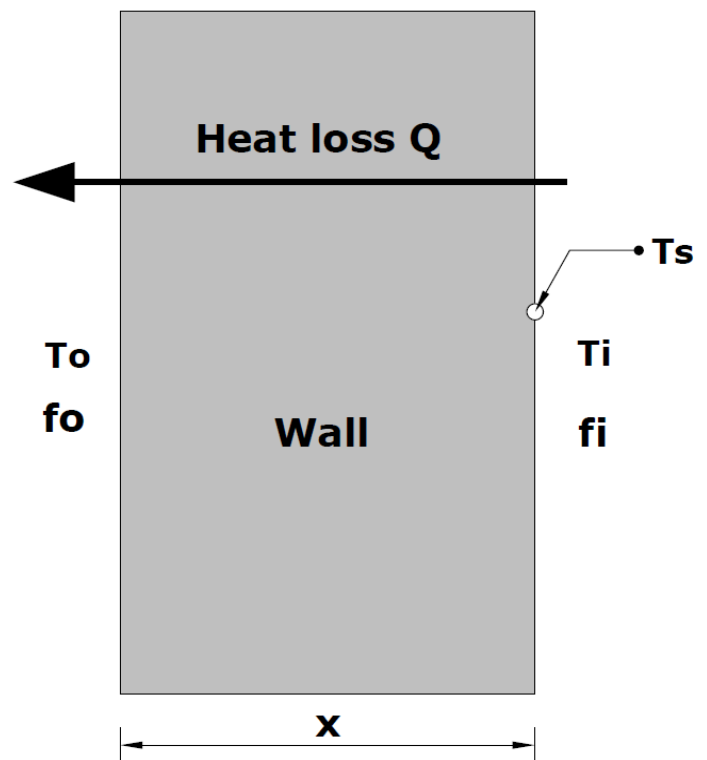
$$\frac{1}{R_t} \times (T_i - T_o) = \frac{1}{R_{fi}} \times (T_i - T_s)$$

$$\frac{R_t}{R_{fi}} = \frac{T_i - T_o}{T_i - T_s}$$

Where;

$$R_t = \frac{1}{f_o} + \frac{x}{k} + \frac{1}{f_i}$$

$$R_{fi} = \frac{1}{f_i}$$





### **12.3 Unconditioned spaces**

Frequently it may be necessary to know the air temperature of an unheated area in Winter, like an *attic* space or of a *storage room*, or any *unconditioned space*. We need to know the approximate temperature in these areas to estimate heat losses through wall partitions separating conditioned rooms from these unconditioned areas. In winter the temperature of unconditioned space can be given by an approximate relation as follows:

$$T_{aj} = T_i - \frac{1}{2}(T_i - T_o)$$

$T_{aj}$  : Temperature of adjacent space.

$T_o$  : Outside air temperature.

$T_i$  : inside air temperature.

In winter the ***floor temperature*** is required to calculate heat loss through the floor. In the absence of any information about the floor temperature, it can be taken as a daily mean ambient temperature in the coldest month of the region considered. For example, in the south of Iraq, the floor temperature can be taken in the range of ***10 °C to 14 °C***.

#### **Example12.1:**

Room walls are composed of common bricks and outside and inside plaster layers with an overall heat transfer coefficient of  $2.5 \text{ W/m}^2 \text{ }^\circ\text{C}$ . The outside air temperature is  $-15^\circ\text{C}$  (DBT) and the inside design conditions



are 23 °C DBT, 55% RH. Is condensation occur on the inside surface or not? Take  $f_i = 8.26 \text{ W/m}^2 \text{ }^\circ\text{C}$ .

### Solution:

Heat loss through the walls per unit area is calculated by:

$$Q = U \times (T_i - T_o) = 2.5 \times \{23 - (-15)\} = 95 \text{ W/m}^2$$

The heat transfer between the inside surface and the inside temperature is:

$$Q = f_i \times (T_i - T_s)$$

$$95 = 8.26 \times (23 - T_s)$$

$$T_s = 11.5 \text{ }^\circ\text{C}$$

The dew point temperature at the inside conditions (23 °C and 55% RH.) is  $T_{dp} = 13.5 \text{ }^\circ\text{C}$ .

*Since the inside surface temperature less than the dew point temperature of the inside air, condensation will occur on the inside surface.*

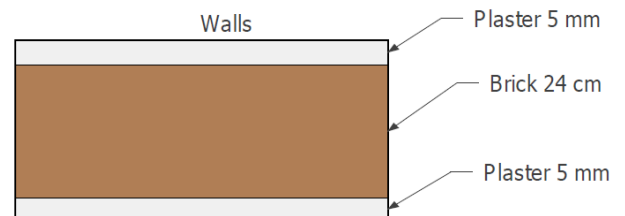
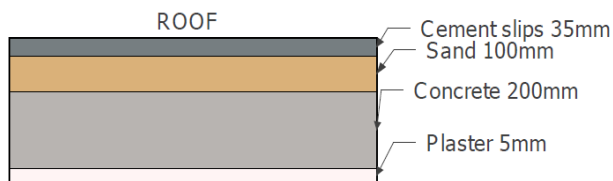
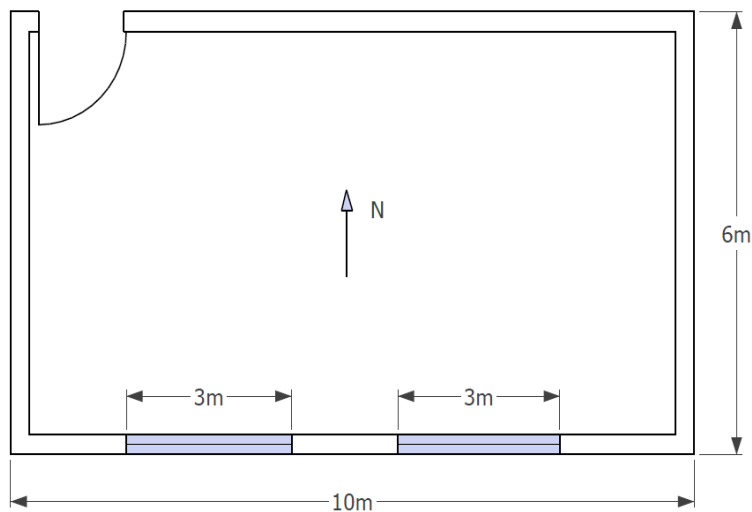
### Example12.2:

40 students' capacity classroom size (10 × 6 × 4)m lies in the south of Iraq. The outside conditions are 5°C DBT and 80% RH. The inside design conditions are 24 °C (DBT) and 50% RH. The classroom has two windows of (3 × 2)m and a door of (2 × 2.5)m. Neglect heat loss from



the door, calculate the heating load of the classroom in January. The thermal conductivity of the structure's components is listed in the table below.

Material	Thickness mm	Thermal conductivity W/m.K
Concrete	200	0.9
Brick	240	0.7
Sand	100	2
Cement slabs	35	0.71
Plaster	5	0.75
Glass	6	1.5







## Solution:

### 1-Walls

$$R_t = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} = \frac{1}{9.37} + \frac{0.005}{0.75} + \frac{0.24}{0.7} + \frac{0.005}{0.75} + \frac{1}{22.7}$$

$$R_t = 0.506 \text{ m}^2 \cdot \text{°C/W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.506} = 1.976 \text{ W/m}^2 \text{ °C}$$

$$Q = U \times A \times (T_i - T_o)$$

The total area of the walls  $A = 2(6 \times 4) + (10 \times 4) - (2.5 \times 2) + (10 \times 4) - 2(3 \times 2) = 111 \text{ m}^2$

$$Q = U \times A \times (T_i - T_o) = 1.976 \times 111 \times (24 - 5) = \mathbf{4167.384 \text{ W}}$$

### 2-Roof

$$R_t = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{1}{f_o}$$

$$R_t = \frac{1}{9.37} + \frac{0.005}{0.75} + \frac{0.2}{0.9} + \frac{0.1}{2} + \frac{0.035}{0.71} + \frac{1}{22.7} = 0.478 \text{ m}^2 \cdot \text{°C/W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.478} = 2.092 \text{ W/m}^2 \text{ °C}$$

$$Q = U \times A \times (T_i - T_o) = 2.092 \times (10 \times 6) \times (24 - 5) = \mathbf{2384.88 \text{ W}}$$



### 3-Windows

$$R_t = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} = \frac{1}{9.27} + \frac{0.006}{1.5} + \frac{1}{22.7} = 0.154 \text{ m}^2 \cdot \text{°C/W}$$

$$U = \frac{1}{R_t} = \frac{1}{0.154} = 6.494 \text{ W/m}^2 \text{ °C}$$

$$Q = U \times A \times (T_i - T_o) = 6.494 \times 2(3 \times 2) \times (24 - 5) =$$

**1480.632W**

### 4.Infiltration

$ACH = 1.5$  , *Table 27*

$$\dot{V}_{inf} = \text{space volume} \times \frac{ACH}{3600} = 10 \times 6 \times 4 \frac{1.5}{3600} = 0.1 \text{ m}^3/\text{s}$$

$$Q_{inf} = 1.2 \times \dot{V}_{inf} \times (h_i - h_o)$$

$$h_o = 15.85 \text{ kJ/kg} ; [5 \text{ °C(DBT), 80\% RH}]$$

$$h_i = 47.79 \text{ kJ/kg} ; [24 \text{ °C(DBT), 50\% RH}]$$

$$Q_{inf} = 1.2 \times 0.1 \times (47.79 - 15.85) = \mathbf{3.833 \text{ kW}}$$

### 5.floor

The floor temperature is considered here as 14 °C

$$Q = f_i \times A \times (T_i - T_o) = 9.37 \times (10 \times 6) \times (24 - 14) = \mathbf{5622W}$$

The total space heating load is:



$$Q_t = 4167.384 + 2384.88 + 1480.632 + 3833 + 5622 = 17487.896 = 17.488 \text{ kW} = 4.972 \text{ TH}$$

## 5. Ventilation

Ventilation requirements according to the activity in the space is 5 (L/s)/person.

$$\dot{V}_{vent} = 40 \times 5 = 200 \text{ L/s} = 0.2 \text{ m}^3/\text{s}, \text{ Table 28}$$

$$Q_{vent} = 1.2 \times \dot{V}_{vent} \times (h_i - h_o)$$

The infiltration rate is 0.1 m<sup>3</sup>/s and it covers 50% of the fresh air required for ventilation. Therefore only 0.1 m<sup>3</sup>/s of fresh air needs and the total heat gain due to the ventilation air becomes:

$$Q_{vent} = 1.2 \times 0.1 \times (47.79 - 15.85) = 3.833 \text{ kW}$$

The heating coil capacity required is:

$$Q_c = Q_t + Q_{vent} = 17.488 + 3.833 \text{ kW} = 21.321 \text{ kW} \approx 6 \text{ TH}$$



## 13.1 Duct design

In the absence of viscosity or shear stress, Bernoulli's Equation applies to an ideal fluid as follows:

$$P_s + P_d + \rho g z = P_T = \text{constant}$$

$P_s$  = Static pressure

$P_d$  = Velocity or dynamic pressure

$\rho g z$  = Pressure due to datum level

The dynamic pressure  $P_d$  equal to:

$$P_d = \frac{1}{2} \rho V^2, \quad V \text{ is the air velocity.}$$

The total pressure between two points can be written in the following form:

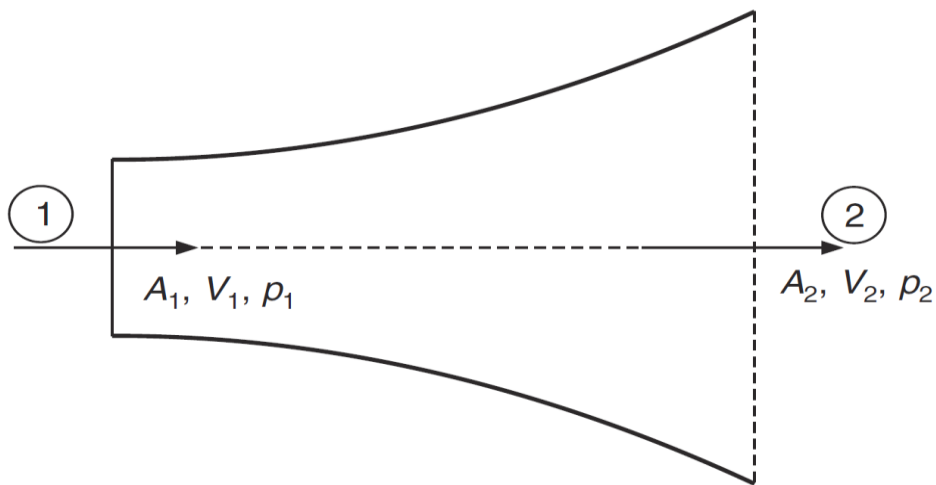
$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2$$

If there is a fan in the duct, which adds energy  $H_f$ , and if  $H_l$  is the loss in energy to overcome frictional effects, then:

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + H_f = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + H_l$$

### Example 13.1:

For the duct shown in the figure below,  $A_1 = 0.4 \text{ m}^2$  and  $A_2 = 1.6 \text{ m}^2$ . The volumetric flow rate is  $200 \text{ m}^3/\text{min}$ . The static pressure at inlet  $P_1 = 30 \text{ mm}$  gauge of water and the pressure drop in the duct due to friction is  $10 \text{ mm}$  of water. Find the static pressure  $P_2$ .



### Solution:

$$\frac{P_2}{\rho g} = \frac{P_1}{\rho g} + \frac{V_1^2 - V_2^2}{2g} + z_1 - z_2 + H_f - H_l$$

$H_f = 0.0$ ,  $P_1 = 0.03 \text{ m}$  of water gauge,  $H_l = 0.01 \text{ m}$  of water,  $\dot{Q} = 200 \text{ m}^3/\text{min}$ , we find  $V_1$  and  $V_2$  from the volumetric flowrate  $\dot{Q}$  the equation as follows:

$$\dot{Q} = A_1 V_1 = A_2 V_2$$

$$V_1 = \frac{\dot{Q}}{A_1} = \frac{200}{0.4} = 500 \text{ m/min.} = 8.33 \text{ m/s}$$



$$V_2 = \frac{\dot{Q}}{A_2} = \frac{200}{1.6} = 125 \text{ m/min.} = 2.083 \text{ m/s}$$

$$\frac{P_2}{\rho g} = \frac{P_1}{\rho g} + \frac{V_1^2 - V_2^2}{2g} + z_1 - z_2 + H_f - H_l$$

$$\frac{P_2}{\rho g} = 0.03 + \frac{8.33^2 - 2.083^2}{2 \times 9.81} + 0 - 0 + 0 - 0.01 = 0.024 \text{ m of water}$$

$$P_2 = \rho \times g \times 0.024 = 1000 \times 9.81 \times 0.024 = 235.44 \text{ Pa}$$

## 13.2 Pressure loss in ducts

The drops in the air duct are due to:

1. Fluid friction
2. Momentum change due to change of direction and/or velocity

The frictional **pressure drop (or loss)** is denoted as  $\Delta P_f$ . and the pressure drop due to momentum change is known as **dynamic pressure drop (or loss)**,  $\Delta P_d$ . the total pressure drop  $\Delta P_t$  can be given as:

$$\Delta P_t = \Delta P_f + \Delta P_d$$

The pressure drop due to friction in the duct may be obtained by using **the Darcy** formula or **Fanning** equation as follows:

$$\Delta P_f = \frac{1}{2} \rho f \frac{LV^2}{D}$$

Where;

$\Delta P_f$  = pressure loss due to friction in *Pa*.



$f$  = friction factor depending on the surface of the duct(dimensionless).

$V$  = mean air velocity in the duct in  $m/s$ .

$L$  = length of the duct in  $m$ .

$D$  = diameter of a circular duct and hydraulic diameter of a non-circular duct in  $m$ .

$\rho$  = air density in  $kg/m^3$ .

The friction factor  $f$  for smooth duct may be obtained by using the following relations:

1-for laminar flow:

$$f = \frac{64}{Re_D}$$

2-for turbulent flow:

$$f = \frac{0.3164}{Re_D^{0.25}}$$

In the case of rough duct surface and turbulent flow, the friction factor can be estimated from the empirical relation as follows:

$$\frac{1}{\sqrt{f}} = -2 \log \left[ \frac{K_s}{3.7 D} + \frac{2.51}{\sqrt{f} Re_D} \right]$$

$K_s$  = roughness of the inner surface of the duct.



the last equation can be solved by using the trial-and-error method or any recent software like MATLAB. The roughness of galvanized Iron (GI) sheet that widely used in air duct fabrication is equal to ***0.00015 m***

The frictional pressure drop in a circular duct may be obtained by using the following expression:

$$\Delta P_f = \frac{0.022243 \dot{Q}^{1.852} \times L}{D^{4.973}}$$

$\dot{Q}$  = volumetric flow rate of air,  $m^3/s$

$D$  = duct diameter,  $m$

$L$  = duct length,  $m$

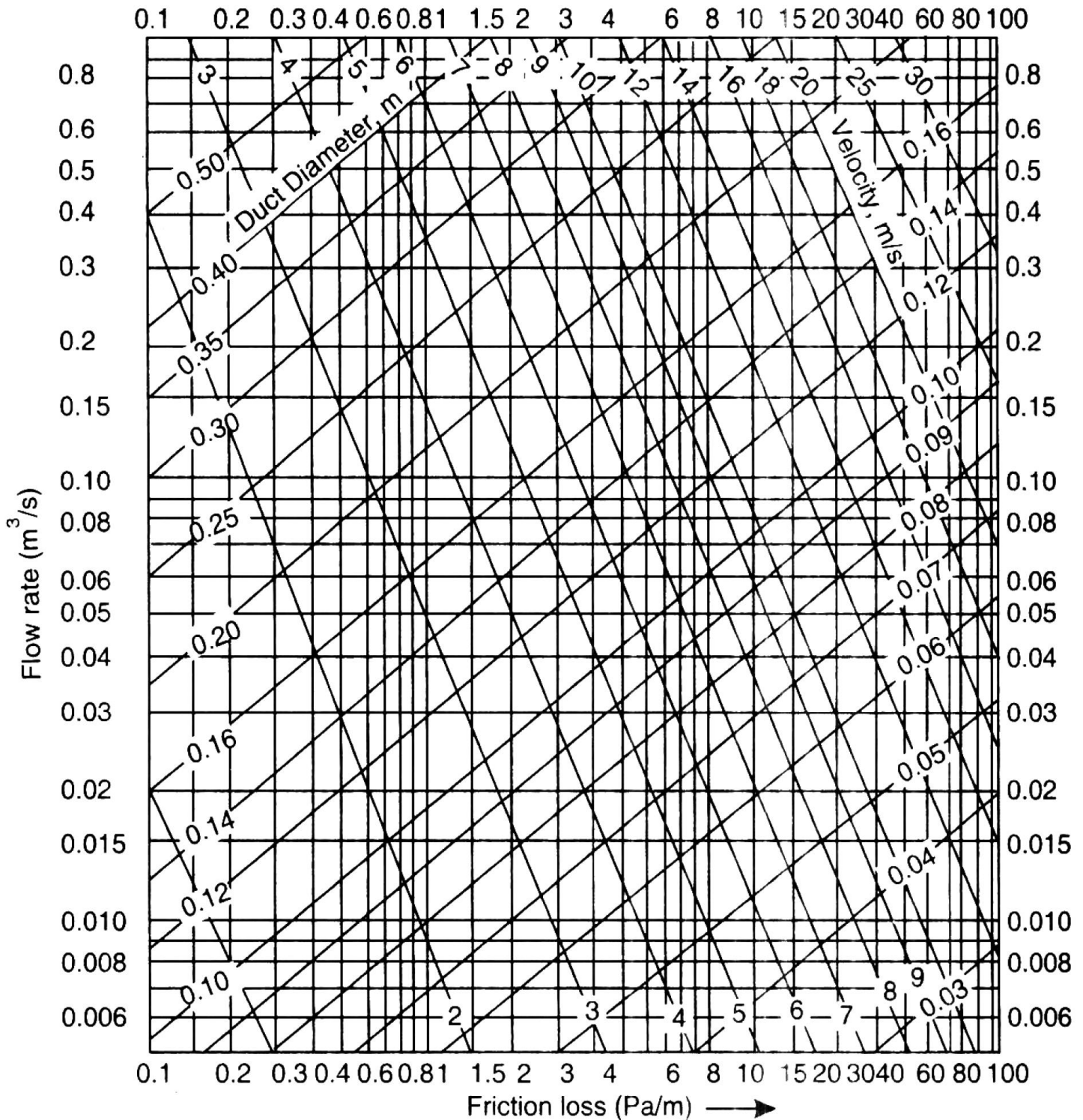
The frictional pressure drop can be also estimated from the chart for circular ducts made of GI sheets at standard air conditions (1 atm, 20 °C).

The dynamic pressure loss in the duct is caused due to the change in direction or magnitude of air velocity in the duct. It is given by:

$$\Delta P_d = K \frac{\rho v^2}{2}$$

$K$  = dynamic pressure loss coefficient.





**Friction chart for circular ducts**



### 13.3 CONVERSION FROM CIRCULAR TO RECTANGULAR DIMENSIONS

The circular cross-section has the minimum value of perimeter to the area of cross-section, i.e.,  $P/A$ . Hence, the frictional resistance offered by walls is minimum for ducts with a circular cross-section. Circular ducts are not convenient since they occupy large areas. Hence, ducts with rectangular cross-sections are used in air conditioning.

A rectangular duct is equivalent to a circular duct, if the volumetric flow rate  $\dot{Q}$  and frictional pressure drop per unit length ( $\Delta P_f/L$ ) are same for the both. The equivalent diameter is given by:

$$D = 1.3 \frac{(a \times b)^{0.625}}{(a+b)^{0.25}}$$

Where;  $a$  and  $b$  are the rectangular sides.

The above equation is found to be *valid for an aspect ratio less than or equal to 8:1*.

The two rectangular sides of the duct can be easily estimated by using the table on the next page.

#### **Example 13.2:**

Airflow in a circular duct of 506mm at a volumetric flow rate of 1500 m<sup>3</sup>/hr. calculate the friction pressure loss and converse the circular duct into a rectangular duct at an aspect ratio not less than 2:1.

### Solution:

$$\frac{\Delta P_f}{L} = \frac{0.022243 \dot{Q}^{1.852}}{D^{4.973}} = \frac{0.022243 \times (1500/3600)^{1.852}}{(0.506)^{4.973}} = 0.13 \text{ Pa/m}$$

From the table of circular equivalent duct, select ***a*** side of the rectangular as 300mm, thus the ***b*** side is 750mm, then the aspect ratio is  $750/300=2.5$ .

Table Circular Equivalents of Rectangular Duct for Equal Friction and Capacity<sup>a</sup>

Lgth Adj. <sup>b</sup>	Length of One Side of Rectangular Duct ( <i>a</i> ), mm																			
	100	125	150	175	200	225	250	275	300	350	400	450	500	550	600	650	700	750	800	900
	Circular Duct Diameter, mm																			
100	109																			
125	122	137																		
150	133	150	164																	
175	143	161	177	191																
200	152	172	189	204	219															
225	161	181	200	216	232	246														
250	169	190	210	228	244	259	273													
275	176	199	220	238	256	272	287	301												
300	183	207	229	248	266	283	299	314	328											
350	195	222	245	267	286	305	322	339	354	383										
400	207	235	260	283	305	325	343	361	378	409	437									
450	217	247	274	299	321	343	363	382	400	433	464	492								
500	227	258	287	313	337	360	381	401	420	455	488	518	547							
550	236	269	299	326	352	375	398	419	439	477	511	543	573	601						
600	245	279	310	339	365	390	414	436	457	496	533	567	598	628	656					
650	253	289	321	351	378	404	429	452	474	515	553	589	622	653	683	711				
700	261	298	331	362	391	418	443	467	490	533	573	610	644	677	708	737	765			
750	268	306	341	373	402	430	457	482	506	550	592	630	666	700	732	763	792	820		
800	275	314	350	383	414	442	470	496	520	567	609	649	687	722	755	787	818	847	875	
900	289	330	367	402	435	465	494	522	548	597	643	686	726	763	799	833	866	897	927	984
1000	301	344	384	420	454	486	517	546	574	626	674	719	762	802	840	876	911	944	976	1037
1100	313	358	399	437	473	506	538	569	598	652	703	751	795	838	878	916	953	988	1022	1086
1200	324	370	413	453	490	525	558	590	620	677	731	780	827	872	914	954	993	1030	1066	1133
1300	334	382	426	468	506	543	577	610	642	701	757	808	857	904	948	990	1031	1069	1107	1177
1400	344	394	439	482	522	559	595	629	662	724	781	835	886	934	980	1024	1066	1107	1146	1220
1500	353	404	452	495	536	575	612	648	681	745	805	860	913	963	1011	1057	1100	1143	1183	1260
1600	362	415	463	508	551	591	629	665	700	766	827	885	939	991	1041	1088	1133	1177	1219	1298
1700	371	425	475	521	564	605	644	682	718	785	849	908	964	1018	1069	1118	1164	1209	1253	1335
1800	379	434	485	533	577	619	660	698	735	804	869	930	988	1043	1096	1146	1195	1241	1286	1371
1900	387	444	496	544	590	633	674	713	751	823	889	952	1012	1068	1122	1174	1224	1271	1318	1405
2000	395	453	506	555	602	646	688	728	767	840	908	973	1034	1092	1147	1200	1252	1301	1348	1438
2100	402	461	516	566	614	659	702	743	782	857	927	993	1055	1115	1172	1226	1279	1329	1378	1470
2200	410	470	525	577	625	671	715	757	797	874	945	1013	1076	1137	1195	1251	1305	1356	1406	1501
2300	417	478	534	587	636	683	728	771	812	890	963	1031	1097	1159	1218	1275	1330	1383	1434	1532
2400	424	486	543	597	647	695	740	784	826	905	980	1050	1116	1180	1241	1299	1355	1409	1461	1561
2500	430	494	552	606	658	706	753	797	840	920	996	1068	1136	1200	1262	1322	1379	1434	1488	1589
2600	437	501	560	616	668	717	764	810	853	935	1012	1085	1154	1220	1283	1344	1402	1459	1513	1617
2700	443	509	569	625	678	728	776	822	866	950	1028	1102	1173	1240	1304	1366	1425	1483	1538	1644
2800	450	516	577	634	688	738	787	834	879	964	1043	1119	1190	1259	1324	1387	1447	1506	1562	1670
2900	456	523	585	643	697	749	798	845	891	977	1058	1135	1208	1277	1344	1408	1469	1529	1586	1696



## 13.4 Methods for determination of duct size

The following three methods for determination of duct size:

- 1-Velocity reduction method
- 2-Equal pressure drop(Equal friction method EFM)
- 3-Static regain method

### **1-Velocity reduction method**

In this method, the velocity in the ducts is assumed such that they progressively decrease as the flow proceeds. The pressure drop is calculated for these velocities for respective branches and main duct. The duct size is determined for assumed velocities and known air flow rate through the respective ducts. This method is used for very simple duct systems.

### **2-Equal pressure drop(Equal friction method EFM)**

In this method, the size of the duct is decided to give equal pressure drop(or friction loss) per meter length in all ducts, i.e.,

$$\left(\frac{\Delta P_f}{L}\right)_a = \left(\frac{\Delta P_f}{L}\right)_b = \left(\frac{\Delta P_f}{L}\right)_c = \dots$$

the velocities are automatically reduced in the branch ducts as the flow is decreased.

The procedure of duct sizing by using equal friction methods as follows:

- 1-Select air velocity in the main ducts(directly after air the handling unit) according to the recommended air supply velocities by ASHRAE standard.



2-from the given volumetric flow rate in the main duct and the selected velocity, calculate the duct diameter as follows:

$$\dot{Q} = A \times V = \frac{\pi}{4} D^2 \times V$$

3- The pressure drop can be calculated from the chart or the following expression:

$$\frac{\Delta P_f}{L} = \frac{0.022243 \dot{Q}^{1.852}}{D^{4.973}}$$

4-Duct size for any branch now is determined using the pressure drop calculated in step3 and volumetric flow rate  $\dot{Q}$  in each branch.

5-Converse the circular duct into a rectangular or square duct by using the table or the relation:

$$D = 1.3 \frac{(a \times b)^{0.625}}{(a+b)^{0.25}}$$

### 3-Static regain method

The Static Regain method of duct sizing is based on Bernoulli's equation, which states that when a reduction of velocities takes place, a conversion of dynamic pressure into static pressure occurs.

In this method, the size of the duct is decided to give equal pressure at all outlets. This may be done by equalizing the pressure losses in various branches. In this method, the length of all ducts and pressure losses in elbows, air diffusers, grilles, and any other obstacles should be known.

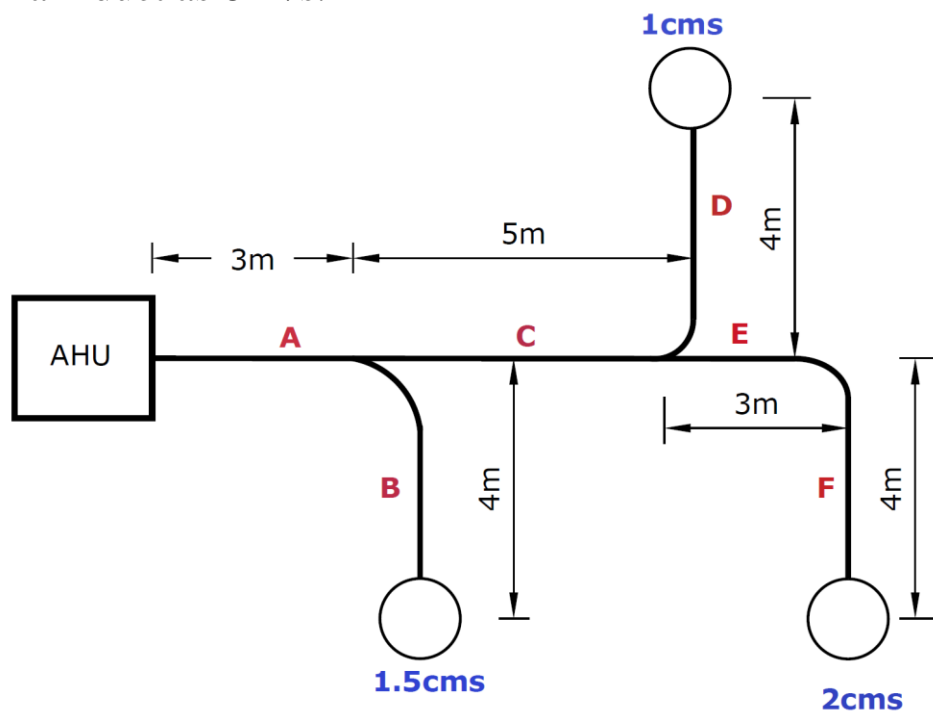
## 14.1 Duct Design(continue)

Table: Recommended maximum air velocity in low -velocity duct system(m/s)

Application	Main ducts		Branch ducts	
	Supply	Return	Supply	Return
Residences Apartments Hotel bed rooms	5	4	3	3
Hospital bed rooms	7.5	6.5	6	5
Office, Libraries	10	7.5	8	6
Theaters	6.5	5.5	5	4
Stores, Banks	6.5	5.5	5	4
Restaurants	10	7.5	8	6
Cafeterias	10	7.5	6	6

### Example 14.1

Determine the dimensions of all the ducts in the system shown in the figure by using the equal friction method. Take the velocity at the inlet of the main duct as 8 m/s.





### **Solution:**

The total volumetric flow rate = 1.5+1+2=4.5 cms(m<sup>3</sup>/s)

The diameter is calculated as follows:

$$\dot{Q} = A_A \times V_A$$

$$4.5 = \frac{\pi}{4} D_A^2 \times 8$$

$$D_A = 0.85 \text{ m} \equiv 1300 \times 500 \text{ mm}$$

The friction pressure drop can be calculated by using the following relation:

$$\frac{\Delta P_f}{L} = \frac{0.022243 \dot{Q}^{1.852}}{D^{4.973}} = \frac{0.022243 \times 4.5^{1.852}}{0.85^{4.973}} = 0.809 \text{ Pa/m}$$

### **Duct B**

$$\frac{\Delta P_f}{L} = \frac{0.022243 \dot{Q}^{1.852}}{D^{4.973}}$$

$$0.809 = \frac{0.022243 \times 1.5^{1.852}}{D^{4.973}}$$

$$D = 0.565 \text{ m} \equiv 525 \times 500 \text{ mm}$$

### **Duct C**

$$0.809 = \frac{0.022243 \times 3^{1.852}}{D^{4.973}}$$

$$D = 0.731 \text{ m} = 900 \times 500 \text{ mm}$$

### **Duct D**

$$0.809 = \frac{0.022243 \times 1^{1.852}}{D^{4.973}}$$

$$D = 0.485 \text{ m} \equiv 700 \times 300 \text{ mm}$$



## Duct E

$$0.809 = \frac{0.022243 \times 2^{1.852}}{D^{4.973}}$$

$$D = 0.628m \equiv 650 \times 500mm$$

Duct Design by equal friction method can be improved by using *balancing pressure loss*. This method assuming that the pressure loss in any duct pathway from AHU to the supplied air to the space is equal.

For example: let the pressure loss in the duct elbow is 5Pa and the pressure loss in each air diffuser is 20Pa. Branch B and D are re-designed as follows:

First of all, we should select the duct path of the highest pressure loss in the system which is called *index run*.

In the figure, the index run is *ACEF*. The total pressure loss along this path is:(see the figure on the next page)

$$\Delta P_t = \Delta P_f \times (ACEF \text{ length}) + \Delta P_{elbow} + \Delta P_{diffuser}$$

$$\Delta P_t = 0.809 \times (3 + 5 + 3 + 4) + 5 + 20 = 37.135Pa$$

## Duct B

The pressure loss from duct *A* is:

$$\Delta P_A = 0.809 \times 3 = 2.427 Pa$$

Thus, the available pressure in the branching point of duct *B* is:

$$\Delta P = 37.135 - 2.427 = 34.708Pa$$

The pressure loss in the branch *B* is:

$$\Delta P_B = 34.708 - 5 - 20 = 9.708 Pa$$

the new pressure drop per meter in branch *B* is calculated as:

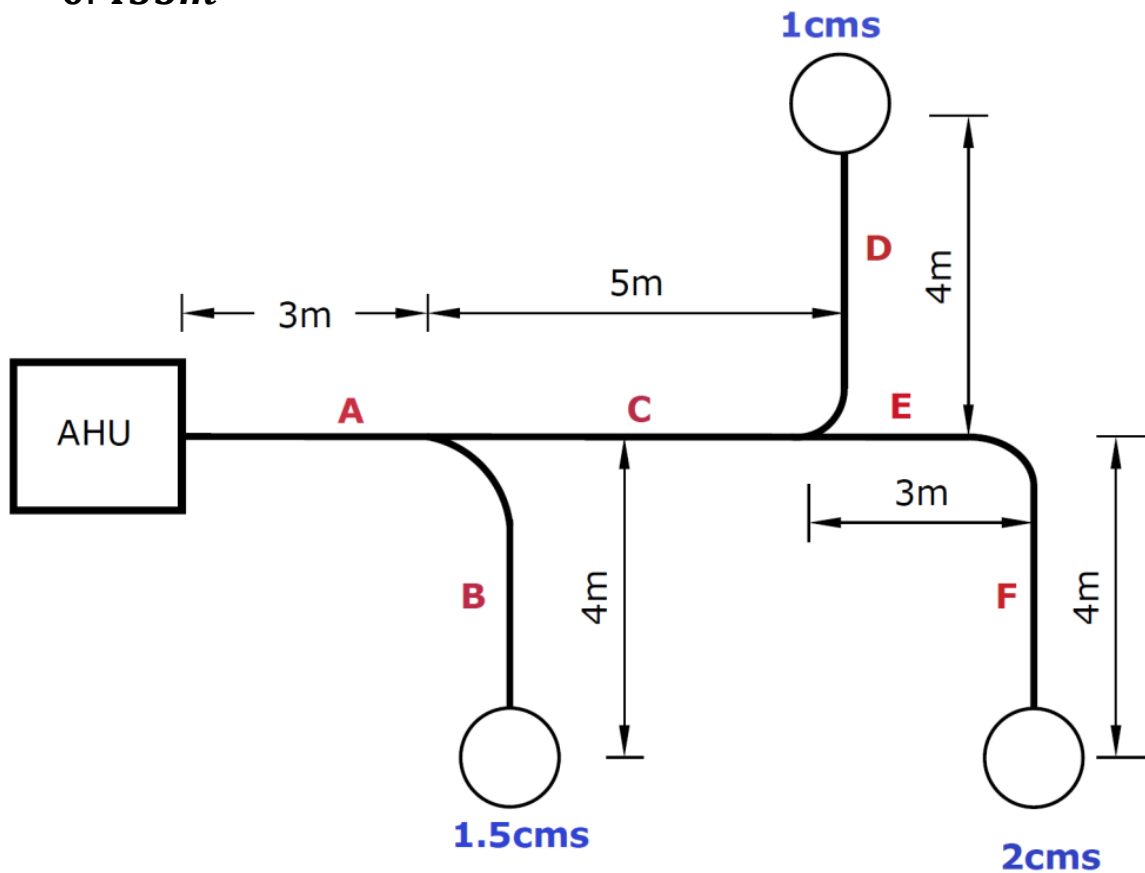


$$\frac{\Delta P_B}{L_B} = \frac{9.708}{4} = 2.427 Pa/m$$

Recalculate the diameter of duct **B** with the new pressure drop

$$2.427 = \frac{0.022243 \times 1.5^{1.852}}{D^{4.973}}$$

$$D = 0.453m$$



### Duct D

The pressure loss from the duct **A** and **C** is:

$$\Delta P_A = 0.809 \times (3 + 5) = 6.472 Pa$$

Thus, the available pressure in the take-off point of the **B** branch is:

$$\Delta P = 37.135 - 6.472 = 30.663 Pa$$



The pressure loss in the branch  $D$  is:

$$\Delta P_D = 30.663 - 5 - 20 = 5.663 \text{ Pa}$$

the new pressure drop per meter in branch  $D$  is:

$$\frac{\Delta P_D}{L_D} = \frac{5.663}{4} = 1.416 \text{ Pa/m}$$

$$1.416 = \frac{0.022243 \times 1^{1.852}}{D^{4.973}}$$

$$D = 0.433 \text{ m}$$

## 14.2 Pressure loss in bends

The pressure loss in the elbow, fitting,...etc. are expressed in terms of an equivalent length  $L_e$  by the following formula:

$$\Delta P_d = C_o P_d = \frac{1}{2} C_o \rho V^2 = \frac{4 f L_e P_d}{D}$$

$$P_d = \frac{1}{2} \rho V^2 \text{ , (dynamic pressure)}$$

$f$  = friction factor

$D$  = duct diameter

The relationship between the dynamic loss coefficient  $C_o$  and the equivalent length  $L_e$  is:

$$C_o = \frac{4 f L_e}{D}$$

**Table: dynamic loss coefficient  $C_o$**

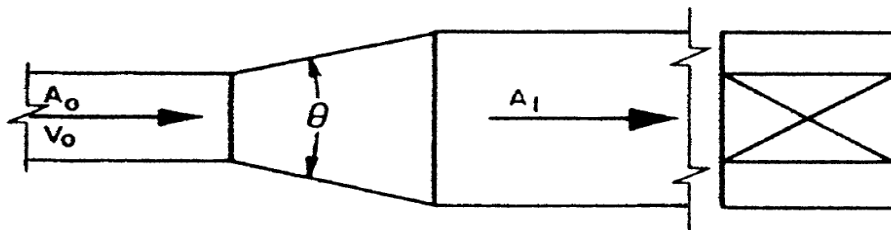
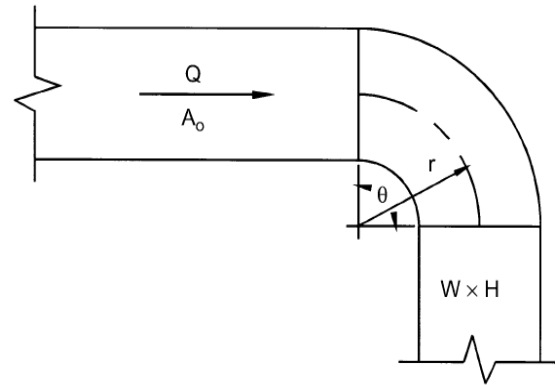
**CR3-1 Elbow, Smooth Radius, Without Vanes**

		$C_p$ Values										
		$H/W$										
$r/W$		0.25	0.50	0.75	1.00	1.50	2.00	3.00	4.00	5.00	6.00	8.00
0.50		1.53	1.38	1.29	1.18	1.06	1.00	1.00	1.06	1.12	1.16	1.18
0.75		0.57	0.52	0.48	0.44	0.40	0.39	0.39	0.40	0.42	0.43	0.44
1.00		0.27	0.25	0.23	0.21	0.19	0.18	0.18	0.19	0.20	0.21	0.21
1.50		0.22	0.20	0.19	0.17	0.15	0.14	0.14	0.15	0.16	0.17	0.17
2.00		0.20	0.18	0.16	0.15	0.14	0.13	0.13	0.14	0.14	0.15	0.15

		Angle Factor $K$										
$\theta$		0	20	30	45	60	75	90	110	130	150	180
$K$		0.00	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

$C_o = KC_p$  where  $K$  = angle factor

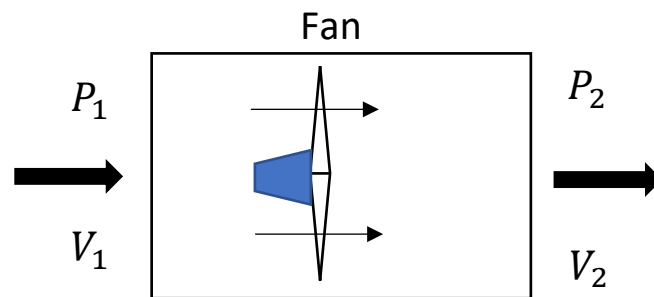


		$C_o$									
		$\theta$ , degrees									
$A_o/A_1$		10	15	30	30	45	60	90	120	150	180
0.06		0.26	0.27	0.40	0.56	0.71	0.86	1.00	0.99	0.98	0.98
0.1		0.24	0.26	0.36	0.53	0.69	0.82	0.93	0.93	0.92	0.91
0.25		0.17	0.19	0.22	0.42	0.60	0.68	0.70	0.69	0.67	0.66
0.5		0.14	0.13	0.15	0.24	0.35	0.37	0.38	0.37	0.36	0.35
1		0	0	0	0	0	0	0	0	0	0
2		0.23	0.20	0.20	0.20	0.24	0.28	0.54	0.78	1.0	1.1
4		0.81	0.64	0.64	0.64	0.88	1.1	2.8	4.4	5.7	6.6
6		1.8	1.4	1.4	1.4	2.0	2.5	6.6	10	13	15
10		5.0	5.0	5.0	5.0	6.5	8.0	19	29	37	43

### 14.3 Fans

The movement of the air through ductwork is done by a fan. The fan induces air motion in the room. The selection of a fan is very crucial for the efficiency of the air distribution system, control of noise level, indoor air quality, and comfort to the occupants.

If 1 and 2 denote the states at the inlet and outlet of the fan, then the total pressure rise across the fan denoted by **FTP** is given in pascals as follows:



$$FTP = \Delta P_t = FSP + FdP$$

$$FTP = \Delta P_t = P_2 - P_1 + \frac{1}{2}\rho(V_2^2 - V_1^2)$$

$FSP$  = fan static pressure

$FdP$  = fan dynamic pressure

The fan efficiency is given as:

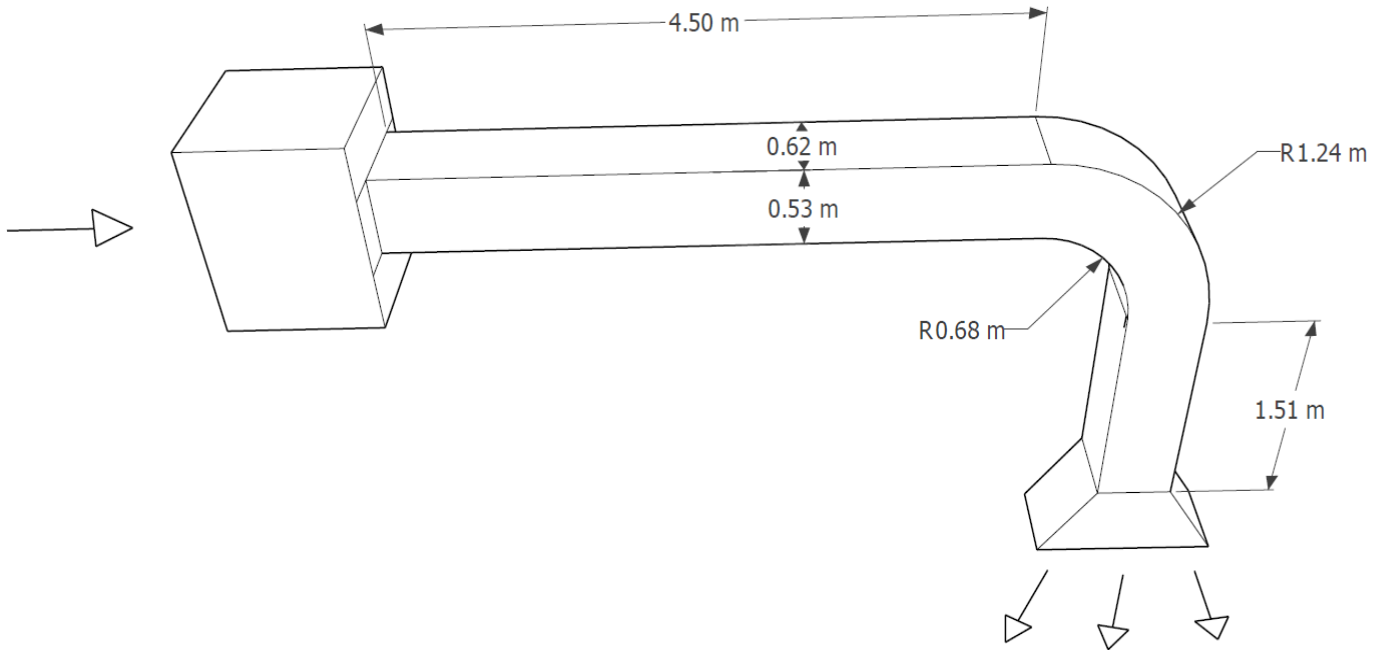
$$\eta = \frac{\dot{Q} \times FTP}{W}$$

Or the fan power is:

$$W = \frac{\dot{Q} \times FTP}{\eta}$$

### Example 14.2:

Axial fan supplying air with a volumetric flow rate of  $10000 \text{ m}^3/\text{hr}$ . to the duct system shown in the figure below. Find the power of the fan. Assume that the fan efficiency is 80%.



### Solution:

The hydraulic diameter of the duct is:

$$D = 1.3 \frac{(a \times b)^{0.625}}{(a+b)^{0.25}} = 1.3 \frac{(0.62 \times 0.53)^{0.625}}{(0.62+0.53)^{0.25}} = 0.626 \text{ m}$$

$$\frac{\Delta P_f}{L} = \frac{0.022243 \dot{Q}^{1.852}}{D^{4.973}} = \frac{0.022243 \left(\frac{10000}{3600}\right)^{1.852}}{0.626^{4.973}} = 1.52 \text{ Pa/m}$$

The average bend radius is:

$$R = \frac{R_o + R_i}{2} = \frac{1.24 + 0.68}{2} = 0.96 \text{ m}$$

The total length of the duct is:

$$L_t = 4.5 + \frac{\pi}{2} \times 0.96 + 1.51 = 7.518 \text{ m}$$



The friction pressure loss is:

$$\Delta P_f = 1.52 \times 7.518 = \mathbf{11.43 Pa}$$

the dynamic pressure loss in the bend is calculated as follows:

$$\Delta P_d = C_o P_d = \frac{1}{2} C_o \rho V^2$$

The dynamic loss coefficient  $C_o$  as follows:

$$\text{The ratios } \frac{R}{w} = \frac{0.96}{0.62} = \mathbf{1.55}, \text{ and } \frac{H}{w} = \frac{0.53}{0.62} = \mathbf{0.855}$$

From bends table  $C_o \approx \mathbf{0.18}$

The velocity in the duct is:

$$V = \frac{\dot{Q}}{A} = \frac{10000/3600}{0.62 \times 0.53} = \mathbf{8.453 m/s}$$

The pressure loss in the bend is:

$$\Delta P_d = \frac{1}{2} C_o \rho V^2 = \frac{1}{2} \times 0.18 \times 1.2 \times 8.453^2 = \mathbf{7.72 Pa}$$

The change in the dynamic pressure across the fan is:

$$\Delta P_{df} = \frac{1}{2} \rho (V^2 - V_o^2)$$

$V_o$  is initial air velocity in the fan inlet which equals zero, thus  $\Delta P_{df}$  is:

$$\Delta P_{df} = \frac{1}{2} \rho V^2 = \frac{1}{2} \times 1.2 \times 8.453^2 = \mathbf{42.872 Pa}$$

$$\mathbf{FTP} = 11.43 + 7.72 + 42.872 = \mathbf{62.02 Pa}$$

The power of the fan is:

$$W = \frac{\dot{Q} \times FTP}{\eta} = \frac{10000}{3600} \times 62.02 = \mathbf{215.35 W}$$

## 1. Systems of Air Conditioning

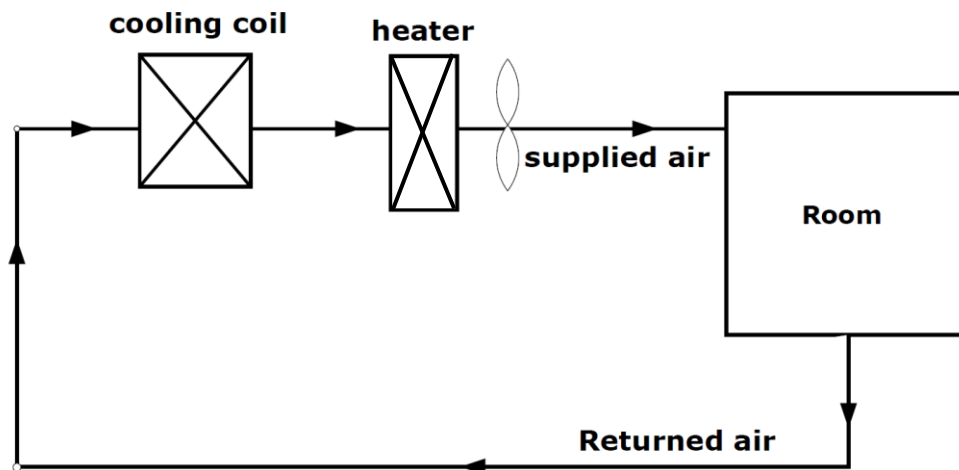
Cooled or heated air is supplied to the air-conditioned spaces through a three-air distribution system:

- 1- All return air system
- 2- Mixed air system
- 3- All fresh-air system

Selection of any one of the above systems depends on the special requirements of the spaces and the fund of the project.

### 1.1 All return air system

In this system, all supplied air to the conditioned space is returned to the air conditioning unit or air handling unit (AHU) without any fresh air. The fresh air required is taken from the infiltration process through windows, door...., etc. A good example of this system is split units which are well known in the residential air conditioning.



All return air system

### Example1.1

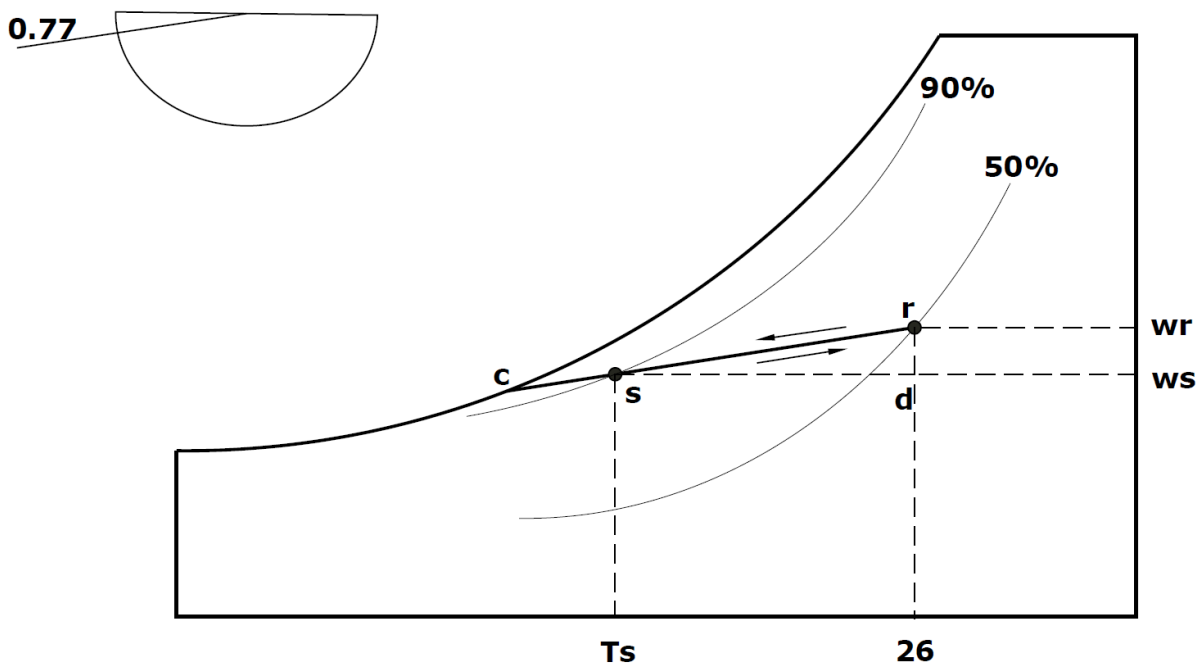
Air-conditioned space has a sensible and latent cooling load of 50 kW and 15 kW respectively. The inside conditions of the space are 26 °C and 50% RH. The air conditioning system uses all return-air methods. Find the conditions, the volumetric flow rate of the supplied air, and the cooling capacity of the coil.

### Solution

The sensible heat ratio can be calculated as follows:

$$SHR = \frac{50}{50 + 15} = 0.77$$

If no information on the bypass factor that specified exactly the supplied air conditions, the relative humidity of the supplying air can be taken as 90%.







Now sketch a line started from point **r** parallel to the sensible heat ratio line (SHR=0.77) and extending it to touch 100% RH line. The point **s** shown in the figure represents the air supplied conditions to the room.

From the psychrometric chart:

$$T_s = 14.5 \text{ }^\circ\text{C}$$

$$h_s = 38.2 \text{ kJ/kg}$$

$$w_s = 0.0093 \text{ kg/kg of dry air}$$

$$Q_s = 1.22 \dot{V}(T_r - T_s)$$

$$50 = 1.22\dot{V}(26 - 14.5)$$

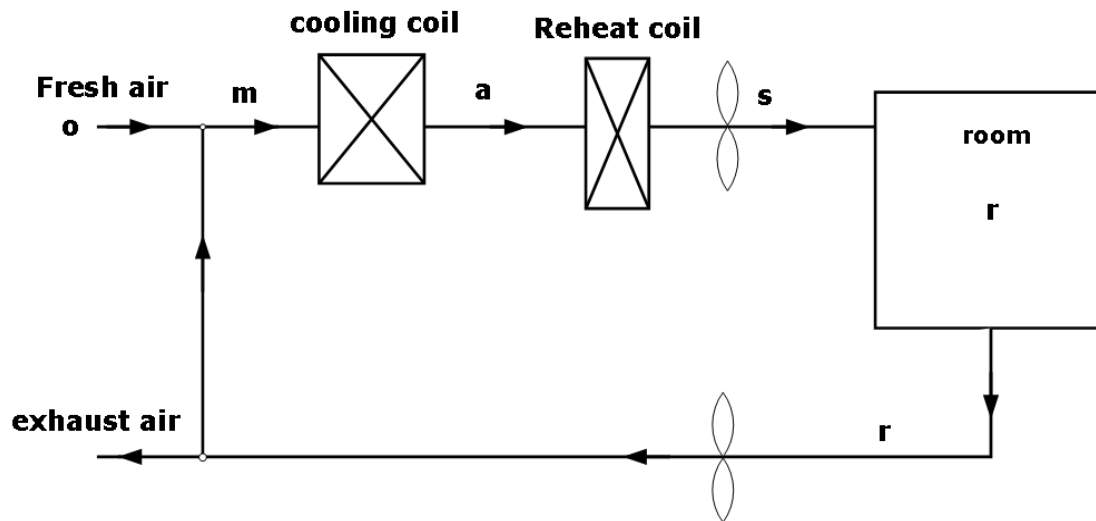
$$\dot{V} = 3.564 \text{ m}^3/\text{s}$$

In this case the space cooling load  $Q_T$  is equal to the coil cooling capacity  $Q_c$ :

$$Q_c = Q_T = Q_s + Q_L = 50 + 15 = 65 \text{ kW}$$

## 1.2 Mixed air system

This system is the conventional way that widely uses for air supplying to the conditioned spaces. It is included a quantity of fresh air mixed with a percentage of return air from the space. The mixture passing through a cooling or heating coil before it supplying to space. *The coil capacity is higher than the space load because of the additional cooling or heating load required for ventilation.*



### Example1.2

The room sensible and latent heat loads of an air-conditioned space are 84 MJ/hr. and 21MJ/hr., respectively. The room condition is 25 °C DBT and 50% RH. The outdoor condition is 43 °C DBT and 27 °C WBT. The ventilation requirement is such that on a mass flow rate basis 20% of fresh air is introduced and 80% of supply air is circulated. The bypass factor of the cooling coil is 0.2. Determine:

- Coil ADP
- Condition of air entering and leaving the coil.
- Fresh air.
- The capacity of the coil in TR.



**Solution:**

$$\text{Sensible cooling load of the room } Q_s = \frac{84 \times 10^6}{3600 \times 1000} = 23.23 \text{ kW}$$

$$\text{Latent cooling load of the room } Q_L = \frac{21 \times 10^6}{3600 \times 1000} = 5.83 \text{ kW}$$

$$\text{Room sensible heat ratio } \text{RSHR} = \frac{84}{84+21} = 0.8$$

Mark outside condition O and room condition R on the psychrometric chart. Draw the line RSHR through R. Point 3 indicates the mixture condition, at this condition the air enters the cooling coil. The temperature of the air at point 3 is calculated as follows:

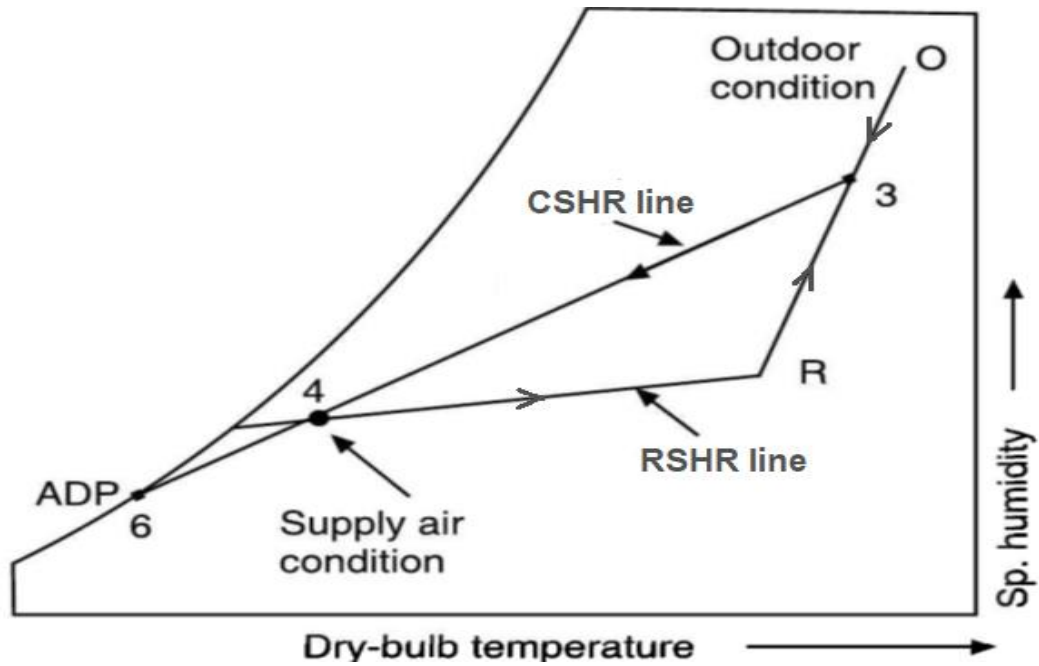
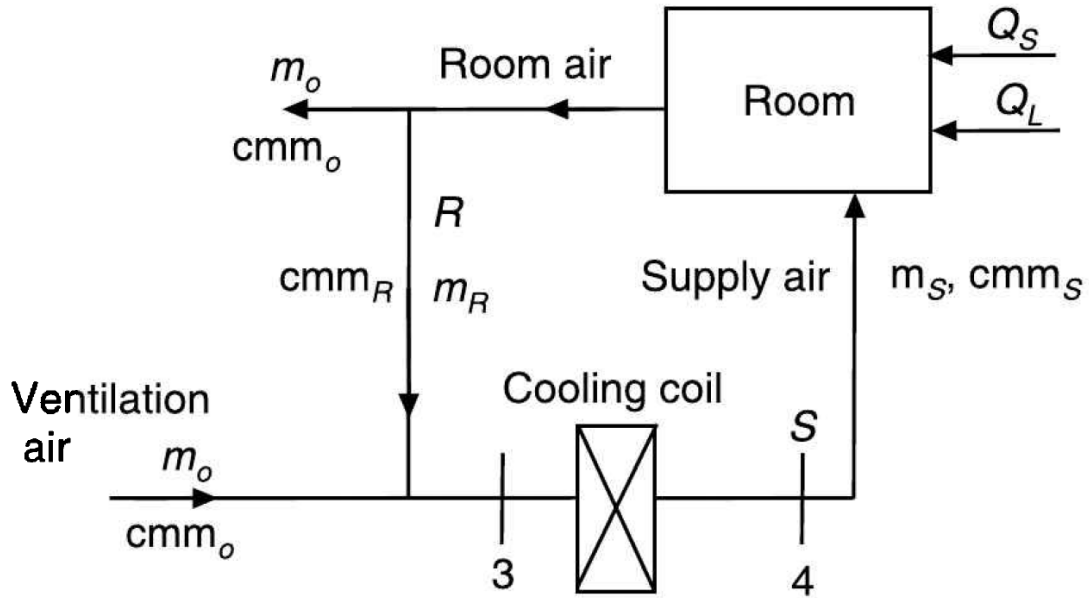
$$T_3 = 0.2 \times 43 + 0.8 \times 25 = 28.6 \text{ }^\circ\text{C}$$

Condition of air entering the coil: 28.6 °C DBT, 19.8 °C WBT. Draw a line 3-ADP, so that its point of intersection 4 with the RSHR line is such that

$$\text{BF} = \frac{\text{length of line 4-ADP}}{\text{length of line 3-ADP}} = 0.2$$

- a) Coil ADP=10.6 °C,  $T_4 = 14.2 \text{ }^\circ\text{C}$
- b) Point 4 indicates the condition of air leaving the cooling coil 14.2 °C DBT, 12.7 °C WBT.

Supply air absorbs room sensible heat while changing condition from 4 to R:



$$\dot{v} = \frac{Q_s}{1.22 (T_R - T_4)} = \frac{23.33}{1.22 (25 - 14.2)} = 1.77 \text{ m}^3/\text{s} = 106.24 \text{ m}^3/\text{min}.$$

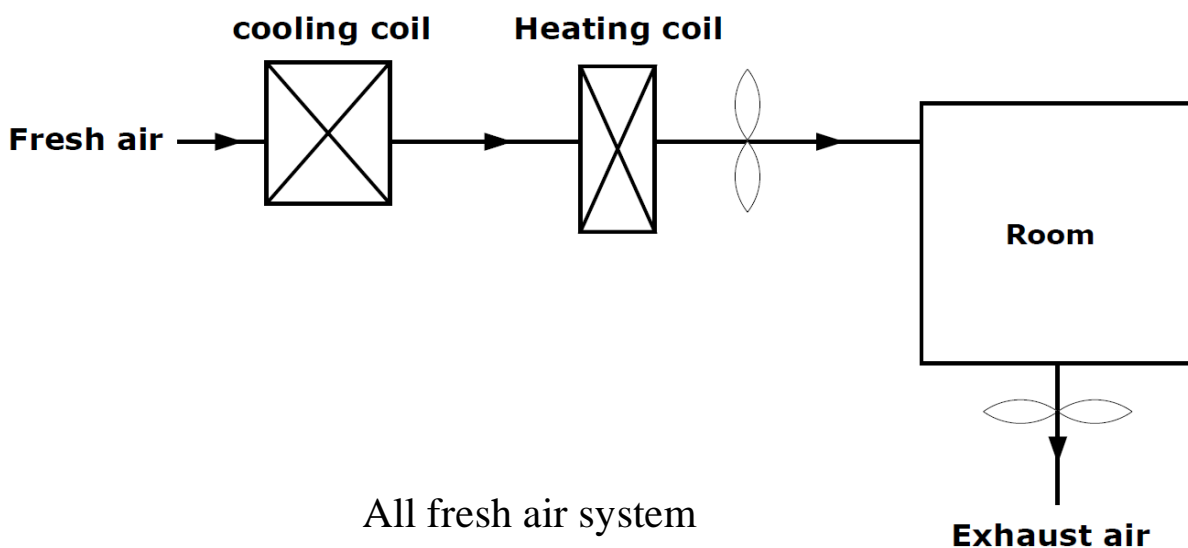
c) Fresh air quantity =  $0.2 \times 1.77 = 0.354 \text{ m}^3/\text{s} = 21.24 \text{ m}^3/\text{min}.$

d) capacity of the coil  $Q_c$  is:

$$Q_c = 1.2 \times \dot{v} \times (h_3 - h_4) = 1.2 \times 1.77 \times (57 - 36) \\ = 44.604 \text{ kW} = 12.7 \text{ TR}$$

### 1.3 All fresh air system

The air distribution system in some special buildings required 100% fresh air like operations rooms in the hospital and the places of cooking. No returned air circulated and all fresh air supplied to spaces is exhausted to the outdoor. This system is of high cost since the coil capacity is much more than the space cooling load. To reduce the cooling and heating demands in buildings, a heat recovery system may be used. The heat recovery system draws the heat from the exhausted air and passes it to the fresh air supplied to space.



### Example 1.3

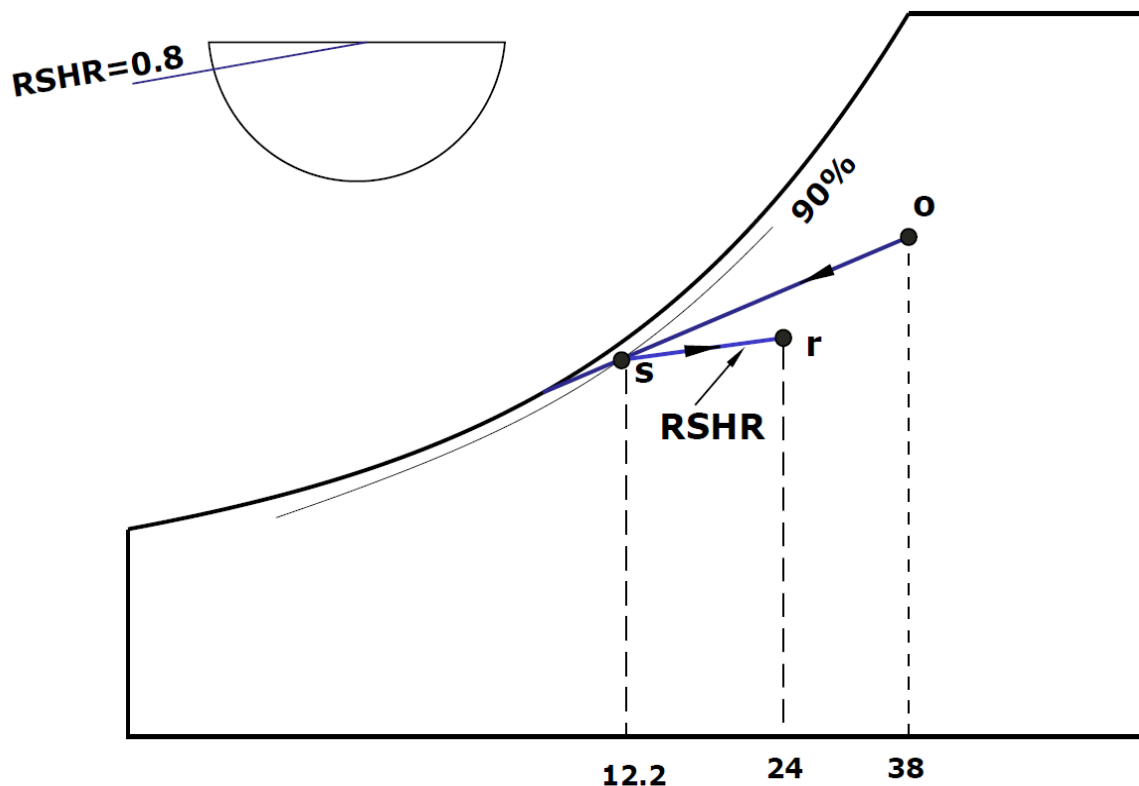
Space has a sensible and latent heat of 24 kW and 6 kW respectively. The inside condition is 24 °C and 50% RH. the special required full fresh air supplying of 38 °C DBT and 27 °C WBT. Determine:

- Quantity of air supplied.
- The capacity of the coil.

#### Solution:

- Draw a line through **r** and parallel to RSHR to intersect the 90% RH line at point **s**, then connected the points **o** and **s** with a straight line.

$$RSHR = \frac{24}{24+6} = 0.8$$





From the psychrometric chart, the temperature of air supplied to the room at point s is equal to 12.2 °C . The quantity of air supplied is :

$$\dot{v} = \frac{Q_s}{1.22 (T_r - T_s)} = \frac{24}{1.22 (24 - 12.2)} = 1.667 \text{ m}^3/\text{s}$$

b) The coil capacity is:

$$\begin{aligned} Q_c &= 1.2 \times \dot{v} \times (h_o - h_s) \\ &= 1.2 \times 1.667 \times (84.5 - 32.6) = 103.82 \text{ kW} \end{aligned}$$

*It is noted that the capacity of the cooling coil (103.82 kW) is much more than the room cooling load(30 kW) so a full fresh air system is very costly as well as of high energy consumption.*