DETAILED SOLUTIONS

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ESE 2027 : Prelims Exam
CLASSROOM TEST SERIES

MECHANICAL ENGINEERING

Test 2

Section A : Thermodynamics [All Topics]

Section B : Refrigeration and Air-Conditioning [All Topics]

Answer Key

1.	(c)	16.	(b)	31.	(c)	46.	(b)	61.	(b)
2.	(c)	17.	(c)	32.	(d)	47.	(c)	62.	(c)
3.	(c)	18.	(b)	33.	(b)	48.	(d)	63.	(c)
4.	(c)	19.	(b)	34.	(c)	49.	(d)	64.	(b)
5.	(b)	20.	(a)	35.	(d)	50.	(d)	65.	(d)
6.	(c)	21.	(a)	36.	(c)	51.	(b)	66.	(a)
7.	(c)	22.	(b)	37.	(b)	52.	(c)	67.	(c)
8.	(b)	23.	(c)	38.	(b)	53.	(a)	68.	(b)
9.	(a)	24.	(b)	39.	(b)	54.	(a)	69.	(d)
10.	(d)	25.	(c)	40.	(c)	55.	(c)	70.	(c)
11.	(b)	26.	(a)	41.	(b)	56.	(b)	71.	(b)
12.	(a)	27.	(d)	42.	(d)	57.	(b)	72.	(c)
13.	(c)	28.	(d)	43.	(c)	58.	(a)	73.	(c)
14.	(c)	29.	(a)	44.	(b)	59.	(d)	74.	(c)
15.	(d)	30.	(a)	45.	(c)	60.	(a)	75.	(d)

Section A : Thermodynamics

1. (c)

- It allows macroscopic thermodynamic and flow properties to be treated as continuous point functions.
- At very high altitude or vacuums, the continuum hypothesis ceases to be valid because the mean free path may become large.

2. (c)

- Brine: NaCl + Water, liquid solution only → Pure Substance
- True solution: Single homogeneous phase → pure substance
- Emulsion: Dispersed liquid in liquid medium → Not pure substance
- Aerosol: Dispersed liquid/solid in gas medium → Not pure substance
- Azeotropic refrigerant: liquid + vapour → pure substance

3. (c)

The important Characteristics of mollier diagram are

1. If the temperature increases, the slope of isobar will increases.
2. The isobars diverges from one another as the temperature and entropy increases.
3. The isotherms become straight and almost horizontal in the superheated region, specially at low pressure.
4. The line of constant dryness fraction are parallel to the saturation line.

4. (c)

5. (b)

Given : $V = 0.6\ m^3$; $P = 6$ MPa $= 6000$ kPa; $T = 300$ K; $z = 0.69$

$$\because \quad z = \frac{PV}{mRT}$$

$$\Rightarrow \quad 0.69 = \frac{6 \times 1000 \times 0.6}{m \times 0.25 \times 300}$$

$$\Rightarrow \quad m = 69.565 \text{ kg} \simeq 69.6 \text{ kg}$$

6. (c)

Given, mass of Nitrogen $(m_{N_2}) = 3$ kg ; Mass of carbondioxide $(m_{CO_2}) = 8$ kg ;

Pressure $(P) = 300$ kPa; Temperature $(T) = 27°C = 300$ K

The total mass of the system $(m) = m_{N_2} + m_{CO_2} = 3 + 8 = 11$ kg

The equivalent gas constant of the mixture $(R) = \dfrac{m_{N_2} \times \dfrac{\bar{R}}{M_{N_2}} + m_{CO_2} \times \dfrac{\bar{R}}{M_{CO_2}}}{m}$

$$\Rightarrow \quad R = \frac{3\times\frac{8.314}{28}+8\times\frac{8.314}{44}}{11} = 0.218 \text{ kJ/kmol K or kJ/kg K}$$

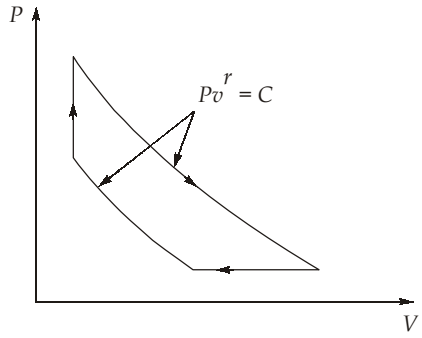
Using, $PV = mRT$

$$300\times V = 11\times 0.218\times 300$$

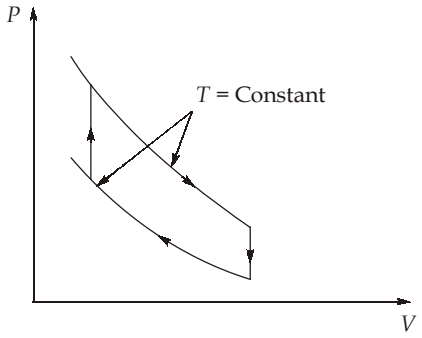
$$\Rightarrow \quad V = 2.390 \text{ m}^3 \approx 2.4 \text{ m}^3$$

7. (c)

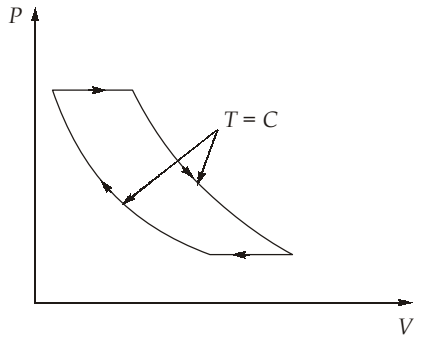
Atkinson cycle



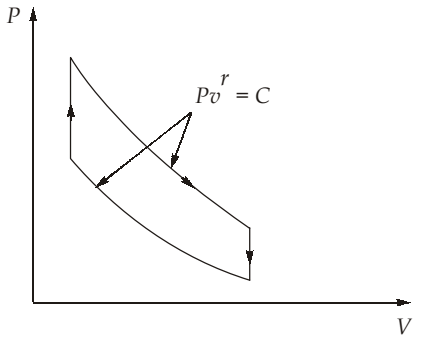
Stirling cycle



Ericsion cycle



Otto cycle



8. (b)

Given : $V_s = 8000 \text{ cm}^3$; $V_c = 2000 \text{ cm}^3$, $\gamma = 1.5$

$\therefore$ The compression ratio, $(r) = 1+\frac{V_s}{V_c} = 1+\frac{8000}{2000} = 5$

$\therefore$ The air standard efficiency $(\eta_{otto}) = 1-\frac{1}{(r)^{\gamma-1}} = 1-\frac{1}{(5)^{1.5-1}} = 1-\frac{1}{(5)^{0.5}}$

$= 0.5528$ or 55.28%

9. (a)

10. (d)

Given : $V_2 = 8V_1$; $P \propto D^2$

For a sphere of diameter D,

$$V = \frac{\pi}{6}D^3$$

$$\Rightarrow \quad D \propto V^{1/3}$$

We know $V \propto D^3$, for spherical balloon

$$\therefore \quad P \propto V^{2/3}$$

$$\Rightarrow \quad P = CV^{2/3}$$

The boundary work is given as,

$$W = \int_{V_1}^{V_2} P\,dV = \int_{V_1}^{8V_1} CV^{2/3}\,dV$$

$$= C\left[\frac{V^{5/3}}{5/3}\right]_{V_1}^{8V_1} = C \times \frac{3}{5} \times \left[8^{5/3} - 1\right]$$

$$= \frac{3}{5} \times 31 \times CV_1^{5/3} = \frac{3}{5} \times 31 \cdot V_1 \times CV_1^{2/3}$$

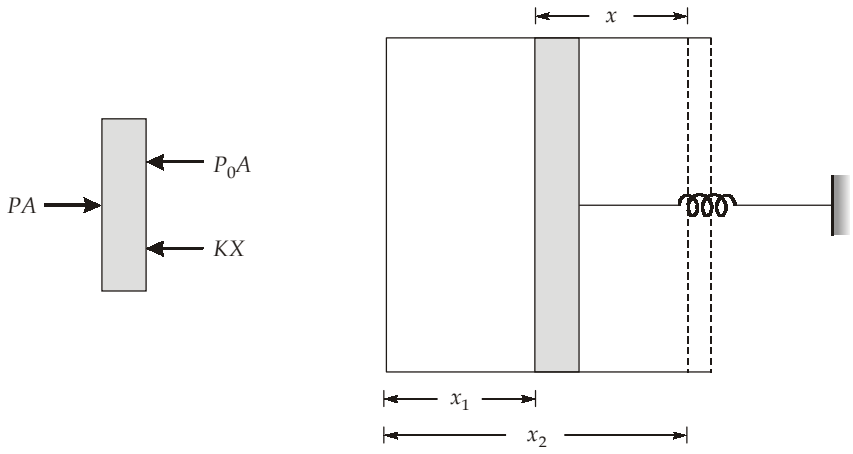
$$\because \quad P_1 = CV_1^{2/3}$$

$$\Rightarrow \quad W = \frac{93}{5}P_1V_1 = 18.6\,P_1V_1$$

11. (b)

Given : $V_1 = 0.1\ \text{m}^3$, $V_2 = 0.5\ \text{m}^3$, $P_1 = 300\ \text{kPa}$, $P_0 = 100\ \text{kPa}$

At equilibrium state,



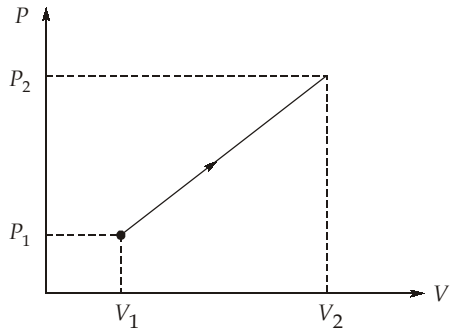
where, A = cross-sectional area of piston, x_1 = Initial compression in spring

$$PA = P_0 A + K X$$

$$P = P_0 + \frac{K}{A}X$$

$\because$ Pressure of gas inside the cylinder is linear function of piston displacement.

At initial condition



$$P_1 = P_0 + \frac{K}{A}X_1$$

$$\Rightarrow \quad K = \frac{(P_1 - P_0)A}{X_1} = \frac{200A}{\frac{0.1}{A}} = 2000A^2$$

And at final condition, $$P2 = P_0 + \frac{K}{A}X_2$$

$$= P_0 + \frac{2000A^2}{A} \times \frac{0.5}{A} = 1100 \text{ kPa}$$

$\because$ The total boundary work done by the gas is

$$= \frac{1}{2}(P_1 + P_2)(V_2 - V_1)$$

$$= \frac{1}{2} \times (1100 + 300)(0.5 - 0.1) = 280 \text{ kJ}$$

12. (a)

$$(\Delta S)_{\text{resistor}} = \frac{\delta Q}{T} + s_{gen}$$

$$s_{\text{gen}} > 0 \qquad \left[\because \frac{\delta Q}{T} = 0\right]$$

$$(\delta S)_{\text{resistor}} > 0$$

13. (c)

Given : $A_p = 0.1\ \text{m}^2, \Delta X = 0.2\ \text{m}, P_0 = 100\ \text{kPa}, W_{\text{net}} = 3.4\ \text{kJ}, V = 28\ \text{V}, I = 0.8\ \text{A}, \eta = 75\%$

$\therefore$ The net work done on the fluid = Work done by the stirrer on the fluid + Work done by fluid against the atmospheric pressure

$$\Rightarrow \quad W_{\text{net}} = W_{\text{stirrer}} - W_{\text{atm}}$$

$$\Rightarrow \quad 3.4 = W_{\text{stirrer}} - P_0 \times \Delta V$$

$$3.4 = W_{\text{stirrer}} - 100 \times (A_p \times \Delta X)$$

$$3.4 = W_{\text{stirrer}} - 100 \times 0.1 \times 0.2$$

$$\Rightarrow \quad W_{\text{stirrer}} = 5.4\ \text{kJ}$$

Also, $W_{\text{stirrer}} = (V \cdot I \cdot \eta) \times t$

$\because$ $28 \times 0.8 \times 0.75 \times t = 5.4 \times 1000$

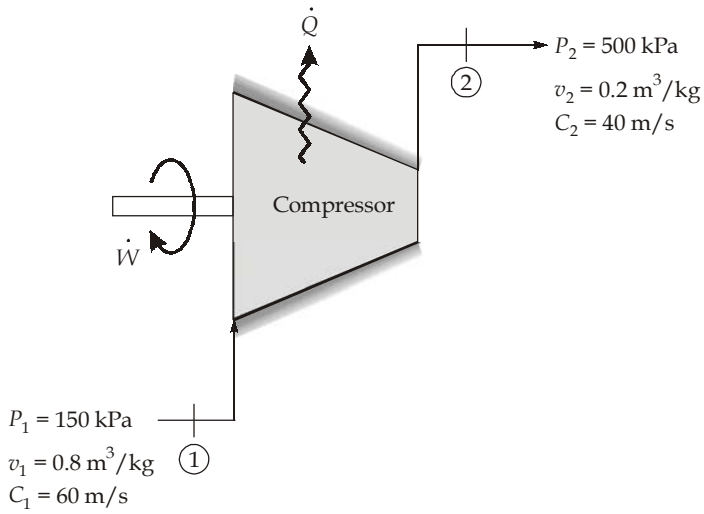
$\Rightarrow$ $t = 321.43$ sec or 5.36 min

14. (c)

- Transit energies means energy in transition i.e. the energy possessed by the system is capable of crossing the boundaries of the system.
- The internal energy of a system is the energy stored within the body resulting from the kinetic and potential energy of its molecules, thus it is related to molecular structure and degree of molecular activities.

15. (d)

Given : $\dot{m} = 0.5\ \text{kg/s}; \Delta u = u_2 - u_1 = 60\ \text{kJ/kg}$



Now applying steady flow energy equation,

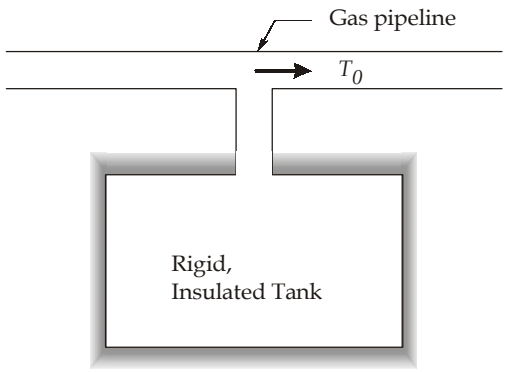
$$\left[P_1v_1 + u_1 + \frac{V_1^2}{2}\right]\dot{m} + \dot{Q} = \left[P_2v_2 + u_2 + \frac{V_2^2}{2}\right]\dot{m} + \dot{W}_{CV}$$

$$\Rightarrow \quad \dot{W}_{cv} = \dot{m}\left[P_1v_1 - P_2v_2 + u_1 - u_2 + \frac{V_1^2 - V_2^2}{2}\right] + \dot{Q}$$

$$= 0.5\left[150 \times 0.8 - 500 \times 0.2 - 60 + \frac{60^2 - 40^2}{2000}\right] - 20 = -39.5 \text{ kW}$$

Negative sign indicates work is supplied to the compressor.

16. (b)



Applying conservation of mass,

$$\dot{m}_2 - \dot{m}_1 = \dot{m}_i - \dot{m}_e$$

$$\Rightarrow \quad \dot{m}_2 = \dot{m}_i$$

where,

$T_i = T_0$ = Temperature of gas in the pipeline

T_2 = final temperature of gas in the tank.

Applying conservation of energy,

$$U_2 - U_1 = \dot{m}_i h_i + \dot{Q} - \dot{m}_e h_e - \dot{W}_{cv}$$

$$\Rightarrow \quad \dot{m}_2 u_2 = \dot{m}_i h_i + 0 - 0 - 0$$

$$\Rightarrow \quad U_2 = h_i$$

and $$c_v T_2 = c_p T_0$$

$$T_2 = \gamma T_0$$

- The filling of an evacuated space is highly irreversible, which generates entropy. Hence, the specific entropy increases.

17. (c)

Given :

State 1 : $P_1 = 4$ MPa, $T_1 = 300$ K, $V_1 = 0.5\ m^3$, $R = 0.25$ kJ/kg K, $\gamma = 1.4$

State 2 : $P_2 = 1$ MPa, $T_2 = 250$ K, $V_2 = V_1 = 0.5\ m^3$

State 3 : $T_3 = 300$ K

The mass of oxygen remaining in the tank after use is

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{1 \times 1000 \times 0.5}{0.25 \times 250} = 8 \text{ kg}$$

$$\text{The amount of heat absorb} = m_2 c_v (T_3 - T_2)$$

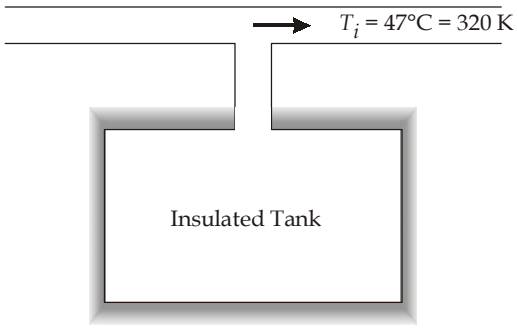
$$= \frac{mR}{\gamma - 1}[T_3 - T_2] = \frac{8 \times 0.25}{1.4 - 1}(300 - 250)$$

$$= 250 \text{ kJ}$$

18. (b)

Given : Initial state of tank-1: $T_1 = 37°C = 310$ K; $m_1 = 1$ kg

Final state of tank-2: $T_2 = ?, m_2 = 2m_1 = 2$ kg



Applying mass conservation:

$$m_2 - m_1 = m_i - m_e$$

$$\Rightarrow \quad 2 - 1 = m_i - 0$$

$$\because \quad m_i = 1 \text{ kg}$$

Applying Energy conservation:

$$U_2 - U_1 = m_i h_i + Q - m_e h_e - W_{cv}$$

$$m_2u_2 - m_1u_1 = m_ih_i + 0 - 0 - 0$$

$$\Rightarrow \quad m_2c_vT_2 - m_1c_vT_1 = m_ic_pT_i$$

$$\Rightarrow \quad m_2T_2 - m_1T_1 = m_i\frac{c_p}{c_v}T_i$$

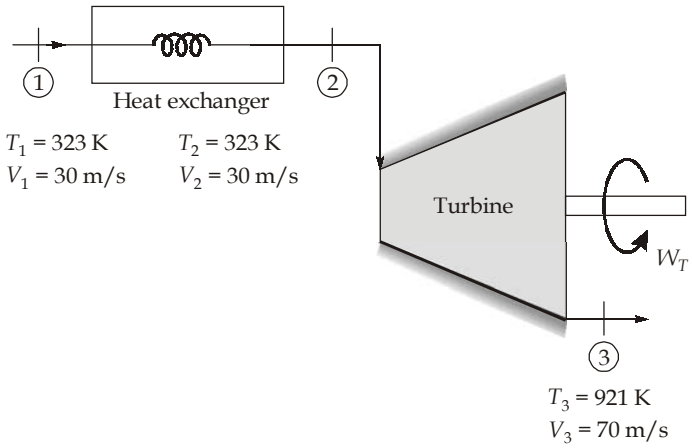
$$\Rightarrow \quad 2\times T_2 - 1\times 310 = 1\times 1.4\times 320$$

$$\therefore \quad T_2 = 379 \text{ K or } 106°\text{C}$$

19. (b)

Given : $T_1 = 50°\text{C} = 323$ K, $V_1 = 30$ m/s , $T_2 = 800°\text{C} = 1073$ K, $V_2 = 30$ m/s ,

$T_3 = 648°\text{C} = 921$ K, $V_3 = 70$ m/s



For heat exchanger:

$$h_1 + \frac{V_1^2}{2} + q = h_2 + \frac{V_2^2}{2} + w_{cv}$$

$$\Rightarrow \quad q = h_2 - h_1 = c_p(T_2 - T_1)$$

$$= 1.005\times(1073 - 323) = 753.75 \text{ kJ/kg}$$

For turbine :

$$h_2 + \frac{V_2^2}{2000} + Q = h_3 + \frac{V_3^2}{2000} + w_T$$

$$\Rightarrow \quad w_T = (h_2 - h_3) + \frac{V_2^2 - V_3^2}{2000}$$

$$= 1.005(1073 - 921) + \frac{30^2 - 70^2}{2000} = 150.76 \text{ kJ/kg}$$

$$\because \quad w_T = \frac{753.75}{1050.76} = 4.999 \simeq 5$$

20. (a)

$$\text{Workdone } (W) = P\Delta V$$
$$= 100 \times 10^3 \times (5 - 2)$$
$$= 300000 \text{ J} = 300 \text{ kJ}$$
$$\text{Change in internal energy } (\Delta U) = Q - W$$
$$= 1500 - 300 = 1200 \text{ kJ}$$

21. (a)

The efficiency of the Carnot engine is represented by :

$$\eta = 1 - \frac{T_L}{T_H}$$

Now, keeping the temperature of the sink (T_L) constant.

$$\frac{\partial \eta}{\partial T_H} = \frac{T_L}{T_H^2} \quad \ldots(i)$$

and, keeping the temperature of the source (T_H) constant:

$$\frac{\partial \eta}{\partial T_L} = -\frac{1}{T_H} \quad \ldots(ii)$$

$\therefore \quad T_H > T_L$

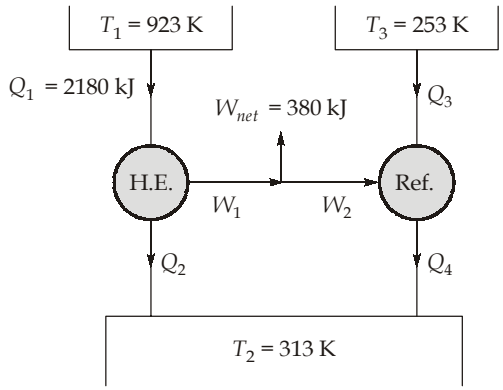
$\because$ From equation (i) and (ii),

$$\frac{\partial \eta}{\partial T_H} < \frac{\partial \eta}{\partial T_L}$$

$\Rightarrow$ The efficiency can be increased effectively by decreasing T_L than increasing T_H.

22. (b)

Given : $T_1 = 923 \text{ K}, T_2 = 313 \text{ K}, T_3 = 253 \text{ K}, Q_1 = 2180 \text{ kJ}, W_{\text{net}} = 380 \text{ kJ}$



For heat engine, $\eta = 1 - \frac{T_2}{T_1} = \frac{W_1}{Q_1}$

$$W_1 = Q_1\left(1-\frac{T_2}{T_1}\right) = 2180\times\left(1-\frac{313}{923}\right)$$

$$= 1440.74 \text{ kJ}$$

Applying first law of thermodynamics for heat engine,

$$Q_1 - Q_2 = W_1$$

$$\Rightarrow \quad Q_2 = Q_1 - W_1 = 2180 - 1440.74 = 739.26 \text{ kJ}$$

and $$W_{net} = W_1 - W_2$$

$$380 = 1440.74 - W_2$$

$$\Rightarrow \quad W_2 = 1060.74 \text{ kJ}$$

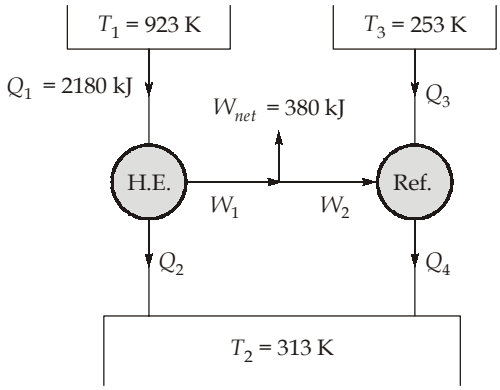
$$\text{COP} = \frac{T_3}{T_2 - T_3} = \frac{Q_3}{W_2}$$

$$\Rightarrow \quad Q_3 = W_2 \times \frac{T_3}{T_3 - T_2}$$

$$= 1060.74 \times \frac{253}{313 - 253} = 4472.79 \text{ kJ}$$

23. (c)

Given : $T_1 = 923$ K, $T_2 = 313$ K, $T_3 = 253$ K, $Q_1 = 2180$ kJ, $W_{net} = 380$ kJ



For heat engine, $$\eta = 1-\frac{T_2}{T_1} = \frac{W_1}{Q_1}$$

$$W_1 = Q_1\left(1-\frac{T_2}{T_1}\right) = 2180\times\left(1-\frac{313}{923}\right)$$

$$= 1440.74 \text{ kJ}$$

Applying first law of thermodynamics for heat engine,

$$Q_1 - Q_2 = W_1$$

$$\Rightarrow \quad Q_2 = Q_1 - W_1 = 2180 - 1440.74 = 739.26 \text{ kJ}$$

and $$W_{net} = W_1 - W_2$$

$$380 = 1440.74 - W_2$$

$$\Rightarrow \quad W_2 = 1060.74 \text{ kJ}$$

$$\text{COP} = \frac{T_3}{T_2 - T_3} = \frac{Q_3}{W_2}$$

$$\Rightarrow \quad Q_3 = W_2 \times \frac{T_3}{T_3 - T_2}$$

$$= 1060.74 \times \frac{253}{313 - 253} = 4472.79 \text{ kJ}$$

Applying first law of thermodynamics for the refrigerator,

$$W_2 = Q_4 - Q_3$$

$$Q_4 = W_2 + Q_3 = 1060.74 + 4472.79$$

$$= 5533.53 \text{ kJ}$$

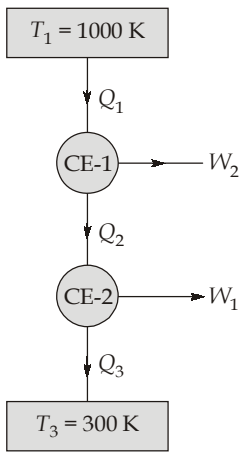
$\because$ The heat absorbed from the reservoir at -20°C is $= 4472.79 \text{ kJ} \approx 4473 \text{ kJ}$

and the net heat supplied to reservoir at 40°C $= Q_2 + Q_4$

$$= 6272.79 \text{ kJ} \simeq 6273 \text{ kJ}$$

24. (b)

Given : $T_1 = 1000$ K, $T_3 = 300$ K



$\therefore \quad W_1 = W_2$ (Given)

$\Rightarrow \quad Q_1 - Q_2 = Q_2 - Q_3$

$\Rightarrow \quad 2Q_2 = Q_1 + Q_3$

$\Rightarrow \quad 2 = \frac{Q_1}{Q_2} + \frac{Q_3}{Q_2}$ ...(i)

We know for Carnot engine:

$$\eta = 1 - \frac{T_L}{T_H} = 1 - \frac{Q_L}{Q_H}$$

$\Rightarrow \quad \frac{Q_L}{Q_H} = \frac{T_L}{T_H}$

$\because \quad \frac{Q_2}{Q_1} = \frac{T_2}{T_1}$ and $\frac{Q_3}{Q_2} = \frac{T_3}{T_2}$

From equation (i)

$$2 = \frac{T_1}{T_2} + \frac{T_3}{T_2}$$

$\Rightarrow \quad T_2 = \frac{T_1 + T_3}{2} = 650 \text{ K}$

$\because$ Efficiency of Carnot engine-1 and 2 are η_1 and η_2 respectively,

$\Rightarrow \eta_1 = 1 - \frac{T_2}{T_1}$ and $\eta_2 = 1 - \frac{T_3}{T_2}$

$\eta_1 = 1 - \frac{650}{1000}$ and $\eta_2 = 1 - \frac{300}{650}$

$\eta_1 = 0.35$ or 35% and $\eta_2 = 0.538$ or 53.8%

25. (c)

Given : $T_H = 1000$ K and $T_L = 600$ K

$\because$ The initial efficiency of carnot engine is

$$\eta_{\text{initial}} = 1 - \frac{T_L}{T_H} = 1 - \frac{600}{1000} = 0.40 \text{ or } 40\%$$

The New efficiency of the engine is,

$$\eta_{\text{new}} = 1.3 \times \eta_{\text{initial}} = 1.3 \times 0.4 = 0.52$$

Let ΔT be the temperature change in source and sink temperature

$$\eta_{\text{new}} = 1 - \frac{T_L - \Delta T}{T_H - \Delta T}$$

Both temperature are decreased by same amount because if the source and sink temperature are change by same amount and efficiency increases then there is a decrease in temperature of both source and sink.

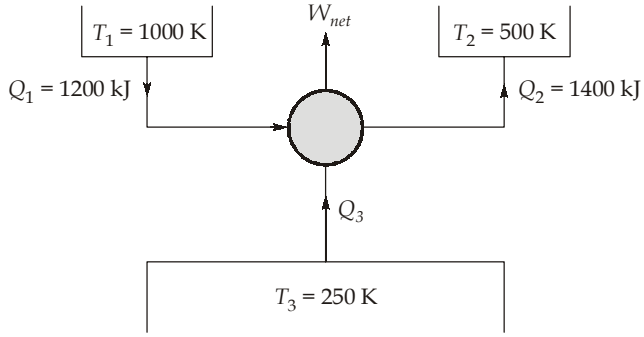
$$0.52 = 1 - \frac{600 - \Delta T}{1000 - \Delta T}$$

$$\Delta T = 230.769°C \simeq 230.8°C$$

26. (a)

Slope of constant volume curve is more than slope of constant pressure curve on T-s.

27. (d)



Let Q_3 be the amount of heat supplied by reservoir-3

Now using Clausius theorem for reversible device,

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

$$\Rightarrow \quad \frac{Q_1}{T_1} + \frac{Q_2}{T_2} + \frac{Q_3}{T_3} = 0$$

$$\Rightarrow \quad \frac{1200}{1000} - \frac{1400}{500} + \frac{Q_3}{250} = 0$$

$$\therefore \quad Q_3 = 400 \text{ kJ}$$

i.e., Heat is supplied from reservoir to the device.

Now applying 1st law of thermodynamics.

$$W_{net} = \Sigma Q_i = Q_1 + Q_2 + Q_3 = 1200 - 1400 + 400$$

$$= +200 \text{ kJ (i.e. work producing devices)}$$

28. (d)

- Entropy is a thermodynamic property and an exact differential i.e. it is a state function. But entropy generation is not the property due to internal dissipations i.e. it is a path function.
- An adiabatic process need not be isentropic, since entropy can also increase due to friction etc. but if the process is adiabatic and reversible, it must be adiabatic. also if an isentropic process is adiabatic it cannot be reversible.

29. (a)

Using T-ds equation :

$$Tds = dh - vdp$$

$$\Rightarrow \quad ds = \frac{dh}{T} - \frac{v\,dp}{T}$$

$$= \frac{C_p\,dT}{T} - \frac{R\,dp}{P}$$

$\because$ from station A to station B

$$\int_{S_A}^{S_B} ds = \int_{T_A}^{T_B} \frac{C_p\,dT}{T} - \int_{P_A}^{P_B} \frac{R\,dp}{P}$$

$$S_B - S_A = 1.005\ln\left(\frac{T_B}{T_A}\right) - 0.287\ln\left(\frac{P_B}{P_A}\right)$$

$$= 1.005\ln\left(\frac{273+27}{273+127}\right) - 0.287\ln\left(\frac{100}{1600}\right)$$

$$= 1.005\ln(0.75) - 0.287\ln\left(\frac{1}{16}\right)$$

$$= -0.28791 + 0.7957$$

$$(\Delta S)_{\text{system}} = 0.5066\ \text{kJ/kgK}$$

$\because$ The duct is insulated, therefore $(\Delta S)_{\text{surr}} = 0$

$$\Rightarrow \quad (\Delta s)_{\text{univ}} = (\Delta S)_{\text{surr}} + (\Delta S)_{\text{system}} = +0.5066$$

$$\Rightarrow \quad (\Delta s)_{\text{univ}} > 0$$

Which is impossible, hence flow is from station A to station B.

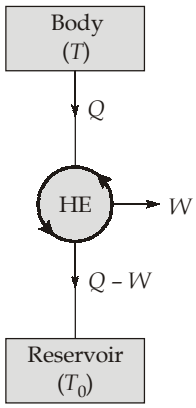
30. (a)

The second law states that while the entropy of a system can decrease, it must be accompanied by an increase in the entropy of the surrounding such that the total entropy of the universe always increases. This is the principle of entropy generation.

$$\Delta S_{\text{sys}} + \Delta S_{\text{surr}} \geq 0$$

If a system undergoes external irreversibility, such as thermal resistance during heat transfer, it cannot achieve Carnot efficiency. The Carnot efficiency only be attained when both internal and external irreversibilities are eliminated. External irreversibilities cause entropy generation in the surroundings, leading to deviation from the ideal Carnot performance.

31. (c)



The change in entropy of the finite body is given as

$$(\Delta S)_{body} = \int_{T}^{T_0} C\frac{dT}{T} = C\ln\frac{T_0}{T}$$

The change in entropy of the heat engine is zero i.e. $(\Delta S)_{HE} = 0$

$\therefore$ The amount of heat rejected is $(Q - W)$

$\because$ The change in entropy of the thermal reservoir is,

$$(\Delta S)_{TR} = \frac{Q-W}{T_0}$$

The change in entropy of the universe,

$$(\Delta S)_{univ} = (\Delta S)_{body} + (\Delta S)_{HE} + (\Delta S)_{TR}$$

$$(\Delta S)_{univ} = C\ln\frac{T_0}{T} + 0 + \frac{Q-W}{T_0}$$

Now, for maximum obtainable work the system should be reversible and for reversible system

$\Rightarrow$ $$C\ln\frac{T_0}{T} + \frac{Q-W}{T_0} = 0$$

$\because$ $$W_{max} = \left[\frac{Q}{T_0} + C\ln\frac{T_0}{T}\right]T_0$$

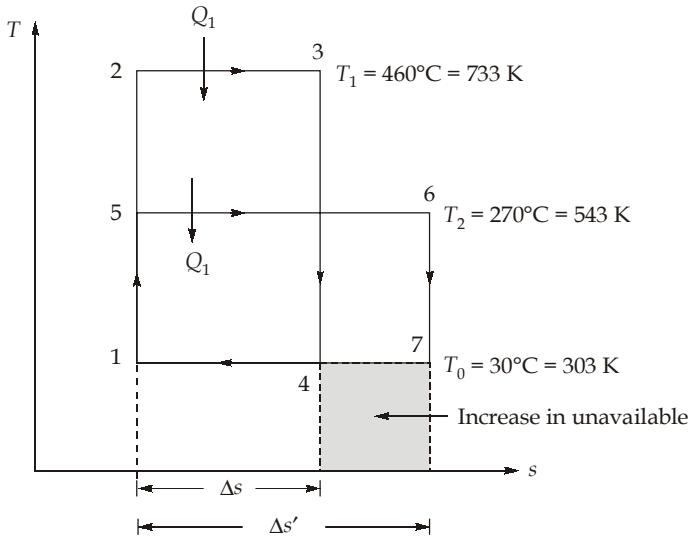
The amount of heat transferred from body to heat engine is given by

$$Q = C(T - T_0)$$

$\Rightarrow$ $$W_{max} = C\left[(T - T_0) - T_0\ln\frac{T}{T_0}\right]$$

32. (d)

33. (b)



$$\therefore \quad Q_1 = T_1\Delta S = T_2\Delta S'$$

$$\Rightarrow \quad \Delta S'/\Delta S = T_1/T_2$$

Now, work done in cycle 12341 is:

$$W = (T_1 - T_0)\Delta S$$

and workdone in cycle 15671 is:

$$W' = (T_2 - T_0)\Delta S'$$

The fraction of energy that become unavailable due to irreversible heat transfer

$$= \frac{W - W'}{W} = \frac{(T_1 - T_0)\Delta S - (T_2 - T_0)\Delta S'}{(T_1 - T_0)\Delta S} = \frac{T_0\left(\frac{\Delta S'}{\Delta S} - 1\right)}{T_1 - T_0}$$

$$= \frac{T_0\left[\frac{T_1}{T_2} - 1\right]}{T_1 - T_0} = \frac{303\left(\frac{733}{543} - 1\right)}{733 - 303} = 0.246 \approx 0.25$$

34. (c)

Given : $T_1 = 327°C$, $T_2 = 27°C$, $V_1 = V$; $P_1 = 500$ kPa, $P_2 = 100$ kPa, $V_2 = 2V$; $m = 2$ kg

We know, the maximum obtainable work is given by:

$$W_{max} = \phi_1 - \phi_2$$

$$= (U_1 - T_0S_1) - (U_2 - T_0S_2)$$

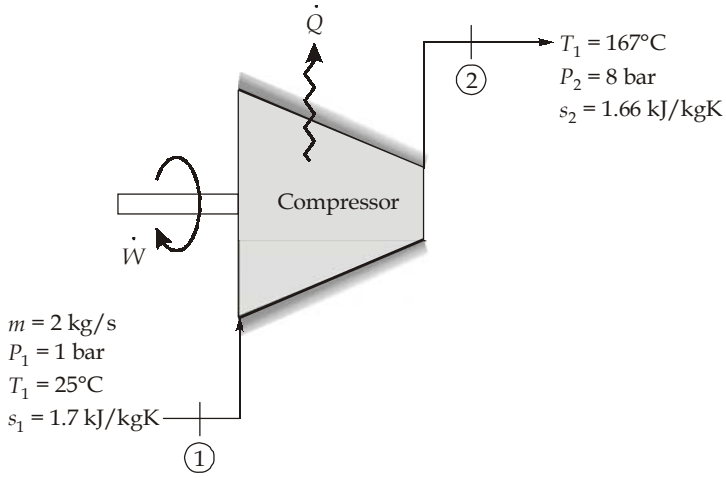
$$= U_1 - U_2 - T_0(S_1 - S_2)$$

$$= mC_v(T_1 - T_2) - T_0 m\left[C_v \ln\frac{T_1}{T_2} + R\ln\frac{V_1}{V_2}\right]$$

$$= 2\times 0.718(327 - 27) - 300\times 2\times[0.718\ln 2 + 0.287\ln\frac{1}{2}]$$

$$= 430.8 - 179.247 = 251.55 \text{ kJ}$$

35. (d)



Applying SFEE, to obtain the actual power supplied to the compressor,

$$\dot{m}\left(h_1 + \frac{V_1^2}{2} + gz_1\right) + \dot{Q} = \dot{m}\left(h_2 + \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{act}$$

$$\Rightarrow \quad \dot{W}_{act} = \dot{m}(h_1 - h_2) + \dot{Q}$$

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

$$= 2\times 1.005(25 - 167) + (-100)$$

$$= -385.42 \text{ kW or } 385.42 \text{ kW}$$

Now the minimum/reversible power supplied to the compressor is,

$$\dot{W}_{rev} = \phi_1 - \phi_2$$

$$= (H_1 - T_0S_1) - (H_2 - T_0S_2)$$

$$= \dot{m}C_p(T_1 - T_2) - T_0\dot{m}(s_1 - s_2)$$

$$\Rightarrow \quad \dot{W}_{rev} = 2\times 0.005(25-167) - 298\times 2(1.7-1.66)$$

$$= -309.27 \text{ kW}$$

$\because$ The second law efficiency of the compressor is defined as

$$\eta_{II} = \frac{\text{Minimum work consumed}}{\text{Actual work consumed}} \times 100$$

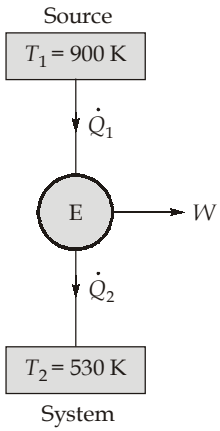
$$= \frac{309.27}{385.42} \times 100 = 80.24\%$$

36. (c)

$$(\Delta S)_{uni} = (\Delta S)_{sys} + (\Delta S)_{surr}$$

$$= \frac{4000}{400} + \frac{-4000}{400} = 0 \text{ kJ/K}$$

37. (b)



$\because$ The efficiency of the cycle is

$$\eta = 1 - \frac{T_2}{T_1} = 1 - \frac{530}{900} = 0.411$$

and, also $\quad \eta = 1 - \dfrac{\dot{Q}_2}{\dot{Q}_1}$

$$\Rightarrow \quad \dot{Q}_1 = \frac{\dot{Q}_2}{1-\eta} = \frac{3000}{1-0.411} = 5093.38 \text{ kJ/min}$$

The rate of entropy production is

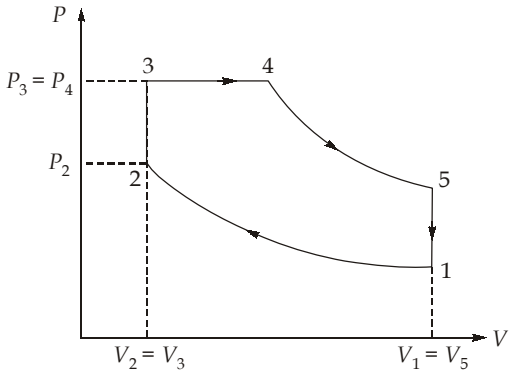
$$(\dot{S})_{gen} = (\dot{S})_{source} + (\dot{S})_{system} + (\dot{S})_{engine}$$

$$= -\frac{\dot{Q}_1}{T_1} + \frac{\dot{Q}_2}{T_2} + 0$$

$$= -\frac{5093.38}{900} + \frac{3000}{530} + 0 \simeq 0$$

38. (b)

Given : $\frac{V_5}{V_4} = 10,\ \alpha = 1.4$



As heat addition at constant pressure ceases at 4% of the stroke.

$$0.4(V_1 - V_2) = (V_4 - V_3)$$

$$0.4 = \frac{V_4 - V_3}{V_1 - V_3} \qquad [\text{As } V_2 = V_3]$$

$$0.4 = \frac{\frac{V_4}{V_3} - 1}{\frac{V_1}{V_3} - 1}$$

$$0.4 = \frac{\rho - 1}{r - 1}$$

$$0.4 = \frac{\frac{r}{10} - 1}{r - 1} \qquad [\text{As } \rho \times r_e = r \Rightarrow \rho = \frac{r}{r_e} = \frac{r}{10}]$$

On solving, $r = 16$

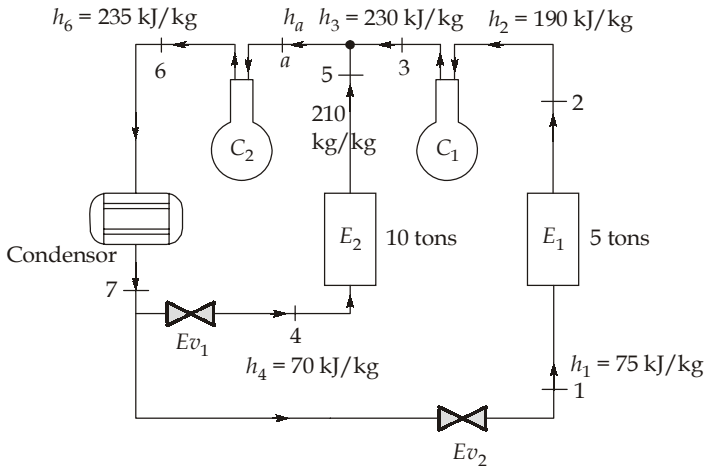
Section B : Refrigeration and Air-Conditioning

39. (b)

They are generally used only in larger sizes because in the larger sizes they are much lower in first cost, low running cost and requires considerably less space than reciprocating compressors.

40. (c)

Given : $RE_1 = 5$ Tons $= 5 \times 3.5 = 17.5$ kW, $RE_2 = 10$ Tons $= 10 \times 3.5 = 35$ kW



$$RE_1 = \dot{m}_1(h_2 - h_1) \qquad RE_2 = \dot{m}_2(h_5 - h_4)$$

$$17.5 = \dot{m}_1(190 - 75) \qquad 35 = \dot{m}_2(210 - 70)$$

$$\dot{m}_1 = 0.152 \text{ kg/sec} \qquad \dot{m}_2 = 0.25 \text{ or } \frac{1}{4} \text{ kg/sec}$$

Energy Conservation at point '*A*',

$$\dot{m}_1 h_3 + \dot{m}_2 h_5 = h_a(\dot{m}_1 + \dot{m}_2)$$

$$0.152 \times 230 + 0.25 \times 210 = h_a(0.152 + 0.25)$$

$$h_a = 217.56 \text{ kJ/kg}$$

41. (b)

Power required to drive the compressor 1 :

$$= \dot{m}_1(h_3 - h_2)$$

$$= 0.152(230 - 190) = 6.08 \text{ kW}$$

For compressor 2 :

$$= (\dot{m}_1 + \dot{m}_2)(h_6 - h_a)$$

$$= 0.402 \times (235 - 217.56)$$

$$= 7.011 \text{ kW}$$

$$\text{Total power required} = 6.08 + 7.011 = 13.09 \text{ kW}$$

$$\text{COP} = \frac{\text{Total refrigeration load}}{\text{Total work}} = \frac{15 \times 3.5}{13.09}$$

$$\text{COP} = 4.01$$

42. (d)

Advantages :

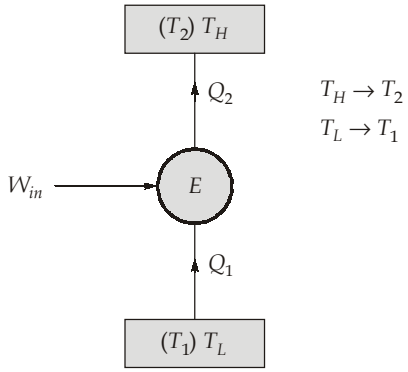
1. As working cycle is near the Carnot cycle, the C.O.P. is quite high. The C.O.P. of the vapour cycle lies between 3 and 4 whereas the C.O.P. of air cycle is always less than one.
2. Running cost of vapour refrigeration system is only 20% of air refrigeration system used on ground level.
3. As heat is carried away by the latent heat of vapour, the amount of liquid circulated is considerably less per ton of refrigeration, so that the size of evaporator is smaller for same refrigerating effect.
4. Just by adjusting the throttle valve of the same unit, the required temperature of the evaporator can be achieved.

Disdvantages :

1. First investment cost is high.
2. Major problem is to prevent leakage of the refrigerant.

43. (c)

Given :



when $\text{COP} = 2$

$$T_{H'} = T_H + 20$$

$$T_{L'} = T_L - 20$$

$$\text{COP} = \frac{T_L}{T_H - T_L} = 4$$

$$4 = \frac{T_L}{T_H - T_L}$$

$$\Rightarrow \quad T_H - T_L = \frac{T_L}{4}$$

Increase in higher temperature or decrease in lower temperature = 20°C.

$$\text{COP} = \frac{T_L - 20}{(T_H + 20) - (T_L - 20)} = 2$$

$$\frac{T_L - 20}{T_H - T_L + 40} = 2$$

$$T_L - 20 = 2\left(\frac{T_L}{4} + 40\right)$$

$$T_L - 20 = \frac{T_L}{2} + 80$$

$$\frac{T_L}{2} = 100 \text{ K}$$

$$\Rightarrow \quad T_L = 200 \text{ K}$$

$$\Rightarrow \quad T_L = -73°\text{C}$$

$$T_H = T_L + \frac{T_L}{4} = \frac{5T_L}{4} = \frac{5 \times 200}{4} = 250 \text{ K}$$

$$T_H = -23°\text{C}$$

44. (b)

Given : $T_g = 127 + 273 = 400$ K; $T_c = 17 + 273 = 290$ K; $T_e = -13 + 273 = 260$ K

$$\text{COP} = \frac{T_e}{T_g}\left(\frac{T_g - T_c}{T_c - T_e}\right) = \frac{260}{400}\left(\frac{400 - 290}{290 - 260}\right)$$

$$\text{COP}_1 = 2.383$$

Refrigeration temperature is decreased to –33°C and heating temperature is increased to 187°C,

So

$$T_g = 187 + 273 = 460 \text{ K}$$

$$T_e = -33 + 273 = 240 \text{ K}$$

$$T_c = 290 \text{ K}$$

$$\text{COP}_2 = \frac{T_e}{T_g}\left(\frac{T_g - T_c}{T_c - T_e}\right) = \frac{240}{460}\left(\frac{460 - 290}{290 - 240}\right) = 1.774$$

$$\%\text{decrease in COP} = \frac{2.383 - 1.774}{2.383} = 0.2556 = 25.56\%$$

45. (c)

Given : $C_2Cl_2F_4$

The chemical formula is $C_mH_nCl_pF_q$

$$m = 2$$

$$n = 0$$

$$p = 2$$
$$q = 4$$

For saturated Hydrocarbons :

$$n + p + q = 2(m+1)$$
$$0 + 2 + 4 = 2(2+1)$$
$$6 = 6$$

It satisfies the balance equation, so the chemical formula is correct

$$\text{Designation} = R_{(m-1)(n+1)q}$$
$$= R_{(2-1)(0+1)4} = R_{114}$$

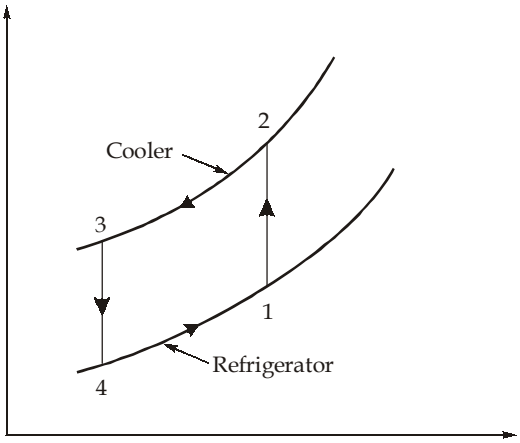
Comparing from R_{xyz} so

$$x = 1$$
$$y = 1$$
$$z = 4$$

So, $x + yz = 1 + 1 \times 4 = 5$

46. (b)

Given : $P_1 = 1$ bar, $\gamma = 1.5, C_p = 1$ kJ/kg, $P_2 = 8$ bar, $RE = 6$ Tons



$$T_1 = 7 + 273 = 280 \text{ K}$$
$$T_3 = 17 + 273 = 290 \text{ K}$$
$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 280\left(\frac{8}{1}\right)^{\frac{1.5-1}{1.5}}$$

$$T_2 = 280\left[(2)^3\right]^{\frac{1}{3}} = 560 \text{ K}$$

$$T_4 = \frac{T_3}{\left[(2)^3\right]^{\frac{1}{3}}} = \frac{290}{2} = 145 \text{ K}$$

$$RE = 6 \times 3.5 \text{ kW} = 21 \text{ kW}$$

$$\dot{m}_a(h_1 - h_4) = 21$$

$$\dot{m}_a C_p(T_1 - T_4) = 21$$

$$\dot{m}_a \times 1(280 - 145) = 21$$

$$\dot{m}_a = \frac{21}{135} = 0.156 \text{ kg/sec}$$

Now,

Heat gained by water in the cooler = Heat lost by air in the cooler.

$$\dot{m}_w C_{pw}(T_{wo} - T_{wi}) = \dot{m}_a C_{pa}(T_2 - T_3)$$

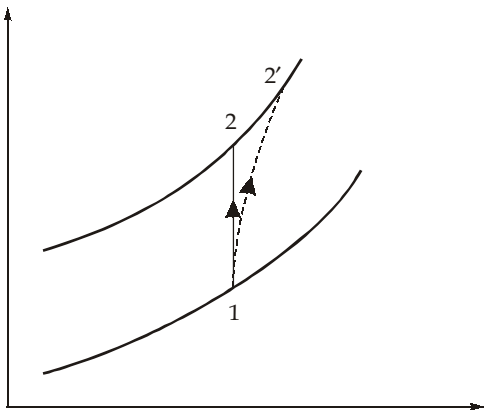
$$\dot{m}_w \times 4.2 \times 25 = 0.156 \times 1 \times (560 - 290)$$

$$\dot{m}_w = 0.4011 \text{ kg/sec}$$

$$\dot{m}_w = 1444.11 \text{ kg/hr}$$

47. (c)

Given : $T_1 = 27 + 273 = 300$ K ; $P_1 = 1$ bar; $P_2 = 27$ bar; $\eta_c = 0.9$; $T_{2'} - T_2 = ?$



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 300\left(\frac{27}{1}\right)^{\frac{0.3}{1.3}}$$

$$= 300 \times 2.14 = 642 \text{ K}$$

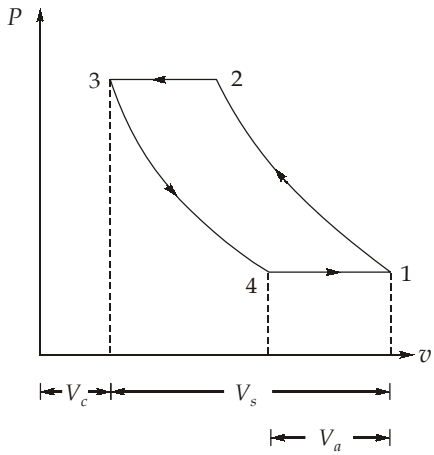
$$\eta_c = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{642 - 300}{T_{2'} - 300} = 0.9$$

$$T_{2'} = \frac{342}{0.9} + 300 = 680 \text{ K}$$

$$T_{2'} - T_2 = 680 - 642 = 38 \text{ K}$$

48. (d)

Given : $P_1 = 2$ bar; $P_2 = 16$ bar; Number of cylinder (K) = 2



$$L = D = 40 \text{ mm} = 0.04 \text{ m}$$

$$C = \frac{V_c}{V_s} = 0.05;\ N = 1800 \text{ RPM};\ \gamma = 1.5$$

$$\eta_v = 1 + c - c\left(\frac{P_H}{P_L}\right)^{\frac{1}{\gamma}} = 1 + 0.05 - 0.05\left(\frac{16}{2}\right)^{\frac{1}{1.5}} = 0.85$$

and

$$\eta_a = \eta_v \times \frac{\pi}{4} D^2 L \times \frac{N}{60} \times K$$

$$V_a = 0.85 \frac{\pi}{4} (0.04)^2 \times 0.04 \times \frac{1800}{60} \times 2$$

$$= 0.85 \times \frac{\pi}{4} \times (0.04)^2 \times 0.04 \times 60$$

$$= 0.0256 \text{ m}^3/\text{sec} = 0.1538 \simeq 0.154 \text{ m}^3/\text{min}$$

49. (d)

50. (d)

Hunting can be minimised, eliminated or avoided by :

(i) Correct arrangement of suction piping.

(ii) Correct location of the remote bulb.

(iii) Correct location of external equaliser line connection

(iv) Use of the recommended power assembly charge for the range of Thermo-expansion valve operation.

51. (b)

Flooded type is always filled with liquid, whereas the amount of liquid present in the dry expansion evaporator will vary with load on the evaporator. When the load on the evaporator is light, the amount of liquid in the evaporator is small. As the load on the evaporator increases, the amount of liquid in the evaporator increases to accommodate the greater load.

52. (c)

Given : DBT = 27°C, $P_{vs} = 3570\ \text{N/m}^2$, $h_{fg} = 2400\ \text{kJ/kg}$, DPT = 15°C, $P_v = 1785\ \text{N/m}^2$,

$C_{pa} = 1\ \text{kJ/kg}$,

$$\text{Total pressure } P = 99.96\ \text{kPa},\ C_{pw} = 4.20\ \text{kJ/kg}$$

$$\text{Relative humidity, } \phi = \frac{P_v}{P_{vs}} = \frac{1785}{3570} = 0.5$$

The enthalpy per kg of water vapour (h_{wv})

$$= C_{pw} \times \text{DBT} + h_{fg} = 4.2 \times 27 + 2400$$

$$= 2513.4\ \text{kJ/kg of water vapour}$$

$$\simeq 2513\ \text{kJ/kg of water vapour}$$

53. (a)

Given : DBT = 27°C, $P_{vs} = 3570\ \text{N/m}^2$, $h_{fg} = 2400\ \text{kJ/kg}$, DPT = 15°C, $P_v = 1785\ \text{N/m}^2$,

$C_{pa} = 1\ \text{kJ/kg}$,

$$\text{Total pressure } P = 99.96\ \text{kPa},\ C_{pw} = 4.20\ \text{kJ/kg}$$

$$\text{Specific humidty, } \omega = 0.622\frac{P_v}{P - P_v} = 0.622 \times \frac{1785}{99960 - 1785}$$

$$\Rightarrow \quad \omega = 0.622 \times \frac{1785}{55 \times 1785}$$

$$\therefore \quad \omega = 0.0113\ \text{kg/kg of dry air}$$

Enthalpy of mixture per kg of dry air = h_{da}

$$h_{da} = h_a + \omega h_{wv}$$
$$= C_p \text{DBT} + \omega h_{wv} = 1 \times 27 + 0.0113 \times 2513.4$$
$$= 55.4 \text{ kJ/kg of dry air}$$
$$\simeq 55 \text{ kJ/kg of dry air}$$

54. (a)

From Psychrometric chart,

$h_a = 83$ kJ/kg, DBT = 36°C, RH = 0.5

$h_b = 45$ kJ/kg, DBT = 21°C, RH = 0.55

$h_c = 60$ kJ/kg ; $V_{sa} = 0.9$ m^3/kg ; $V_{sb} = 0.85$ m^3/kg

$$\text{Volume} = 3 \text{ m}^3/\text{sec} = 3 \times 60 = 180 \text{ m}^3/\text{min}$$

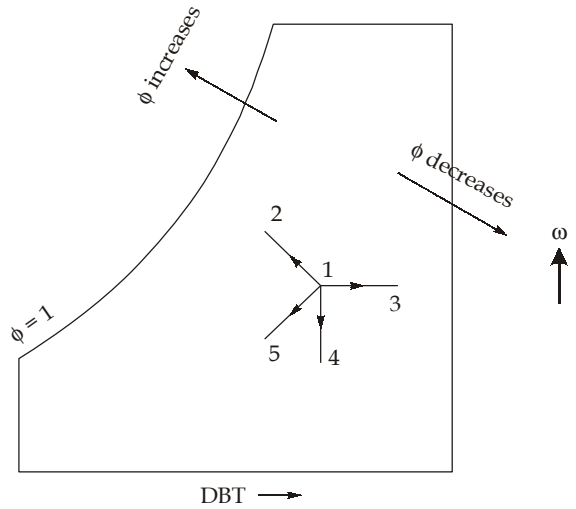
$$\text{The mass of air circulated per minute} = \frac{180}{0.9} = 200 \text{ kg/min}$$

$$\text{Sensible heat removed} = \dot{m}(h_c - h_b) = 200(60 - 45) = 200 \times 15 = 3000 \text{ kJ/min}$$

$$\text{Latent heat removed} = \dot{m}(h_a - h_c) = 200(83 - 60) = 200 \times 23 = 4600 \text{ kJ/min}$$

$$\text{SHR} = \frac{\text{Sensible Heat}}{\text{Total heat}} = \frac{3000}{3000 + 4600} = \frac{3000}{7600}$$
$$= 0.395$$

55. (c)



Process 1-2 → Cooling with humidification (ϕ-increases, ω-increases)

Process 1-3 → Sensible heating (ϕ-decreases and ω = constant)

Process 1-4 → Dehumidification (ϕ-decreases, ω-decreases)

Process 1-5 → Cooling with dehumidification (ϕ-increases, ω-decreases)

56. (b)

On psychrometry chart, specific volumes lines have highest inclination from than enthalpy lines and wet bulb temperature line. enthalpy lines and wet bulb temperature line are not exactly same inclination. Wet Bulb temp line have inclination slightly higher than enthalpy line.

57. (b)

Given : DBT = 31°C, DBT = 25°C, $RH_1 = 0.5$, $RH_2 = ?$; $P_{vs_1} = 0.0450$ bar, $P_{vs_2} = 0.0317$ bar

According to Question $\omega_1 = \omega_2$

$$\phi = \frac{P_{v1}}{P_{vs_1}}$$

$$P_{v1} = 0.5 \times 0.0450$$

$$= 0.0225 \text{ bar}$$

$$\omega_1 = \omega_2$$

$$\omega = 0.622\frac{P_v}{P - P_v}$$

P = constant and ω = constant

So, P_v = also remain constant from above expression.

$$P_{v1} = P_{v2} = 0.0225 \text{ bar}$$

$$\phi_2 = \frac{P_{v2}}{P_{vs_2}} = \frac{0.0225}{0.0317}$$

$$\phi_2 = 0.7097 \simeq 0.71$$

58. (a)

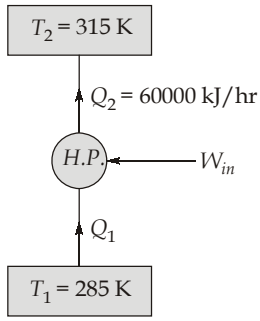
T_1 = 12°C = 12 + 273 = 285 K

T_2 = 42°C = 42 + 273 = 315 K

Actual COP = $0.6(\text{COP})_{\text{rev carnot}}$

$$\text{COP} = \frac{T_2}{T_2 - T_1} \quad \text{[Rev. carnot cycle]}$$

$$\text{COP} = \frac{315}{315 - 285} = \frac{315}{30} = 10.5$$



$$\text{Actual COP} = 10.5 \times 0.6 = 6.3$$

$$\text{COP} = \frac{\text{Heating capacity}}{\text{Power input}}$$

$$P_{in} = \frac{\text{Heating capacity}}{\text{COP}} = \frac{60000}{6.3} \text{ kJ/hr}$$

$$P_{in} = \frac{60000}{6.3} \times \frac{1}{3600} (\text{kJ/s or kW})$$

$$P_{in} = \frac{100}{37.8} = 2.64 \text{ kW}$$

59. (d)

The ASHRAE precisely defines the parameters to be considered in an indoor environment and they are :

1. Indoor air temperature (IAT) and mean radiant temperature (MRT).
2. Indoor relative humidity (IRH).
3. Outdoor ventilation rate provided.
4. Air cleanliness.
5. Air path and movement.
6. Sound level.
7. Pressure difference between space and surroundings.

60. (a)

- Heat store in the body is positive when the temperature of body rises and is negative when the temperature of the body falls below equilibrium temperature.
- In hot summer, temperature of body rises so, heat store in the body is always positive.
- Most comfort feeling exists when heat store in body is zero. ($H_s = 0$).

61. (b)

Given:- Bypass factor of an air conditioner $BF_n = 0.3$

$$BF_1 = 0.9$$

$$n = \text{Number of depth}$$

Bypass factor of n depth coil is

$$BF_n = (BF_1)^n$$

$$0.3 = (0.9)^n$$

Taking natural log both sides,

$$\ln 0.3 = n \ln (0.9)$$

$$\ln \frac{3}{10} = \ln \frac{9}{10}$$

$$\frac{\ln 3 - \ln 10}{\ln 9 - \ln 10} = n$$

$$\frac{1.099-2.303}{2\times1.099-2.303} = n$$

$$\Rightarrow \quad n = 11.47$$

62. (c)

Given:

Inlet :

$$h_1 = 55 \text{ kJ/kg}$$
$$\text{DBT} = 25°\text{C}$$
$$v = 0.85 \text{ m}^3$$
$$V = 1275 \text{ m}^3/\text{min}$$

Exit :

$$h_2 = 25 \text{ kJ/kg}$$
$$\text{DBT} = 12°\text{C}$$

Total mass flow rate passing through the cooling coil

$$= \frac{1275}{0.85} = 1500 \text{ kg/min}$$

$$\dot{m} = 25 \text{ kg/sec}$$

$$\text{Total cooling load (TCL)} = 25(h_1 - h_2) = 25(55-25) = 25\times30 = 750 \text{ kW}$$

$$= \frac{750}{3.5} = 214.285 \simeq 214.3 \text{ Tons}$$

63. (c)

The basic components of the steam jet refrigeration system are : Steam ejector, Flash chamber fitted with spray nozzle and baffles, steam condensers, chilled water circulating pump, condensate removal pump and air removal ejector for steam condenser.

64. (b)

$$\text{RE} = 1 \text{ Ton} = 3.5 \text{ kW}$$

$$W_{in} = 0.7 \text{ kW}$$

$$\text{COP}_R = \frac{\text{RE}}{W_{in}} = \frac{3.5}{0.7} = 5$$

$$\text{COP}_{HP} = \text{COP}_R + 1 = 5 + 1$$

$$= 6$$

65. (d)

Given :

COP $R_1 \rightarrow c_1 = 3$

$R_2 \quad c_2 = 4$

$R_3 \quad c_3 = 5$

$$\text{Combined COP of cascade system} = \frac{c_1 c_2 c_3}{1 + c_1 + c_2 + c_3 + c_1 c_2 + c_2 c_3 + c_3 c_1}$$

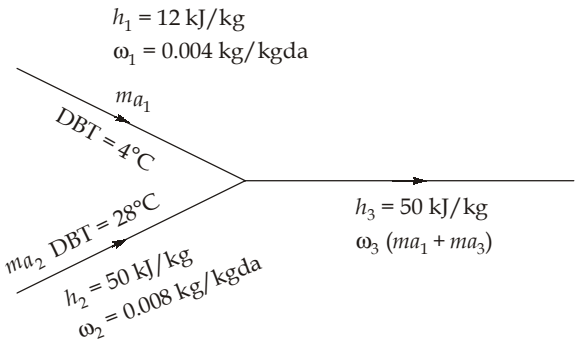
$$= \frac{3 \times 4 \times 5}{1 + 3 + 4 + 5 + 3 \times 4 + 4 \times 5 + 5 \times 3}$$

$$= \frac{60}{1 + 12 + 12 + 20 + 15} = \frac{60}{60}$$

$$C = 1$$

66. (a)

Given :



Mass of dry air per unit mass of moist air ,

$$ma_1 = \frac{1}{1 + \omega_1} = \frac{1}{1 + 0.004} = 0.996 \text{ kg}$$

Mass of dry air per two-units of moist air ,

$$ma_2 = \frac{2}{1 + \omega_2} = \frac{2}{1 + 0.008} = 1.984 \text{ kg}$$

After mixing:

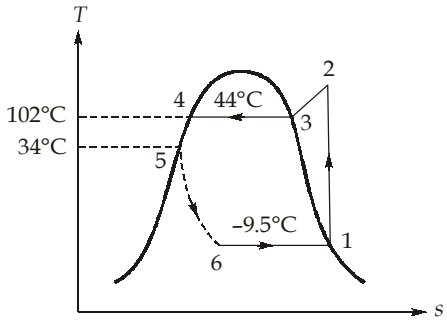
$$ma_1 h_1 + ma_2 h_2 = (ma_1 + ma_2) h_3$$

$$h_3 = \frac{0.996 \times 12 + 1.984 \times 50}{0.996 + 1.984}$$

$$h_3 = 37.3 \text{ kJ/kgda}$$

67. (c)

Given : From table,



$h_1 = 460$ kJ/kg; $h_2 = 540$ kJ/kg; $h_4 = 135$ kJ/kg; $h_5 = h_6$

$$h_5 = h_4 - C_{pl}(T_4 - T_5) = 135 - 1.6(44 - 34)$$

$$= 135 - 1.6 \times 10 = 135 - 16$$

$$= 119 \text{ kJ/kg}$$

$$\text{COP} = \frac{h_1 - h_6}{h_2 - h_1} = \frac{460 - 119}{540 - 460} = \frac{341}{80}$$

$$\text{COP} = 4.26$$

68. (b)

$$\text{Heat rejection factor} = \frac{\text{Heat rejected in condenser}}{\text{Refrigeration effect}}$$

$$\text{Heat rejected in condenser} = 1.2 \times 2100 = 2520 \text{ kJ/min}$$

$$\text{COP} = \frac{\text{Refrigeration effect}}{w_{in}} = \frac{2100}{420} = 5$$

69. (d)

- High boiling point of the refrigerant at atmospheric pressure. reduces the capacity of the system and lowers the operating cost.
- Refrigeration must have a freezing point well below the operating evaporator temparature.

Latent Heat of Refrtigerant : High latent heat of refrigerant at evaporator temperature is desirable because the refrigerating effect per kg of refrigerant will be high. The weight of refrigeration required to be circulated in the system per ton of refrigeration will be low if the latent heat of the refrigerant is high. This also reduce the initial cost of the refrigerant used in the system. The refrigerants with low latent heats can be used by increasing the size, number and speed of the cylinder if other properties are satisfactory.

70. (c)

- The high pressure steam supplied from the boiler to drive the ejector is termed as motive steam.
- The low pressure vapour drawn from the flash chamber/evaporator by entrainment is termed as suction vapour.

71. (b)

The process in which high velocity motive steam transfer momentum to the low velocity water vapour generated in the flash chamber is known as entrainment.

72. (c)

- Nozzle efficiency: It is defined as the ratio of actual enthalpy drop to isentropic enthalpy drop of the motive steam passing through the nozzle.
- Entrainment Efficiency: During the entrainment steam will lose some energy, and the process of entrainment is very inefficient so to taken into account by a factor known as entrainment efficiency.

73. (c)

The nozzles are of converging, diverging or converging-diverging type as per the design. An efficient nozzle is designed to have higher velocity, greater mass flow and minimum inlet losses.

74. (c)

Saturation temperature of the water vapour in the moist air is known as dew point temperature.

75. (d)

The assumptions are made in analysis the magnitude of refrigeration effect and work done are:

1. Heat absorption and rejection take place only at the junctions.
2. Thermal conductivity and electrical resistance do not change over the temperature range under consideration.
3. No energy exchange between the conductor through space separating them.
4. Thomson coefficient is negligible.

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