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# CLASS TEST 2019-2020

## MECHANICAL ENGINEERING

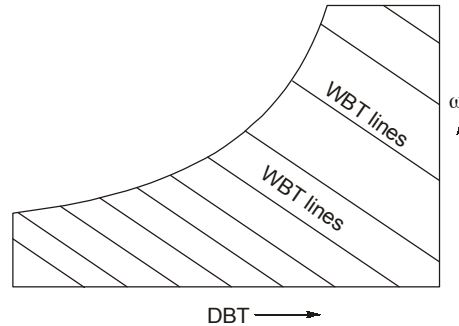
Date of Test : 05/08/2019

### ANSWER KEY > Refregeration & Air Conditioning

- |        |         |         |         |         |
|--------|---------|---------|---------|---------|
| 1. (d) | 7. (a)  | 13. (d) | 19. (a) | 25. (a) |
| 2. (d) | 8. (b)  | 14. (a) | 20. (b) | 26. (b) |
| 3. (d) | 9. (c)  | 15. (b) | 21. (b) | 27. (d) |
| 4. (b) | 10. (d) | 16. (c) | 22. (b) | 28. (a) |
| 5. (b) | 11. (b) | 17. (d) | 23. (b) | 29. (d) |
| 6. (b) | 12. (c) | 18. (b) | 24. (a) | 30. (a) |

## DETAILED EXPLANATIONS

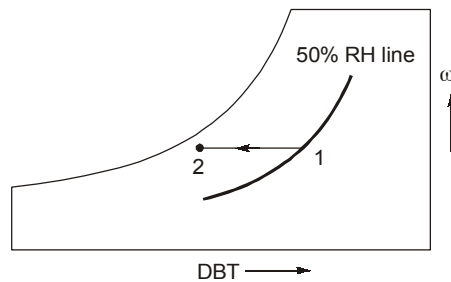
1. (d)



2. (d)

In thermodynamics, a critical point (or critical state) is the end point of a phase equilibrium curve. In the vicinity of the critical point, the physical properties of the liquid and the vapour changes dramatically with both phases becoming ever more similar.

3. (d)



for the process 1-2 as shown above:

- (i) DBT decreases
- (ii) WBT decreases
- (iii) Relative humidity (RH) increase
- (iv) humidity ratio remains constant

Therefore, best option is (d).

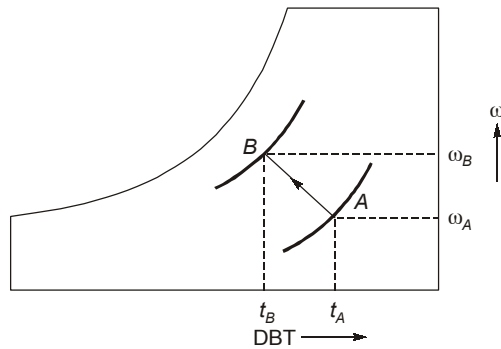
4. (b)

$$(\text{COP})_{\text{HP}} = (\text{COP})_{\text{Ref}} + 1 = 4 + 1 = 5$$

$$\eta_{\text{HE}} = \frac{1}{(\text{COP})_{\text{HP}}} = \frac{1}{5} = 0.2 \text{ or } 20\%$$

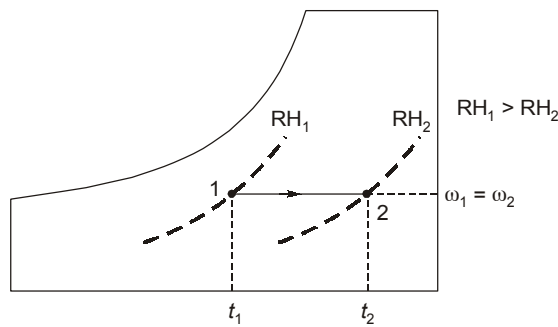
**Note:** It is advisable to try other method also.

5. (b)



We can see that  $t_A > t_B$  and  $\omega_B > \omega_A$ . Therefore, correct answer is (b).

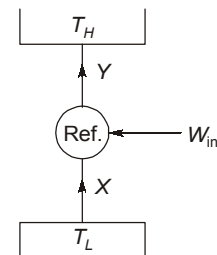
6. (b)



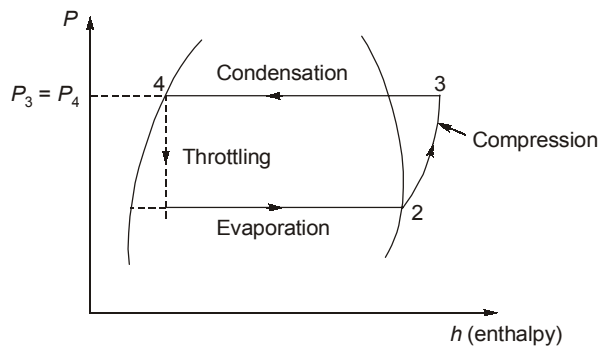
7. (a)

$$\text{COP} = \frac{\text{Heat extracted}}{\text{Work input}}$$

$$= \frac{X}{Y - X}$$



8. (b)



11. (b)

At very low evaporator temperature, the COP becomes very low and also the size of the compressor becomes large (due to small volumetric refrigeration effect).

12. (c)

$$\text{Mass flow rate of MAS 1} = \frac{85}{0.85} = 100 \text{ kg of dry air/min}$$

$$\text{Mass flow rate of MAS 2} = \frac{90}{0.9} = 100 \text{ kg of dry air/min}$$

$$\begin{aligned} \text{Enthalpy of mixed stream, } h &= \frac{m_1 h_1 + m_2 h_2}{m_1 + m_2} = \frac{100 \times 52 + 100 \times 90}{100 + 100} \\ &= 71 \text{ kJ/kg of dry air} \end{aligned}$$

13. (d)

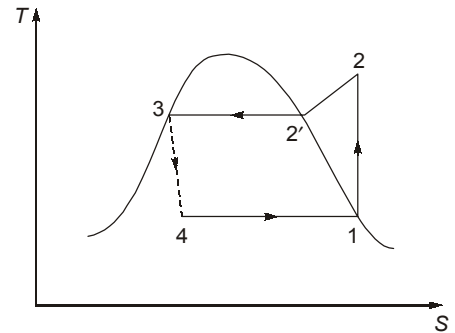
Given:  $h_1 = 407.1 \text{ kJ/kg}$ ;  $h_3 = h_4 = 271.62 \text{ kJ/kg}$ ;  $S_1 = 1.72 \text{ kJ/kgK}$ ;  $S_{2'} = 1.7072 \text{ kJ/kgK}$ ;  $h_{2'} = 423.4 \text{ kJ/kg}$

For the isentropic process  $1 \rightarrow 2$ :

$$\begin{aligned} S_1 &= S_2 = S_{2'} + C_p \frac{T_2}{T_{2'}} \\ 1.72 &= 1.7072 + 1.3 \ln \frac{T_2}{(273 + 50)} \\ T_2 &= (273 + 50) \exp \left[ \frac{1.72 - 1.7072}{1.3} \right] \\ T_2 &= 326.2 \text{ K} \end{aligned}$$

Enthalpy at discharge,  $h_2 = h_{2'} + c_p(T_2 - T_{2'})$

$$\begin{aligned} h_2 &= 423.4 + 1.3(326.2 - 323) \\ &= 427.56 \text{ kJ/kg} \end{aligned}$$



14. (a)

$$\text{Relative humidity, } \phi = \frac{P_v}{P_s}$$

$$P_v = 0.048 \times 0.65 = 0.0312 \text{ bar}$$

$$\begin{aligned} \text{Degree of saturation, } \mu &= \frac{\omega}{\omega_s} = \frac{P_v(P - P_s)}{P_s(P - P_v)} \\ &= \frac{0.0312(1.0132 - 0.048)}{0.048(1.0132 - 0.0312)} \\ &= \frac{0.03011424}{0.047136} = 0.63888 \end{aligned}$$

15. (b)

Specific humidity,  $\omega = 0.0186 \text{ kg vapour/kg of dry air}$

$$\omega = 0.622 \frac{P_v}{P - P_v} \quad (\text{where } P_v \text{ is vapour pressure})$$

$$0.0186 = 0.622 \left( \frac{P_v}{1.0132 - P_v} \right)$$

$$1.0132 \times 0.0186 - 0.0186 P_v = 0.622 P_v$$

$$P_v = \frac{1.0132 \times 0.0186}{0.6406}$$

$$P_v = 0.0294 \text{ bar}$$

$$\text{Relative humidity, } \phi = \frac{P_v}{P_s} = \frac{0.0294}{0.0563} = 0.5222 \text{ or } 52.22\%$$

16. (c)

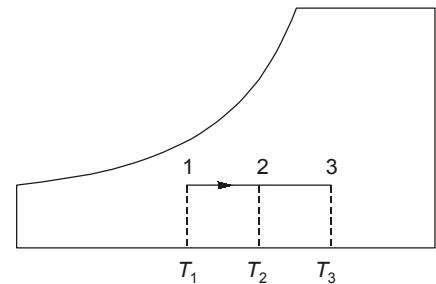
$$(\text{COP})_1 = 5, (\text{COP})_2 = 5.5$$

$$\begin{aligned} \text{Combined COP} &= \frac{(\text{COP})_1 \times (\text{COP})_2}{1 + (\text{COP})_1 + (\text{COP})_2} \\ &= \frac{5 \times 5.5}{1 + 5 + 5.5} = \frac{27.5}{11.5} = 2.39 \end{aligned}$$

17. (d)

$$T_1 = 15^\circ\text{C}, T_2 = 25^\circ\text{C}, T_3 = 34^\circ\text{C}$$

$$\begin{aligned} \text{BPF} &= \frac{T_3 - T_2}{T_3 - T_1} \\ &= \frac{34 - 25}{34 - 15} \\ &= \frac{9}{19} = 0.474 \end{aligned}$$



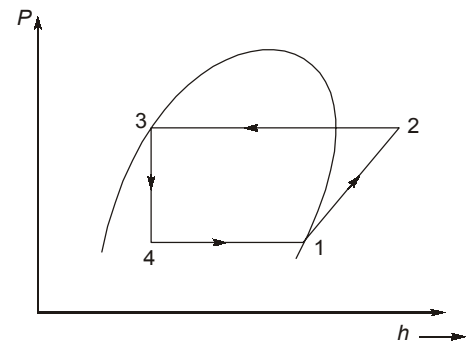
18. (b)

$$h_1 = 210 \text{ kJ/kg}$$

$$h_3 = h_4 = 74 \text{ kJ/kg}$$

$$\text{mass flow rate, } \dot{m} = 1 \text{ kg/s}$$

$$\begin{aligned} \text{Cooling load, } Q &= \dot{m}(h_1 - h_4) \\ &= 1(210 - 74) \\ &= 136 \text{ kW} \end{aligned}$$



20. (b)

For a two-stage cascade system working on Carnot cycle, the optimum cascade temperature at which the COP will be maximum,  $T_{cc, \text{opt}}$  is given by:

$$T_{cc, \text{opt}} = \sqrt{T_e \cdot T_c}$$

where  $T_e$  and  $T_c$  are the evaporator temperature of low temperature cascade and condenser temperature of high temp. cascade, respectively.

$$T_e = -90^\circ\text{C} = -90 + 273 = 183 \text{ K}$$

$$T_c = 50^\circ\text{C} = 50 + 273 = 323 \text{ K}$$

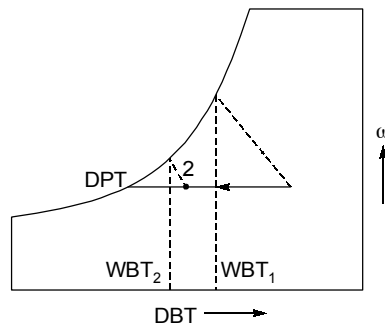
$$T_{cc, \text{opt}} = \sqrt{183 \times 323} = 243.123 \text{ K or } -29.87^\circ\text{C} \approx -30^\circ\text{C}$$

**Note:** Limitation of multi-stage vapour compression refrigeration (VCR) system are overcome by using cascade refrigeration system. In a cascade system, a series of refrigerants with progressively lower boiling points are used in a series of single stage units. The condenser of lower stage system is coupled to the evaporator of the next higher stage system and so on.

21. (b)

The use of hermetic compressors is ideal in smaller refrigeration systems, which use capillary tubes as expansion devices and are critically charged systems. Hermetic compressors are normally not serviceable. They are not flexible as it is difficult to vary their speed to control the cooling capacity.

22. (b)



We can see that  
DPT = constant  
 $WBT_2 < WBT_1$   
 $DBT_1 > DBT_2$

23. (b)

Based on Gibbs' phase rule, the thermodynamic state of moist air is uniquely fixed if the barometric pressure and two other independent properties are known. So, statement 4 is correct (not 3).

24. (a)

Given,

$$COP = 3.6 = \frac{T_2 + T}{(T_1 - T) - (T_2 + T)}$$

$$T_2 = -40^\circ\text{C} = -40 + 273 = 233 \text{ K}$$

$$T_1 = 30^\circ\text{C} = 30 + 273 = 303 \text{ K}$$

$$3.6 = \frac{233 + T}{(303 - T) - (233 + T)}$$

$$3.6 = \frac{233 + T}{70 - 2T}$$

$$233 + T = 252 - 7.2 T$$

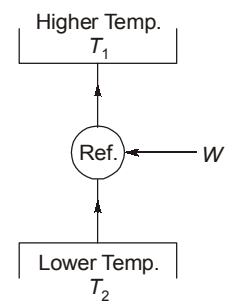
$$T + 7.2 T = 252 - 233$$

$$T = \frac{19}{8.2} = 2.317 \text{ K} \approx 2.32 \text{ K}$$

New temperature are,

$$T_{2'} = T_2 + T = 233 + 2.32 = 235.32 \text{ K or } -37.68^\circ\text{C}$$

$$T_{1'} = T_1 - T = 303 - 2.32 = 300.68 \text{ K or } 27.68^\circ\text{C}$$



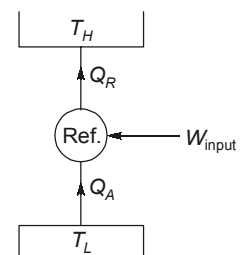
26. (b)

$$COP = \frac{Q_A}{W_{in}} = \frac{Q_A}{Q_R - Q_A}$$

$$Q_R - Q_A = \frac{Q_A}{COP}$$

$$= \frac{9}{3.5} = 2.57 \text{ kJ}$$

$$Q_R = 9 + 2.57 = 11.57 \text{ kJ}$$



28. (a)

$$\text{COP} = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{(5)^{\frac{0.4}{1.4}} - 1} = 1.713$$

29. (d)

$$\omega = 0.015 \text{ kg w.v/kg d.a.}$$

$$P = 740 \text{ mm of Hg}$$

$$\omega = 0.622 \left( \frac{P_v}{P - P_v} \right)$$

Where  $P_v$  = partial pressure of water vapour

$$0.015 = 0.622 \left( \frac{P_v}{740 - P_v} \right)$$

$$0.015 \times 740 - 0.015 P_v = 0.622 P_v$$

$$P_v = \frac{0.015 \times 740}{(0.622 + 0.015)} = 17.42 \text{ mm Hg}$$

Partial pressure of given air sample,  $P_a = P - P_v = 740 - 17.42 = 722.58 \text{ mm Hg}$

$$\text{Specific volume of air, } v_a = \frac{R_a T}{P_a}$$

$$= \frac{287 \times (273 + 21)}{\frac{722.58}{1000} \times 13.6 \times 1000 \times 9.81} = 0.8752 \text{ m}^3/\text{kg.d.a}$$

30. (a)

Given:  $h_1 = 190 \text{ kJ/kg}$ ;  $h_2 = 215 \text{ kJ/kg}$ ;  $h_3 = h_4 = 75 \text{ kJ/kg}$

$$\text{Refrigerant flow, } \dot{m} = \frac{Q}{q_0}$$

$$\text{Refrigerant effect, } q_0 = h_1 - h_4 = 190 - 75 = 115 \text{ kJ/kg}$$

$$Q = 12 \times 3.5 \text{ kJ/s}$$

$$\dot{m} = \frac{12 \times 3.5}{115} = 0.3652 \text{ kg/s}$$

$$\begin{aligned} \text{Power consumption} &= \dot{m}(h_2 - h_1) \\ &= 0.3652 (215 - 190) = 9.13 \text{ kW} \end{aligned}$$

$$\text{Horse power/ton of refrigeration} = \frac{9.13 \times 1000}{746 \times 12} = 1.02 \text{ HP/TR}$$

