TED(21) – 5022 REVISION 2021 MODEL QUESTION PAPER

Refrigeration & Air conditioning

Time :3Hours | Max. Marks :75

Answer Key

PART A

1. (a) actual COP/theoretical COP
2. (b) always more than unity.
   [Note : Typically, the Coefficient of Performance (COP) of Vapour compression refrigeration Systems (VCRS) is greater than one, although it can also be equal to one or less than one]
3. (b) Ammonia
4. (b) expansion
5. (d) All of the above
6. Coil, Shell
7. (c) less
8. (d) gas liquefaction
9. (b) less than

PART B

1. COP of a refrigerator is defined as, "The ratio of the cooling effect to the work input."

A tonne of refrigeration (TR) is a unit of heat extraction capacity of the refrigeration or cooling equipment. Generally, TR is defined as the amount of heat transferred to freeze or melt 1 short ton of ice at 0 deg. C in 24 hours.

$$1\text{TR} = \frac{907.185\ \text{kg}\ \times 334.6\,\frac{\text{kJ}}{\text{kg}}}{24\ \text{h} \times 60\,\frac{\text{min}}{\text{h}}} \approx 210\ \frac{\text{kJ}}{\text{min}} = 3.5\ \text{kW}$$

2. Desirable Properties of ideal refrigerant –absorbent combination in Vapour Absorption system is given below

(i). Refrigerant should be much more volatile than absorbent.

(ii).Refrigerant should have high latent heat to reduce mass flow

(iii). Both should be chemically stable at all operating conditions

(iv). Absorbent should have strong affinity for refrigerant.

(v) Both should not cause corrosion in the range of conditions same as VC

(vi) Should not be toxic and inflammable.

3. A decrease in the suction pressure of a vapour compression cycle, such as in a refrigeration or air conditioning system, can have several significant effects on the system's performance. Three key effects of a decrease in suction pressure are:

Reduced Cooling Capacity:

When the suction pressure decreases, the evaporator's ability to absorb heat from the surrounding space is reduced. This results in a decrease in the cooling capacity of the system.

Lower Efficiency:
A decrease in suction pressure can lead to lower system efficiency. This is because the compressor has to work harder to maintain the same level of cooling or refrigeration, which increases the power consumption.

Risk of Compressor Damage:
Operating the compressor at a lower suction pressure than it is designed for can be detrimental to the compressor's reliability and longevity.

4. ***Properties of a good refrigerants:***

- Low boiling Point
- High Critical Temperature
- High latent heat of vaporisation
- Low specific heat of liquid
- Low specific volume of vapour
- Non-corrosive to metal
- Non-flammable
- Non-explosive
- Non-toxic
- Low cost
- Easy to liquify at moderate pressure and temperature
- Easy to locating leaks by odour or suitable indicator
- Mixes well with oil.

5 The need for substitutes for CFC (chlorofluorocarbon) refrigerants is primarily driven by environmental concerns. CFCs are known to deplete the ozone layer, leading to adverse effects on human health and ecosystems. Substitutes are essential to:

Protect the Ozone Layer: CFCs release chlorine atoms when they break down in the atmosphere, which destroy ozone molecules. Ozone layer depletion allows harmful UV radiation to reach the Earth's surface, increasing the risk of skin cancer and other health issues.

Mitigate Climate Change: CFCs are potent greenhouse gases, contributing to global warming. By replacing them with more environmentally friendly alternatives, we can reduce their impact on climate change.

Comply with International Agreements: International agreements like the Montreal Protocol mandate the phase-out of CFCs. Substitutes help countries meet their commitments to protect the ozone layer and combat climate change.

Ensure Sustainability: CFCs are no longer manufactured, and their availability is limited. Substitutes are necessary for the continued operation of refrigeration and air conditioning systems, ensuring comfort and safety.

In summary, substitutes for CFC refrigerants are vital to protect the ozone layer, combat climate change, and meet international agreements, while also ensuring the sustainability of cooling and refrigeration technologies.

6. Hermetic sealed compressors offer several advantages, including:

Energy Efficiency: Hermetic compressors are typically more energy-efficient than their semi-hermetic or open counterparts, making them a cost-effective choice for various applications.

Reliability: The sealed design minimizes the risk of refrigerant leaks and external contaminants, enhancing the compressor's durability and reliability.

Compact Size: Hermetic compressors are compact and lightweight, making them suitable for space-constrained installations and portable appliances.

7. The advantages of forced convection evaporators include:

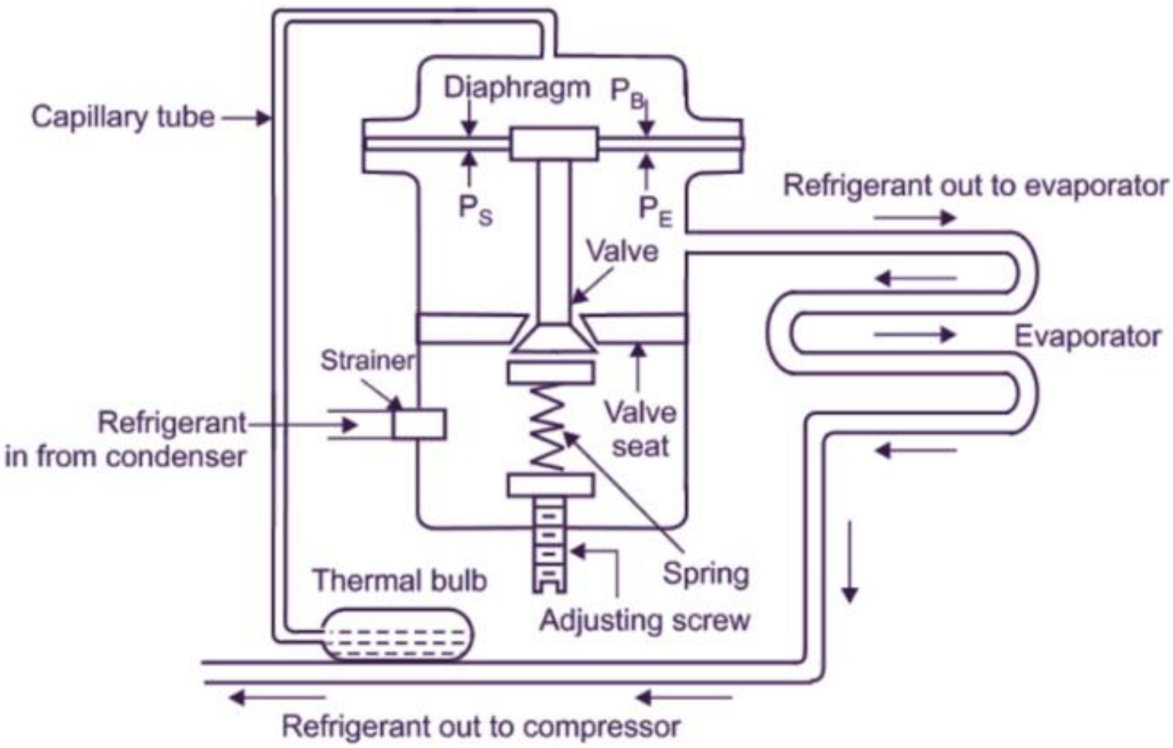
Faster evaporation rates: Mechanical circulation speeds up the process.

Improved heat transfer: Enhanced efficiency and energy savings.

Better control and consistency: Allows precise adjustments and handles a wide range of liquids.

8.

Thermostatic Expansion Valve



9.

I. Refrigeration and Air Conditioning: Cooling systems for homes and businesses.

II. Food Preservation: Extending the shelf life of perishable goods.

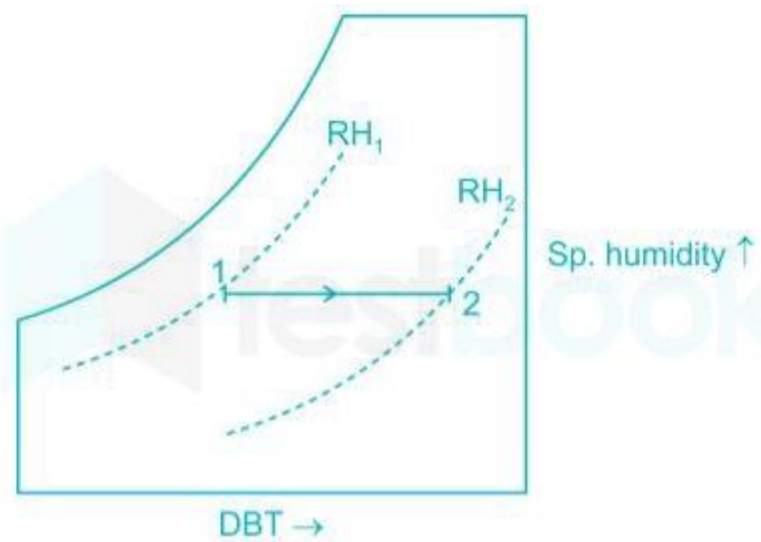
III. Cryogenics: Extremely low temperatures for medical, industrial, and scientific purposes.

IV. Biomedical and Pharmaceutical Storage: Preserving biological samples and medicines.

V. Material Science and Research: Conducting experiments in cryogenic conditions.

VI. Space Exploration: Operating equipment in the extreme cold of outer space and celestial bodies.

10. Sensible heating is a process in which the temperature of air increases without a change in its moisture content. In other words, when air undergoes sensible heating, it becomes warmer, but its humidity (moisture content) remains constant. This process is typically associated with heating systems, where air is heated without adding or removing moisture.

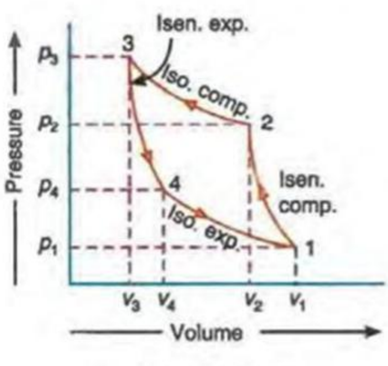


Part C

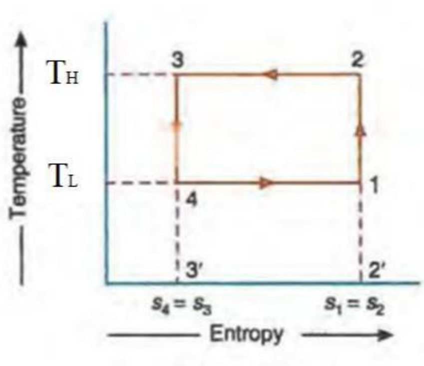
III

Reversed Carnot Air Refrigeration Cycle

1.



(a) p-v diagram.



(b) T-s diagram.

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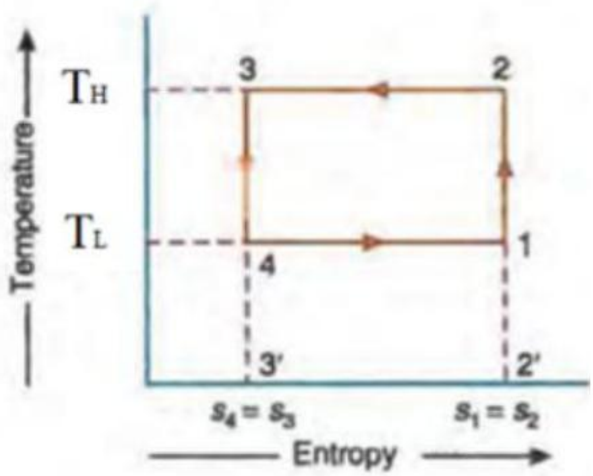
Four processes are

- **Isentropic compression process (1 – 2)**
- **Isothermal compression process (2 – 3)**
- **Isentropic expansion process (3 – 4)**
- **Isothermal expansion process (4 – 1)**

Heat absorbed by the refrigerant $Q_L = T_L (S_1 - S_4)$

Heat rejected from the refrigerant $Q_H = T_H(S_2 - S_3)$



Net Work input = Heat Rejected – Heat absorbed

$W = Q_H - Q_L = T_H(S_2 - S_3) - T_L (S_1 - S_4)$

$W = (T_H - T_L)(S_1 - S_4)$

$RE = Q_L = T_L (S_1 - S_4)$

$$COP = \frac{RE}{W} = \frac{T_L (S_1 - S_4)}{(T_H - T_L)(S_1 - S_4)}$$

$$COP = \frac{T_L}{(T_H - T_L)} = \frac{1}{\left(\frac{T_H}{T_L} - 1\right)}$$

T- Temperature in Kelvin

2. To compute the average rate of cooling in kJ/hr and the tons of refrigeration (TR) provided by the ice, we can use the given information:

Given:

Mass of ice m = 200 kg
Initial temperature of ice, T_i = -10°C
Final temperature of water, T_f = 5°C
Time (t) = 24 hours
Specific heat of ice, $C_{p,i}$ = 1.94 kJ/kg°C
Specific heat of water, $C_{p,w}$= 4.1868 kJ/kg°C
Latent heat of fusion of ice at 0°C, L = 335 kJ/kg
First, we need to calculate the energy required to raise the temperature of the ice from -10°C to 0°C and then melt it at 0°C:

Energy required for sensible heating of Ice:
$Q_1 = m\ C_{p,i}\ (0 - T_i) = 200\ kg \times 1.94\ kJ/kg°C \times 10°C = 3880\ kJ$

Energy required for melting the ice:
$Q_2 = m \times L = 200\ kg \times 335\ kJ/kg = 67000\ kJ$

Energy required for sensible heating of water:
$Q_3 = m\ C_{p,w}\ (T_f - 0) = 200\ kg \times 4.1868\ kJ/kg°C \times 5\ °C = 4186.8\ kJ$

Now, add the three energy values together to find the total energy required:

Total energy (Q_{total}) = $Q_1 + Q_2 + Q_3$
Q_{total} = 3880 kJ + 67000 kJ +4186.8 kJ = 75066.8 kJ

To find the average rate of cooling Q_ in kJ/hr, divide the total energy by the time in hours:

$Q_{average} = Q_{total} / t$
$Q_{average}$ = 75066.8 kJ / 24 hr = 3127.78 kJ/hr

Now, let's calculate the tons of refrigeration (TR) provided by the ice. One ton of refrigeration is equivalent to 3.5 kW
1 ton of refrigeration = 3.5 kW

$Q_{average}$ (in kW) = $Q_{average}$ (in kJ/hr) / 3600
$Q_{average}$ (in kW) = 3127.78 kJ/hr / 3600 = 0.8688 kW

Now, to find the TR:

TR = $Q_{average}$ (in kW) / 3.517
TR = 0.8688 kW / 3.5 = 0.248 TR

So, the average rate of cooling is approximately 3127.78 kJ/hr, and the ice provides approximately 0.248 TR.

3. Vapour Absorption Refrigeration (VAR) systems offer several advantages over Vapor Compression Refrigeration (VCR) systems in certain applications. Here are some of the key advantages of Vapour Absorption Refrigeration:

Environmentally Friendly: VAR systems typically use water as the refrigerant and an absorbent solution (e.g., lithium bromide) as the absorbent. This eliminates the need for synthetic refrigerants, such as chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs), which are known to contribute to ozone depletion and have a high global warming potential. VAR systems are more environmentally friendly and have a lower impact on climate change.

Energy Efficiency: VAR systems can be more energy-efficient than VCR systems under certain conditions, particularly when waste heat or low-grade heat sources are readily available. These systems are often used in industrial processes where waste heat can drive the absorption process, making them more energy-efficient in some applications.

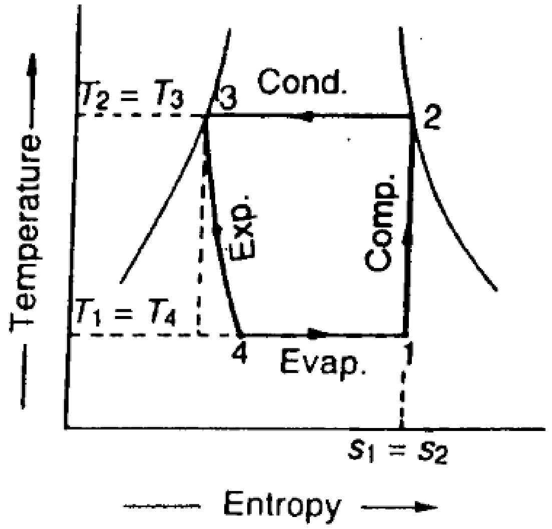
Quiet Operation: VAR systems tend to operate more quietly than VCR systems because they do not require compressors, which can produce significant noise. This makes VAR systems suitable for applications where noise levels are a concern, such as in hotels or libraries.

High Temperature Applications: Vapour Absorption Refrigeration systems can operate effectively at higher temperature levels compared to Vapor Compression systems. They are commonly used in industrial and commercial applications where higher temperature heat sources are available, such as in combined heat and power (CHP) systems.

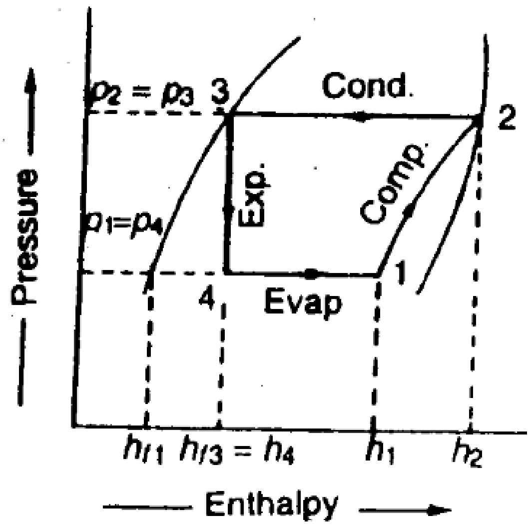
Reduced Electrical Load: In VAR systems, the primary energy input is typically in the form of heat, reducing the demand for electricity compared to VCR systems, which require electricity to operate compressors. This can lead to cost savings and reduced strain on electrical grids.

Remote and Off-Grid Applications: Vapour Absorption Refrigeration systems are suitable for remote or off-grid locations where electricity supply may be limited or unreliable. They can be powered by various heat sources, such as natural gas, solar thermal energy, or waste heat.

4.



(*a*) *T-s* diagram.



(*b*) *p-h* diagram.

The theoretical Vapour Compression Cycle with Dry Saturated Vapor after Compression is a simplified model of a refrigeration cycle:

Compression: The refrigerant is compressed into a high-pressure, high-temperature vapour with no moisture.

Condensation: The hot vapour releases heat and becomes a high-pressure liquid.

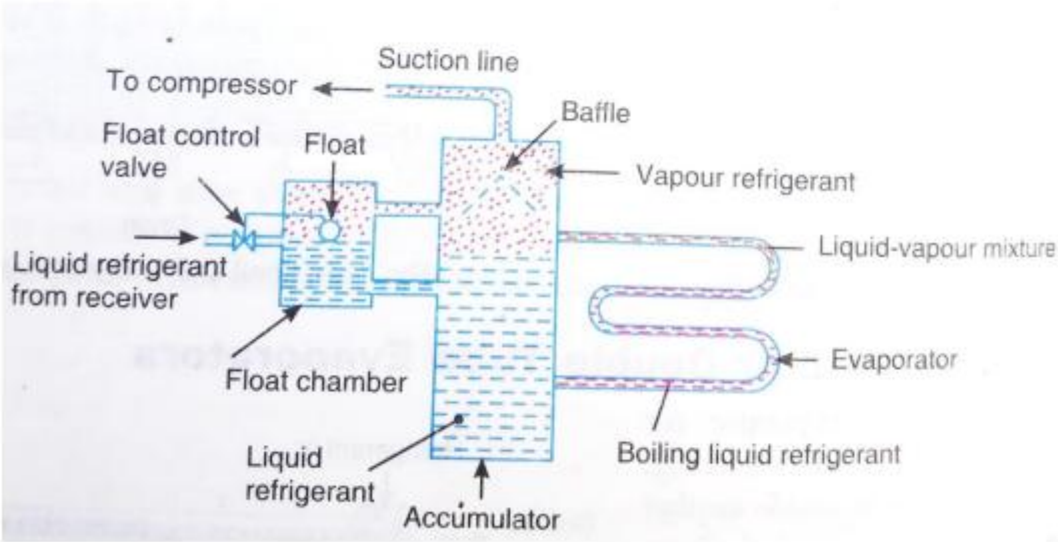
Expansion: The liquid passes through an expansion device, lowering its pressure, temperature, and becoming a mix of liquid and vapour.

Evaporation: In the evaporator, it absorbs heat, turning back into a low-pressure vapour, ready for compression again.

This cycle removes heat from the target area (cooling) and rejects it elsewhere (condenser). It's driven by the compressor and serves as the basis for many refrigeration and air conditioning systems.

5.

Flooded evaporators :

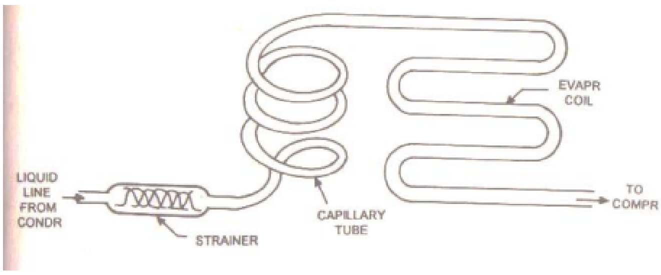


In this evaporator, a constant liquid refrigerant level is always maintained. A float control valve is used as an expansion device which maintains a constant liquid refrigerant level in the evaporator. The liquid refrigerant from the receiver passes thorough a low side float control valve and accumulator before entering the evaporator coil. The accumulator or the surge tank serve as a storage tank for the liquid refrigerant. Due to the heat supplied by the substance to be cooled, the liquid refrigerant in the evaporator coil vaporizes and thus the liquid level falls down. The accumulator supplies more liquid to the evaporator in order to keep the liquid refrigerant in the evaporator at proper level. In this way, the level of liquid refrigerant in the accumulator also falls down. Now the float valve in the float chamber will open and the liquid refrigerant from the receiver is admitted in to the accumulator. As the liquid level in the accumulator rises and reaches to the constant level , float also rises with it until the float control valve closes.

The vapor refrigerant formed by the evaporation being lighter, rises up passes on to the top of the accumulator from where it is supplied to the suction side of the compressor. The baffle plate arrests any liquid present in the vapor.

6.

Capillary tube:



Capillary tube is used as an expansion device in small capacity hermetic sealed refrigeration units such as in domestic refrigerators, water coolers, room air conditioners and freezers. It is a small copper tube of small internal diameter and of varying length depending up on the application. The inside diameter of the tube may varies from 0.5 mm to 2.5 mm and length varies from 0.5 m to 5 m. It is installed between condenser and evaporator. A fine mesh screen is provided at the inlet of the tube in order to protect it from contaminants.

Liquid refrigerant from the condenser enters the capillary tube. Due to the frictional resistance offered by small diameter tube, the pressure drops. Since the frictional resistance is directly proportional to the length and inversely proportional to the diameter, therefore longer the capillary tube and smaller its internal diameter, greater is the pressure drop created in the refrigerant flow. The diameter and length of capillary tube once selected for a given set of conditions and load cannot be operated efficiently at other conditions.

Advantages of Capillary Tube:

Simplicity: Capillary tubes are simple in design, with no moving parts or external power requirements. This makes them highly reliable and cost-effective.

Compact: Capillary tubes are small and can be easily coiled or bent to fit within limited space in refrigeration systems.

Maintenance-Free: With no moving parts, capillary tubes require minimal maintenance, reducing the need for servicing.

Low Cost: Capillary tubes are inexpensive components, which helps keep the overall cost of refrigeration systems down.

Good for Low-Capacity Systems: They are particularly suitable for low-capacity or small-scale refrigeration systems where precision isn't a primary concern.

Longevity: Due to their simple design, capillary tubes can have a long operational life.

7.

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

Relative humidity

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

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

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

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

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

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

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

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

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

We know that the relative humidity,

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

Specific humidity

We know that specific humidity,

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

Dew point temperature

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour (P_v), therefore from steam tables, we find that corresponding

to 9.62 mm of Hg or 9.62 x 133.3 = 1282.3 N/m^2 = 0.012 823 bar, the dew point temperature is,

$$t_{dp} = 10.6° \text{ C Ans.}$$

8. Given: t_d = 30°C ,P_b = 740 mm of Hg . Assume Wet Bulb Temperature, t_w = 20 ^{0}C

1.Dew point temperature

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

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

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

We know that barometric pressure,

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

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

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

∴ Partial pressure of water vapour,

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

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

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

Since the dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour (P_v), therefore from steam tables, we find that corresponding to pressure 0.017 01 bar, the dew point temperature is

$$t_{dp} = 15°\text{C Ans}$$

2. Relative humidity

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

$$P_s = 0.042\,42 \text{ bar}$$

We know the relative humidity,

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

3. *Specific humidity*

We know that specific humidity,

$$W = \frac{0.622\, p_v}{p_b - p_v} = \frac{0.622 \times 0.01701}{0.986\,42 - 0.01701}$$

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

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

9.

Solution Given: $m_a = 100$ kg/min; $t_{dt} = 35°C$; $\phi = 50\%$; ADP = 5°C; BPF = 0.15

Outlet state of moist air

Let t_{d2}, and ϕ_2 = Temperature and relative humidity of air leaving the cooling coil.

First of all, mark the initial condition of air, i.e. 35° C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 1, as shown in Fig. 22. From the psychrometric chart, we find that the dew point temperature of the entering air at point 1,

$$t_{dpt} = 23°C$$

Since the coil or apparatus dew point (ADP) is less than the dew point temperature of entering air, therefore it is a process of cooling and dehumidification.

We know that by-pass factor,

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

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

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

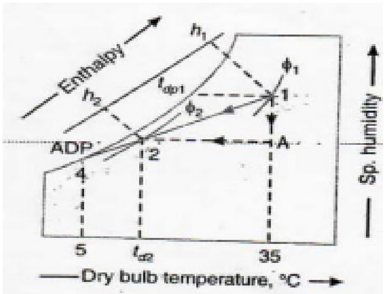


Fig.22

From the psychrometric chart, we find that the relative humidity corresponding to a dry bulb temperature (t_{d2},) of 9.5°Con the line 1-4 is $\phi_2 = 99\%$. Ans.

Cooling capacity of the coil

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

$$h_1 = 81 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 28 \text{ kJ/kg of dry air}$$

We know that cooling capacity of the coil

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

$$= 5300/210 = 25.24 \text{ TR Ans. ....}(\because 1 \text{ TR} = 210 \text{ kJ/min})$$

10.

Solution: Given: t_{dt} = 16° C; $\varnothing_1$ = 25% ; t_{d2} = 30° C; 42 = 50%

Heat added to the air

First of all, mark the initial condition of air i.e. at 16°C dry bulb temperature and 25% relative humidity on the psychrometric chart at point 1, as shown in Fig. 16.47. Then mark the final condition of air at 30° C dry bulb temperature and 50% relative humidity on the psychrometric chart at point 2. Now locate the point A by drawing horizontal line through point 1 and vertical line through point 2. From the psychrometric chart, we find that enthalpy of air at point 1,

$$h_1 = 23 \text{ k1/kg of dry air}$$

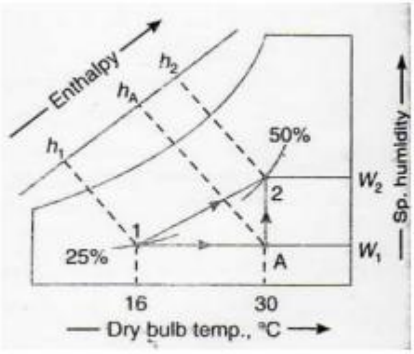


Fig.26

Enthalpy of air at point A,

$$h_A = 38 \text{ kJ/kg of dry air}$$

and enthalpy of air at point 2,

$$h_2 = 64 \text{ la/kg of dry air}$$

∴ Heat added to the air

$$= h_2 - h_1 = 64 - 23 = 41 \text{ kJ/kg of dry air Ans.}$$

Moisture added to the air

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

$$W_1 = 0.0026 \text{ kg/kg of dry air}$$

and specific humidity in the air at point 2,

$$W_2 = 0.0132 \text{ kg /kg of dry air}$$

∴ Moisture added to the air

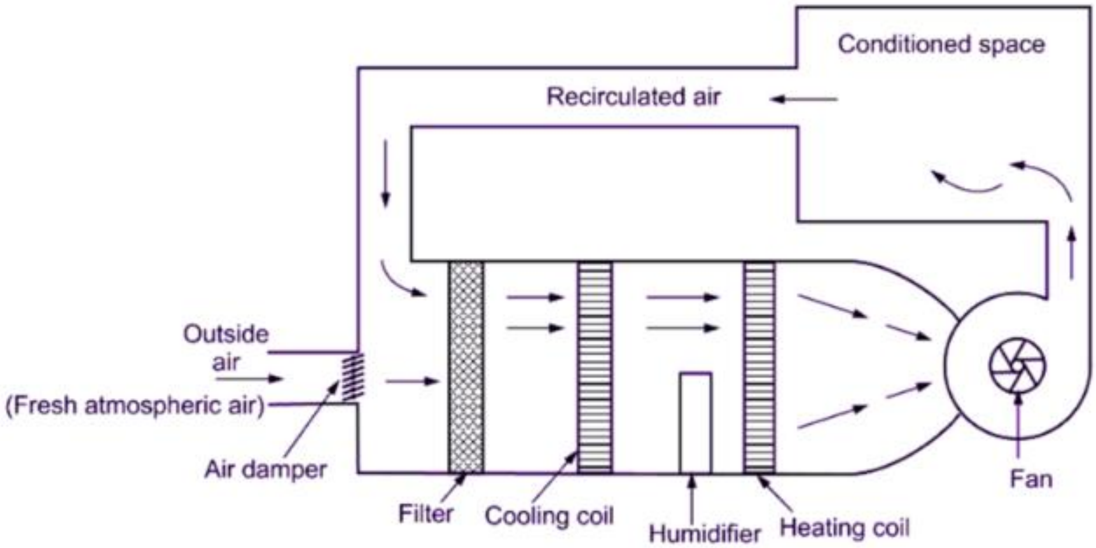
$$= W_2 – W_1 = 0.0132 - 0.0026 = 0.0106 \text{ kg/kg of dry air Ans.}$$

Sensible heat factor of the process

We know that sensible heat factor of the process,

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

11. Year round air conditioning system



A year-round air conditioning system is designed to provide both cooling and heating functions to maintain a comfortable indoor environment throughout all seasons.

In winter, the cooling coil is made inoperative. Heating coil operates to heat the air. Humidification is obtained with the help of spray type air washer (humidifier). In summer, the heating coil and humidifier are made inoperative. The cooling coil operates to cool the air. Dehumidification is obtained by cooling the air below its dew point temperature.

12.

Solution: Given $v_1 = 300\ m^3/min$; $t_{dt,} = 35°C$; $Ø_1 = 55\%$; $t_{d2} = 20°C$; $Ø_2 = 60\%$

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

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

∴ Mass of air supplied,

Sensible heat removed from the air

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

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

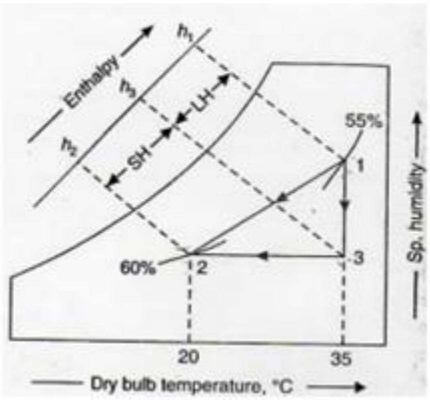


Fig.5

Enthalpy of air at point 2,

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

and enthalpy of air at point 3,

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

We know that sensible heat removed from the air,

$$SH = m_a (h_3 - h_2)$$

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

Latent heat removed from the air

We know that latent heat removed from the air,

$$LH = m_a (h_1 - h_3)$$

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

Sensible heat factor for the system

We know that sensible heal factor for the system,

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