

2020

RANK *Improvement* **WORKBOOK**



**Detailed Explanations of
Objective & Conventional *Questions***

Mechanical Engineering
Refrigeration and Air-Conditioning



MADE EASY
Publications

1

Basic Concepts, Heat Engine, Refrigerator & Reversed Carnot Cycle

LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

13. (c)
14. (d)
15. (c)
16. (a)
17. (a)
18. (c)
19. (b)
20. (c)

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LEVEL 3 Conventional Questions

Solution : 21

Capacity of reverse carnot refrigeration cycle, R.C. = 1 TR = 3.5 kW

From the given table:

$$h_2 = 1490.42 \text{ kJ/kg}, \quad h_3 = 390.587 \text{ kJ/kg},$$

$$h_r = 1450.22 \text{ kJ/kg}, \quad h_{4'} = 154.056 \text{ kJ/kg}$$

As in carnot cycle, process 1 – 2 is isentropic compression.

$$\text{So,} \quad s_2 = s_1 = s_{4'} + x_1(s_r - s_{4'})$$

$$\text{From table} \quad 5.1558 = 0.82965 + x_1(5.755 - 0.82965)$$

$$x_1 = 0.8783$$

$$\text{So} \quad h_1 = h_{4'} + x_1(h_r' - h_{4'})$$

$$\Rightarrow \quad h_1 = 1292.476 \text{ kJ/kg}$$

Process 3 – 4 is isentropic expansion process so

$$s_3 = s_4 = s_{4'} + x_4(s_r - s_{4'})$$

$$\Rightarrow \quad 1.64377 = 0.82965 + x_4(5.755 - 0.82965)$$

$$x_4 = 0.1653$$

$$\Rightarrow \quad h_4 = h_{4'} + x_4(h_r - h_{4'})$$

$$\Rightarrow \quad h_4 = 154.056 + 0.1653(1450.22 - 154.056)$$

$$h_4 = 368.311 \text{ kJ/kg}$$

So,

(i) Let \dot{m} bet the mass flow rate of refrigerant

$$\Rightarrow \quad \dot{m}(h_1 - h_4) = \text{R.C.} = 3.5$$

$$\Rightarrow \quad \dot{m} = \frac{3.5}{1292.533 - 368.311} = 0.003787 \text{ kg/s}$$

$$\text{or} \quad \dot{m} = 0.2272 \text{ kg/min}$$

(ii) Work done by compressor, $W_C = \dot{m}(h_2 - h_1) = 0.003787(1490.42 - 1292.476) = 0.7496 \text{ kW}$

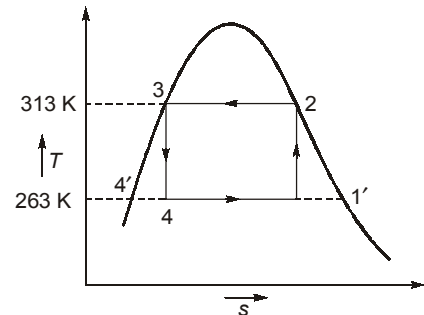
$$\text{or} \quad W_C = 749.6 \text{ W}$$

(iii) Condenser heat rejection = $\dot{m}(h_2 - h_3) = 0.003787(1490.42 - 390.587) = 4.165 \text{ kW}$

(iv) Expansion work = $\dot{m}(h_3 - h_4)$

$$= \frac{0.2283}{60}(390.587 - 368.22) = 84.7564 \text{ watt}$$

$$\text{COP} = \frac{RC}{W_{\text{net}}} = \frac{3.5167 \times 1000}{749.6 - 84.7564} = 5.2895$$



Solution : 22

For summer :

$$t_k = 45^\circ\text{C} + 15^\circ\text{C} = 60^\circ\text{C}$$

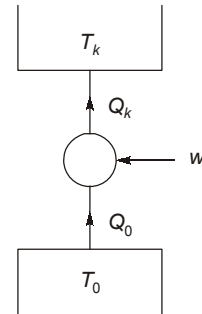
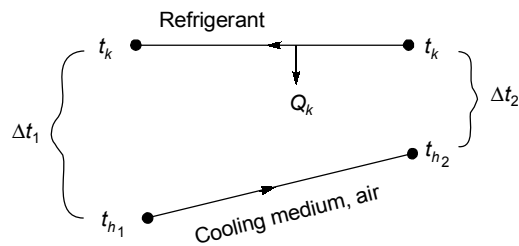
$$t_o = 5^\circ\text{C}$$

$$\text{COP} = \frac{273 + 5}{60 - 5} = \frac{278}{55} = 5.05$$

for winter : $t_k = 55^\circ\text{C}$

$$t_o = 15 - 5 = 10^\circ\text{C}$$

$$\text{COP} = \frac{273 + 55}{55 - 10} = \frac{328}{45} = 7.3$$

**Solution : 23**Given $t_k = 55^\circ\text{C}$, $t_{h1} = 43^\circ\text{C}$, $t_{h2} = 53^\circ\text{C}$

$$\Delta t_1 = 55 - 43 = 12^\circ\text{C}$$

$$\Delta t_2 = 55 - 53 = 2^\circ\text{C}$$

$$\text{COP} = \frac{273 + 5}{55 - 5} = \frac{278}{50} = 5.56$$

$$\text{Power consumption} = \frac{\text{Refrigerating effect}}{\text{COP}} = \frac{1 \text{ TR}}{\text{COP}}$$

$$\dot{W} = \frac{3.5167}{5.56} = 0.63 \text{ kW}$$

Heat rejected

$$\dot{Q}_k = \dot{Q}_0 + \dot{W} = 3.5167 + 0.63 = 4.15 \text{ kW}$$

Mass flow rate of cooling air,

$$\dot{m}_a = \frac{\dot{Q}_k}{C_p \Delta t_h} = \frac{4.15}{1.005(53 - 43)} = 0.413 \text{ kg/s}$$

LMTD of heat exchanger,

$$\Delta T_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \left(\frac{\Delta t_1}{\Delta t_2} \right)} = \frac{12 - 2}{\ln 6} = 5.58^\circ\text{C}$$

Area of the heat exchanger,

$$A = \frac{\dot{Q}_k}{U \Delta T_m} = \frac{4.15 \times 10^3}{250 \times 5.58} = 2.975 \text{ m}^2$$



2

Vapour Compression Refrigeration Cycle

LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

12. (b)
13. (a)
14. (b)
15. (c)
16. (c)
17. (c)
18. (a)
19. (d)
20. (c)
21. (a)
22. (b)
23. (c)
24. (b)
25. (a)
26. (c)
27. (d)
28. (d)

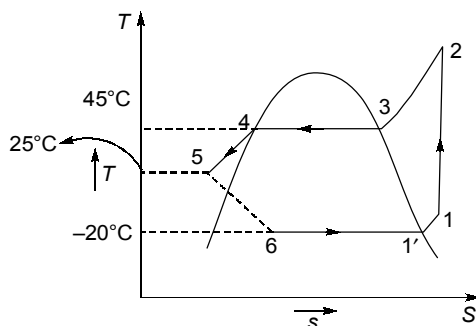
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LEVEL 3 Conventional Questions

Solution : 29

Given that

$$RC = 10 \text{ TR} = 35 \text{ kW}$$



Refrigerant vapour is being used to subcool the liquid from condenser by means of a heat exchanger.

$$\text{As given } h_r = 178.7 \text{ kJ/kg}, \quad h_3 = 204.9 \text{ kJ/kg},$$

$$h_4 = 79.7 \text{ kJ/kg},$$

$$h_4 = h_5 + C_{pf}(T_4 - T_5) \Rightarrow 79.7 = h_5 + 1.02(45 - 25)$$

$$h_5 = 59.3 \text{ kJ/kg} = h_6$$

In heat exchanger, using heat balance:

$$h_4 - h_5 = h_1 - h_r$$

$$79.7 - 59.3 = h_1 - 178.7$$

$$h_1 = 199.1 \text{ kJ/kg}$$

As at constant pressure

$$h_1 = h_r + C_{pge}(T_1 - T_r) \Rightarrow 199.1 = 178.7 + 0.61(T_1 + 20)$$

⇒

$$T_1 = 13.442^\circ\text{C} \text{ or } 286.442 \text{ K}$$

Entropy at 1: ⇒

$$s_1 = s_r + C_{pg} \ln \frac{T_1}{T_r} = 0.7088 + 0.61 \ln \frac{286.442}{253}$$

⇒

$$s_1 = 0.78453 \text{ kJ/kgK}$$

As process 1 – 2 is isentropic compression, so

$$s_1 = s_2 = s_3 + C_{pgc} \ln \frac{T_2}{T_3}$$

⇒

$$0.78453 = 0.6812 + 0.755 \ln \frac{T_2}{318}$$

⇒

$$T_2 = 364.64 \text{ K}$$

Compressor discharge temperature,

$$t_2 = 91.64^\circ\text{C}$$

$$\text{Enthalpy } h_2 = h_3 + C_{pgc}(t_2 - t_3)$$

$$= 204.9 + 0.755 (91.64 - 45)$$

$$h_2 = 240.114 \text{ kJ/kg}$$

$$\text{Refrigeration capacity} = \dot{m}(h_1 - h_6) = 35$$

$$\Rightarrow \dot{m} = \frac{35}{(178.7 - 59.3)}$$

$$\Rightarrow \text{mass flow rate, } \dot{m} = 0.2931 \text{ kg/s}$$

$$\begin{aligned} \text{Compressor work, } W_C &= \dot{m}(h_2 - h_1) \\ &= 0.2931(240.114 - 199.1) \end{aligned}$$

$$\Rightarrow W_C = 12.021 \text{ kW}$$

$$\text{Condenser heat rejection, } Q_{CR} = \dot{m}(h_2 - h_4)$$

$$\begin{aligned} \Rightarrow Q_{CR} &= 0.2931(240.114 - 79.7) \\ Q_{CR} &= 47.017 \text{ kW} \end{aligned}$$

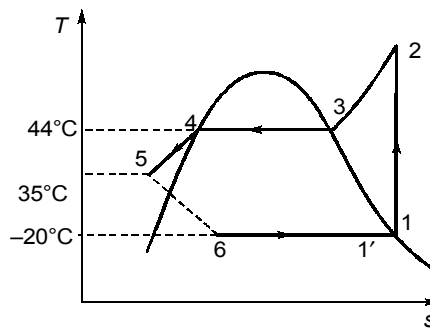
$$\text{COP} = \frac{RC}{W_C} = \frac{35}{12.021} = 2.911$$

Solution : 30

Given that:

For compressor Bore $d = 10 \text{ cm} = 0.1 \text{ m}$, Stroke, $l = 12 \text{ cm} = 0.12 \text{ m}$, Speed $N = 2800 \text{ rpm}$

Clearance factor, $C = 0.04$



$$\text{As given } C_{pg} = 0.95 \text{ kJ/KgK}, \quad C_{pf} = 1.30 \text{ kJ/Kg-K}$$

$$h_1 = 397.53 \text{ kJ/kg}, \quad h_3 = 416.43 \text{ kJ/kg},$$

$$h_4 = 254.30 \text{ kJ/kg}$$

$$\Rightarrow h_4 = h_5 + C_{pf}(T_4 - T_5) \Rightarrow h_5 = 254.30 - 1.30 (44 - 35)$$

$$\Rightarrow h_5 = 242.6 \text{ kJ/kg} = h_6$$

Process 1 – 2 is isentropic compression

$$\Rightarrow s_2 = s_1 = s_3 + C_{pg} \ln \frac{T_2}{T_3}$$

$$\Rightarrow 1.7841 = 1.6943 + 0.95 \ln \frac{T_2}{317} \Rightarrow T_2 = 348.427 \text{ K}$$

So,

$$h_2 = h_3 + C_{pg}(T_2 - T_3)$$

$$h_2 = 416.43 + 0.95(75.427 - 44)$$

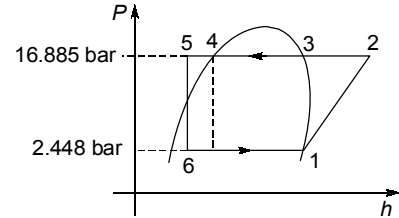
$$h_2 = 446.285 \text{ kJ/kg}$$

(i) Temperature of vapour at compressor exit is

$$T_2 = 348.427 \text{ K}$$

⇒

$$T_2 = 75.427^\circ\text{C}$$

(ii) Volumetric efficiency $\eta_v = 1 + C - C \left(\frac{P_2}{P_1} \right)^{1/n}$ Where n is polytropic index of compressor

$$\text{So, } \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = \frac{T_2}{T_1} \text{ or } \left(\frac{16.885}{2.448} \right)^{\frac{n-1}{n}} = \frac{348.427}{253}$$

$$n = 1.1986$$

$$\Rightarrow \eta_v = 1 + 0.04 - 0.04 \left(\frac{16.885}{2.448} \right)^{\frac{1}{1.1986}}$$

$$\Rightarrow \eta_v = 83.96\%$$

(iii) Volumetric efficiency, $\eta_v = \frac{\dot{m} v_1}{\frac{\pi}{4} d^2 l \times K \times N / 60}$

$$\Rightarrow 0.8396 = \frac{\dot{m} \times 0.0928}{\frac{\pi}{4} \times 0.1^2 \times 0.12 \times 1 \times \frac{2800}{60}}$$

$$\dot{m} = 0.3979 \text{ kg/s or } 23.875 \text{ kg/min}$$

(iv) Cooling capacity = $\dot{m}(h_1 - h_6)$

$$\text{RC} = 0.3979(397.53 - 242.6)$$

⇒ Refrigeration capacity (RC) = 61.646 kW

or R.C. = 17.613 TR

(v) Power of compressor $W_C = \dot{m}(h_2 - h_1) = 0.3979(446.285 - 397.53)$

$$W_C = 19.4 \text{ kW}$$

(vi) $\text{COP} = \frac{\text{RC}}{W_C} = \frac{61.646}{19.4} = 3.177$

Solution : 31

Given data:

$$\dot{m} = 30 \text{ kg/min} = 0.5 \text{ kg/s}$$

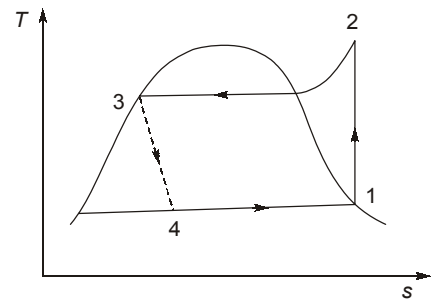
$$h_4 = 74.6 \text{ kJ/kg}$$

$$h_1 = 185.4 \text{ kJ/kg}$$

$$h_2 = 208 \text{ kJ/kg}$$

$$\begin{aligned} \text{COP} &= \frac{\text{Heat extracted}}{\text{Power input}} = \frac{h_1 - h_4}{h_2 - h_1} \\ &= \frac{185.4 - 74.6}{208 - 185.4} = \frac{110.8}{22.6} = 4.9 \end{aligned}$$

$$\text{Power input} = \text{mass flow rate} \times (h_2 - h_1) = 0.5(208 - 185.4) = 0.5 \times 22.6 = 11.3 \text{ kW}$$



Solution : 32

The cycle of refrigeration is represented on T-s diagram of figure.

Enthalpy at '2',
$$h_2 = h_2' + c_p(T_2 - T_2')$$

From the given table:

$$h_2' = 183.2 \text{ kJ/kg}$$

$$(T_2 - T_2') = \text{Degree of superheat as the vapour enters the compressor} = 7^\circ\text{C}$$

$$h_2 = 183.2 + 0.733 \times 7 = 188.33 \text{ kJ/kg}$$

Also, entropy at '2',
$$s_2 = s_2' + c_p \log_e \frac{T_2}{T_2'} = 0.7020 + 0.733 \log_e \left(\frac{270}{263} \right) = 0.7212 \text{ kJ/kgK}$$

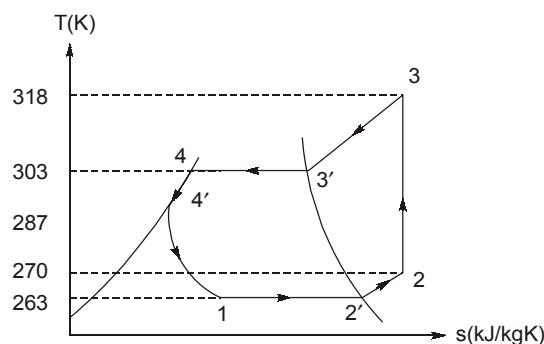
For isentropic process 2-3

$$\text{Entropy at '2'} = \text{Entropy at '3'}$$

$$0.7212 = s_3' + c_p \log_e \left(\frac{T_3}{T_3'} \right) = 0.6854 + 0.733 \log_e \left(\frac{T_3}{303} \right)$$

$$\therefore \log_e \left(\frac{T_3}{303} \right) = 0.0488$$

i.e.,
$$T_3 = 318 \text{ K}$$



Now, enthalpy at '3',
$$h_3 = h_3' + c_p(T_3 - T_3') = 199.6 + 0.733(318 - 303) = 210.6 \text{ kJ/kg}$$

Also, enthalpy at 4', $h_{f_{4'}} = h_{f_4} - (c_p)_{\text{liquid}}(T_4 - T_{4'}) = 64.6 - 1.235 \times 6 = 57.19 \text{ kJ/kg}$

For the process 4' - 1,

Enthalpy at 4' = enthalpy at 1 = 57.19 kJ/kg

For specific volume at 2,

$$\frac{v_2'}{T_2'} = \frac{v_2}{T_2}$$

$$\therefore v_2 = \frac{v_2'}{T_2'} \times T_2 = 0.0767 \times \frac{270}{263} = 0.07874 \text{ m}^3/\text{kg}$$

(i) Refrigerating effect per kg

$$= h_2 - h_1 = 188.33 - 57.19 = 131.14 \text{ kJ/kg}$$

(ii) Mass of refrigerant to be circulated per minute for producing effect of 2400 kJ/min.

$$= \frac{2400}{131.14} = 18.3 \text{ kg/min}$$

(iii) Theoretical piston displacement per minute

$$= \text{Mass flow/min.} \times \text{specific volume at suction}$$

$$= 18.3 \times 0.07874 = 1.441 \text{ m}^3/\text{min}$$

(iv) Theoretical power required to run the compressor

$$= \text{Mass flow of refrigerant per sec.} \times \text{compressor work/kg}$$

$$\text{or} \quad = \frac{18.3}{60} \times (h_3 - h_2) = \frac{18.3}{60} \times (210.6 - 188.33) = 6.79 \text{ kW}$$

(v) Heat removed through the condenser per min.

$$= \text{Mass flow of refrigerant} \times \text{heat removed per kg of refrigerant}$$

$$= 18.3(h_3 - h_{f_{4'}}) = 18.3(210.6 - 57.19) = 2807.4 \text{ kJ/min}$$

(vi) Theoretical bore (d) and stroke (l):

Theoretical piston displacement per cylinder

$$= \frac{\text{Total displacement per minute}}{\text{Number of cylinder}} = \frac{1.441}{2} = 0.7205 \text{ m}^3/\text{min}$$

Also, length of stroke = 1.25 × diameter of piston

$$\text{Hence,} \quad 0.7205 = \frac{\pi}{4} d^2 \times (1.25 d) \times 1000$$

$$\text{i.e.,} \quad d = 0.09 \text{ m or } 90 \text{ mm}$$

$$\text{and} \quad l = 1.25 d = 1.25 \times 90 = 112.5 \text{ mm}$$



3

VARS, Gas Cycle Refrigeration & other Refrigeration System

LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

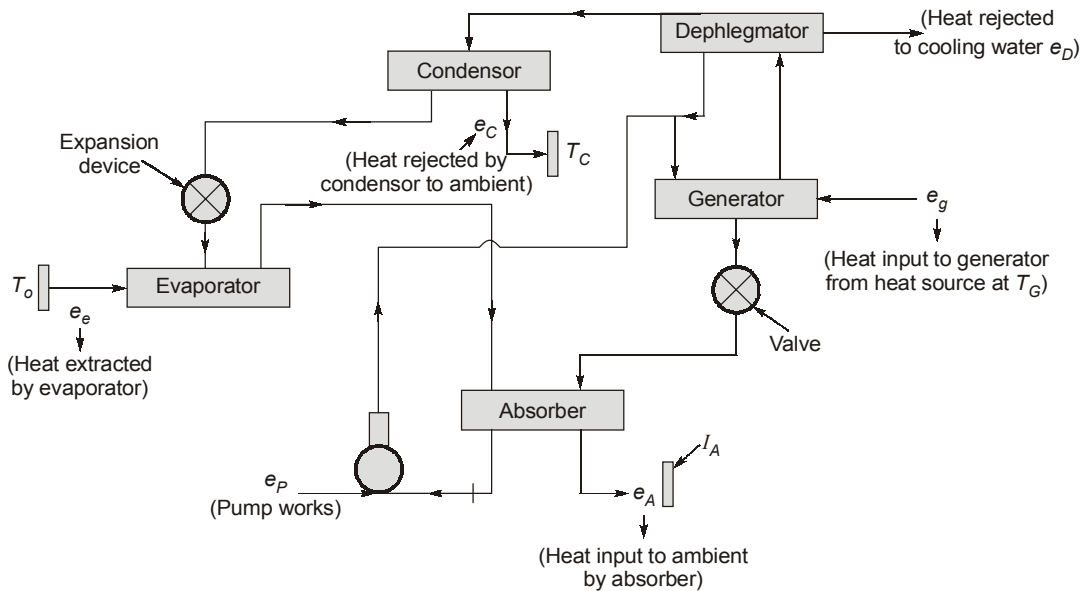
14. (c)
15. (a)
16. (b)
17. (a)
18. (b)
19. (d)
20. (a)
21. (b)

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LEVEL 3 Conventional Questions

Solution : 22

Vapour absorption system with aqua-ammonia as refrigerant is used to utilize available heat energy as an alternative to compressor work. General schematic is as shown below:



As compared to a vapour compression system differences are:

- **Absorber:** Refrigerant vapours of ammonia are absorbed in a weak solution with water.
- **Pump:** Used to pump aqua-ammonia solution to condenser pressure.
- **Generator:** Distillation of ammonia vapour from the rich solution leaving weak solution for recirculation.
- **Dephlegmator:** Further concentration of vapour to separate water from ammonia vapour.

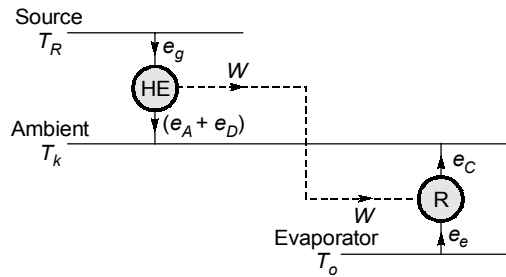
Nature of heat/work interactions as required are mentioned in the figure.

Refrigerant vapour of NH_3 formed as a result of heat extraction in evaporator goes to absorber where it is dissolved in a weak aqua-ammonia solution. Ammonia vapour loses heat e_a which is rejected by the absorber and forms a concentrated solution which is pumped to condenser pressure by pump. Pump work $= (-vdp)$, is usually negligible since specific volume- v of liquid is very less. In generator distillation of vapour of NH_3 takes place by supplying heat from source- e_g at T_h . Weak solution is recirculated through a valve to absorber. Further condensation of ammonia vapours takes place in dephlegmator along with rejection of heat e_d to cooling water.

Maximum COP of cycle

Aqua-ammonia cycle is a heat operated refrigeration machine. It can be approximated as a combination of a heat engine operated on aqua solution circuit and a refrigerator operated on ammonia cycle.

For heat source temperature T_R , ambient temperature ($T_A = T_C = T_K$) and evaporator temperature T_o , above system can be shown as:



Work output of heat engine is used to power the refrigerator.

Energy balance of above system can be shown as below

$$e_e + e_p + e_g = e_c + e_D + e_A = e_K \quad (\text{net heat rejected})$$

$$\text{COP of cycle} = \frac{e_e}{e_g} = \frac{e_e}{W} \times \frac{W}{e_g}$$

$$= \text{COP}_R \times \eta_{\text{HE}}$$

$$\text{COP}_R = \text{COP of refrigerator}$$

$$\eta_{\text{HE}} = \text{efficiency of heat engine}$$

For max COP - Assumptions

- (a) Heat engine and refrigerator operate on reversible cycle as Carnot engine and refrigerator.
- (b) No pressure drop or heat loss during any of the process.

$$\text{COP}_{\text{max}} = \text{COP}_R \times \text{COP}_{\text{HE}} = \left(\frac{T_0}{T_K - T_0} \right) \times \left(1 - \frac{T_K}{T_R} \right)$$

From given data, we get,

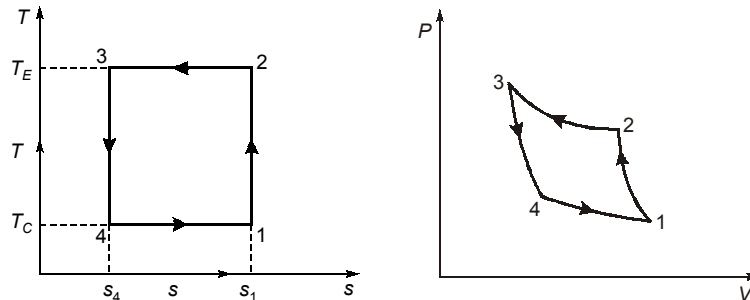
$$T_K = 27^\circ\text{C} = 300 \text{ K}$$

$$T_h = 127^\circ\text{C} = 400 \text{ K}$$

$$T_0 = -13^\circ\text{C} = 260 \text{ K}$$

$$\therefore \text{COP}_{\text{max}} = \left(\frac{260}{300 - 260} \right) \times \left(1 - \frac{300}{400} \right) = 1.625$$

Solution : 23



- 1-2 → Isentropic compression
- 2-3 → Isothermal Heat Rejection
- 3-4 → Isentropic expansion

4-1 → Isothermal Heat addition

As we can see that in p-v diagram, both in process 1-2, and 2-3 the pressure is increasing so two compressors are required.

Similarly in process 3-4 and 4-1 the pressure is decreasing and volume is increasing so two expanders are required.

Work done by compressors (per kg) = $w_{12} + w_{23}$

$$= \frac{\gamma}{\gamma-1} R(T_1 - T_2) + RT_E \log_e \frac{V_3}{V_2} = c_p(T_C - T_E) + RT_E \log_e \frac{V_3}{V_2}$$

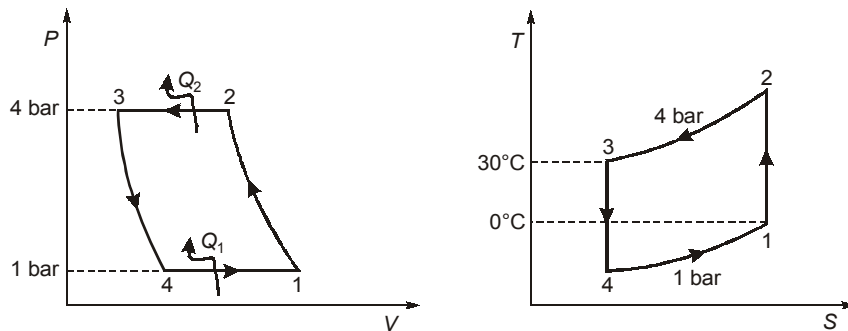
$$\text{Work done by expanders} = c_p(T_3 - T_4) + RT_C \log_e \left(\frac{V_1}{V_4} \right) = c_p(T_E - T_C) + RT_C \log_e \left(\frac{V_1}{V_4} \right)$$

Alternatively Net work required = Heat rejected – Heat absorbed

$$= T_E(s_1 - s_4) - T_C(s_1 - s_4) = (T_E - T_C)(s_1 - s_4)$$

$$(COP)_R = \frac{\text{Heat absorbed}}{\text{Work done}} = \frac{T_C(s_1 - s_4)}{(T_E - T_C)(s_1 - s_4)} = \frac{T_C}{T_E - T_C}$$

Solution : 24



Pressure ratio: $r_p = \frac{p_2}{p_1} = \frac{p_3}{p_4}$

Assuming 1-2 and 3-4 as isentropic

$$\therefore T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = T_1 \cdot r_p^{\left(\frac{\gamma-1}{\gamma} \right)}$$

$$\therefore T_3 = T_4 \cdot r_p^{\left(\frac{\gamma-1}{\gamma} \right)}$$

Coefficient of performance,

$$COP = \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{c_p(T_1 - T_4)}{c_p(T_2 - T_1) - c_p(T_3 - T_4)}$$

$$COP = \frac{(T_1 - T_4)}{(T_2 - T_1) - (T_3 - T_4)} \quad \dots(i)$$

$$= \frac{\left(T_1 - \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} \right)}{T_1 \left(r_p^{\frac{\gamma-1}{\gamma}} - 1 \right) - T_3 \left(1 - \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}} \right)} = \frac{\left[T_1 - \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} \right]}{\left[T_1 - \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} \right] \left[r_p^{\frac{\gamma-1}{\gamma}} - 1 \right]}$$

$$COP = \frac{1}{\left(r_p^{\frac{\gamma-1}{\gamma}} - 1 \right)}$$

Temperature at all points

$$\therefore T_2 = 273(4)^{\frac{0.4}{1.4}} = 405.676 \text{ K}$$

$$T_4 = \frac{303}{4^{\frac{2}{7}}} = 203.9 \text{ K}$$

Volume flow rate of air for 1 TR capacity

$$\therefore m_a c_p (T_1 - T_4) = 1 \text{ TR} = 3.5$$

$$m_a = \frac{3.5}{1.005 \times (273 - 203.9)} = 0.050402 \text{ kg/s}$$

At inlet to compressor density of air

$$\rho_1 = \frac{p_1}{RT_1} = \frac{10^5}{287 \times 273} = 1.276 \text{ kg/m}^3$$

Volume flow rate at inlet to compressor

$$V_1 = \frac{m}{\rho} = \frac{0.050402}{1.276} = 0.039 \text{ m}^3/\text{s} = 2.37 \text{ m}^3/\text{min}$$

Volume flow rate at exit to turbine

$$V_4 = \frac{mRT_4}{P_4} = \frac{0.0504 \times 0.287 \times 203.9}{100} = 0.0294 \text{ m}^3/\text{s} = 1.76 \text{ m}^3/\text{min}$$

Solution : 25

Given that:

$$\text{Cooling capacity } RC = 1 \text{ TR} = 3.5 \text{ kW}$$

$$T_1 = 10^\circ\text{C} = 283 \text{ K},$$

$$T_3 = 27^\circ\text{C} = 300 \text{ K},$$

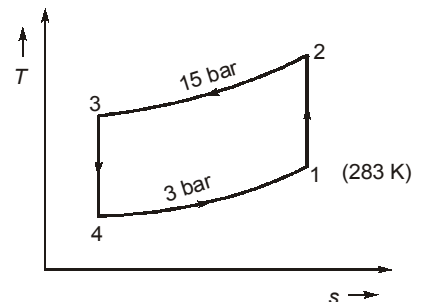
$$\gamma = 1.4,$$

$$R = 0.281 \text{ kJ/Kg-K},$$

$$C_p = 1.005 \text{ kJ/kg-K}$$

As process 1-2 and 3-4 are isentropic so:

$$\frac{T_2}{T_1} = \left(\frac{15}{3} \right) \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{15}{3} \right)^{1.4} \Rightarrow T_2 = 448.221 \text{ K}$$



Similarly
$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = 189.415 \text{ K}$$

(i) As refrigeration capacity

$$\dot{m} \times C_p (T_1 - T_4) = 3.5$$

$$\dot{m} \times 1.005 \times (283 - 189.415) = 3.5$$

$$\Rightarrow \dot{m} = 0.03721 \text{ Kg/s} = 2.233 \text{ kg/min}$$

(ii) At compressor inlet $P_1 \dot{V}_1 = \dot{m} \times R \times T_1 \Rightarrow 300 \times V_1 = 0.03721 \times 0.281 \times 283$

$$\dot{V}_1 = 0.09863 \text{ m}^3/\text{s}$$

At turbine exit: $P_4 \dot{V}_4 = \dot{m} R T_4 \Rightarrow 300 \times V_4 = 0.03721 \times 0.281 \times 189.415$

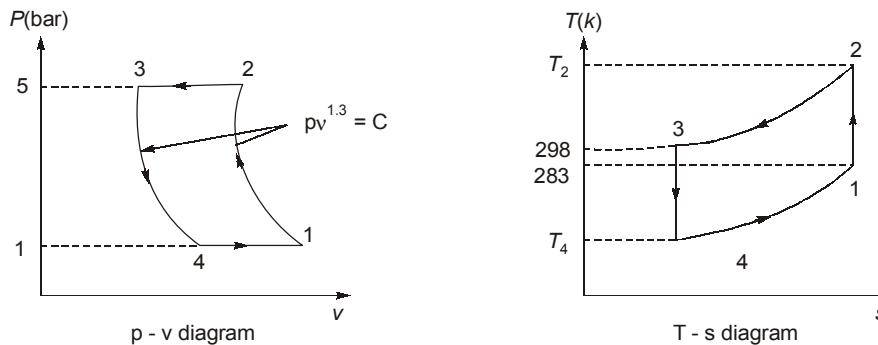
$$\Rightarrow V_4 = 6.602 \times 10^{-3} \text{ m}^3/\text{s}$$

(iii) Compressor work $W_C = \dot{m} \times C_p \times (T_2 - T_1)$
 $= 0.03721 \times 1.005 \times (448.221 - 283)$
 $W_C = 6.178 \text{ kW}$

Turbine work $W_T = \dot{m} C_p (T_3 - T_4)$
 $= 0.03721 \times 1.005 \times (300 - 189.415)$
 $W_T = 4.135 \text{ kW}$

(iv)
$$\text{COP} = \frac{RC}{W_{\text{net}}} = \frac{3.5}{(6.178 - 4.135)} = 1.713$$

Solution : 26



Given :

$$P_1 = P_4 = 1 \text{ bar}$$

$$P_2 = P_3 = 5 \text{ bar}$$

$$T_1 = 10^\circ\text{C} = 10 + 273 = 283 \text{ K}$$

$$T_2 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K}$$

Polytropic process exponent,

$$n = 1.3$$

$$\text{Speed, } N = 300 \text{ rpm}$$

$$\eta_v = 85\%$$

$$\frac{\text{length}}{\text{diameter}} = 1.5$$

$$C_p = 1.005 \text{ kJ/kgK}$$

$$C_v = 0.71 \text{ kJ/kgK}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \left(\frac{5}{1}\right)^{\frac{1.3-1}{1.3}} = 5^{0.23} = 1.45$$

$$T_2 = T_1 \times 1.45 = 283 \times 1.45 = 410.3 \text{ K}$$

similarly,

$$\frac{T_3}{T_4} = (5)^{0.23} = 1.45$$

$$T_4 = \frac{T_3}{1.45} = \frac{298}{1.45} = 205.5 \text{ K}$$

Heat extracted from the refrigerating system per kg in air

$$= C_p(T_1 - T_4) = 1.005(283 - 205.5) = 78 \text{ kJ/kg}$$

$$R = C_p - C_v = 1.005 - 0.71$$

$$= 0.295 \text{ kJ/kgK}$$

Work done during compression,

$$W_{1-2} = \frac{n}{n-1} R(T_2 - T_1) = \frac{1.3}{1.3-1} \times 0.295(410.3 - 283)$$

$$W_{1-2} = 162.7 \text{ kJ/kg}$$

Workdone during expansion,

$$W_{3-4} = \frac{n}{n-1} R(T_3 - T_4) = \frac{1.3}{1.3-1} \times 0.295(298 - 205.5)$$

$$= 118.246 \text{ kJ/kg}$$

$$\text{Net workdone} = W_{1-2} - W_{3-4} = 162.7 - 118.2$$

$$= 44.5 \text{ kJ/kg}$$

$$\text{Theoretical COP} = \frac{\text{Heat extracted}}{\text{Work input}} = \frac{78}{44.5} = 1.75$$

$$\text{Actual COP} = 50\% \text{ of theoretical COP}$$

$$= 0.5 \times 1.75 = 0.875$$

$$\text{Actual heat extracted} = 44.5 \times 0.875 \approx 39 \text{ kJ/kg}$$

Refrigerating capacity of the system

$$= 10 \text{ TR} = 10 \times 210 = 2100 \text{ kJ/min}$$

$$\text{Mass of air circulated per minute} = \frac{2100}{39} = 53.8 \text{ kg/min}$$

Volume of air at point 1, $v_1 = \frac{m_a RT_1}{P_1} = \frac{53.8 \times 295 \times 283}{1 \times 10^5} = 45 \text{ m}^3/\text{min}$

$$v_1 = \left(\frac{\pi D^2}{4} \times L \right) N \eta_v$$

$$45 = \frac{3.14}{4} \times D^2 \times 1.5D \times 300 \times 0.85$$

$$D^3 = \frac{45}{300} = 0.15$$

$$D = 0.53 \text{ m} = 530 \text{ mm}$$

$$L = 1.5 \times D = 1.5 \times 530 = 795 \text{ mm}$$

Solution : 27

Given data:

$$T_1 = 5^\circ\text{C} = (5 + 273)\text{K} = 278 \text{ K}$$

$$T_3 = 30^\circ\text{C} = (30 + 273)\text{K} = 303 \text{ K}$$

$$r_p = \frac{p_2}{p_1} = 4$$

$$p_2 = 6 \text{ bar} = 600 \text{ kPa}$$

$$p_1 = \frac{p_2}{4} = \frac{600}{4} = 150 \text{ kPa}$$

Cooling capacity

$$Q_2 = 2\text{TR} = 2 \times 3.5 \text{ kW} = 7 \text{ kW}$$

(i) For reversible adiabatic process 1–2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_2}{278} = (4)^{\frac{1.4-1}{1.4}} = 4^{0.2857} = 1.4859$$

or $T_2 = 1.4859 \times 278 = 412.08 \text{ K}$

For reversible adiabatic process 3–4,

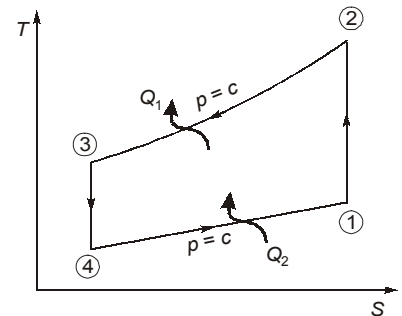
$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

$$\frac{303}{T_4} = (4)^{\frac{1.4-1}{1.4}} = 1.4859$$

or $T_4 = \frac{303}{1.4859} = 203.91 \text{ K}$

(ii) Cooling capacity: $Q = mc_p(T_1 - T_4)$
 $7 = m \times 1.005(278 - 203.91)$

or $m = 0.094 \text{ kg/s}$



Applying equation of state at point 1,

$$p_1 V_1 = mRT_1$$

where p_1 is in kPa

V_1 is in m^3/s ; m is in kg/s; $R = 0.287$ kJ/kgK T_1 is in K.

$$\therefore 150 \times V_1 = 0.094 \times 0.287 \times 278$$

$$\text{or } V_1 = 0.0499 \text{ m}^3/\text{s}$$

Applying equation of state at point 4,

$$p_4 V_4 = mRT_4$$

$$150 \times V_4 = 0.094 \times 0.287 \times 203.91$$

$$\therefore p_4 = p_1$$

$$\text{or } V_4 = 0.03667 \text{ m}^3/\text{s}$$

Solution : 28

Solar Energy for Refrigerations:

- The schematic of a single-stage absorption system using ammonia as the refrigerant and ammonia-water as the absorbent is shown in figure (1).
- The condenser, throttle and evaporator operate in the exactly the same manner as for the vapor compression system. In place of the compressor, however, the absorption system uses a series of three heat exchanger (absorber, regenerating intermediate heat exchanger and a generator) and a small solution pump. Ammonia vapor exiting the evaporator (State 6) is absorbed in a liquid solution of water-ammonia in the absorber.

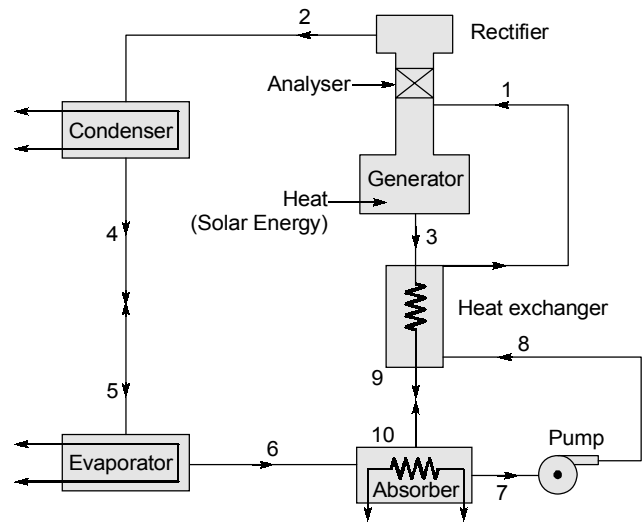


Fig. (1)

The absorption of ammonia vapor into the water-ammonia solution is analogous to a condensation process. The process is exothermic and so cooling water is required to carry away the heat of absorption.

- The ammonia-rich liquid solution leaving the absorber (State 7) is pumped to a higher pressure, passed through a heat exchanger and delivered to the generator (State 1). The power requirement for the pump is much smaller than that for the compressor since v , the specific volume of the liquid solution, is much smaller than the specific volume of a refrigerant vapor.
- In the generator, the liquid solution is heated, which promotes desorption of the refrigerant (ammonia) from the solution. Unfortunately, some water also is desorbed with the ammonia, and it must be separated from the ammonia using the rectifier. With the use of a rectifier, water exits at State 2 with the ammonia and travels to the evaporator, where it increases the temperature at which refrigeration can be provided.
- This solution temperature needed to derive the desorption process with ammonia - water is in the range between 120°C to 130°C (248°F to 266°F). Temperatures in this range can be obtained using low cost non-tracking solar collectors. The overall efficiency of a solar refrigeration system is the product of the solar collection efficiency and the coefficient of performance of the absorption system.

$$T_0 = 303 \text{ K}$$

$$T_E = 251 \text{ K}$$

$$T_g = 403 \text{ K}$$

Now for reversible system

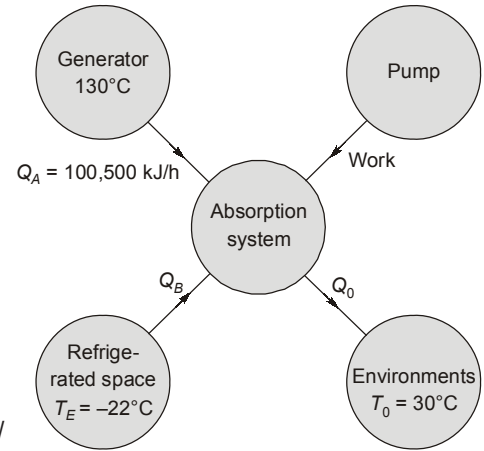
$$\frac{Q_G}{Q_E} = \frac{\left(\frac{T_0 - T_E}{T_E}\right)}{\left(\frac{T_G - T_0}{T_G}\right)} = \frac{(303 - 251) \cdot 251}{(403 - 303) \cdot 403}$$

$$= 0.835$$

$$Q_E = \frac{Q_g}{0.835}$$

⇒

$$= \frac{100,500}{3600 \times 0.835} = 33.433 \text{ kW}$$



Solution : 29

Q_G = Heat added to absorption fluid

W_P = Pump work added to absorption fluid system by pump

Q_E = Heat added to absorption fluid system in the evaporator

Q_0 = Heat reflected from absorption fluid system at condenser, absorber and heat exchangers to environments at temperature to assumed constant

Applying the energy balance equation,

$$Q_0 = Q_E + Q_G + W_P$$

Where pump work is very small and neglected

$$\therefore Q_0 = Q_E + Q_G \quad \dots (i)$$

Clausius's inequality for reversible cycle,

$$\oint \frac{\delta Q}{T} = 0$$

$$\frac{Q_G}{T_G} + \frac{Q_E}{T_E} - \frac{Q_0}{T_0} = 0$$

$$\frac{Q_G}{T_G} + \frac{Q_E}{T_E} - \frac{(Q_E + Q_G)}{T_0} = 0$$

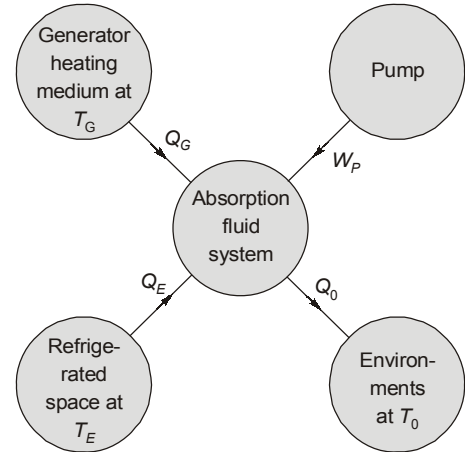
$$\frac{Q_G}{T_G} - \frac{Q_G}{T_0} + \frac{Q_E}{T_E} - \frac{Q_E}{T_0} = 0$$

$$Q_G \left[\frac{1}{T_G} - \frac{1}{T_0} \right] = Q_E \left[\frac{1}{T_0} - \frac{1}{T_E} \right] = Q_E \left[\frac{T_E - T_0}{T_0 T_E} \right]$$

or

$$Q_G \left[\frac{T_G - T_0}{T_G} \right] = Q_E \left[\frac{T_0 - T_E}{T_E} \right]$$

$$\frac{Q_E}{Q_G} = \left(\frac{T_E}{T_0 - T_E} \right) \left(\frac{T_G - T_0}{T_G} \right)$$



$$\therefore \quad COP = \frac{Q_E}{Q_G} = \left(\frac{T_E}{T_0 - T_E} \right) \left(\frac{T_G - T_0}{T_G} \right)$$

Now it is given that $T_G = 383 \text{ K}, T_E = 268 \text{ K}, T_0 = 308 \text{ K}$

$$\Rightarrow \quad COP = \left(\frac{268}{308 - 268} \right) \left(\frac{383 - 308}{383} \right)$$

$$COP = 1.31$$

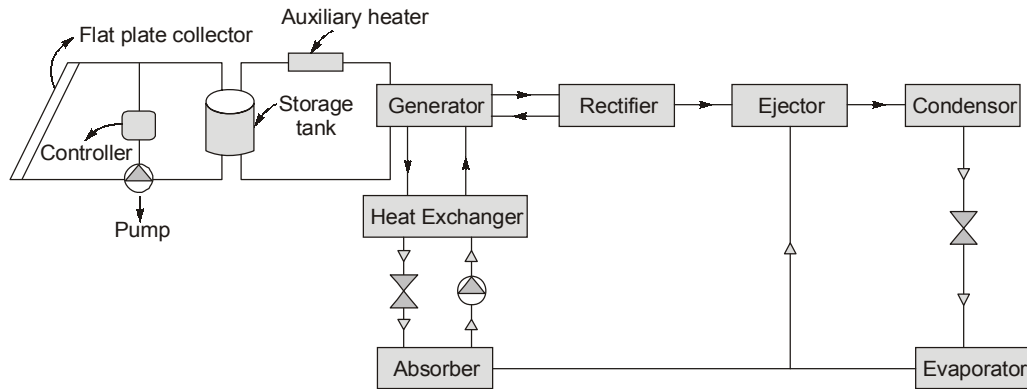
Solution : 30

Vapour Absorption Refrigeration System

Vapour absorption refrigeration system is based on heat energy. It is mostly preferred in those areas where the cost of electricity is very high or large amount of heat energy is available.

In the vapour-absorption system, the function of compressor is accomplished in a three-step process by the use of the absorber, heat exchanger and generator.

VARs system based on solar heating supplies heat to the generator of VARs system. Solar heating system consists solar flat collector, controllers, storage tank, pump and an auxiliary heater which supplies heat to the generator. Schematic diagram of the whole system is as given below:



As shown in above diagram, solar heating system supplies heat to the generator. Normally $\text{NH}_3\text{-H}_2\text{O}$ combination is used as refrigerant and absorber in VAR system.

- ⇒ The heat from the surrounding is absorbed in evaporator by which low pressure ammonia vapour from the evaporator comes in contact in the absorber with the weak solution (the concentration of NH_3 is low in H_2O) coming from the generator, it is readily absorbed, releasing the latent heat of condensation.
- ⇒ Strong solution rich in ammonia, is pumped to the generator where the heat (Q_a) is supplied from auxiliary heater.
- ⇒ In condenser, the ammonia vapour gets condensed, and throttled by the expansion valve, then sent to the evaporator where it absorbs latent heat of vapourisation from the surrounding.
- ⇒ Water vapour going to the condenser along with the ammonia vapour after condensation may get frozen to ice and block the expansion valve. So an analyzer-rectifier combination is used to eliminate water vapour going into the condenser.
- ⇒ The analyzer is a direct contact type heat exchanger consisting of a series of trays mounted above the generator. The strong solution from the absorber flows downward over the trays to cool the outgoing vapours. since the saturation temperature of water is higher than that of ammonia at a given pressure,

it is the water vapour that condenses first. As the vapour passes upward through the analyzer, it is cooled and enriched by ammonia, and the liquid is heated. Thus the vapour going to the condenser is lower in temperature and richer in ammonia, and the heat input to the generator is decreased.

⇒ The final reduction in the percentage of water vapours in the ammonia going to the condenser occurs in the rectifier which is a water-cooled heat exchanger which condense water vapour and returns it to the generator through the drip line.

COP of VARS system:

COP of a VARS system is given by following expression:

$$\text{COP} = \frac{T_E}{T_G} \left[\frac{T_G - T_O}{T_O - T_E} \right]$$

Given:

$$T_E = 5^\circ\text{C} = 278 \text{ K}, T_G = 90^\circ\text{C} = 363 \text{ K} \text{ and } T_O = 40^\circ\text{C} = 313 \text{ K}$$

⇒

$$\text{COP} = \frac{278}{363} \left[\frac{363 - 313}{313 - 278} \right] \Rightarrow \text{COP} = 1.094$$

■ ■ ■ ■

4

Refrigerants & Refrigeration Equipments

LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

10. (c)
11. (b)
12. (a)
13. (c)
14. (a)

■■■■

LEVEL 3 Conventional Questions

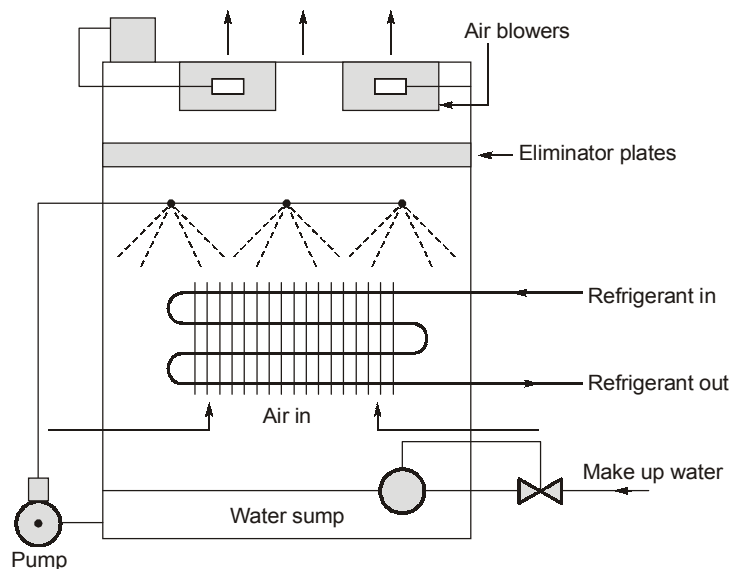
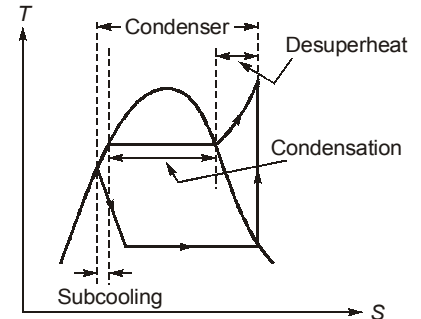
Solution : 15

Functions of condenser in a refrigerating machine are as follows:

- To enable rejection of heat from the refrigerant in a refrigeration cycle.
- Enables desuperheat, condensation and subcooling of refrigerant to get vapour to condense to subcooled liquid stage which can then be fed to expansion device.

Different types of condensers are :

- **Air-cooled condensers**-for small cooling load operation like domestic refrigerators and air conditioners. This can be further classified as:
 - ♦ Natural convection type
 - ♦ Forced convection type
- **Water-cooled condenser:** Water is used to extract heat from the refrigerant. Further classified as:
 - ♦ Double tube type
 - ♦ Shell and tube type
 - ♦ Shell and coil type
- **Evaporative condensers:** Combines the features of both a cooling tower and a water-cooled condenser in a single unit.



Consider and evaporative condenser as shown above. Water is pumped from the sump and sprayed on tubes carrying refrigerant. Water droplets on tube get evaporated by extracting latent heat of vaporization from refrigerant. Water is continuously recirculated and make up water is added as required.

Air is continuously recirculated using blowers and used to cool the refrigerant tubes which are connected to

a plate to enhance area of heat exchange as shown above. The role of air primarily to enhance evaporation of water. Evaporative condensers are used in medium to large capacity systems. These are normally cheaper than water cooled condensers which require a separate cooling tower.

Solution : 16**Types of compressors used in VCR plants:**

Compressor is used to compress the refrigerant to a pressure that is higher than in the evaporator (upto 8–10 times) so that condensation process can take place at a temperature that is compatible with readily available cold source, typically outside air.

Different types of compressors which are used in refrigeration are as given below:

(a) Reciprocating compressors: These are divided into hermetic, semi-hermetic and open type and are used above all for applications with very high cooling capacity requirements. In general these are noisy compressors which creates vibrations that can be felt around the entire circuit.

Reciprocating compressors are called hermetic compressors when the casing is welded and sealed and the cylinder heads can not be accessed for inspection or maintenance.

Semi-hermetic means the compressor itself and the motor are housed in the same casing, which is designed to be opened for inspection and servicing. These are made in such a way as to avoid air or dust from entering the mechanism.

Reciprocating compressors are defined as open when one end of the crankshaft protrudes outside of casing that houses the pistons and mechanisms inside compressor and can be opened.

(b) Rotary compressors: It includes rotary vane, scroll, screw and centrifugal compressors. These are hermetic compressors used in many applications and different operating ranges.

Rotary vane compressors consist of a cylindrical casing, two openings - one suction and one discharge and a rotor positioned eccentrically with respect to the casing. In this compression occurs by refrigerant flowing into the chamber where, due to eccentric rotation, there is a reduction in the desired volume.

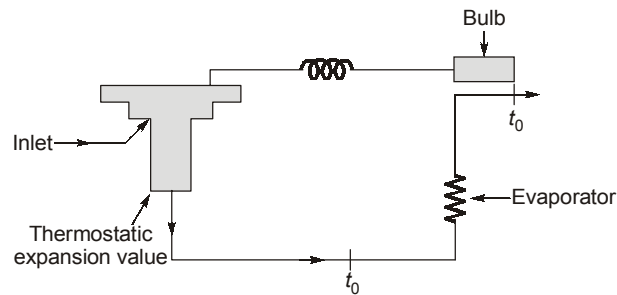
Scroll compressors use two scrolls, one fixed and other moving and coupled to the motor. The scrolls are interleaved so as to be in contact with one another in various places and thus form a series of gradually thinner pockets air towards the centre. The fluid is compressed by rotation of the orbiting scroll around the fixed scroll until being discharged in the centre. These are widely used in home and commercial air conditioning systems.

Screw compressors are based on a mechanism made up of two intermeshing screws, called rotors. As the rotors revolve, the fluid is drawn in through the inlet port, filling the volume between two lobes. When the space between the lobes are full of fluid, suction ceases, the fluid between the lobes then forced due to rotation into an increasingly small space, causing compression.

Centrifugal compressor feature a cast iron casing, a steel shaft and a cast aluminium alloy impeller. The fluid is drawn in by the impeller near its axis and due to centrifugal force is pushed to the edge of the compressor casing. The fluid leaves the impeller with significant kinetic energy, which is then converted into pressure energy in the diffuser. Its use is limited to high cooling capacity and low compression ratios (usually large chillers running on R134a). Cooling capacity can be varied by using fins that change the angle at which fluid enters the impeller.

Solution : 17

Consider the thermostatic expansion valve as shown in figure:



$$\text{Evaporator inlet temperature} = t_0$$

$$\text{Evaporator outlet temperature} = t'_0$$

$$\text{Degree of superheat} = t'_0 - t_0$$

This degree of superheat result into a pressure difference ($P'_0 - P_0$) which actuates the valve, where P'_0 and P_0 are respective saturation pressure.

Given: **Case I:** $t_0 = 0^\circ\text{C}$, $t'_0 = 10^\circ\text{C}$

$$P_0 = 2.928 \text{ bar } P'_0 = 4.146 \text{ bar}$$

We have,

$$P'_0 - P_0 = 1.218 \text{ bar}$$

Case II:

$$t_0 = -30^\circ\text{C}, P_0 = 0.8438 \text{ bar}$$

Since valve follow up spring setting remains the same, ($P'_0 - P_0$) will remain the same.

$$P'_0 - P_0 = 1.218 \text{ bar}$$

⇒

$$P'_0 = 0.8438 + 1.218 = 2.0618 \text{ bar}$$

Hence approximate refrigerant outlet temperature corresponding to this saturation pressure will be -10°C as per the given table.

Hence, degree of superheat required to maintain evaporator temperature of

$$-30^\circ\text{C} = -10^\circ\text{C} - (-30^\circ\text{C}) = 20^\circ\text{C}$$

Solution : 18

When the refrigerant enters the space to be cooled, it absorbs heat from the surroundings. This process is usually accompanied by a phase change process which maximizes the cooling effect.

Good qualities of refrigerant:

- (i) It should be non-poisonous
- (ii) It should be non-explosive
- (iii) It should be non-corrosive
- (iv) It must be non-inflammable
- (v) Leakage should be non-inflammable
- (vi) It should have a well balanced enthalpy of evaporation per unit mass.
- (vii) A minimum difference between the vaporising pressure and condensing pressure is desirable.

The refrigerant which does not destroy ozone layer is R-134a Tetra fluoro ethane ($\text{C}_2\text{H}_2\text{F}_4$).

Refrigerant which do not contain chlorine does not harm the ozone layer.

Solution : 19**1. The common items of control involved in air-conditioning are given below:**

- (i) Expansion devices
 - (a) Automatic expansion valve
 - (b) Thermostatic expansion valve
- (ii) Cut outs for refrigeration units
 - (a) High pressure cut-out
 - (b) Low pressure cut-out
- (iii) Back pressure valves
- (iv) Liquid level regulating devices
 - (a) High-side float valve
 - (b) Low-side float valve
- (v) Flow-regulating devices
 - (a) Solenoid valves
 - (b) Check valves
 - (c) Water and Brine valves
- (vi) Thermostats
 - (a) Evaporator thermostat
 - (b) Room thermostat

2. Adiabatic saturation temperature is the temperature at which the air becomes adiabatically saturated by the evaporation of water into air.

If the bulb of wet bulb thermometer is covered by a wick thoroughly wetted by water, the temperature which is measured by the thermometer is the **wet bulb temperature**.

Both these temperature are equivalent when the air is fully saturated.

3. Metabolism: When food is digested and converted to energy, heat is released. This process is known as metabolism.

For an average healthy person the metabolic rate is of the order of 50W – 75W. This value may be increased during hard work.

We are interested in the metabolic rate of the occupants of a building because human comfort condition cannot be achieved until the heat generated by metabolism is accounted and taken away.

Solution : 20

A refrigerant is said to be ideal if it has all of following properties:

- Low boiling point
- High critical temperature
- high latent heat of vaporization
- Low specific heat of liquid
- Low specific volume of vapour
- Non-corrosive to metal
- Non-flammable and non-explosive
- Non-toxic
- Low cost
- Mixes well with oil
- Low viscosity

- Freezing temperature should be low
- Low pressure ratios
- Easy to liquefy at moderate pressure and temperature
- Ease of locating leaks by odour or suitable indicator.

Solution : 21

Refrigerant number	Chemical name	Chemical formula
<i>R</i> - 11	Trichloromonofluoro methane	CCl ₃ F
<i>R</i> - 12	Dichlorodifluoro methane	CCl ₂ F ₂
<i>R</i> - 13	Monochlorotrifluoro methane	CClF ₃
<i>R</i> - 22	Monochlorodifluoro methane	CHClF ₂
<i>R</i> - 717	Ammonia	NH ₃



LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

11. (a)
12. (a)
13. (a)
14. (c)
15. (c)
16. (c)
17. (d)
18. (d)
19. (a)
20. (b)
21. (c)
22. (b)

■ ■ ■ ■

LEVEL 3 Conventional Questions
Solution : 23

For process 1-2 which is sensible heating process

$$\text{Heat absorbed} = 50 \text{ kW}$$

$$\text{i.e.} \quad = \dot{m}(h_2 - h_1)$$

$$\text{At state 1-2} \quad w_1 = 0.0035 \text{ (From psychrometric chart)}$$

$$\text{Specific volume} \quad = 0.792 \text{ m}^3/\text{kg},$$

$$h_1 = 14 \text{ kJ/kg}$$

$$\text{Volume flow rate} = 100 \text{ m}^3/\text{min} = 1.667 \text{ m}^3/\text{s}$$

$$\text{Mass flow rate} = \frac{1.667}{0.792} = 2.104 \text{ kg/s}$$

Process 1-2 : Heating in chamber

$$\text{Now heat transferred,} \quad Q = m(h_2 - h_1) = 2.104 (h_2 - 14)$$

$$50 = 2.104 (h_2 - 14)$$

$$h_2 = 37.76 \text{ kJ/kg}$$

Now at state 2, from psychrometric chart

$$w_2 = 0.0035;$$

and

$$h_2 = 37.76 \text{ kJ/kg}$$

After humidification by steam injection

Process 2-3 : Heating and humidification with steam injection process

$$w_3 = w_2 + \frac{m_s}{m_a} = 0.0035 + \frac{35}{3600 \times 2.1} = 0.0081$$

$$\text{Enthalpy of leaving air,} \quad h_3 = h_2 + \frac{m_s}{m_a} (h_v)$$

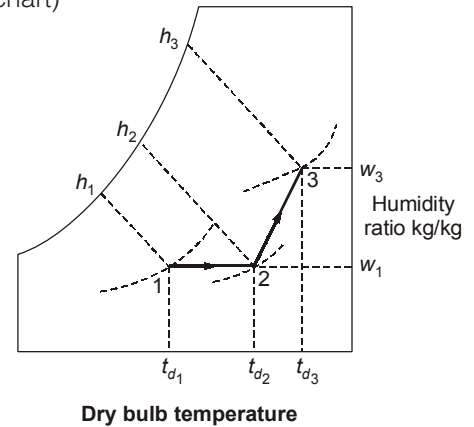
$$h_v = 2691.3 \text{ kJ/kg at } 110^\circ\text{C}$$

$$h_3 = 37.76 + \frac{35}{3600 \times 2.104} \times 2691.3 = 50.19 \text{ kJ/kg}$$

At state (3)

$$t_{d3} = 31^\circ\text{C}$$

$$t_{w3} = 18.3^\circ\text{C}$$

**Solution : 24**

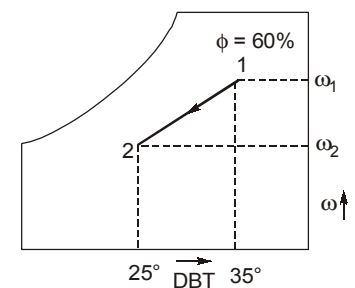
Atmospheric condition: Relative humidity, $\phi = 60\%$,

Dry bulb temperature, $t_1 = 35^\circ\text{C}$,

Atmospheric pressure, $P = 100 \text{ kPa}$ and $\Delta\omega = 0.005 \text{ Kg/kg dry air}$

$$\text{As} \quad \phi_1 = 60\% \text{ or } 0.6 = \frac{P_{v1}}{P_{vs1}} \Rightarrow P_{v1} = 0.6 \times 5.63$$

$$\Rightarrow \quad P_{v1} = 3.378 \text{ kPa}$$



[From table $P_{vs1} = 5.63 \text{ kPa}$]

So,
$$\omega_1 = \frac{0.622 P_{v1}}{P - P_{v1}} = \frac{0.622 \times 3.378}{100 - 3.378}$$

$\Rightarrow \omega_1 = 0.021746 \text{ kg/kg d.a.}$

As given
$$\Delta\omega = \omega_1 - \omega_2$$

$\Rightarrow 0.005 = 0.021746 - \omega_2$

$\Rightarrow \omega_2 = 0.016746 \text{ kg/kg dry air}$

As
$$\omega_2 = \frac{0.622 P_{v2}}{P - P_{v2}}$$

$\Rightarrow 0.016746 = \frac{0.622 P_{v2}}{100 - P_{v2}} \Rightarrow P_{v2} = 2.6217 \text{ kPa}$

(i) From table, P_{vs2} corresponding to 25°C is 3.17 kPa

Relative humidity at final stage,

$$\phi_2 = \text{relative humidity} = \frac{P_{v2}}{P_{vs2}}$$

$\Rightarrow \phi_2 = \frac{2.6217}{3.17} \Rightarrow \phi_2 \approx 82.7\%$

\Rightarrow Relative humidity = 82.7%

(ii) As the dew point temperature is the saturation temperature corresponding to partial pressure of vapour, so let saturation temperature corresponding P_{v2} be t_{DPT} then from table it is clear that this temperature will lie between 20°C and 25°C ; so interpolating the value from the table;

$$\frac{t_{ADP} - 20}{25 - 20} = \frac{2.6217 - 2.34}{3.17 - 2.34}$$

$\Rightarrow t_{ADP} = 21.7^\circ\text{C}$

Solution : 25

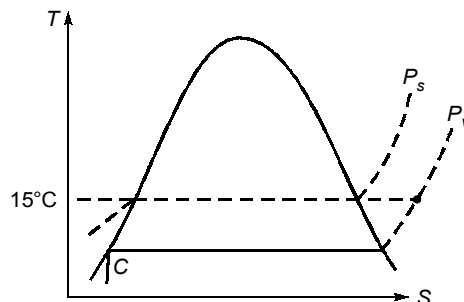
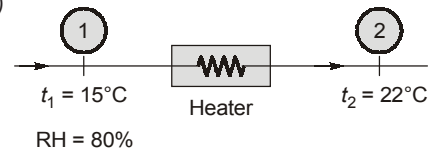
Consider heating of air as shown in figure between states (1) and (2)

Saturation pressure at 15°C DBT = $P_s = 0.017 \text{ bar}$

\therefore Vapour pressure @ 80% relative humidity

$$P_v = P_s \times 0.8 = P_s \times \phi$$

$$= 0.8 \times 0.017 = 0.0136 \text{ bar}$$



$$\therefore \text{Specific humidity of air} = \omega_1 = \omega_2 = 0.622 \cdot \frac{P_v}{P - P_v} \text{ kg/kg d.a.}$$

$$= 0.622 \cdot \frac{0.0136}{1.013 - 0.0136} = 8.464 \times 10^{-3} \text{ kg/kg d.a.}$$

$$\begin{aligned} \text{Specific heat of air, } C_{p,m} &= C_{p,a} + \omega C_{p,v} = 1.005 + 1.88 \omega \\ &= 1.005 + 1.88 \times 8.464 \times 10^{-3} = 1.0209 \text{ kJ/kg d.a. K} \end{aligned}$$

$$\text{Volume flow rate of air, cmm} = 100 \text{ m}^3/\text{min}$$

$$\text{Mass flow rate of air} = \frac{\text{cmm}}{60} \cdot \rho = \frac{100}{60} \times 1.2 \text{ kg/s} = 2 \text{ kg/s}$$

$$\begin{aligned} \therefore \text{Heat added in sensible heating} &= \dot{m}_a C_{p,m} (t_2 - t_1) = 2 \times 1.0209 \times (22 - 15) \text{ K} \\ &= 14.2926 \text{ kW} \approx 14.29 \text{ kW, Heat added/min} = 857.556 \text{ kJ/min} \end{aligned}$$

Solution : 26

Given

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

$$WBT = 15^\circ\text{C}$$

$$P = 1 \text{ bar}$$

$$\begin{aligned} P_v &= (P_{vsat})_{t_{wbt}} - \frac{(P - P_{vsat})(dbt - wbt) \times 1.8}{2854 - 1.325(1.8t_{dbt} + 32)} \\ &= 0.017307 - \frac{(1 - 0.017307)(25 - 15) \times 1.8}{2854 - 1.325(1.8 \times 25 + 32)} = 0.017307 - \frac{17.43}{2751.975} \\ &= 0.0106 \text{ bar} \end{aligned}$$

Psychrometric properties

$$\begin{aligned} 1. \text{ Specific humidity: } w &= 0.622 \frac{P_v}{P - P_v} = 0.622 \frac{0.01097}{1 - 0.01097} \\ &= 0.0066638 \text{ kg vapour/kg of dry air} \end{aligned}$$

2. Relative humidity (ϕ)

$$\phi = \frac{P_v}{P_{vs}} \times 100 = \frac{0.0106}{0.03166} \times 100 = 33.48\%$$

3. Degree of saturation (μ)

$$\mu = 0.3348 \times \left[\frac{1 - 0.03166}{1 - 0.0106} \right] = 0.32767 \Rightarrow 32.77\%$$

4. Enthalpy of moist air (h)

$$h = c_{pa}t + w[2500 + 1.88t]$$

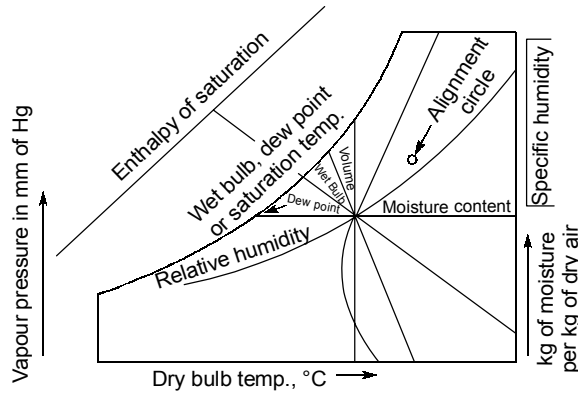
where

$$c_{pa} = 1.005 \text{ kJ/kgK}$$

$$t = DBT = 25^\circ\text{C}$$

$$\begin{aligned} h &= (1.005 \times 25) + 0.0066638 [2500 + (1.88 \times 25)] \\ &= 25.125 + 0.0066638(2547) \\ &= 42.098 \text{ kJ/kg of dry air} \end{aligned}$$

Solution : 27



Solution : 28

Air
30°C
105 kPa
8 m³

$$P_a = P_t - P_s$$

$$= 105 - 4.2469 = 100.7531 \text{ kPa}$$

$$m_a = \frac{P_a V}{R_a T} = \frac{100.7531 \times 8}{0.287 \times 303} = 9.27 \text{ kg}$$

$$P_v = P_g = P_{s@30^\circ\text{C}} = 4.2469 \text{ kPa}$$

$$\omega = \frac{0.622 P_v}{P_t - P_v} = \frac{0.622 \times 4.2469}{105 - 4.2469}$$

$$= 0.0262 \text{ kg H}_2\text{O/kg dry air}$$

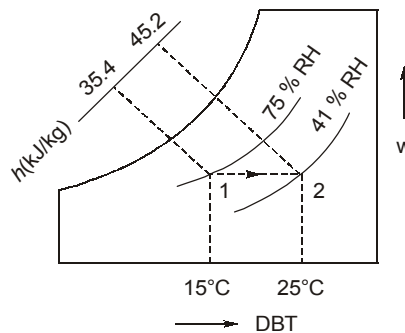
The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + wh_v \approx C_p T + wh_{g@30^\circ\text{C}}$$

$$h = 1.005 \times 30 + 0.0262 \times 2555.6$$

$$= 97.1 \text{ kJ/kg of dry air}$$

Solution : 29



- Locate point 1 on the psychrometric chart at the intersection of 15°C DBT and 75% RH lines.
- Through point 1, draw a horizontal line to cut 25°C DBT line and get point 2.
- Read the following values from the psychrometric chart

$$h_1 = 35.4 \text{ kJ/kg}$$

$$h_2 = 45.2 \text{ kJ/kg}$$

$$v_{s1} = 0.8267 \text{ m}^3/\text{kg}$$

- (i) RH of heated air = 41 % (read from chart)
- (ii) WBT of heated air = 16.1°C (read from chart)
- (iii) Mass of air circulated per minute,

$$\dot{m}_a = \frac{200}{0.8267} = 241.9 \text{ kg}$$

$$\begin{aligned} \text{Heat added to air/min} &= \dot{m}_a(h_2 - h_1) = 241.9(45.2 - 35.4) \\ &= 2370.6 \text{ kJ} \end{aligned}$$



6

Air Conditioning Systems & Comfort Conditions

LEVEL 1 Objective Questions

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

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LEVEL 2 Objective Questions

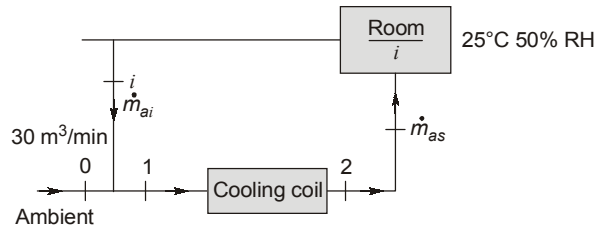
11. (b)
12. (a)
13. (d)
14. (d)
15. (c)
16. (d)
17. (a)
18. (c)
19. (c)
20. (d)
21. (b)
22. (d)
23. (c)
24. (b)

■■■■

LEVEL 3 Conventional Questions

Solution : 25

Schematic of air conditioning system can be shown in this figure



Various processes in the system can be represented on psychrometric chart as below:

Given:

Saturation pressure for 25°C, $P_{s@25^\circ\text{C}} = 3.1693 \text{ kPa}$

RH for room air = 50%

\therefore Vapour pressure for room air = $P_s \times 0.5$

$\Rightarrow P_v = 1.58465 \text{ kPa}$

Humidity ratio or specific humidity for room air,

$$\begin{aligned}\omega_i &= 0.622 \frac{P_v}{P - P_v} \\ &= 9.88 \times 10^{-3} \text{ kg/kg d.a.}\end{aligned}$$

Similarly for outdoor air:

$$P_{\text{sat}@43^\circ\text{C}} = 8.6495 \text{ kPa}$$

$$\phi = \text{RH} = 30\%$$

$$\therefore P_v = P_s \times \phi = 2.59485$$

$$\therefore \text{Humidity ratio for outdoor air, } \omega_0 = 0.622 \cdot \frac{2.59485}{101.325 - 2.59485} = 16.35 \times 10^{-3} \text{ kg/kg d.a.}$$

Density of outdoor air and recirculated air can be taken as 1.2 kg d.a./m^3 .

\therefore Mass flow rate of dry air in outdoor air taken in

$$\dot{m}_{a_o} = \dot{Q}_f \times \rho = \frac{30}{60} \times 1.2 = 0.6 \text{ kg/s}$$

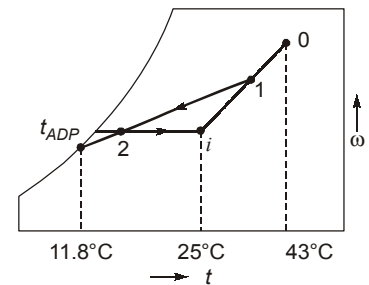
Assume supply air rate as

Ventilation load:

$$\text{Outside air sensible heat, (OASH)} = \dot{m}_{s_o} \cdot C_{p,m}(t_0 - t_i) = 0.6 \times 1.0216 \times (43 - 25) = 11.03 \text{ kW}$$

$$\text{Outside air latent heat, (OALH)} = \dot{m}_a \cdot 2500(\omega_o - \omega_i) = 0.6 \times 2500 \times (16.35 - 9.88) \times 10^{-3}$$

$$\Rightarrow \text{OALH} = 9.705 \text{ kW}$$



Effective room sensible heat, (ERSH) = RSH + BPF × OASH = 100 + 0.15 × 11.03 = 101.66 kW

Effective room latent heat = RLH + BPF × OALH = 15 + 0.15 × 9.705 = 16.46 kW

We know,

$$ERSH = \dot{m}_{as} \cdot C_{p,m}(t_i - t_{ADP})$$

$$\Rightarrow \dot{m}_{as} = \frac{101.66}{1.0216 \times (25 - 11.8)} = 7.54 \text{ kg/s}$$

∴ Volume flow rate of supply air,

$$\frac{\dot{m}_{as}}{\rho} = \frac{7.54}{1.2} = 6.283 \text{ m}^3/\text{s} = 377 \text{ m}^3/\text{min}$$

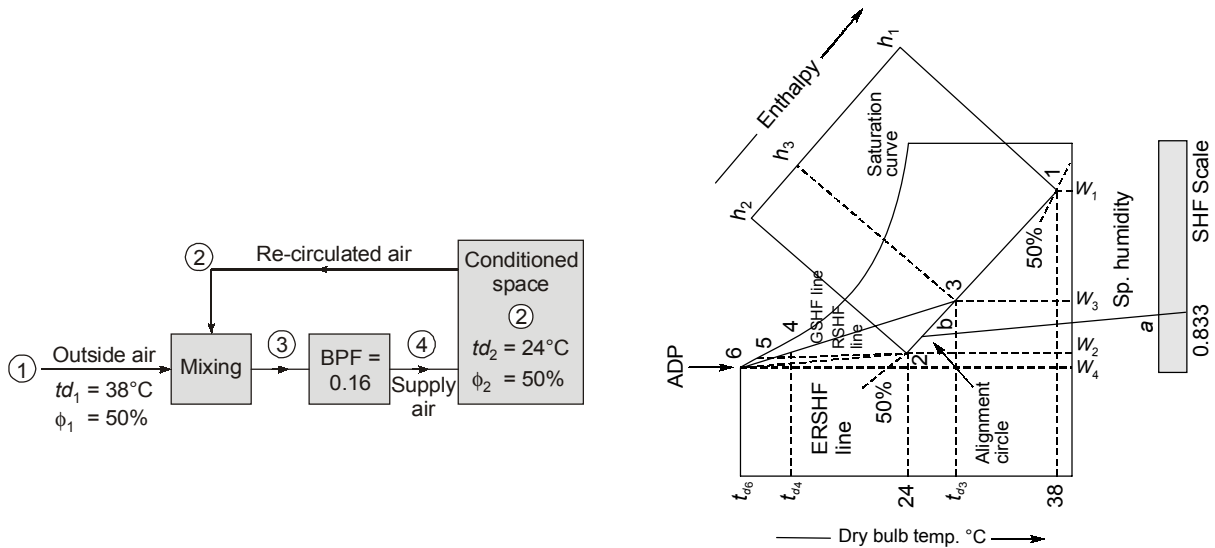
∴ Temperature at the inlet of cooling coil,

$$\dot{m}_{as} t_1 = \dot{m}_{a0} t_0 + (\dot{m}_{as} - \dot{m}_{a0}) \cdot t_i = 26.43^\circ\text{C}$$

$$\text{By pass factor} = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = 0.15$$

$$\Rightarrow t_2 = 11.8 + 0.15 (26.43 - 11.8) = 13.995 \approx 14^\circ\text{C}$$

Solution : 26



Given:

$$\begin{aligned} RSH &= 24 \text{ kW}, \\ RLH &= 6 \text{ kW} \\ t_{d1} &= 38^\circ\text{C} \\ \phi_1 &= 50\%; & t_{d2} &= 24^\circ\text{C} \\ \phi_2 &= 50\%; & BPF &= 0.16 \end{aligned}$$

First of all, mark the initial condition of air at 38°C DBT and 50% RH as point 1. Now mark the room condition at 24°C DBT/50% RH as point 2.

$$\text{We know, } RSHF = \frac{RSH}{RSH + RLH} = \frac{24}{30} = 0.8$$

Now mark this calculated value of $RSHF = 0.8$ on the sensible heat factor scale as point a and join with point b which is the alignment circle (i.e. 26°C DBT and $50\% RH$) from point 2, draw a line 2 - 5 parallel to this line ab . The line 2 - 5 is called $RSHF$ line. Since 20% of fresh air or outside air is mixed with 80% of supply air, therefore the condition of air entering the cooling coil after mixing process is marked on the line 1 - 2 by point 3, such that

$$\text{Length } 2 - 3 = \text{Length } 1 - 2 \times 0.2$$

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

$$\frac{\text{Length } 4 - 6}{\text{Length } 3 - 6} = BPF = 0.6$$

1. Supply air flow rate

Let

$V =$ supply air flow rate in m^3/min

$t_{d4} =$ DBT of air leaving the cooling coil

$t_{d6} =$ ADP (Apparatus Dew Point)

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

$$t_{d3} = 28.5^\circ\text{C}$$

$$BPF = 0.16 = \frac{t_{d4} - ADP}{t_{d3} - ADP} = \frac{t_{d4} - t_{d6}}{28.5 - t_{d6}}$$

By trial and error, we find that $t_{d4} = 13.72^\circ\text{C}$, $t_{d6} = 11^\circ\text{C}$

$$RSH = 0.02044 V (t_{d2} - t_{d4})$$

$$24 = 0.02044 \times V (24 - 13.72)$$

$$V = 114.2187 \text{ m}^3/\text{min}$$

2. Outside air sensible heat

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

$$V_0 = 0.2V = 22.84 \text{ m}^3/\text{min}$$

$$\begin{aligned} \text{Output side air sensible heat, } OASH &= 0.02044 V_0 (t_{d1} - t_{d2}) = 0.025044 \times 22.84 (38 - 24) \\ &= 6.537 \text{ kW} \end{aligned}$$

3. From the psychrometric chart, we find that specific humidity of outside air at point 1 and 2.

$$W_1 = 0.021 \text{ kg/kg d.a}$$

$$W_2 = 0.009 \text{ kg/kg d.a}$$

$$\text{Outside air latent heat, } OALH = 50 V_0 (W_1 - W_2) = 50 \times 22.84 \times 0.012 = 13.704 \text{ kW}$$

4. Grand total heat

$$\text{Total sensible heat, } TSH = RSH + OASH = 24 + 6.537 = 30.537 \text{ kW}$$

5. Effective room sensible heat factor

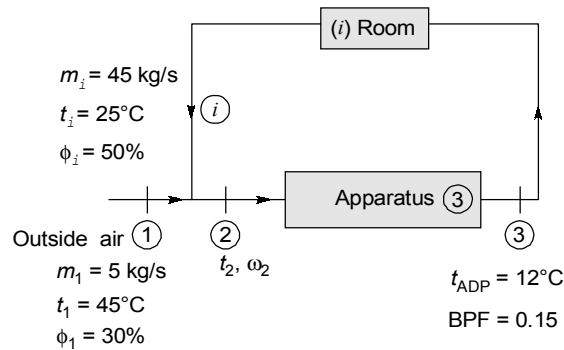
$$ERSH = RSH + OASH \times BPF = 24 + 6.537 \times 0.16 = 25.046 \text{ kW}$$

$$ERLH = RLH + OALH \times BPF = 6 + 13.704 \times 0.16 = 8.2 \text{ kW}$$

$$ERSHF = \frac{25.046}{25.046 + 8.2} = 0.753$$

Solution : 27

Given that:



$t^\circ\text{C}$	$P_{vs}(\text{bar})$
12°C	0.014016
25°C	0.03166
45°C	0.09584

From mass conservation:

$$m_1 + m_i = m_2 = 45 + 5 = 50 \text{ kg/s}$$

$$m_i \times t_i + m_1 \times t_1 = m_2 \times t_2 \Rightarrow t_2 = \frac{45 \times 25 + 5 \times 45}{50}$$

$$t_2 = 27^\circ\text{C}$$

As

$$\phi_i = \frac{P_{vi}}{P_{vsi}} \Rightarrow P_{vi} = 0.50 \times 0.03166 = 0.01583 \text{ bar}$$

$$\omega_i = \frac{0.622 P_{vi}}{P - P_{vi}} \Rightarrow \omega_i = \frac{0.622 \times 0.01583}{1.01325 - 0.01583} = 0.0098717 \text{ kg/kg dry air}$$

Also,

$$\phi_1 = \frac{P_{v1}}{P_{vs1}} = 0.30 = \frac{P_{v1}}{0.09584}, P_{v1} = 0.028752 \text{ bar}$$

$$\omega_{\text{ADP}} = \frac{0.622 \times 0.014016}{1.01325 - 0.014016} = 8.7246 \times 10^{-3} \text{ kg / kg dry air}$$

$$\omega_1 = \frac{0.622 P_{v1}}{P - P_{v1}} = \frac{0.622 \times 0.028752}{1.01325 - 0.028752} \Rightarrow \omega_1 = 0.0181653 \text{ kg/kg dry air}$$

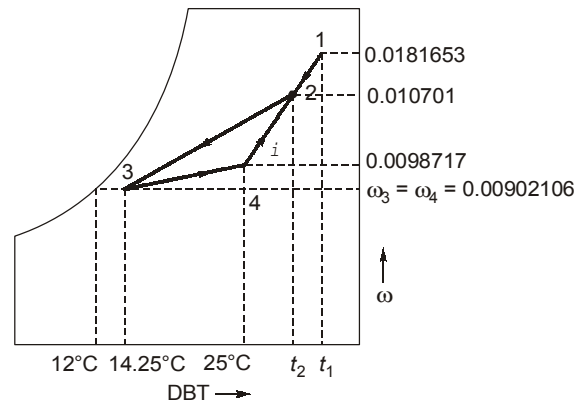
$$\omega_2 = \frac{5 \times 0.0181653 + 45 \times 0.0098717}{50} = 0.010701 \text{ kg/kg dry air}$$

⇒

$$\text{BPF} = 0.15 = \frac{\omega_3 - 0.0087246}{(0.010701 - 0.0087246)} \Rightarrow \omega_3 = 9.02106 \times 10^{-3} \text{ kg/kg dry air}$$

At point 3,

$$\text{BPF} = 0.15 = \frac{t_3 - 12}{27 - 12} \Rightarrow t_3 = 14.25^\circ\text{C}$$



Calculation of enthalpy:

$$h_1 = 1.005 t_1 + \omega_1 (2500 + 1.88 t_1) = 92.175 \text{ kJ/kg}$$

$$h_i = 1.005 t_i + \omega_i (2500 + 1.88 t_i) = 50.268 \text{ kJ/kg}$$

$$\begin{aligned} h_2 &= 1.005 t_2 + \omega_2 (2500 + 1.88 t_2) \\ &= 1.005 \times 27 + 0.010701 (2500 + 1.88 \times 27) = 54.43 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} h_3 &= 1.005 t_3 + \omega_3 (2500 + 1.88 t_3) \\ &= 1.005 \times 14.25 + 0.00902106 (2500 + 1.88 \times 14.25) \end{aligned}$$

$$h_3 = 37.115 \text{ kJ/kg}$$

$$h_4 = 1.005 t_4 + \omega_4 (2500 + 1.88 t_4)$$

As $t_4 = t_i = 25^\circ\text{C}$ and $\omega_4 = \omega_3 = 0.00902106$

$\Rightarrow h_4 = 48.101 \text{ kJ/kg}$

So, Room sensible heat, $\text{RSH} = \dot{m} (h_4 - h_3) = 50(48.101 - 37.115)$

$\Rightarrow \text{RSH} = 549.332 \text{ kW}$

Room latent heat, $\text{RLH} = \dot{m} (h_i - h_4) = 50(50.268 - 48.101)$

$$\text{RLH} = 108.35 \text{ kW}$$

Condensate rate = $50 \times (0.010701 - 0.00902106)$

$$= 0.084 \text{ kg of vapour/sec}$$

Solution : 28

Reading from the psychrometric chart, we find that

$$w_{\text{ambient}} = 0.01039 \text{ kg/kg of dry air at } T_{\text{ambient}} = 32^\circ\text{C}, \phi_{\text{ambient}} = 35\%$$

$$w_{\text{room}} = 0.01024 \text{ kg/kg of dry air at } T_{\text{room}} = 24^\circ\text{C}, \phi_{\text{room}} = 55\%$$

Here, infiltration of ambient air will cause the air in the storage room to be changed 1.8 times every hour.

$$\rho_{\text{ambient}} = \frac{p_0}{RT_0} = \frac{101.325}{0.287 \times (32 + 273)} = 1.158 \text{ kg/m}^3$$

mass flow rate of air entering the cold storage:

$$\dot{m}_{\text{air}} = (\rho_{\text{ambient}})(\text{Volume of space}) \text{ACH}$$

$$\dot{m}_{\text{air}} = 1.158 \times 20 \times 13 \times 3 \times 1.8 = 1625.83 \text{ kg/h}$$

$$\dot{m}_{\text{air}} = \frac{1625.83}{3600} = 0.4516 \text{ kg/s}$$

Sensible heat load of infiltration

$$\begin{aligned} Q_s &= \dot{m} c_p (T_{\text{ambient}} - T_{\text{room}}) \\ &= 0.4516 \times 1.005(32 - 24) = 3.63 \text{ kW} \end{aligned}$$

Latent heat load of infiltration,

$$\begin{aligned} Q_L &= \dot{m} (W_{\text{ambient}} - W_{\text{room}}) h_{fg} \\ &= 0.4516(0.01039 - 0.01024) \times 2444.1 = 0.166 \end{aligned}$$

Total heat load of infiltration,

$$Q_{\text{Total}} = 3.63 + 0.166 = 3.80 \text{ kW}$$

