Refrigeration & Air Conditioning

Psychrometry & Air Conditioning

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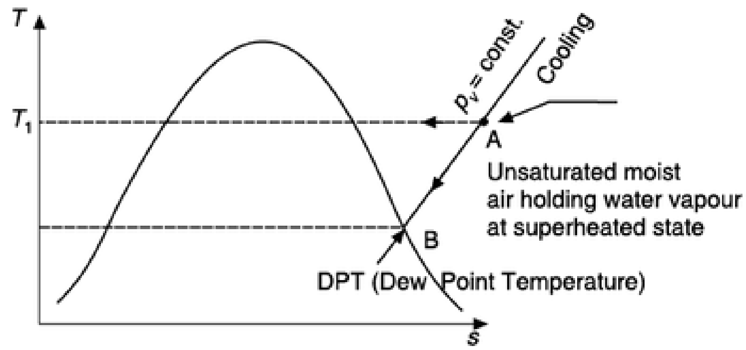
Psychrometry & Air Conditioning

Psychrometry

This branch of science deals with the study of properties of moist air and its behaviour under different conditions. The properties of moist air include Dry-bulb Temperature (**DBT**), Wet-bulb Temperature (**WBT**), humidity. Relative Humidity (**RH**). etc.

Let us consider moist air represented by the state A on T-s diagram show in Figure 8.1.



Water vapour in the air at state A is in the superheated condition and it has a partial pressure p_v. Suppose one adds water vapour to such air till the air becomes fully saturated with water vapour (point B), the partial pressure of water vapour contained in the air would be p_s. The addition of water vapour to the air can be by spraying water in the air. At state point B, air contains the maximum amount of water vapour corresponding to its temperature. For saturated air, the maximum specific humidity (w_s or Wmax) is given as

$$w_{max} = 0.622 \frac{p_s}{p - p_s}$$

Here, p_s = partial pressure of water vapour in a saturated air corresponding to DBT and p is the total pressure of moist air.

Dry-Bulb Temperature (DBT)

It is the temperature of air measured or recorded by a thermometer. It is denoted by T or T_{db} or DBT.

Wet-Bulb Temperature (WBT)

It is the temperature of air recorded by a thermometer when its bulb is covered with wet wick or cloth over which air is moving at a velocity at 2.5 to l0 m/s.

Dew Point Temperature (DPT)

It is the temperature of air recorded by a thermometer when the moisture present in it starts condensing. It is denoted by T_{dp} or DPT.

Consider that a certain sample of unsaturated moist air shown by state A in given figure, is cooled at constant pressure slowly by passing over the cooling coil. Its temperature goes on decreasing till it reaches a temperature DPT, at which the first drop of dew will be formed. It means that the water vapour in the air starts condensing. In the case of **dehumidification** of air, it is required to maintain the temperature of cooling coil well below DPT. During the cooling process, the partial pressure of water vapour and the specific humidity *w* remain constant until the vapour starts condensing.

The DPT (saturated temperature) can be found from the *steam table corresponding to the partial pressure of water vapour* p_v.

Humidity Ratio (Specific Humidity)

It is the ratio of mass of water vapour to the mass of dry air contained in the sample air. It is denoted by V. It is normally expressed in g/kg of dry air.

w = mass of water vapour in air/mass of dry air in air

$= m_v/m_a$

Let p_a, v_a T, m_a and R_a be the pressure, specific volume, DBT in K, mass and gas constant respectively.

Let p_v, v_v T, m_v and R_v be the pressure, specific volume, DBT in K, mass and gas constant respectively.

Using equation

$$pv = mRT$$

For dry air

$$p_a v_a = m_a R_a T$$

For water vapour

$$p_v v_v = m_v R_v T$$

As air and water vapour have the same volume and temperature, we get

$$\frac{p_v}{p_a} = \frac{m_v r_v}{m_a r_a}$$

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

Gas constant R = 287 J/kg-K, R_v = 461 J/kg-K

∴ $$w = \frac{m_v}{m_a} = \frac{p_v x 287}{p_a x 461} = \frac{0.6225\, x\, p_v}{p_a}$$

where p_v and p_a are partial pressure of water vapour and dry air respectively.

$\therefore \qquad w = \dfrac{0.6229\, x\,(p_v)}{(p - p_v)}$ kg/kg dry air.

Example

The dry-bulb temperature and dew point temperature of atmospheric air are 30°C and 14°C, respectively. If the barometer reading is 758 mm of Hg, determine the humidity ratio.

Solution

Use of steam table

From steam tables (will be provided in AMIE exam hall on request), the partial pressure of water vapour corresponding to DP temperature 14^0C is 0.015973 bar.

Atmospheric pressure is 758 mm Hg (758 x 0.001332 = 1.0105656 bar).

Humidity ratio

Humidity ratio will be

$$w = \frac{0.6229\, p_v}{(p - p_v)} = \frac{0.6229(0.015873)}{(1.0105656 - 0.015973)} = 0.01605 \text{ kg/kg dry air}$$

Absolute Humidity

It is the *mass of water vapour present in one cubic metre of dry air*. It is expressed in terms of grains per cubic metre of dry air (g/m^3 of dry air). Many a time it is expressed in terms of grains per m^3 of dry air. One kg of water vapour is equal to 15,430 grains.

$$p_vV = m_vR_vT$$

where

p_v = vapour pressure in air or saturation pressure at dew point

V= volume of air, which is also of water vapour

T = dry bulb temperature

R_v = gas constant of water vapour (462 kJ/kg-K)

m_v = mass of water vapour in kg

$\therefore$ Vapour density

$$= m_v/V = p_v/R_vT$$

Example

Find the absolute humidity of the air sample which has a dew point temperature of 16°C.

Solution

From steam tables, the vapour pressure for the saturation temperature of 16°C is 0.018168 bar. Let the volume V be 1 m^3.

$$R = \frac{R_0}{M_{wv}} = \frac{8314.14}{18.015} = 461.52\ J/kg - K$$

Mass of water vapour per m^3 is

$$\rho = \frac{p_v}{RT} = \frac{1816.8x1}{461.52x289} = 0.01362\ \text{kg/m}^3$$

Degree of saturation (m)

It is the mass of water vapour in a sample of air to the mass of water vapour in the same air when it is saturated at the same temperature.

Mathematically,

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

where w_v and w_s are specific humidity of air and saturated air, respectively

$$\mu = \frac{0.622.\dfrac{p_v}{p - p_v}}{0.622.\dfrac{p_s}{p - p_s}} = \frac{p_v}{p_s}\left(\frac{p - p_s}{p - p_v}\right)$$

where

p_s = partial pressure of water vapour when air is separated. It is obtained from steam tables corresponding to DBT (T_{db}).

P_v = partial pressure of water vapour in a moist air.

p = total pressure of moist air.

Relative humidity, RH = p_v/p_s = 0 when moist air is totally dry, i.e. which does not contain water vapour.

If the moist air is saturated, then $p_v = p_s$ then RH = 1 and μ = 1. It shows that the degree of saturation varies between 0 and 1.

Pressure

In air conditioning terms, air means a mixture of water vapour and remaining gases.

So by Dalton's law of partial pressure

$$p = p_a + p_v$$

where

p = total pressure of air

p_a = partial pressure of dry air

p_v = partial pressure of water vapour.

The partial pressure of water vapour can be found out by Carrier's equation

$$p_v = p_w - \frac{(p - p_w)(T_{db} - T_{wb})}{2800 - 1.3(1.8Tdb + 32)}$$

where

p_w = saturation pressure of water vapour corresponding to wet bulb temperature (from steam tables)

p_v = atmospheric pressure of moist air

T_{db} = dry bulb temperature

T_{wb} = wet bulb temperature

Relative humidity (RH)

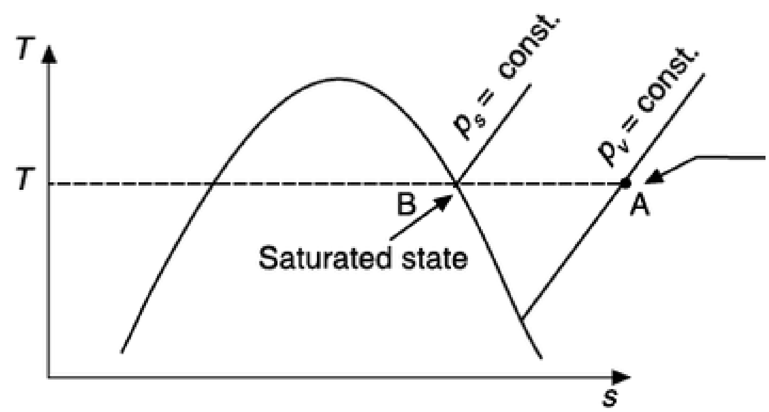
It is the ratio of mass of water vapour in a given volume of air at any temperature and pressure to the maximum amount of mass of water vapour which the same volume of air can hold at the same temperature conditions. The air contains maximum amount of water vapour at the saturation conditions.

Let v_v and v_s be the specific volumes of water vapour in the actual and moist saturated air at temperature T and in a volume V.

$$RH = \frac{(V / v_v)}{(V / v_s)} = \frac{v_s}{v_v}$$

Applying ideal gas equation to the state points A and B of following figure,

$$p_v v_v = p_s v_s$$



Relative humidity is therefore defined as the ratio of vapour pressure in a sample of air to vapour pressure of saturated air at the same temperature, i.e.

RH = vapour pressure of water vapour/vapour pressure of saturated air at the same temperature

Relative humidity is measured in *percentage*. It has great influence on evaporation of water in the air and therefore on the *comfort* of human beings.

Example (AMIE Summer 20111)

Prove that relative humidity ϕ is given by

$$\phi = \frac{\mu}{1-(1-\mu)(p_{vs}/p)}$$

where μ is degree of saturation; p_{vs} is saturation pressure of vapour in moise air and p_t is the total pressure of moist air.

Solution

$$\mu = \frac{w}{w_s} = \frac{0.622\dfrac{p_v}{p_t - p_v}}{0.622\dfrac{p_{vs}}{p - p_{vs}}} = \frac{p_v}{p_{vs}}\left(\frac{p - p_{vs}}{p - p_v}\right)$$

But $\phi = \dfrac{p_v}{p_{vs}}$

$\therefore$ $\mu = \phi\left(\dfrac{p_t - p_{vs}}{p - p_{vs}\phi}\right)$

or $\mu p - \mu\phi p_{vs} = \phi p - \phi p_{vs}$

or $\mu p = \phi p - \phi p_{vs} + \mu\phi p_{vs}$

$= \phi[p_t - p_{vs} + \mu p_{vs}]$

Dividing by p_t

$$\mu = \phi\left(1 - \frac{p_{vs}}{p} + \mu\frac{p_{vs}}{p}\right)$$

$$= \phi\left(1 - (1-\mu)\frac{p_{vs}}{p}\right)$$

$\therefore$

$$\phi = \frac{\mu}{1-(1-\mu)\left(\dfrac{p_{vs}}{p}\right)}$$

Enthalpy of air (h)

Air is a homogeneous mixture of dry air and water vapour. Therefore, enthalpy of air is found taking the *sum of enthalpy of dry air and enthalpy of water vapour in the moist air*.

Enthalpy of air/kg of dry air = Enthalpy of dry air + enthalpy of w kg of water vapour

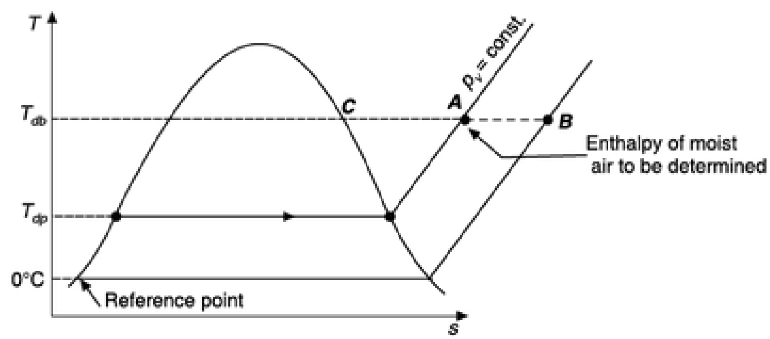
$$= h_a + wh_v$$

Considering the change in enthalpy of perfect gas as a function of temperature only, the enthalpy of dry air part, above a datum of 0°C, can be found as

$$h_a = c_{pa}T_{db} = 1.005T_{db} \text{ kJ/kg}$$

Assuming enthalpy of saturated liquid at 0°C as zero, the enthalpy of water vapour at point A in given figure, is expressed as

$$h_v = c_{pw}T_{dp} + (h_{fg})_{dp} + c_{pv}(T_{db} - T_{dp})$$



where

c_{pw} = specific heat of water vapour (kJ/kg-K)

T_{db} = dry-bulb temperature

T_{dp} = dew point temperature

$(h_{fg})_{dp}$ = latent heat of vaporization at dew point temperature

c_{pv} = specific heat of water vapour (kJ/kg-K).

From reference state as 0°C,

$$h = h_a + wh_v$$

$$\therefore \quad h = c_{pa}T_{db} + w[c_{pw}T_{dp} + (h_{fg})_{dp} + c_{pv}(T_{db} - T_{dp})] \quad (1)$$

As the reference state of water vapour is at 0°C , T_{dp} = 0, instead of $(h_{fg})_{dp}$ one has take latent heat of vaporisation at 0°C and equal to 2501 kJ/kg

$$\therefore \quad h = c_{pa}T_{db} + [w(2501 + c_{pv}(T_{db} - 0)]$$

$$\therefore \quad h = 1.005T_{db} + w(2501 + 1.88T_{db})] \quad (2)$$

Eq. (1) can also be simplified as

$$h = c_{pa}T_{db} + w[c_{pw}T_{dp} + (h_{fg})_{dp} + c_{pv}(T_{db} - T_{dp})]$$

$= (c_{pa} + wc_{pv})T_{db} + w(h_g - c_{pv}T_{dp})$ (3)

$= c_{pm}T_{db} + w(hg - c_{pv}T_{dp})$ (4)

where

w = specific humidity in kg/kg of dry air

h_g = enthalpy of saturated water vapour at dew point temperature in kJ/lcg

T_{db} = dry-bulb temperature in °C

T_{dp} = dew point temperature in °C

c_{pa} = specific heat of dry air = 1.005 kJ/kg-K

c_{pv} = specific heat of water vapour = 1.88 kJ/kg-K

c_{pm} = specific heat of moist air in kJ/kg of dry air-K.

Specific volume

It is the volume of air per unit mass of dry air. It is measured in m^3/kg of dry air. Air flow is measured by anemometer as volume rate of flow and the heat added or cooling requires mass flow rate. So specific volume is essential to relate the two.

Example

For a dry-bulb temperature of 25°C and a relative humidity of 50%, calculate the following for air when the barometric pressure is 740 mm Hg. Find without using psychrometric chart:

(a) Partial pressure of water vapour and dry air

(b) Dew point temperature

(c) Specific humidity

(d) Specific volume

(e) Enthalpy

Solution

Given values

Dry-bulb temperature, T_{db} = 25°C

Relative humidity, RH = 0.50

Atmospheric (barometric) pressure,

p_b = 740 mm Hg

= 740 x 133 Pa = 98,420 N/m^2 = 98.420 kPa

Using steam table

From steam tables (will be available in exam hall), saturation pressure of water vapour corresponding to dry-bulb temperature,

$T_{db} = 25^0C$, $p_s = 3.17$ kPa.

Relative humidity, RH = p_v/p_s

$\therefore \quad 0.5 = p_v/3.17$

Partial pressure

Partial pressure of water vapour, $p_v = 1.585$ kPa

$p_b = p_a + p_v$

Partial pressure of dry air, $p_a = 98.420 – 1.585 = 96.835$ kPa

Corresponding to $p_v = 1.585$, the saturation temperature from steam tables = 14^0C.

Dew point temperature

$T_{dp} = 14^0C$

Specific humidity

$$w = 0.622\frac{p_v}{p_a} = 0.622x\frac{1.585}{96.835} = 0.01018 \text{ kg/kg dry air}$$

Specific volume

$$\upsilon = \upsilon_a = \frac{R_aT}{p_a} = \frac{287.3x(25+273)}{96.835x10^3} = 0.8841 \text{ m}^3\text{/kg dry air}$$

Enthalpy of moist air

$$h = h_a + wh_v$$

$= 1.005T_{db} + w(2501 + 1.88T_{db})$

$= 1.005 \text{ x } 25 + 0.01018(2501 + 1.88 \text{ x } 25)$

= 51.0636 kJ/kg dry air.

Example

A sample of moist air has a dry-bulb temperature of 43°C and a wet-bulb temperature of 29°C. Calculate the following without making use of the psychrometric chart.

(a) Partial pressure of water vapour

(b) Specific humidity

(c) Relative humidity

(d) Dew point temperature

(e) Humid specific heat

(f) Enthalpy

(g) Degree of saturation

Solution

Given data

Dry-bulb temperature (DBT), T_{db} = 43°C

Wet-bulb temperature (WBT), T_{wb} = 29°C

Assuming barometric pressure = 760 mm Hg = 1.01325 bar

Partial pressure of water $w =$

$$p_v = p_w = \frac{(p - p_w)(T_{db} - T_{wb})(1.8)}{2800 - 1.3(1.8T_{db} + 32)} \quad \text{[at WBT = 29}^0\text{C, } p_w = 4.013 \text{ kPa]}$$

$$= 4.013 - \frac{(101.325 - 4.013)(43 - 29)(1.8)}{2800 - 1.3(1.8x43 + 32)} = 3.090\, kPa = 23.23\, mm\, Hg$$

Corresponding to T_{db} = 43^{0}C, saturation pressure, p_s = 8.65 kPa is obtained from steam tables.

Specific humidity

$$w = 0.622\frac{p_v}{p_a} = 0.622x\frac{3.090}{(101.325 - 3.090)} = 0.01956\, kg / kg\, dry\, air$$

Relative humidity

$$RH = \frac{p_v}{p_s}x100 = \frac{3.090}{8.65}x100 = 35.72\%$$

Dew point temperature

Dew point temperature corresponding to partial pressure p_v = 23.23 mm Hg

= 3.0920 kPa at 24.1^{0}C is obtained from steam tables.

Humid specific heat

$$c_p = c_{pa} + wp_v = 1.005 + 0.01956x1.88 = 1.0417\, kJ / kg - C$$

Enthalpy of moist air

$$h = h_a + wh_v = c_{pa}xT + w(h_{fg0} + 1.887)$$

$= 1.005 \times 43 + 0.01956(2501 + 1.88 \times 43)$

$= 93.7157$ kJ/kg dry air

Degree of saturation

$$\mu = RH\left(\frac{1 - p_s / p_b}{1 - p_v / p_b}\right) = 0.3572\left(\frac{1 - 8.65/101.325}{1 - 3.09/101.325}\right) = 0.3369$$

Example

A sample of air has dry and wet-bulb temperatures of 35°C and 25°C respectively. The barometric pressure is 760 mm Hg. Calculate without using psychrometric chart:

(a) Humidity ratio, relative humidity and enthalpy of the sample.

(b) Humidity ratio, relative humidity and enthalpy, if the air were adiabatically saturated. Only the use of steam tables is permitted.

Solution

Part (a)

Do it yourself.

Answer: 0.01584 kg/kg dry air; 44.70%; 75.8331 kJ/kg dry air.

Part (b)

Humidity ratio when air is adiabatically saturated

$$w = 0.622\frac{p_s}{p_b - p_s} = 0.622\frac{5.63}{(101.325 - 5.63)} = 0.03659 \text{ kg/kg dry air}$$

Relative density, RH = 100%

Enthalpy of moist air,

$h = c_{pa}T_{db} + w(h_{fg0} + 1.88T_{db})$

$= 1.005 \times 35 + 0.03659(2501 + 1.88 \times 35)$

$= 129.09$ kJ/kg dry air

Problem

A sling psychrometer reads 40^0C DBT and 28^0C WBT when the atmospheric pressure is 75 cm of Hg. Calculate the following:

(a) specific humidity

(b) relative humidity

(c) dew point temperature

(d) enthalpy

(e) vapour density

Answer: 0.0193 kg/kg of dry air; 40.7%; 24.1^{0}C; 89.7 kJ/kg of dry air; 0.02076 kg/m^3

Example

The temperature of air entering an adiabatic saturator is 42°C and that of the air leaving is 30°C. Compute the humidity ratio and relative humidity of the entering air.

Solution

DBT = 42^{0}C $\therefore$ p_s = 8.20 kPa

WBT = 30^{0}C $\therefore$ p_w = 4.246 kPa

$$p_v = p_w - \frac{(p - p_w)(T_{db} - T_{wb})(1.8)}{2800 - 1.3(1.8T_{db} + 32)}$$

$$= 4.246 - \frac{(101.325 - 4.246)(42 - 30)(1.8)}{2800 - 1.3(1.8x42 + 32)} = 3.467\ kPa$$

Humidity ratio

$$w = 0.622\frac{p_v}{p - p_v}$$

$$= 0.622\frac{3.467}{(101.325 - 3.467)}$$

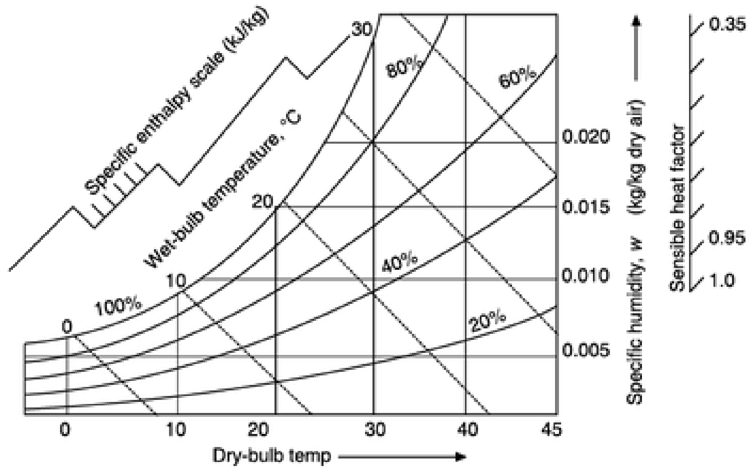
= 0.02204 kg/kg dry air

Relative humidity

$$RH = \frac{p_v}{p_s}x100 = \frac{3.467}{8.20}x100 = 42.28\%$$

PSYCHROMETRIC CHART

A psychrometric chart is a graphical representation of the thermodynamic properties of moist air. These properties of moist air vary with atmospheric pressure and altitude. One such chart for atmospheric pressure of 1.01325 bar at sea level is shown in following figure.



The variables shown on a complete psychrometric chart are: DBT, WBT, relative humidity, total heat; vapour pressure and the actual moisture content of the air.

As shown in this figure the dry-bulb temperature is taken as the x-axis and the mass of water vapour per kg of dry air as the ordinate.

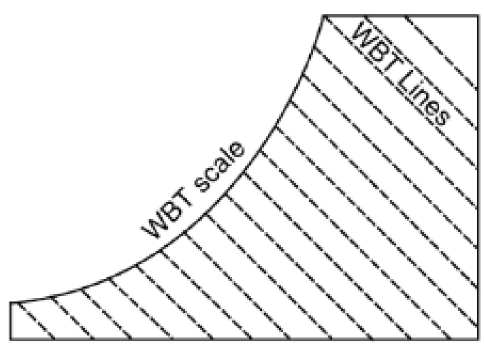
The following illustrations will help in locating the different lines and scales on the chart.

DBT lines

These dry-bulb temperature lines extend vertically upwards and there is one line for each degree of temperature.

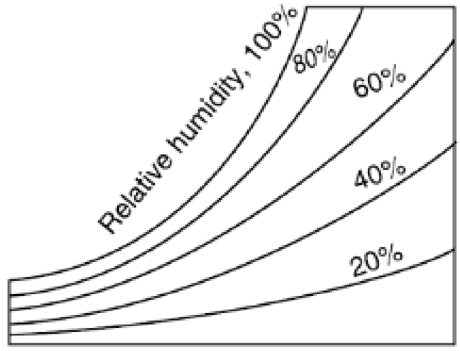
WBT lines

The wet-bulb temperature scale is found along the 'in-step' of the chart extending from the toe to the top. These lines extend diagonally downwards to the right. There is one line for each degree of temperature.



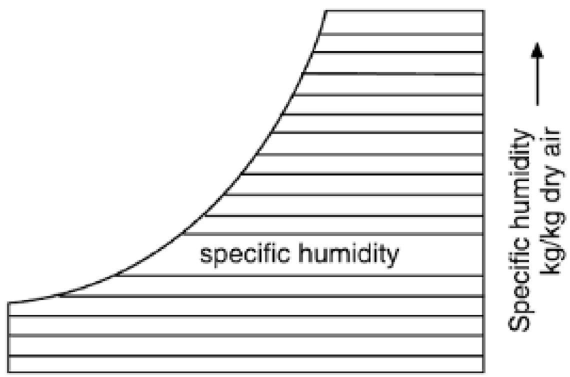
RH lines

On the psychrometric chart, the relative humidity lines are the only curved lines on it. The various relative humidifies are indicated on the lines themselves. The 100% RH line or saturation curve becomes the boundary of the chart on the left side. The region beyond this line is the supersaturated zone or fog zone.



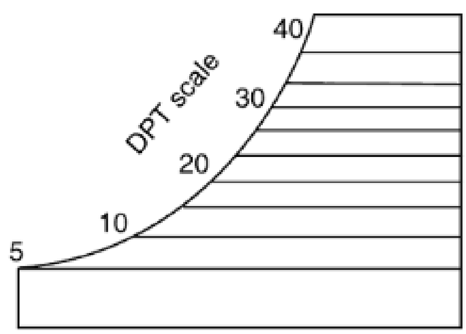
Specific humidity lines

The scale for specific humidity is a vertical scale on the right side of psychrometric chart. The scale is in grams of moisture per kilogram dry air.



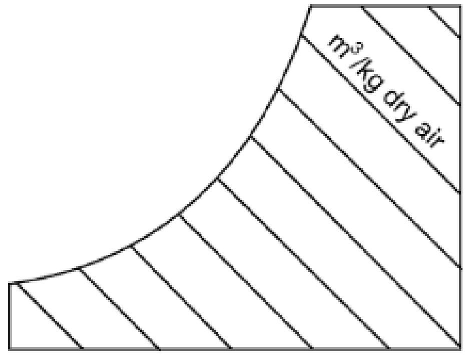
DPT lines

The scale for dew point temperature is identical to the scale of WBT lines. The DPT lines run horizontal to the right.



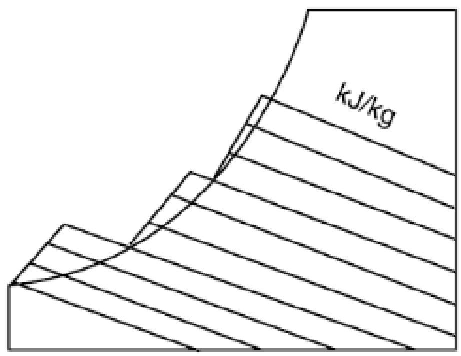
Specific volume lines

The specific volume lines are drawn along the sole chart and they are equally-spaced diagonal lines.



Specific enthalpy lines

The specific enthalpy scale is located along the 'in-step' of the chart. These lines are similar to WBT lines. Specific enthalpy lines indicate the total heat content.



Example

The atmospheric air is at 38°C DBT and 1.01325 bar pressure. If its thermodynamic WBT is 24°C, determine:

(a) The humidity ratio (specific humidity)

(b) Specific enthalpy

(c) Dew point temperature

(d) Relative humidity

(e) Specific volume.

Solution

Locate the state point on the chart at the *intersection* of 38°C DBT and 24°C thermodynamic WBT lines.

Then read the following values:

(a) Humidity ratio =13 g/kg dry air

(b) Specific enthalpy = 70.8 kJ/kg

(c) Dew point temperature = 18°C

(d) Relative humidity = 32%

(e) Specific volume = 0.9 m^3/kg dry air.

TYPICAL AIR CONDITIONING PROCESSES

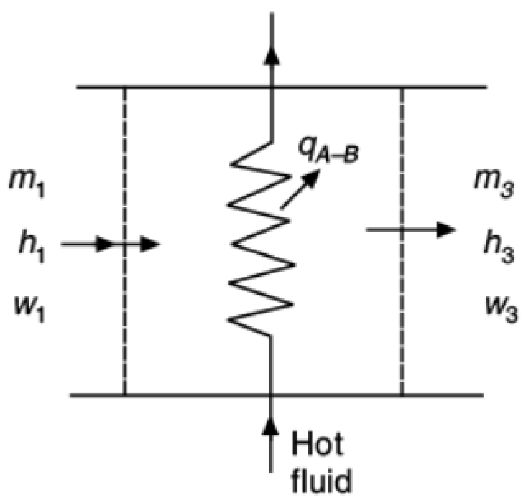
Any two of the foregoing properties of air can help to locate the state of air on the psychrometric chart. The condition of air at any point on the psychrometric chart is *fixed* by *any combination of two properties*. Such combinations are unlimited to locate the point on the chart but normally DBT and DPT or WBT combination is followed.

The various air-conditioning processes can be illustrated on the psychrometric chart by marking the *end conditions* of the air. For all the following processes, air is considered at atmospheric pressure of 1.01325 bar.

Sensible Heating of Air

Sensible heat will be added to the moist air while passing it over the hot dry surface. Normally, the heating surface is steam or hot water coil, whose surface temperature is above DBT of the air. As the air comes in contact with the warm surface, the DBT of air increases and tends to approach the temperature of the heating surface.

Since no moisture is added or removed from the air the specific humidity remains constant. The process is illustrated with a schematic diagram as in following figure.



The process takes place along the constant DPT line. For steady flow conditions, the required rate of heat addition is

$$Q_{1-3} = m_1(h_3 - h_1)$$

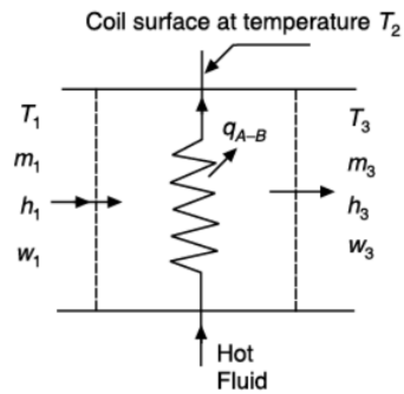
The process takes place along the constant moisture content line, here T_{db2} or T_2 is me heater temperature.

Example

Moist air, saturated at 10°C, flows over a heating coil at the rate of 5000 m^3/h. Air leaves the coil at 40°C. Plot the process on a psychrometric chart and determine the following: (a)WBT of air (b) The sensible heat transferred in kW (c) The total heat transferred in kW.

Solution

Diagram



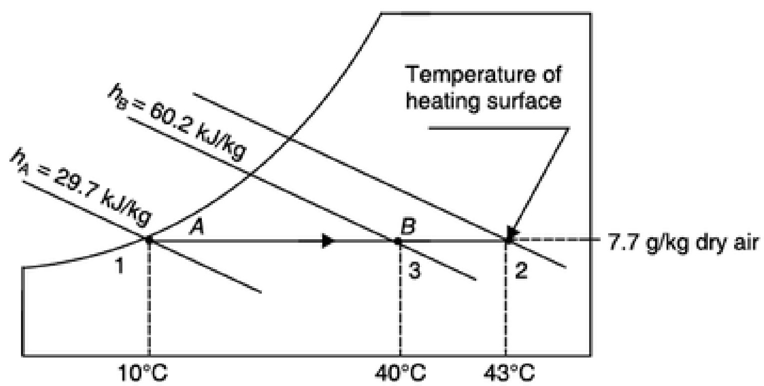
Psychrometric chart

The point A is located on the saturation curve at 10°C (See following figure)

w_A = 7.7 g/kg dry air

h_A = 29.7 kJ/kg dry air

v_a = 0.812 m^3/kg dry air



Locate state B

h_B = 60.2 kJ/kg dry air

$w_b = w_A$ = 7.7 g/kg dry air

WBT of air

WBT at state A = 10°C

WBT at state B = 20.2°C

Sensible heat transfer rate

Mass flow rate of air

m_A = 5000/(0.812 x 3600) = 1.71 kg/s

Sensible heat transfer rate

= $m_A(h_B - h_A)$

= (1.71)(60.2 -29.7)

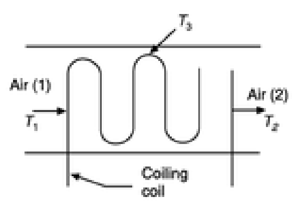
= 52.17 kW

Total heat transferred

In this example sensible heat is also equal to total heat, as there is no change in humidity ratio.

Coil bypass factor

If all the air flowing over the coil comes into contact with the heating surface of the coil and remains in contact for a sufficiently for long time, then the DB temperature of the air leaving the coil and the coil surface temperature will be same. However, practically air does not remain in contact with the heating surface for a long time and also some part of the air does not contact the surface. This means some air will bypass the coil. Therefore, the net effect is that the temperature of air at the outlet is less than the hot surface temperature. This is defined by a factor known as bypass factor.



The inability of coil to heat or cool the air to its temperature is indicated by a factor called **Bypass Factor (BF).**

BF = temperature drop that is not achieved/temperature drop that could be achieved

$$= \frac{T_2 - T_3}{T_2 - T_1}$$

where

T_1 = DBT of air entering the coil

T_3 = DBT of air leaving the coil

T_2 = mean effective temperature of the coil

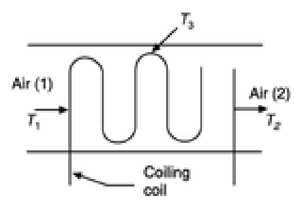
The bypass factor depends upon the following.

- Number of rows of coil: Lesser is the number of rows, the higher is the BF, and greater the number of rows, the lesser is the BE Coils are available from 2 rows to as high as 12 rows.
- Air velocity: For higher air velocity, BF is higher. The normal air velocity range is above 160 m/min.

The bypass factors range from 0.01 to 0.15 as per the application.

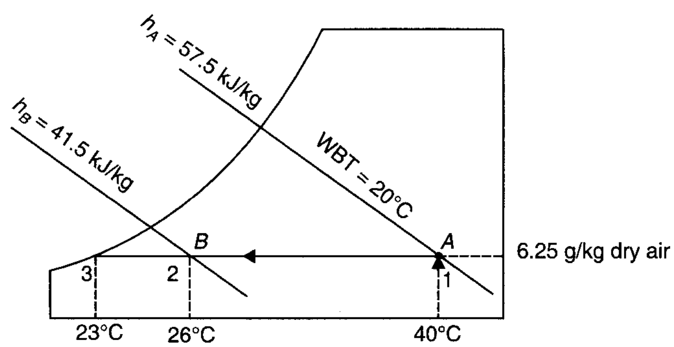
SENSIBLE COOLING OF MOIST AIR

Sensible cooling of moist air can be done by passing it over a cooling coil whose surface temperature is kept below the DBT of entering air and above the DP temperature. In this process, no moisture is added or removed, and DP temperature and latent heat content of air remain the same throughout the cooling process. Therefore, the process is represented as a horizontal line from right to left, depending upon the end conditions of air (see figure).



The total change in heat (enthalpy) content is equal to the change in sensible heat.

Following figure shows the cooling process on psychrometric chart.



The air can be cooled to surface temperature of coil. But this requires contact of air with coil surface for a sufficient period. The cooling of air depends on the number of rows of coil, depth of the coil and the velocity of air approaching the coil.

The heat transfer between the air and the cooling coil indicates that the temperature difference between the air and the coil surface is large at the beginning (first few rows of the coil). At subsequent rows this difference decreases. So; the last few rows are uneconomical due to very small temperature drop achieved. Therefore, the number of rows is limited and the air is then cooled to a temperature higher than T_2.

Low velocity of air allows adequate time for cooling. But low velocity causes laminar flow and also requires a higher cross-section of coil for adequate airflow. So velocity has to be of reasonably high value. Thus, air cannot be cooled to coil temperature by low air velocity.

Cooling capacity of coil = mass flow rate of air x specific heat x ($T_1 - T_2$)

Example

Moist air having DBT and WBT of 40°C and 20°C; respectively flows over a cooling coil at the rate of 7000 m^3/h. Finally, it is cooled to 26°C DBT. Plot the process on psychrometric chart and determine:

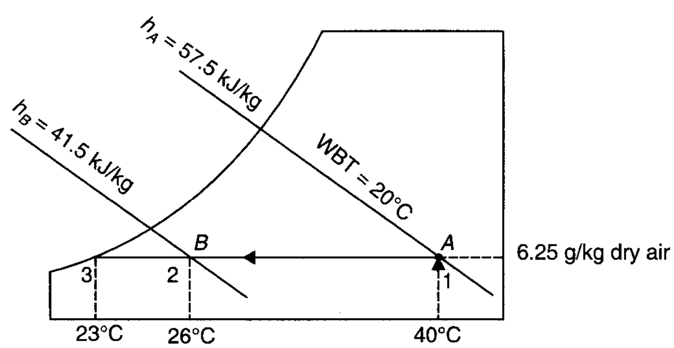
(a) Final WBT of air.

(b) The total heat transferred in kW.

If the cooling coil surface temperature is 22°C. find the bypass factor of the coil.

Solution

The process is as plotted on psychrometric chart shown in Figure (b).



The point A is located on the chart at the intersection of 40°C DBT and 20°C WBT lines. Now the values: h_A = 57.5 kJ/kg dry air and h_B = 41.5 kJ/kg dry, v_1 = 0.396 m^3/kg dry air

WBT of air

WBT of air at outlet = 15.5°C

Mass flow rate of air

$$m_a = 7000/(0.896 \times 3600) = 2.17 \text{ kg/s}$$

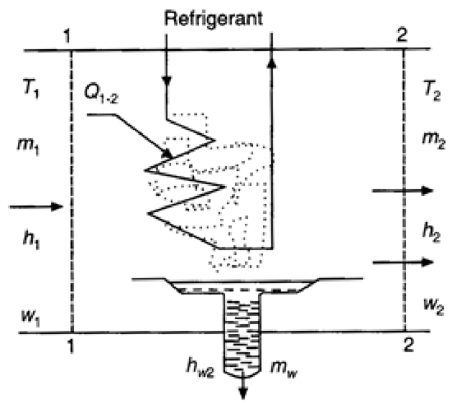
Total heat transfer rate = $m_a(h_A - h_B)$ = 2.17(57.5 – 41.5) = 34.72 kW

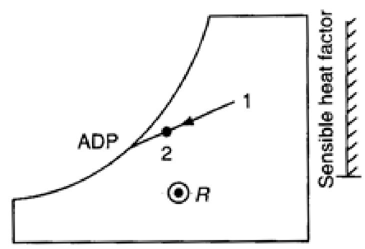
Bypass factor = $\dfrac{T_2 - T_3}{T_1 - T_3} = \dfrac{26-22}{40-22} = 0.22$

COOLING AND DEHUMIDIFICATION OF MOIST AIR

Here air is to be cooled and during cooling water vapour is to be separated from the air. Moisture separation will occur only when moist air is cooled to a temperature below its dew point temperature. Therefore, the effective surface temperature of the cooling coil kept below the initial dew point temperature of the air is called Apparatus Dew Point (ADP).

A cooling coil is shown schematically in Figure (a) and air flows uniformly across the coil. The process of cooling the air from state 1 to state 2 is as shown in Figure 8(b), while the coil surface temperature is kept at ADP.





(a) Device for cooling & dehumidification (b) cooling & dehumidification on chart

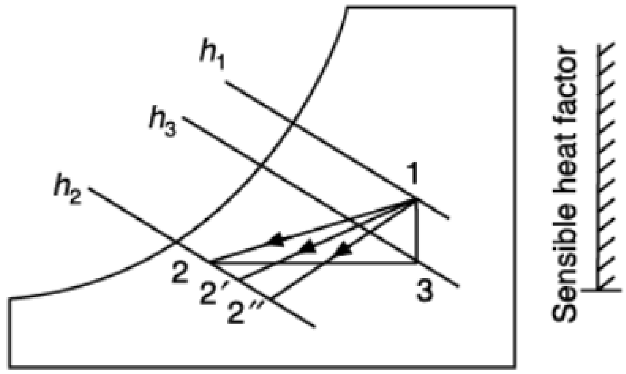
Although moisture separation occurs at various temperatures ranging from the initial dew point to final saturation temperature, it is assumed that the condensate water is cooled to the final air temperature T_2 before it drains from the system under steady state.

Cooling the air to the coil surface temperature becomes uneconomical as the size of the heat exchanger becomes very large as (ΔT) reduces. Therefore, surface area of the coil is so designed that it wall have a certain bypass factor given by following formula,

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

- In any cooling and dehumidification process, both sensible and latent heats need to be rejected and this is carried out by the cooling fluid circulated through the coil.
- The sum of sensible and latent heat is the total heat transferred.
- The ratio of sensible heat to the total heat transfer is known as the **Coil Sensible Heat Factor (CSHF).**

By knowing CSHF, we can find the sensible heat and latent heat quantities.



$$CSHF = \frac{sensible\,heat}{total\,heat} = \frac{Q_s}{Q_t}$$

The enthalpy difference ($h_1 - h_2$) represents the total heat absorbed by the coil. In the processes 1-2, 1-2', 1-2" the total heat absorbed is same but the proportion of sensible heat absorbed to total heat absorbed successively reduces. This is indicated by the fact that $T_1'' > T_2' > T_2$. This proportion is called the Sensible Heat Factor (SHF).

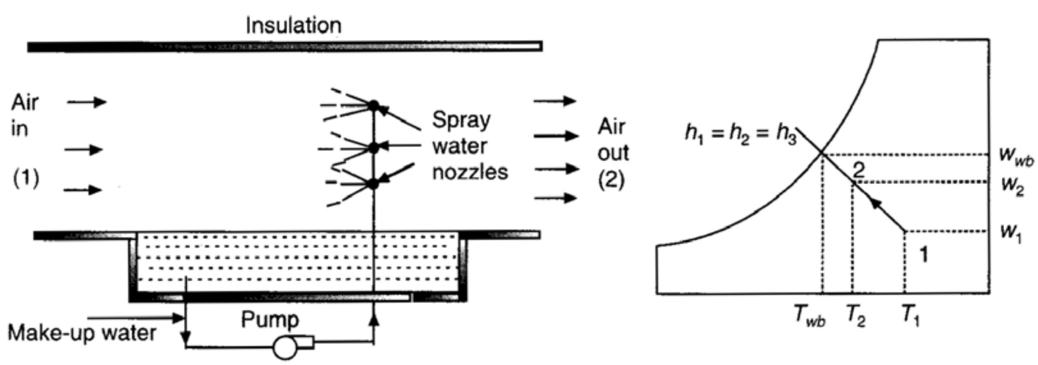
SHF represents the slope of the line representing the process on the psychrometric chart So, from figure it can be said that,

$$SHF = \frac{h_3 - h_2}{h_1 - h_2}$$

ADIABATIC COOLING OR COOLING WITH HUMIDIFICATION PROCESS

Adiabatic cooling is because of the water attaining the wet-bulb temperature of air due to constant evaporation. In some dry areas, the difference between dry-bulb and wet-bulb temperature is high enough to use well water as cooling medium in a coil. As the WB temperature is higher than the dew point of air, the process is sensible cooling.

Let us consider warm air blowing over the water surface as shown in following fifure.



(a) Adiabatic or evaporative cooling **(b) schematic diagram**

The air gets cooled. For cooling, air has to lose the heat it possesses. The air physically contacts the water surface, therefore, in one way it dissipates heat to the water by conduction.

Secondly, 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 actually comes from both air as well as water. Therefore, both air and water get cooled. The heat exchange process is only between the air and the water, so the system is said to be insulated. This process of cooling is known as **adiabatic cooling**. It is also called evaporative cooling or **cooling with humidification**. For this reason, water in an open lake or well is cooler than the surrounding.

The water surface can also be in the form of water spray as shown in Figure (a) to hasten evaporation and cooling.

During an adiabatic process no heat enters or leaves the system. Thus, the process line 1-2 on the psychrometric chart is along the constant enthalpy line.

The lowest possible temperature to which air can be cooled is the wet-bulb temperature (T_{wb}). Due to inefficient spray systems or uneconomical situations of providing a large number of banks of spray to get the cooling to T_{wb}, air is practically cooled to T_2 ($T_2 > T_{wb}$).

The efficiency of spray is defined as

Spray efficiency

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

The water is recirculated with a pump. Make-up water is added to compensate for the water evaporated during the operation. The makeup water also known as humidifier duty in litre/h can be given by the formula,

$$Humidifier\ duty = \frac{m^3/\min}{Specific\ volume} = (w_2 - w_1)x60$$

Example

Moist air at 32°C DBT and 50% RH enters a cooling coil at 10,000 m³/h. It is desired that the air leaving the coil has a DBT of 20°C and WBT of 18°C. Determine the following:

(a) Mean effective surface temperature of the coil

(b) Bypass factor of the coil

(c) Sensible heat factor of the coil

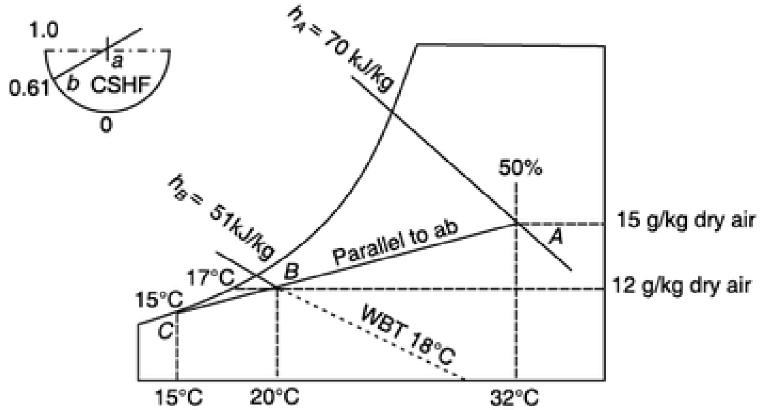
(d) Total heat removed from air

(e) Mass of water vapour condensed.

Solution

Diagram

Following figure shows the schematic solution.



State A is located at the intersection of DBT of 32°C and RH = 50%. Thus, h_A = 70 kJ/kg dry air and w_A = 15.0 g moisture/kg dry air and ν_A = 0.885 m^3/kg dry air.

State B is located on the intersection of DBT of 20°C line and WBT of 18°C line. Therefore, h_B = 51.0 kJ/kg dry air, w_B =12.0 g moisture/leg dry air.

From steam tables h_{wb-sat} corresponding to temperature 17°C DPT = 71.3 kJ/kg.

Mean effective temperature

Extend the line AB to meet the saturation curve at point C which indicates the mean effective surface temperature of 15^0C.

Coil bypass factor

The coil bypass factor = $\frac{T_B - T_C}{T_A - T_C} = \frac{20-15}{32-15} = 0.29$

Sensible heat factor

To get the sensible heat factor of the cooling coil, draw a line parallel to the process line AB through the centre of the protractor given on the churl. Then read the CSHF scale on the protractor, which is 0.61 in this case.

Total heat removed from air

Total heat removed per kilogram air

m_A = (10,000)/(0.88 x 3600) = 3.13 kg/s

q_{A-B} =3.13[(70 – 51) – (0.015 – 0.012) x 71.3] = 58.8 kW

Mass of water vapour condensed

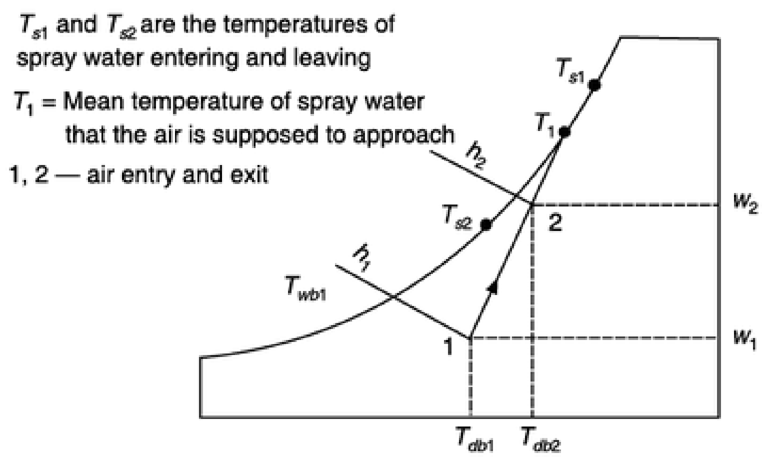
Mass of water vapour condensed

= $m_A(w_A - w_B)$ = (3.13)(0.015 – 0.012) = 0.009 kg/s

HEATING AND HUMIDIFICATION

In this case both heat as well as water vapour are *added* to the air. To achieve this, the temperature of water to be sprayed in the air stream is kept at a temperature greater than the DBT of incoming air so that heat will be transferred to air to heat it. The unsaturated air reaches the condition of saturation and the heat of vaporisation of water is absorbed from the spray water itself so that the spray water gets cooled. The heating and humidification process is shown in following figure.

During this process, the humidity ratio, the dry-bulb temperature, the wet-bulb temperature, the dew point temperature and the enthalpy of air increase while passing through hot spray. The relative humidity may increase or decrease. The spray water is to be heated before being pumped to the spray nozzles. The air enters at state 1 and leaves at state 2, as shown in figure.



The mass balance for water is:

$$(m_{w1} - m_{w2}) = m(w_2 - w_1)$$

where m_{w1}, m_{w2} are the mass flow rates of water entering and leaving in kg/min, respectively, m is the mass flow rate of air in kg/min, and w_1 and w_2 are specific humidity of air entering and leaving respectively.

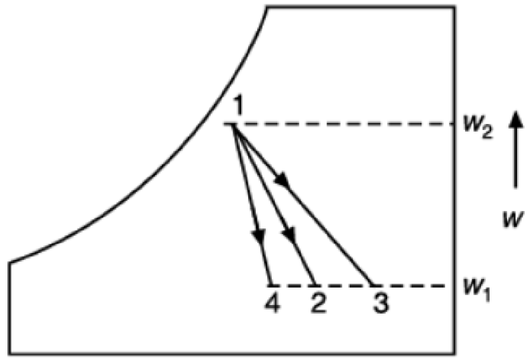
CHEMICAL DEHUMIDIFICATION OR SORBENT DEHUMIDIFICATION

Sorbents are materials that have an ability to attract and hold gases and liquids, other than water vapour - a characteristic that makes them very useful in chemical separation processes. Desiccants are a subset of sorbents; they have a particular affinity for water

In certain applications, the humidity requirement is very low which cannot be achieved by a cooling coil due to limitation of its temperature. In such cases air is passed through sorbents.

The solid sorbents include silica gel, zeolites, synthetic zeolites (molecular sieves), activated alumina, carbons and synthetic polymers.

The chemical dehumidification process shown in following figure takes place along constant enthalpy or WBT line from 1-2. Ideally the latent heat is released by condensation of moisture in the solid adsorbent from the air and picked up by air as sensible heat raising its temperature. Thus, it is an adiabatic process.

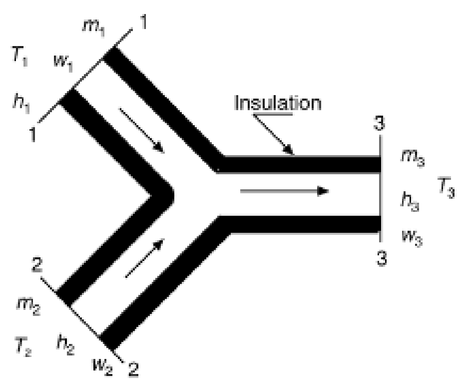


The absorption capacity of the chemical depends on its temperature. The lower the temperature, the higher would be the moisture absorption capacity and vice versa. Ideally the air should absorb all the released heat, but practically part of the heat is absorbed by the chemicals. So the chemical from the sump is cooled and then recirculated for spraying. Then the process would be 1-4 and the enthalpy difference. (h_2 – h_4) or (h_3 – h_4) is the heat absorbed by the chemical which is removed before recirculation.

This process, followed by sensible cooling, can give rise to some conditions unattainable by a cooling coil.

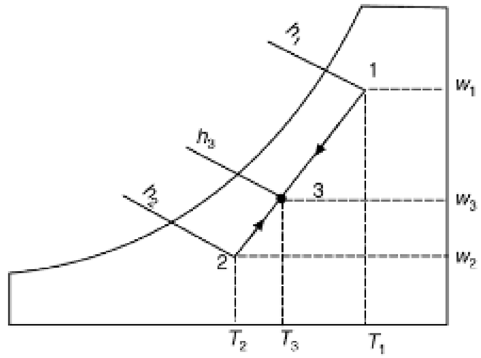
ADIABAT1C MIXING OF AIR STREAMS

See following figure of adiabatic mixing of two air streams.



The pipelines carrying air streams are assumed to be perfectly insulated so that no heat enters or leaves the system.

Now see following figure of schematic diagram.



From mass balance and heat balance, we get

$$\frac{m_1}{m_2} = \frac{w_3 - w_2}{w_1 - w_3}$$

and
$$\frac{m_1}{m_2} = \frac{T_3 - T_2}{T_1 - T_3}$$

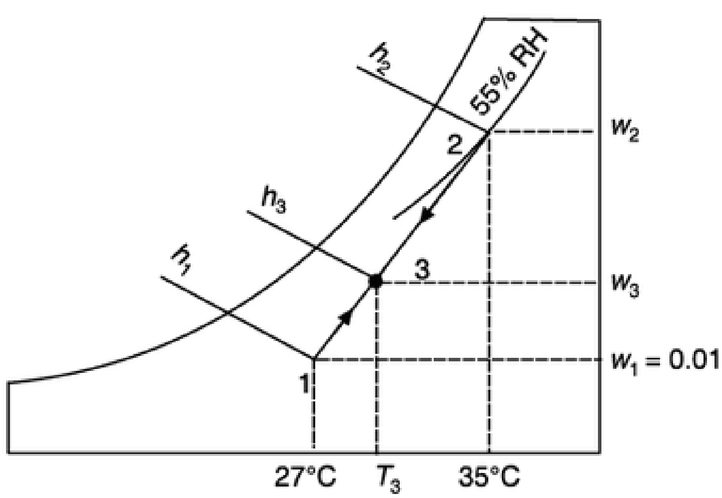
When hot and high humid air is mixed with very cold air, the resulting mixture would contain fog and air mixture and the final condition (point 3) on the psychrometric chart would be to the left or above the saturation curve. The temperature of the fog is corresponding to the wet bulb temperature line passing through point 3.

Example

An air stream of 7000 m3/h at a DBT of 27°C and humidity ratio of 0.010 kg/kg dry air is adiabatically mixed with 120,000 m^3/h of air having 35°C DBT and 55% RH. Find the DBT and WBT of the resulting mixture.

Solution

The process is as shown in following figure where states 1 and 2 are marked and the specific volumes ν_1 and ν_2 are 0.863 and 0.9 m^3/kg, respectively.



Now m_1 = 7000/(0.863 x 3600) = 2.25 kg/s

m_2 = 20,000/(0.9 x 3600) = 6.17 kg/s

$$\frac{Line 1-3}{Line 1-2} = \frac{m_1}{m_3} = \frac{2.25}{8.42} = 0.27$$

Consequently the length of line 1-3 = (0.27)(1 - 2). Accordingly, divide the line segment 1-2 to mark the point 3. The values of DBT and WBT at point 3 are read as 29°C and 20°C. respectively.

Example

500 m^3/min of fresh air at 30°C DBT and 50% RH is adiabatically mixed with 1000 m^3/min of recirculated air at 22°C DBT and 10°C DPT. Calculate the enthalpy, specific volume, humidity ratio and final DBT of the mixture.

Solution

From the psychrometric chart at 22°C DBT and 10°C DPT,

h = 41.5 kJ/kg of dry air

specific volume = 0.846 m^3/kg of dry air

Humidity ratio = w = 0.0077 kg/kg of dry air

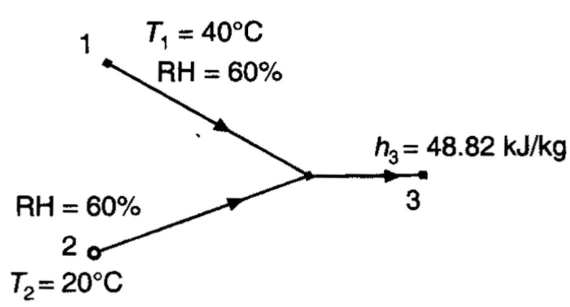
At 30^{0}C DBT and 50% RH

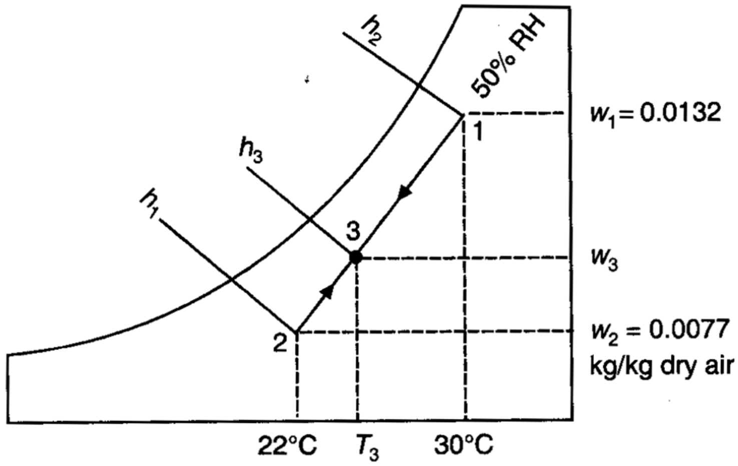
h = 64 kJ/kg of dry air

specific volume = 0.877 m^3/kg of dry air

Humidity ratio = w = 0.0132 kg/kg of dry air

Recirculated air mass rate of flow = 1000/0.846 = 1182 kg/min





Fresh air mass rate of flow = 500/0.877 = 570.1 kg/min

Enthalpy of mixture = $\frac{m_1h_1 + m_2h_2}{m_1 + m_2} = \frac{1182x41.5 + 570.1x64}{1182 + 570.1} = 48.82$ kJ/kg of dry air.

Specific volume = $\frac{m_1\upsilon_1 + m_2\upsilon_2}{m_1 + m_2} = \frac{1182x0.846 + 570.1x0.877}{1182 + 570.1} = 0.856$ m^3/kg of dry air

Humidity ratio = $\frac{m_1w_1 + m_2w_2}{m_1 + m_2} = \frac{1182x0.077 + 570.1x0.0132}{1182 + 570.1} = 0.00948$ kg/kg of dry air

Problem

A stream of moist air at 2°C dry bulb and 80 per cent relative humidity mixes with another stream of moist air at 30°C dry bulb and 10°C dew point in the-ratio by mass of one part of the first to two parts of the second. Calculate the temperature and specific humidity of the air after mixing.

Answer: h = 36.68 kJ/kg dry air; w = 0.0627 kg/kg dry air

Hint: m = 1/(1 + w)

Problem

1 kg of air (A) al 25°C and 0.012 kg/(kg dry air) is mixed with 3 kg of air (B) al 40°C and 0.02 kg/(kg dry air) to form a mixture M. What is the condition of M?

Answer: w_m = 0.018 kg/kg dry air; T_m = 36.25^{0}C; h_m = 83 kJ/kg

Hint: Find w_m *and* T_m *from mixing formula. Now from these values and using psychrometry chart, find* h_A *and* h_B *which comes out to be 56 kJ/kg and 92 kJ/kg respectively. Now find* h_m.

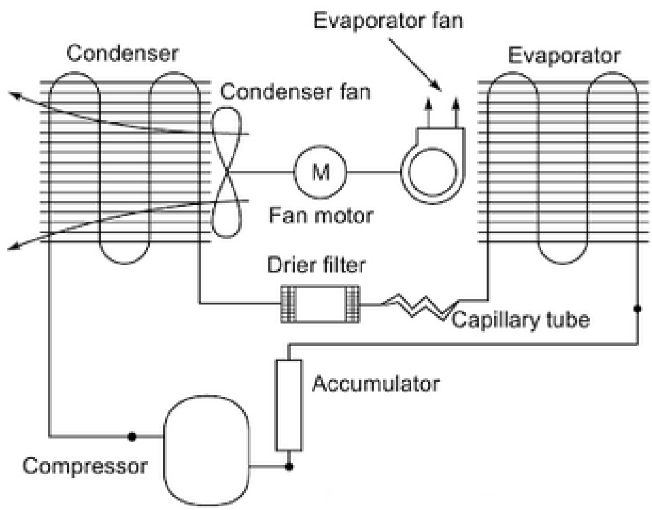
Air Conditioning

UNITARY SYSTEM

Such a unit is designed to be installed in or near the conditioned space. The components are contained in the unit. Unitary systems are standardized for certain applications but minor modifications are possible to suit an application. Heating components are rarely included.

Room Air Conditioners (Window AC)

This is the *simplest* form of an **unitary** air-conditioning system, assembled inside a casing (see figure), and suitable for installation on a window or wall opening.



The assembly incorporates a refrigeration unit and a double shaft fan motor, with fans mounted on both shafts of the motor-one on the evaporator side and the other for the air-cooled condenser. The room (or cooling side) and the outdoor (heat rejection side) of the unit are separated by an insulated partition within the casing. The front panel, with supply and return air grills and a door opening to get access to the control operating panel on the unit face, is attached to the unit on the room side. The other components on the room (or indoor) side of the unit are the cooling coil with an air filler mounted on its front face, the centrifugal evaporator blower, the operating panel consisting of selector switches, thermostat, knobs for fresh air, exhaust levers and condensate drip tray below the cooling coil. The outdoor side of the unit has the compressor, condenser coil, fan motor and propeller fan for the air-cooled condenser. The supply air grill on the front panel has adjustable horizontal louvers for adjusting the direction of the supply air. up or down or horizontally. Vertical adjustable louvers are also provided in many models to adjust the sideways How of the supply air. The distribution of the cool and dehumidified supply air to the room thus can be finely adjusted. Motorised (vertical) deflectors are also fitted in some makes, which change the direction of the air flow continuously and without affecting its uniform distribution.

The unit thus consists of: the refrigeration system, the control system (thermostat and selector switch), electrical protection system (motor overload switches and winding protection thermostat for the compressor motor), air-circulation system (fan motor, centrifugal evaporator blower and propeller fan for air cooled condenser) and ventilation (fresh air) damper and exhaust system.

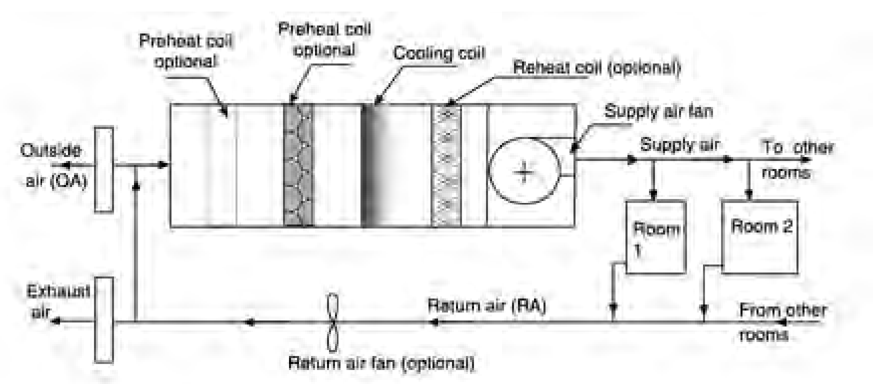
The refrigeration system consists of a hermetic compressor, forced air-cooled tinned condenser coil, finned cooling coil, capillary tube as throttling device and a refrigerant filter/drier. The refrigerant used is R-22.

The evaporator fan sucks the air from the room to be conditioned through the air filter and the cooling coil of the refrigeration unit, and delivers the cool and dehumidified air back to the room. This air mixes with the room air and brings down the temperature and humidity levels in the room to maintain comfortable conditions. Fresh air is admitted through a damper and is mixed with the return-air before passing it over the air filter and the cooling coil.

By incorporating a **reversing valve**, the unit *can be used for heating the room* during winter. The reversing valve is a two-position valve with four ports. In the normal or 'cooling" mode, the compressor discharge port gets connected to the condenser coil and the evaporator coil gets connected to the compressor suction port. In the 'heating' position of the reversing valve, the discharge port of the condenser gets connected to the room side coil which now acts as a condenser coil, rejecting heat into the room. The coil on the outdoor side now acts as an evaporator and is now connected to the suction port of the compressor. Atmospheric air now heats up the refrigerant in the outdoor coil. Thus, the valve reverses the function of the outdoor and indoor coils and facilitates heating of the room using the refrigeration system. This method of reverse cycle heating can work satisfactorily as long as the outside temperature is above 3° to 4°C.

CENTRAL AIR CONDITIONING SYSTEMS

A central air conditioning system can be used for single-zone (a zone consisting of a single room or group of rooms) or multi zone applications. In this section a central AC system, all-air for a single-zone application is discussed and the system is shown in following figure.



A single-zone air conditioning system has one thermostat that automatically controls one heating or cooling unit to maintain proper temperature in a zone comprising a single room or a group of rooms. A window air conditioner is an example of a single-zone air conditioning unit.

The system shown in given figure is for year-round air conditioning to control both temperature and humidity. All the components shown in the figure may not be utilized in all the circumstances.

An air-handling unit (AHU) cools or heats air that is then distributed to the single zone. The supply air fan is necessary to distribute air through the ductwork to the rooms.

- **Cooling coil:** It cools and dehumidifies the air and provides humidity control in summer. Reheat coil is optional and is used when air temperature is to be maintained at the required level, especially in winter. In summer, it may remain idle.
- **Reheating coil:** It heats the cooled air when the room heat gain is less than the maximum, thus providing humidity control in summer. The coil capacity is such that it satisfies the heating needs during winter.
- **Ductwork:** It is arranged so that the system takes in some outside ventilation air (OA), the rest being return air (RA) recirculated from the rooms. The equivalent amount of outside air must then be exhausted from the building. Dampers are provided to vary the rate of ventilation air as per the requirement of fresh air in the rooms. The arrangement of dampers is shown in Figure 10.3. In some applications as in operating theatres, ventilation air can be 100%.
- **Return air fan:** It takes the air from the rooms and distributes it through return air ducts back to the air conditioning unit or to the outdoors. In small systems with little or no return air ducts, the return air fan is not required because the supply fan can be used to draw in the return air.
- **Preheat coil:** The preheat coil may be located either in the outside air or the mixed airstream. It is required in cold climates (below freezing) to increase the temperature of air so that the chilled water cooling coils do not freeze. It is optional in milder climates and when DX (dry expansion) cooling coils are used,
- **Filters:** The filters are required to clean the air.

A room thermostat will control the cooling coil capacity to maintain the desired room temperature.

UNITARY VS. CENTRAL SYSTEMS

As already stated earlier, the classification of air conditioning systems into unitary and central systems, is not according to how the system functions, but how the equipment is arranged.

In a unitary system, the refrigeration and air conditioning components are factory selected and assembled in a package. This includes refrigeration equipment, fan, coils, filters, dampers and controls.

A central system is one where all the components are separate. The engineer has to design and install the central plant and its suitable components are based on the air-conditioning load.

Unitary equipment is usually located in or close to the space to be conditioned whereas the central equipment is usually remote from the space, and each of the components may or may not be remote from each other, depending on the desirability.

Unitary systems are generally all-air systems limited largely to the more simple types such as single-zone units with or without reheat. This is because they are factory assembled on a volume basis.

Central systems can be all-air, all-water or air-water systems and they are generally suitable for multizone units.

HEAT PUMP UNIT

A heat pump is an air conditioner in which the refrigeration cycle can be reversed, producing heating instead of cooling in the indoor environment. They are also commonly referred to as a "reverse cycle air conditioner". The heat pump is significantly more energy efficient than electric resistance heating. Some homeowners elect to have a heat pump system installed as a feature of a central air conditioner. When the heat pump is in heating mode, the indoor evaporator coil switches roles and becomes the condenser coil, producing heat. The outdoor condenser unit also switches roles to serve as the evaporator, and discharges cold air (colder than the ambient outdoor air).

Air-source heat pumps are more popular in milder winter climates where the temperature is frequently in the range of 4–13 °C, because heat pumps become inefficient in more extreme cold. This is because ice forms on the outdoor unit's heat exchanger coil, which blocks air flow over the coil. To compensate for this, the heat pump system must temporarily switch back into the regular air conditioning mode to switch the outdoor evaporator coil back to being the condenser coil, so that it can heat up and defrost. A heat pump system will therefore have a form of electric resistance heating in the indoor air path that is activated only in this mode in order to compensate for the temporary indoor air cooling, which would otherwise be uncomfortable in the winter.

The icing problem becomes much more severe with lower outdoor temperatures, so heat pumps are commonly installed in tandem with a more conventional form of heating, such as a natural gas or oil furnace, which is used instead of the heat pump during harsher winter temperatures. In this case, the heat pump is used efficiently during the milder temperatures, and the system is switched to the conventional heat source when the outdoor temperature is lower.

Absorption heat pumps are a kind of air-source heat pump, but they do not depend on electricity to power them. Instead, gas, solar power, or heated water is used as a main power source. An absorption pump dissolves ammonia gas in water, which gives off heat. Next, the water and ammonia mixture is depressurized to induce boiling, and the ammonia is boiled off, which absorbs heat from the outdoor air.

Some more expensive fixed window air conditioning units have a true heat pump function. However, a window unit may only have an electric resistance heater.

COMFORT AIR CONDITIONING

Since the purpose of most air-conditioning systems is to provide a comfortable indoor environment, the system designer and operator should understand the factors that affect comfort.

Body heat loss

Heat is generated in the human body due to metabolism or digestion of food. This body heat is continually lost to its cooler surroundings. The factor that determines whether one feels hot or cold is the rate of loss of body heat. When the rate of heat loss is within certain limits, one feels comfortable. If the rate of heat loss is too much, one feels cold and if the rate is too low, one feels hot.

The processes by which the body loses heat to the surroundings are convection, radiation and evaporation.

The rate of body heat loss is affected by five conditions:

- Air temperature
- Air humidity
- Air motion
- Temperature of surrounding objects
- Clothing

The system designer and operator can control comfort, primarily by adjusting three conditions: temperature, humidity and air motion. How are they adjusted to improve comfort?

The indoor air temperature may be lowered to increase the body heat loss in summer while in winter it may be raised to decrease the body heat loss.

In winter, humidity may be raised to decrease the body heat loss and in summer humidity may be lowered to increase the body heat loss by evaporation.

Air motion may be raised to increase the body heat loss in summer and lowered to decrease the body heat loss in winter by convection.

The occupants of buildings, of course, have some personal control over their own comfort. For instance, they can control the amount of clothing that they wear, they can use local fans to increase convection and evaporative heat loss, and they can even stay away from cold walls and windows to keep themselves warm in winter.

Indoor air quality

Another factor, air quality, refers to the *degree of purity* of the air. The level of air quality affects both comfort and health. Air quality is worsened by the presence of contaminants such as tobacco smoke and dust particles, biological microorganisms and toxic gases. Cleaning devices such as filters may be used to remove particles. Adsorbent chemicals may be used to remove unwanted gases. Indoor air contaminants can be diluted in concentration by introducing substantial quantities of outdoor air into the building. This procedure is called **ventilation**.

Effective Temperature

Effective temperature (ET) is defined as that temperature of saturated air at which the subject would experience the same feeling of comfort as experienced in the actual unsaturated environment.

Based on the concept of effective temperatures some comfort charts have been developed. To mention one, there is the Fanger's comfort chart. These may be referred to when a compromise in the inside design conditions is to be achieved.

ASSIGNMENT

PSYCHROMETRY

Q.1. (AMIE S12, 3 marks): What is psychrometry? What do you mean by saturated air and unsaturated air?

Q.2. (AMIE S10, 3 marks): Define the following terms:

(i) wet bulb temperature

(ii) relative humidity

(iii) bypass factor

Q.3. (AMIE S16, 5 marks): Explain various processes with the help of psychrometric chart.

Q.4. (AMIE S10, 14, W12, 13, 16, 10 marks): With the help of psychrometric chart, explain the following processes:

(i) sensible cooling

(ii) sensible heating

(iii) cooling and dehumidification

(iv) heating & humidification

(v) chemical dehumidification

Q.5. (AMIE S10, 11, 12, 14, 4 marks): Define specific humidity (w) and prove that

$$w = 0.622\frac{P_v}{P_t - P_v}$$

where P_v = partial pressure of water vapour and P_t = total pressure of moist air.

Q.6. (AMIE S11, 15, 16, 4 marks): Prove that relative humidity, ϕ, is given by

$$\phi = \frac{\mu}{1-(1-\mu)(p_{vs} / p_t)}$$

where μ is degree of saturation; p_{vs}, the saturation pressure of vapour in moist air; and p_t, the total pressure of moist air.

Q.7. (AMIE W10, 4 marks): Define the following terms (i) adiabatic saturation (ii) humidity ratio (iii) relative humidity (iv) dew point temperature.

Q.8. (AMIE S16, 5 marks): Define humidity ratio, relative humidity, bypass factor, apparatus dew point and sensible heat factor.

Q.9. (AMIE W11, 8 marks): Explain the process of cooling with adiabatic humidification. Represent it on a psychrometric chart. What are its limitations.

Q.10. (AMIE W12, 4 marks): Is it possible to obtain saturated air from unsaturated air without adding any moisture? Explain.

Q.11. (AMIE W12, 6 marks): What is the difference between specific humidity and relative humidity?

Q.12. (AMIE W13, 15, 6 marks): Explain the difference between (i) specific humidity and relative humidity (ii) dry bulb temperature (DBT) and wet bulb temperature (WBT). Define dew point temperature (DPT). When do the DBT, WBT and DPT become equal?

Q.13. (AMIE W13, 4 marks): What is the enthalpy of an air-vapour mixture? Why does it remain constant during an adiabatic saturation process?

Q.14. (AMIE S10, 7 marks): A psychrometer reads 40^0 dry bulb temperature and 28^0C wet bulb temperature. Assuming the barometric pressure as 1.013 bar, determine (i) humidity ratio (ii) relative humidity (iii) dew point temperature (iv) enthalpy of the mixture per kg of dry air.

Answer: 0.018 kg water vapour/kg of air; 40.6%; 24^0C; 90.88 kJ/kg

Q.15. (AMIE W10, 10 marks): 100 m^3 of air per minute at 15^0C DBT and 80% RH is heated until its temperature becomes 22^0C. Using steam tables, calculate (i) heat added to air per minute (ii) RH of heated air. Assume the atmospheric pressure as 1.013 bar.

Answer: 864 kJ/min; 51.6%

Q.16. (AMIE S11, 15 marks): In a laboratory test, a psychrometer recorded a dry bulb temperature of 34^0C and wet bulb temperature of 27^0C. Calculate the following: (i) vapour pressure (ii) relative humidity (iii) degree of saturation (iv) dew point temperature (v) enthalpy of mixture. Given barometric pressure as 1.01325 bar.

Answer: 0.03116 bar; 58.57%; 57.16%; 25^0C; 84.68 kJ/kg of air

Q.17. (AMIE W11, 8 marks): Atmospheric air at a dry bulb temperature of 16^0C and 25% relative humidity passes through a furnace and through a humidifier in such a way that the final dry bulb temperature is 30^0C and relative humidity as 50%. Using the psychrometric chart, find the heat and moisture added to air. Also, find the sensible heat factor.

Answer: 41.5 kJ/kg of dry air; 0.011 kg/kg of dry air; 0.339

Q.18. (AMIE W11, 12 marks): The humidity ratio of atmospheric air at 28^0C dry bulb temperature and 760 mm of mercury is 0.016 kg/kg of dry air. Determine (i) partial pressure of water vapour (take 1 mm of mercury = 0.133 kN/m^2) (ii) relative humidity (iii) dew point temperature (iv) specific enthalpy.

Answer: 0.02535 bar; 67.1%; 21.5^0C

Q.19. (AMIE S12, 8 marks): A sample of 450 g of moist air at 22^0C, 101 kPa and 70% relative humidity is cooled to 5^0C, while keeping the pressure constant. Determine the (i) initial humidity ratio (ii) dew point temperature (iii) amount of water vapour that condenses.

Answer: 0.116 kg of water vapour/kg of air; 0.00542 kg/kg of air; 2.781 g

Q.20. (AMIE W12, 10 marks): An air-water vapour mixture enters an adiabatic saturator at 30^0C and leaves at 20^0C which is the adiabatic saturation temperature. The pressure remains constant at 100 kPa. Determine the relative humidity and the specific humidity of the inlet mixture.

Answer: 39.9%; 0.0107 kg of water vapour/kg of dry air.

Q.21. (AMIE W09, 12, 8 marks): Air having DBT of 40^0C and RH 40% has to be cooled and dehumidified such that its DBT is 25^0C and RH is 70%. Determine the amount of moisture removed and the cooling capacity required when the air flow rate is 30 m^3/min. Estimate the bypass factor of the cooling coil.

Answer: 186.68 l/min; 4.6 tons; 0.423

Q.22. (AMIE S13, 15 marks): A sling psychrometer reads 40^0C DBT and 28^0C WBT when the barometer is 0.95 bar. By using steam tables, calculate (i) specific humidity (ii)relative humidity (iii) vapour density in air (iv) dew point temperature (v) enthalpy of mixture per kg of dry air.

Answer: 0.02067 kg water vapour/kg of air; 41.39%; 0.021156 kg/m^3 of dry air; 24.5^0C; 93.43 kJ/kg

Q.23. (AMIE S14, 8 marks): In a laboratory test, a sling psychrometer recorded dry bulb and wet bulb temperature as 30^0C and 25^0C, respectively. Calculate (i) vapour pressure (ii) relative humidity (iii) specific humidity (iv) degree of saturation.

Answer: 0.028424 bar; 67%; 0.018 kg of water vapour/kg of air; 66%

Q.24. (AMIE W13, 10 marks): A room having dimensions of 5 m x 5 m x 3 m, contains air at25^0C and 100 kPa at a relative humidity of 75%. Determine (i) partial pressure of dry air (ii) specific humidity (iii) enthalpy per unit mass of dry air (iv) masses of dry air and water vapour in the room.

Answer: 97.623 kPa; 0.015 kg/kg of dry air; 63.3 kJ/kg of dry air; 1.284 kg

Q.25. (AMIE W14, 10 marks): If 100 kg/s of moist air of 30^0C dry bulb temperature and 20^0C dew point temperature mixes adiabatically with 30 kg/s of moist air of 40^0C dry bulb temperature and 30% relative humidity, determine dry bulb temperature, humidity ratio, relative humidity and specific enthalpy after mixing. Saturation pressures of 20^0C, 30^0C and 40^0C are 2.34 kPa, 4.246 kPa and 7.4 kPa, respectively. If required, interpolate the intermediate data.

Answer: 32.3^0C; 0.0145 kg of vapour/kg air; 46.4%; 69.6 kJ/kg

Q.26. (AMIE W15, 8 marks): 800m^3/min of recirculated air at 22^0C DBT and 10^0C dew point temperature is to be mixed with 300 m^3/min of fresh air at 30^0C DBT and 50% RH. Determine the enthalpy, specific volume, humidity ratio and dew point temperature of the mixture.

Answer: 47.86 kJ/kg; 0.855 m^3/kg of dry air; 0.0092 kg/kg of dry air; 13^0C

Q.27. (AMIE S16, 10 marks): The pressure and temperature of mixture of dry air and water vapour are 736 mm of Hg and 21^0C (saturation pressure = 0.02485 bar). The dew point temperature of the mixture is 15^0C (saturation pressure = 0.01704 bar). Determine partial pressure of water vapour, relative humidity, specific humidity, enthalpy of mixture and specific volume of mixture.

Answer: 0.01704 bar; 68.57%; 0.01098 kg of water vapour/kg of air; 48.99 kJ/kg; 0.874 m^3/kg

Q.28. (AMIE S16, 10 marks): 120 m^3/min of air at 35^0C (saturation pressure = 0.0563 bar) DBT and 23^0C (saturation pressure = 0.02815 bar) dew point temperature is cooled to 20^0C (saturation pressure = 0.0234 bar) DBT by passing through a cooling coil. Determine relative humidity and wet bulb temperature of out coming air, capacity of cooling coil and amount of water vapour removed.

Answer: 100%; 20^0; 3126.9 kJ/min; 0.414 kg/min

Q.29. (AMIE W16, 8 marks): A mixture of dry air and water vapour is at a temperature of 22^0C under a total pressure of 730 mm of Hg. The dew point temperature is 15^0C, find (i) partial pressure of water vapour (ii) relative humidity (iii) specific humidity (iv) enthalpy of air per kg of dry air. Take P_v = 0.017 bar.

Answer: 0.017 bar; 64.4%; 0.011 kg of water vapour/kg of air; 50.05 kJ/kg

Q.30. (AMIE S17, 12 marks): A stream moist air at a state of 21^0C DBT and 14.5^0C WBT mixes with another stream of moist air at a state of 28^0C DBT and 20.2^0C WBT the respective masses of associated dry air being 3 kg and 1 kg. Calculate the final condition of the mixture and draw psychrometric chart.

AIR CONDITIONING

Q.31. (AMIE S12, 4 marks): What are the classification of air conditioning system?

Q.32. (AMIE W15, 4 marks): What is the relation between refrigeration and air-conditioning?

Q.33. (AMIE W15, 6 marks): Explain the difference between summer and winter air conditioning.

Q.34. (AMIE S17, 8 marks): With the help of schematic explain summer air conditioning system.

Q.35. (AMIE W14, 4 marks): Define the general layout of air-conditioning system. Briefly classify the air-conditioning system.

Q.36. (AMIE S16, 5 marks): What is the information provided by comfort chart? Enumerate the factors that affect comfort.

Q.37. (AMIE S10, 12, W11, 13, 14, 4 marks): What is comfort air-conditioning? What do you mean by effective temperature?

Q.38. (AMIE S17, 8 marks): What is effective temperature and discuss various factors affect effective temperature.

Q.39. (AMIE W10, 11, S13, 15, 16, 8 marks): Explain year round central air-conditioning system with a neat sketch.

Q.40. (AMIE W12, 13, 5 marks): How can a heat pump is used for waste heat recovery?

Q.41. (AMIE S11, 3 marks): Define a unitary system. Where is it commonly preferred?

Q.42. (AMIE W13, 5 marks): How is an air-air heat pump used for year round air conditioning.

Q.43. (AMIE S11, 5 marks): Describe a room air-conditioner with a neat sketch.

Q.44. (AMIE W12, 4 marks): How does a heat pump upgrade low grade reject heat? How can a heat pump be used for (i) space heating (ii) year round air conditioning? How can a heat pump be utilised in a certain place where both heating and cooling effects are required in certain operations?

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