India's Best Institute for IES, GATE & PSUs

Detailed Solutions

ESE-2022 Mains Test Series

Mechanical Engineering Test No : 2

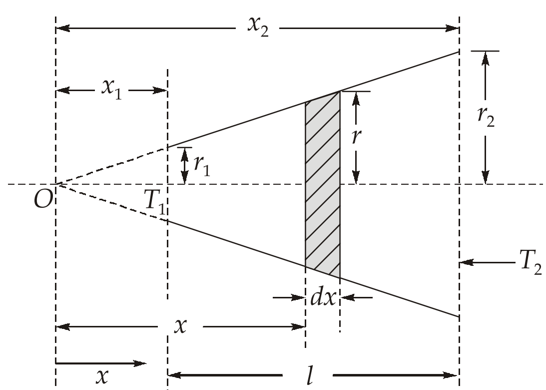
Section A : Heat Transfer + Theory of Machines

Section B : Thermodynamics-1 + Refrigeration and Air Conditioning-1

Section : A

1. (a)

Consider a frustum of constant thermal conductivity (k) and apex at O. An element of thickness dx at a distance x from the apex is taken.



From similar triangle property,

$$r = \left(\frac{r_2 - r_1}{l}\right)x = Cx$$

$\therefore$ Area of element, $A = \pi C^2 x^2$

Heat transfer rate through this element

$$Q = -kA\frac{dT}{dx}$$

$$\frac{dx}{A} = -\frac{k}{Q}\cdot dT$$

Integrating both sides we get

$$\int_{x_1}^{x_2}\frac{dx}{\pi C^2 x^2} = -\frac{k}{Q}\int_{T_1}^{T_2} dT$$

or,
$$\frac{1}{\pi C^2}\left[\frac{1}{x_1}-\frac{1}{x_2}\right] = \frac{k}{Q}(T_1 - T_2)$$

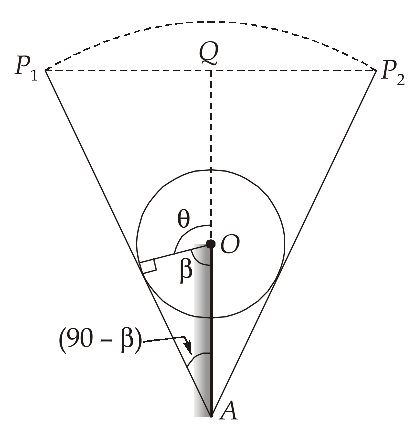
Also,
$$r_1 = Cx_1 \text{ and } r_2 = Cx_2$$

∴
$$\frac{1}{\pi C^2}\left[\frac{C}{r_1}-\frac{C}{r_2}\right] = \frac{k}{Q}(T_1 - T_2)$$

∴ Heat transfer rate,
$$Q = \frac{T_1 - T_2}{\frac{1}{\pi k C}\left(\frac{1}{r_1}-\frac{1}{r_2}\right)} = \frac{T_1 - T_2}{\frac{l}{\pi k(r_2 - r_1)}\left(\frac{1}{r_1}-\frac{1}{r_2}\right)} \quad \left[\because C = \frac{r_2 - r_1}{l}\right]$$

or
$$Q = \frac{T_1 - T_2}{\left(\frac{l}{\pi k r_1 r_2}\right)}$$
Answer

1. (b)



Where,
P_1P_2 = Length of stroke (L_s)
AP_1 = Length of slotted lever (L)

For quick return motion of crank and slotted lever type,

$$\frac{V_{s\max,(\text{return})}}{V_{s\max,(\text{forward})}} = \frac{c+r}{c-r} = 5$$

where, r = Crank radius

c = Distance between fixed centres

So, $$\frac{c+r}{c-r} = 5$$

or $$c + r = 5c - 5r$$

$$6r = 4c$$

$$\Rightarrow \quad \frac{r}{c} = \frac{2}{3}$$

$$\therefore \quad \cos\beta = \frac{2}{3}$$

$$\Rightarrow \quad \beta = 48.189°$$

Ratio of times of cutting and return strokes

$$= \frac{2\theta}{2\beta} = \frac{360 - 2\beta}{2\beta} = \frac{360}{2 \times 48.189} - 1 = 2.735$$ **Answer**

Now, Length of stroke, L_s = 300 mm

$$\therefore \quad \text{Length of slotted lever, } L = \frac{L_s}{2 \times \sin(90 - \beta)} = \frac{300}{2 \times \sin(90 - 48.189)}$$

$$= 224.996$$

$$L \simeq 225 \text{ mm}$$ **Answer**

Maximum cutting velocity per second,

$$V_{smax} = wr \times \frac{L}{c+r} = \left(\frac{2\pi \times 30}{60}\right) \times \frac{225}{1.5r + r} \times r$$

$$= 3.14 \times \frac{225}{2.5} = 282.6 \text{ mm/s}$$ **Answer**

1. (c)

Given: r = 10 cm, $q_g = 4 \times 10^6$ W/m^3, T_a = 150°C, h = 750 W/m^{2}K, k = 50 W/mK

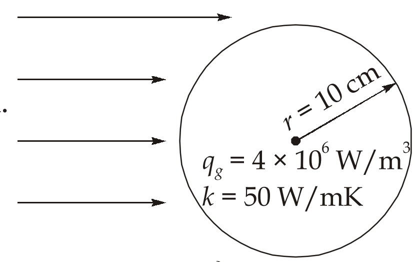
Assumption:

(i) Steady state heat transfer condition.

(ii) Heat conduction takes place only in radial direction.

(iii) Constant thermal conductivity.

(iv) Negligible radiation heat transfer.



$$h = 750 \text{ W/m}^2\text{K}$$

$$T_\infty = 150°\text{C}$$

Now, heat generated in the sphere is convected over the surface.

i.e. $$q_g \times \frac{4}{3}\pi r^3 = h \times 4\pi r^2 \times (T_w - T_a)$$

or, $$q_g \times \frac{r}{3} = h \times (T_w - T_a)$$

∴ $$T_w = \frac{q_g r}{3h} + T_a = \frac{4\times10^6 \times 0.1}{3\times750} + 150$$

$$= 327.777°C$$

Maximum temperature occurs at centre i.e. at $r = 0$; its value is prescribed by the relation.

$$T_{max} = T_w + \frac{q_g R^2}{6k}$$

$$T_{max} = 327.327.777 + \frac{4\times10^6\times0.1^2}{6\times50}$$

∴ $$T_{max} = 461.110°C$$ ....Answer(i)

Temperature at any radius 'r' can be worked out from the relation,

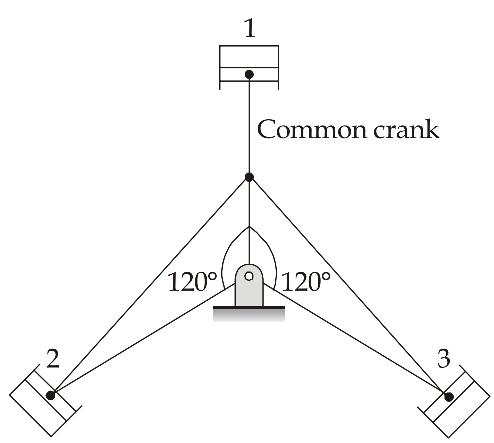
$$\frac{T - T_w}{T_{max} - T_w} = 1 - \left(\frac{r}{R}\right)^2$$

$$\frac{T - 327.777}{461.11 - 327.777} = 1 - \left(\frac{0.08}{0.1}\right)^2$$

$$T = 375.776°C$$ ....Answer(ii)

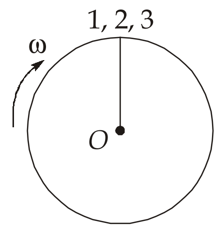
1. (d)

Given: Length of stroke = 180 mm, Crank radius, $r = \frac{0.180}{2} = 0.09$ m, Length of connecting rod, $l = 0.32$ m, Mass of reciprocating parts, $m = 3.2$ kg, $N = 2000$ rpm, $n = \frac{l}{r} = \frac{0.32}{0.9} = 3.55$

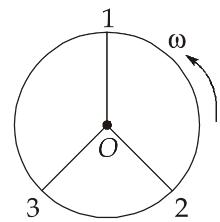


The position of three cylinders is shown in figure.

Primary cranks: The primary direct and reverse crank positions are shown below.



Primary direct



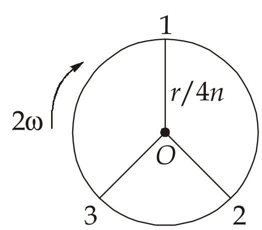
Primary reverse

Since primary reverse cranks form a balanced system and therefore, unbalanced primary force is due to direct cranks only and is given by,

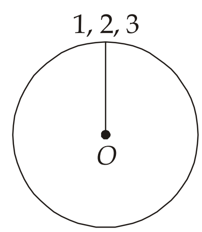
$$F_P = 3\times\frac{m}{2}r\omega^2 = 3\times\frac{3.2}{2}\times 0.09\times\left(\frac{2\pi\times 2000}{60}\right)^2$$

$$= 18.93 \text{ kN}$$ **Answer**

Secondary cranks: The secondary direct and reverse cranks positions are shown below.



Secondary direct



Secondary reverse

Since secondary direct cranks forms a balanced system and therefore, unbalanced secondary force is due to only secondary reverse cranks and is given by,

$$F_S = 3\times\frac{m}{2}\left(\frac{r}{4n}\right)\times(2\omega)^2 = 3\times\frac{3.2\times 0.09}{2\times 3.55}\times\left(\frac{2\pi\times 2000}{60}\right)^2$$

$$= 5.33 \text{ kN}$$ **Answer**

1. (e)

Given: Mean radius of egg, r = 0.025 m, Initial temperature of egg, t_i = 20°C, k = 12 W/mK, h = 125 W/m^{2}K, C = 2 kJ/kgK, ρ = 1200 kg/m^3.

Consider egg to be of spherical shape,

$$\text{Characteristic length, } l_c = \frac{V}{A} = \frac{\frac{4}{3}\pi r^3}{4\pi r^2} = \frac{r}{3}$$

$$\therefore \quad l_c = \frac{0.025}{3} = 8.333\times 10^{-3} \text{ m}$$

$$\text{Biot number, Bi} = \frac{hl_c}{k} = \frac{125 \times 8.333 \times 10^{-3}}{12} = 0.0868$$

Since Bi < 0.1, the lumped parameter model can be adopted. Therefore

$$\frac{t - t_a}{t_i - t_a} = \exp\left(-\frac{hA\tau}{\rho VC}\right)$$

Let, $t_a = 100°C$; temperature of boiling water

and $$\frac{hA\tau}{\rho VC} = \frac{125 \times 3 \times 6 \times 60}{1200 \times 0.025 \times 2000} = 2.25$$

Now, $$\frac{t - 100}{20 - 100} = \exp(-2.25)$$

or $t = 91.568°C$

Now, we have to find time τ′ for temperature values,

$$t_i' = 10°C,\ t_a = 100°C,\ t = 91.568°C$$

∴ $$\frac{91.568 - 100}{10 - 100} = \exp\left(\frac{-125 \times 3 \times \tau'}{1200 \times 0.025 \times 2000}\right)$$

or, $-2.367 = 6.25 \times 10^{-3} \times \tau'$

or, $\tau' = 6.312$ minutes **Answer**

2. (a)

Given: Number of tubes, $N = 100$, Inner diameter, $d_i = 25$ mm, Tube thickness, $t = 2$ mm, $\dot{m}_w = 8\ \text{kg/s}$, $T_{c1} = 25°C$, $T_{c2} = 75°C$, $T_{h1} = T_{h2} = 100°C$, $h_o = 5000\ \text{W/m}^2\text{K}$, $R_{fi} = 0.0002\ \text{m}^2\text{K/W}$.

Now, Heat transfer rate to water,

$$Q = \dot{m}_c c_{pc}\ (T_{c2} - T_{c1}) = 8 \times 4175\ (75 - 25)\ \text{W}$$

∴ $Q = 1670$ kW

$$\text{Flow Reynolds number, Re} = \frac{\rho V d_i}{\mu}$$

$$\text{Mass flow rate inside tube, } \dot{m}_c = \rho \times \frac{\pi}{4} \times d_i^2 \times V \times N$$

or $$V = \frac{\dot{m}_c}{\rho \times \frac{\pi}{4} \times d_i^2 \times N} = \frac{8}{998 \times \frac{\pi}{4} \times 0.025^2 \times 100} = 0.163\ \text{m/s}$$

$$\therefore \qquad \text{Re} = \frac{\rho V d_i}{\mu} = \frac{998 \times 0.163 \times 0.025}{55 \times 10^{-5}} = 7394.272 \text{ (Flow is turbulent)}$$

$$\text{Prandtl number, Pr} = \frac{55 \times 10^{-5} \times 4175}{0.65} = 3.532$$

For turbulent flow,

$$\text{Nu} = \frac{h_i d_i}{k} = 0.023 \times (\text{Re})^{0.8} \times (\text{Pr})^{0.4}$$

$$\therefore \qquad \frac{h_i \times 0.025}{0.65} = 0.023 \times (7394.272)^{0.8} \times (3.532)^{0.4}$$

$$h_i = 1233.178 \text{ W/m}^2\text{K}$$

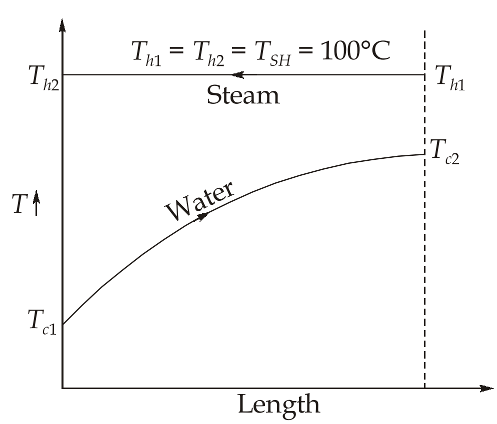
Overall heat transfer coefficient based on inner area.

$$\frac{1}{U_i} = \frac{1}{h_i} + R_{F_i} + \frac{r_i}{r_o} \times \frac{1}{h_o}$$

$$= \frac{1}{1233.178} + 0.002 + \frac{0.0125}{0.0145} \times \frac{1}{5000} = 1.183 \times 10^{-3}$$

$$\therefore \qquad U_i = 845.308 \text{ W/m}^2\text{K}$$ **Answer**

Since, the temperature of one of the fluid remains constant during the flow passage. Both parallel and counter flow would give the same values of LMTD. Assuming counter flow arrangement



$$\theta_m = \frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)}$$

$$\theta_1 = T_{h2} - T_{c1} = 100 - 75 = 25°\text{C}$$

$$\theta_2 = T_{h1} - T_{c2} = 100 - 25 = 75°\text{C}$$

$\therefore \quad \theta_m = \dfrac{25-75}{\ln\left(\dfrac{25}{75}\right)} = 45.511°C$

Now, $\quad Q = U_i A_i \theta_m$

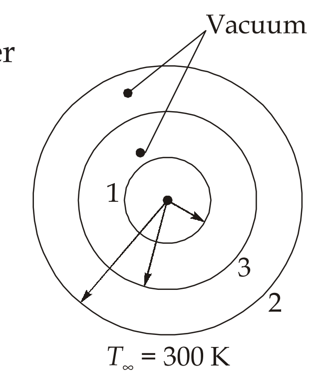
$\therefore \quad A_i = \dfrac{Q}{U_i \theta_m} = \dfrac{1670 \times 10^3}{845.308 \times 45.511} = 43.409\ m^2$

or $\quad N \times \pi d_i \times L = 43.409$

$\therefore \quad L = \dfrac{43.409}{100 \times \pi \times 0.025} = 5.529\ m$ **Answer**

2. (b)

Let the suffix 1, 2 and 3 refers to the inner cylinder, outer cylinder and cylindrical shield. Then



$$d_3 = \frac{d_1 + d_2}{2} = \frac{10+20}{2} = 15\ cm$$

Heat flow from inner cylinder to shield.

$$Q_{13} = (F_g)_{13} A_1 \sigma_b (T_1^4 - T_3^4)$$

Where, $\quad (F_g)_{13} = \dfrac{1}{\dfrac{1}{\varepsilon_1} + \left(\dfrac{1}{\varepsilon_3} - 1\right)\dfrac{A_1}{A_3}} = \dfrac{1}{\dfrac{1}{0.2} + \left(\dfrac{1}{0.3} - 1\right) \times \dfrac{10}{15}}$

$\because \quad \dfrac{A_1}{A_3} = \dfrac{\pi d_1 l}{\pi d_3 l} = \dfrac{10}{15}, \quad F_{13} = 1$

$\therefore \quad (F_g)_{13} = 0.152$

and $\quad Q_{13} = 0.152 \times A_1 \times \sigma_b (1200^4 - T_3^4) \quad$ ... (i)

Likewise, the heat flow from the shield to the outer cylinder is

$$Q_{32} = (F_g)_{32} A_3 \sigma_b (T_3^4 - T_2^4)$$

$$(F_g)_{32} = \frac{1}{\dfrac{1}{\varepsilon_3} + \left(\dfrac{1}{\varepsilon_2} - 1\right)\dfrac{A_3}{A_2}}$$

$\because \quad \dfrac{A_3}{A_2} = \dfrac{\pi d_3 l}{\pi d_2 l} = \dfrac{15}{20}, \quad F_{32} = 1$

$\therefore \quad (F_g)_{32} = \dfrac{1}{\dfrac{1}{0.5} + \left(\dfrac{1}{0.3} - 1\right) \times \dfrac{15}{20}} = 0.266$

and $$Q_{32} = 0.266 \times A_3 \sigma_b \left(T_3^4 - 320^4\right)$$

Under steady state conditions,

$$Q_{13} = Q_{32}$$

$$0.152 \times A_1 \times \sigma_b\,(1200^4 - T_3^4) = 0.266 \times A_3 \times \sigma_b \left(T_3^4 - 320^4\right)$$

$$0.152 \times (1200^4 - T_3^4) = 0.266 \times \frac{A_3}{A_1} \times \left(T_3^4 - 320^4\right)$$

or, $$1200^4 - T_3^4 = \frac{0.266 \times 15}{0.152 \times 10} \times \left(T_3^2 - 320^4\right)$$

$$1200^4 - T_3^4 = 2.625 T_3^4 - 2.625 \times 320^4$$

$\therefore$ $$T_3 = \left[\frac{1200^4 + 2.625 \times 320^4}{1 + 2.625}\right]^{1/4} = 872.541 \text{ K}$$ **Answer**

Heat loss per metre length, $Q_{13} = 0.152 \times \pi \times 0.1 \times 1 \times (5.67 \times 10^{-8})\,(1200^4 - 872.541^4)$

$$Q_{13} = 4043.269 \text{ W}$$ **Answer**

For steady flow, this heat is lost to the surrounding both by convection and radiation from the outer surface of outside cylinder.

$\therefore$ $$4043.269 = h_c A_2\,(T_2 - T_a) + \sigma_b \in_2 A_2 (T_2^4 - T_a^4)$$

$$4043.269 = h_c(\pi \times 0.2 \times 1)(320 - 300) + 5.67 \times 10^{-8} \times 0.3 \times (\pi \times 0.2 \times 1)(320^4 - 300^4)$$

$$4043.269 = 12.56\, h_c + 25.485$$

$$h_c = 319.887 \text{ W/m}^2\text{K}$$ **Answer**

2. (c)

Given: $m = 16$ kg, $l = 1.5$ m, $d = 0.015$ m, $e = 0.5 \times 10^{-3}$ m, $E = 210 \times 10^9$ Pa, $\sigma_{per} = 80 \times 10^6$ Pa.

As the shaft is held in long bearings, it may be assumed to be fixed at the ends.

$\therefore$ Static deflection, $$\Delta = \frac{Mgl^3}{192EI} = \frac{16 \times 9.81 \times 1.5^3}{192 \times 210 \times 10^9 \times \frac{\pi}{64} \times (0.015)^4}$$

$$= 5.289 \times 10^{-3} \text{ m}$$

$\therefore$ $$\omega_n = \sqrt{\frac{g}{\Delta}} = \sqrt{\frac{9.81}{5.289 \times 10^{-3}}} = 43.067 \text{ rad/s}$$

or $$N_C = \frac{43.067 \times 60}{2 \times 3.14} = 411.468 \text{ rpm}$$ **Answer**

Now, $$\frac{M}{I} = \frac{\sigma_{per}}{y}$$

or, $$\frac{\frac{w_1 l}{8}}{\frac{\pi}{64} \times d^4} = \frac{\sigma_{per}}{d/2}$$

$$\frac{\frac{w_1 \times 1.5}{8}}{\frac{\pi}{64} \times 0.015^4} = \frac{80 \times 10^6}{\frac{0.015}{2}}$$

$\Rightarrow$ $$w_1 = \frac{2 \times 80 \times 10^6 \times \pi \times 0.015^4 \times 8}{0.015 \times 64 \times 1.5} = 141.3\text{ N}$$

Additional deflection due to the load,

$$y = \frac{w_1}{w} \times \Delta = \frac{141.3}{16 \times 9.81} \times 5.289 \times 10^{-3}$$

$$= 4.761 \times 10^{-3}\text{ m}$$

or, $$\frac{\pm e}{\left(\frac{N_C}{N}\right)^2 - 1} = 4.761 \times 10^{-3}$$

or, $$\left(\frac{411.468}{N}\right)^2 - 1 = \pm 0.105$$

Taking (+) ve sign, we get,

$$N_1 = \frac{411.468}{\sqrt{1.105}} \Rightarrow N_1 = 391.43\text{ rpm}$$

Taking (–)ve sign, we get,

$$N_2 = \frac{411.468}{\sqrt{0.895}} \Rightarrow N_2 = 434.93\text{ rpm}$$

Thus, the range of unsafe speed is from N_1 to N_2 i.e. 391.43 rpm to 434.93 rpm.

Answer

3. (a)

Given: d = 20 mm, l = 80 mm, k = 27 W/mK, T_o = 165°C, T_∞ = 27°C, h = 25 W/m^{2}K

Assumption:

1. Steady state condition.
2. Fin material having constant thermal conductivity
3. No internal heat generation

For a short fin with insulated tip,

$$Q = kA_c m(T_o - T_a)\tanh ml$$

$$m = \sqrt{\frac{4h}{kd}} = \sqrt{\frac{4 \times 25}{27 \times 0.02}} = 13.608 \text{ m}^{-1}$$

and $ml = 13.608 \times 0.08 = 1.088$

$\therefore$ $$Q = 27 \times \frac{\pi}{4}(0.02)^2 \times 13.608(165 - 27)\tan h(1.088)$$

$$Q = 12.675 \text{ W}$$ **Answer**

Now, let d' be the new diameter. Then for the same material volume.

$$2 \times \frac{\pi}{4} d'^2 \times l = \frac{\pi}{4} d^2 \times l$$

$\therefore$ $$2 \times \frac{\pi}{4} d'^2 \times 0.08 = \frac{\pi}{4}(0.02)^2 \times 0.08$$

$\therefore$ $$d' = 0.0141 \text{ m}$$

Then, $$m' = \sqrt{\frac{4h}{kd'}} = \sqrt{\frac{4 \times 25}{27 \times 0.0141}} = 16.207 \text{ m}^{-1}$$

$\therefore$ $m'l = 16.207 \times 0.08 = 1.296$

Heat loss for two fins, $Q' = 2 \times kA_c m' (T_o - T_a)\tan hm'l$

$$Q' = 2 \times 27 \times \frac{\pi}{4}(0.0141)^2 \times 16.207(165 - 27)\tan h(1.296)$$

$$Q' = 16.222 \text{ W}$$

$\therefore$ Percentage increase in heat flow

$$\Delta Q = \frac{Q' - Q}{Q} \times 100 = \frac{16.222 - 12.675}{12.675} \times 100$$

$$= 27.984\%$$

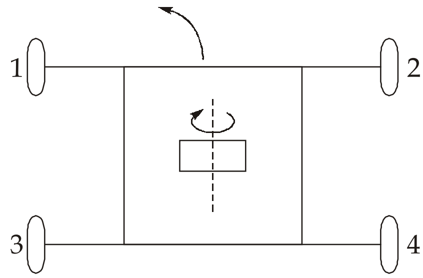
Comment: A thinner or low sectional area fin is a better choice.

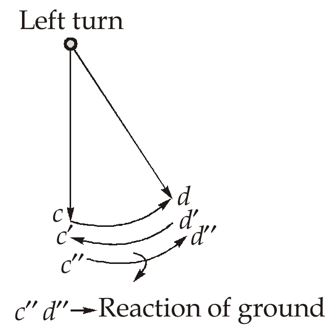
3. (b)

Given: $M = 2000$ kg, $m = 150$ kg, $w = 1.5$ m, $k = 0.14$ m, $b = 2.5$ m, $I_w = 0.7$ kg/m^2,

$r = \frac{1}{2} = 0.5$ m, $R = 100$ m, $v = \frac{72 \times 1000}{3600} = 20$ m/s

(i) Car turning left





(a) Reaction due to weight

$$\text{Total weight} = 2000 \times 9.81$$
$$= 19620 \text{ N}$$

$$\therefore \quad R_{w1,2} = \left(19620 \times \frac{1.5}{2.5}\right) \times \frac{1}{2} = 5886(\text{Upwards})$$

$$R_{w3,4} = \left(19620 \times \frac{1}{2.5}\right) \times \frac{1}{2} = 3924\text{ N}(\text{Upwards})$$

(b) Reaction due to gyroscopic couples

$$C_w = 4I_w \frac{V^2}{rR} = 4 \times 0.7 \times \frac{20^2}{0.5 \times 100} = 22.4\text{ Nm}$$

For outer wheels, $R'_{G2,4} = \frac{C_w}{2w} = \frac{22.4}{2 \times 1.5} = 7.466\text{N}$ Upwards

For inner wheel, $R'_{G1,3} = 7.466$ N (Downwards)

$$I_E = mk^2 = 150 \times 0.14^2 = 2.94 \text{ kgm}^2$$

$$C_E = I_E G w_w \omega_P = I_E \cdot G \cdot \frac{V^2}{rR}$$

$$= 2.94 \times 5 \times \frac{20^2}{0.5 \times 100} = 117.6\text{ Nm}$$

For front wheels, $R''_{G1,2} = \frac{C_E}{2w} = \frac{117.6}{2 \times 2.5} = 23.52\text{ N}$ (Upwards)

For rear wheels, $R''_{G3,4} = 23.52$ N (Downwards)

(c) Reaction due to centrifugal couple,

$$C_C = \frac{Mv^2 \times h}{R} = 2000 \times \frac{20^2}{100} \times 0.6 = 4800\text{ Nm}$$

For outer wheels, $R_{C2,4} = \frac{C_c}{2\omega} = \frac{4800}{2 \times 1.5} = 1600\text{ N}$ (Upwards)

For rear wheels, $R_{C1,3} = 1600\text{ N}$ (Downwards)

Therefore, reaction on wheels,

$$R = R_w + R'_G + R''_G + R_C$$

$$R_1 = 5886 - 7.466 + 23.52 - 1600 = 4302.054\text{ N}$$

$$R_2 = 5886 + 7.466 + 23.52 + 1600 = 7516.986\text{ N}$$

$$R_3 = 3924 - 7.466 - 23.52 - 1600 = 2293.014\text{ N}$$

$$R_4 = 3924 + 7.466 - 23.52 + 1600 = 5507.946\text{ N}$$

(ii) Car turning right

All the reactions due to gyroscopic couples and centrifugal couple change signs. Therefore,

$$R'_1 = 5886 + 7.466 - 23.52 + 1600 = 7469.946\text{ N}$$

$$R'_2 = 5886 - 7.466 - 23.52 - 1600 = 4255.014\text{ N}$$

$$R'_3 = 3924 + 7.466 + 23.52 + 1600 = 5554.986\text{ N}$$

$$R'_4 = 3924 - 7.466 + 23.52 - 1600 = 2340.054\text{ N}$$

3. (c)

(i)

In a centrifugal governor, the resultant of all the external forces which control the movement of the ball can be regarded as a single inward radial force acting at the centre of the ball. The variation of this force F with the radius of rotation of the ball can be studied under static conditions by measuring the outward radial force on the ball which is necessary to keep the ball in a equilibrium at various configurations (i.e. for different values of r). The force F is known as the controlling force and is a function of a single variable r. Thus,

$$F = F(r)$$

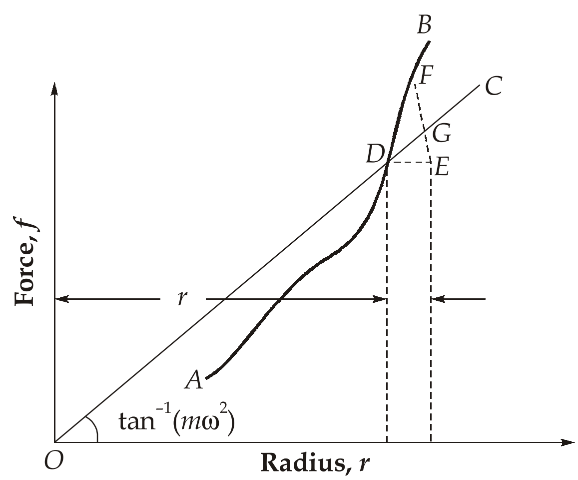
Following figure shows a typical plot of the controlling force characteristic curve (AB). The controlling force is derived from purely statical considerations without reference to the speed of rotation.

Now, let us suppose that the governor ball rotates at a speed ω. The centripetal force needed for maintaining the radius of rotation r is given by $m\omega^2 r$, where m is mass of each ball. The plot of this force against r for a given speed ω will obviously be a straight line passing through the origin as shown by line OC. So, the equilibrium radius for this speed ω will be determined by the intersection of the curve AB with the line OC (at the point D). For this value of r, the controlling force will be equal to the centripetal force. Mathematically, we can express this equilibrium condition as

$$F(r) = m\omega^2 r$$

To determine the condition necessary for stability, let the speed of the governor be ω at the equilibrium position given by the point D. If the speed remains the same, and if the radius changes to $r + \delta r$, the increment in the controlling force EF will be $\left(\frac{dF}{dr}\right)\delta r$, and the corresponding increment in the centripetal force EG will be EG = $m\omega^2\delta r$

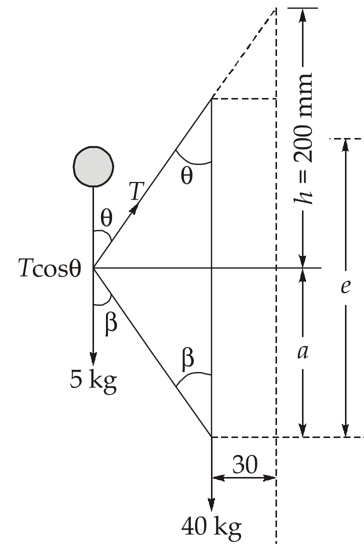
This restoring force should be greater than zero for the equilibrium position to be regained.



Thus, for stable operation, $\frac{dF}{dr} > m\omega^2$; we get finally, $\frac{dF}{dr} > \frac{F}{r}$

The condition for the stability of a governor is that the slope of the curve for the controlling force should be more than that of the line representing the centripetal force at the speed considered.

(ii)



Given: $m = 5$ kg, $M = 40$ kg, $h = 200$ mm, $l = 250$ mm

$$a = \sqrt{250^2 - (190-30)^2}$$
$$= 192.09 \text{ mm}$$

$\therefore$
$$N^2 = \frac{895}{h}\frac{a}{e}\left(\frac{m+M}{m}\right)$$
$$180^2 = \frac{895}{0.2} \times \frac{0.19209}{e}\left(\frac{5+40}{5}\right)$$

or,
$$e = 0.238 \text{ m}$$

Therefore length of the extension links = $e - a$

$$= 238.77 - 192.09 = 46.68 \text{ mm}$$ **Answer**

Now, let T be the tension in upper arms. Considering the vertical component of the forces on the lower link,

$$T\cos\theta = mg + \frac{Mg}{2}$$

$\because$
$$\cos\theta = \frac{0.19209}{0.25} = 0.768$$

$\therefore$
$$T \times 0.768 = 5 \times 9.81 + \frac{40 \times 9.81}{2}$$
$$T = 319.33 \text{ N}$$ **Answer**

4. (a)

Given: $V = 2.5$ m/s, $T_\infty = 20°C$, $T_s = 120°C$, $F_D = 11.75 \times 10^{-3}$, $\rho = 1.029$ kg/m^3, $v = 21.09 \times 10^{-6}$ m^2/s, $c_p = 1.005$ kJ/kgK, Pr = 0.7

(i)

Drag force is,
$$F_D = \bar{C}_f \times \frac{1}{2}\rho A V^2$$

or
$$F_D = \frac{1.328}{\sqrt{Re_l}} \times \frac{1}{2}\rho V^2 (l \times l) \qquad \text{as } (b = l)$$

$$11.75 \times 10^{-3} = \frac{1.328}{\sqrt{\dfrac{2.5 \times l}{21.09 \times 10^{-6}}}} \times \frac{1}{2} \times 1.029 \times l^2 \times 2.5^2$$

$\therefore$
$$l^{3/2} = 0.947$$

or, $l = 0.964$ m

Now, $$\bar{C}_f = \frac{1.328}{\sqrt{Re_l}} = \frac{1.328}{\sqrt{\frac{2.5 \times 0.964}{21.09 \times 10^{-6}}}} = 3.928 \times 10^{-3}$$

From Colburn analogy,

$$\overline{St} \times Pr^{2/3} = \frac{\bar{C}_f}{2}$$

or $$\frac{\bar{h}}{\rho v c_p} \times Pr^{2/3} = \frac{\bar{C}_f}{2}$$

$$\frac{\bar{h} \times (0.7)^{2/3}}{1.029 \times 2.5 \times 1.005 \times 10^3} = \frac{3.928 \times 10^{-3}}{2}$$

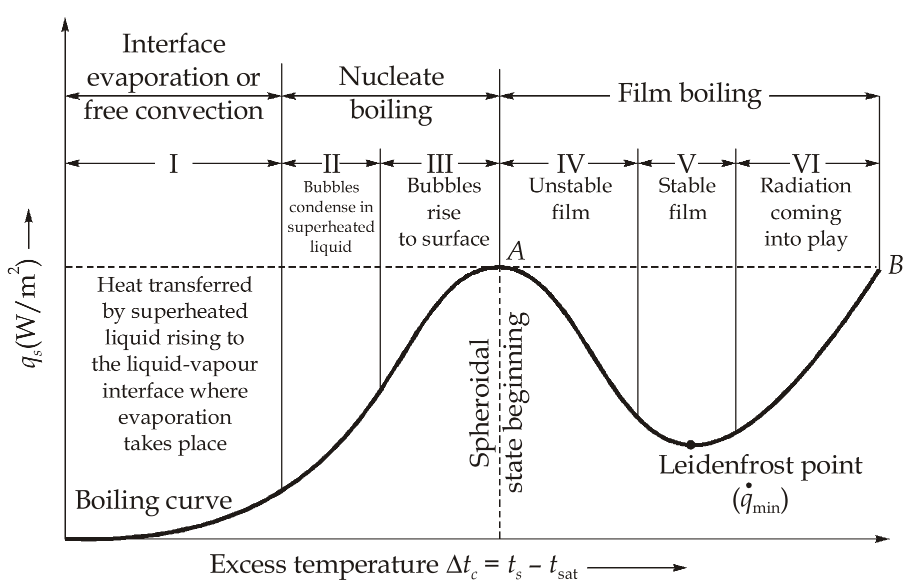
∴ $\bar{h} = 6.44$ W/m^{2}K

Heat loss, $Q = \bar{h} A \Delta T$

$= 6.44 \times 0.964^2 \times (120 - 20)$

$Q = 598.466$ W

(ii)



The boiling curve for water

4. (b)

(i)

	Cycloidal Teeth	Involute Teeth
(a)	Pressure angle varies from maximum at the beginning of engagement, reduces to zero at the pitch point and again increases to maximum at the end of engagement resulting in less smooth running of the gears.	Pressure angle is constant throughout at engagement of teeth. This result in smooth running of the gears.
(b)	It involves double curve for the teeth, epicycloid and hypocycloid. This complicates the manufacture.	It involves single curve for the teeth resulting in simplicity of manufacturing.
(c)	Owing to difficulty of manufacture, these are costlier.	These are simple to manufacture and thus are cheaper.
(d)	Exact centre-distance is required to transmit a constant velocity ratio.	A little variation in the centre distance does not affect the velocity ratio.
(e)	Phenomenon of interference does not occur at all.	Interference can occur if the condition of minimum number of teeth on a gear is not followed.
(f)	The teeth have spreading flanks and thus are stronger.	The teeth have radial flanks and thus are weaker as compared to the cycloidal form for the same pitch.
(g)	In this, a convex flank always has contact with a concave face resulting in less wear.	Two convex surfaces are in contact and thus there is more wear.

(ii)

(a) For maximum efficiency, $\psi_1 = \dfrac{\theta + \phi}{2} = \dfrac{90+6}{2} = 48°$ **Answer**

$$\psi_2 = 90 - 48° = 42°$$ **Answer**

(b) Centre distance, $c = \dfrac{P_n}{2\pi}\left[\dfrac{T_1}{\cos\psi_1} + \dfrac{T_2}{\cos\psi_2}\right]$

or,
$$180 = \frac{10}{2\pi}\left[\frac{T_1}{\cos 48°} + \frac{5T_1}{\cos 42°}\right] = 13.093T_1$$

or
$$T_1 = 13.747$$

Let T_1 to be 14

$\therefore$ $T_2 = 14 \times 5 = 70$

Hence, $T_1 = 14,\ T_2 = 70$

(c) $$C_{exact} = \frac{10}{2\pi}\left[\frac{14}{\cos 48^\circ} + \frac{70}{\cos 42^\circ}\right] = 183.3 \text{ mm}$$ **Answer**

(d) $$d_1 = \frac{P_1 T_1}{\pi} = \frac{P_n T_1}{\cos\psi_1 \times \pi} = \frac{10 \times 14}{\cos 48 \times 3.14} = 66.63 \text{ mm}$$ **Answer**

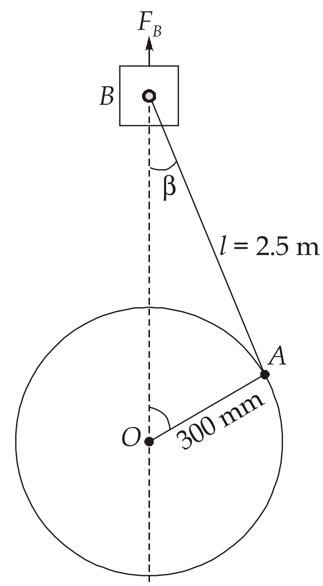
$$d_2 = \frac{P_2 T_2}{\pi} = \frac{P_n T_2}{\cos\psi_2 \times \pi} = \frac{10 \times 70}{\cos 42 \times 3.14} = 299.98 \simeq 300 \text{ mm}$$

Answer

(e) $$\eta_{max} = \frac{\cos(\theta+\phi)+1}{\cos(\theta-\phi)+1} = \frac{\cos(90+6)+1}{\cos(90-6)+1} = 0.81$$ **Answer**

4. (c)

Given: $l = 2.5$ m, $m = 220$ kg, $a = 1000$ mm, $f = 8$, $T = 24$ sec, $r = 300$ mm, $N = 200$ rpm, $\theta = 40°$.



Mass at the crank pin, $$m_a = 220 \times \left(\frac{2.5-1}{2.5}\right) = 132 \text{ kg}$$

Mass at the gudgeon pin, $m_b = 220 - 132 = 88$ kg

$$F_b = m_b r\omega^2\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

$$= 88 \times 0.3 \times \left(\frac{2\pi \times 200}{60}\right)^2\left(\cos 40^\circ + \frac{\cos 80^\circ}{2.5/0.3}\right)$$

$$= 9103.12 \text{ N}$$

As it is a vertical engine, the weight of the portion of the connecting rod at the piston pin also can be combined with this force i.e.

$$\text{Net force} = 9103.12 - 88 \times 9.81 = 8239.84 \text{ N}$$

$\therefore$ Inertia torque, $T_b = Fr\left(\sin\theta + \frac{\sin 2\theta}{2\sqrt{n^2 - \sin^2\theta}}\right)$

$$T_b = 8239.84 \times 0.3\left(\sin 40° + \frac{\sin 80°}{2\sqrt{\left(\frac{2.5}{0.3}\right)^2 - \sin^2 40°}}\right)$$

$$T_b = 8239.84 \times 0.3 \times 0.702$$

or $T_b = 1735.44$ Nm (Counter-clockwise)

We have, $b + \frac{k^2}{b} = L$

Where, $b = 2.5 - 1 = 1.5$ m and 'L' can be found from,

$$t = 2\pi\sqrt{\frac{L}{g}}$$

$$\frac{24}{8} = 2\pi\sqrt{\frac{L}{9.81}} \text{ or } L = 2.238 \text{ m}$$

$\therefore$ $1.5 + \frac{k^2}{1.5} = 2.238$

or $k^2 = 1.108$

or $k = 1.052$

or radius of gyration, $k = 1052$ mm

$$\alpha_c = -\omega^2 \sin\theta\left[\frac{n^2 - 1}{\left(n^2 - \sin^2\theta\right)^{3/2}}\right]$$

$$= -\left(\frac{2\pi \times 200}{60}\right)^2 \sin 40\left[\frac{8.33^2 - 1}{\left(8.33^2 - \sin^2 40\right)^{3/2}}\right]$$

$$= -33.62 \text{ rad/s}^2$$

$\therefore$ Correction couple, $\Delta T = m\alpha_c b(l - L) = 220 \times (-33.62) \times 1.5 \times (2.5 - 2.238)$

$$= -2906.78 \text{ Nm}$$

∴ Correction torque on the crankshaft,

$$T_c = \frac{\Delta T \cos\theta}{\sqrt{n^2 - \sin^2\theta}} = \frac{-2906.78 \cos 40°}{\sqrt{8.33^2 - \sin^2 40}}$$

$$T_c = -268.11 \text{ Nm (Counter clockwise)}$$

Now, torque due to weight of mass at A,

$$T_a = M_a g r \sin\theta = 132 \times 9.81 \times 0.3 \times \sin 40°$$

$$= 249.7 \text{ Nm (clockwise)}$$

∴ Total inertia torque on crankshaft, i.e.

$$T_{net} = T_b - T_c + T_a = 1735.44 - (-268.11) - 249.7$$

$$T_{net} = 1753.85 \text{ Nm}$$

Section : B

5. (a)

(i)

For an isothermal process, to prove:

$$\int_1^2 PdV = -\int_1^2 VdP$$

Isothermal process is defined as, PV = Constant = C

Differentiating both sides, we get,

$$PdV + VdP = 0$$

or

$$PdV = -VdP$$

When system undergoes change in state from state (1) to state (2).

$$\int_1^2 PdV = -\int_1^2 VdP$$

(ii)

Given: P = 105 kPa, T = 21°C = 294 K, ΔV = 0.95 m^3, Δt = 200 sec

$$\text{Volume flow rate, } \dot{V} = \frac{dV}{dt} \approx \frac{\Delta V}{\Delta t} = \frac{0.95}{200} = 4.75 \times 10^{-3} \text{ m}^3/\text{s}$$

Assuming CO_2 as an ideal gas (Close to room conditions)

Now,

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{105 \times 4.75 \times 10^{-3}}{0.1889 \times 294}$$

$$\dot{m} = 8.98 \times 10^{-3} \text{ kg/s}$$

5. (b)

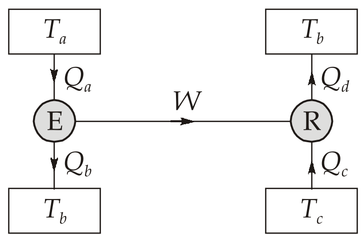
(i)

$$\eta_{\text{engine}} = \frac{W}{Q_a} = \frac{Q_a - Q_b}{Q_a} = \frac{T_a - T_b}{T_a}$$

$$W = Q_a\left(\frac{T_a - T_b}{T_a}\right) \quad \text{... (i)}$$

$$(\text{COP})_{\text{refrigerator}} = \frac{Q_c}{W} = \frac{Q_c}{Q_d - Q_c} = \frac{T_c}{T_b - T_c}$$

$$W = Q_c\left(\frac{T_b - T_c}{T_c}\right) \quad \text{... (ii)}$$



Now, from equations (i) and (ii), we get

$$\frac{Q_c}{Q_a} = \left(\frac{T_a - T_b}{T_a}\right)\left(\frac{T_c}{T_b - T_c}\right) = \frac{T_c}{T_a}\left(\frac{T_a - T_b}{T_b - T_c}\right)$$

(ii) If $Q_a = Q_c$, $T_a = 300°\text{C} = 573$ K, $T_c = 20°\text{C} = 253$ K

As $Q_c = Q_a$ so from above part we get,

$$1 = \left(\frac{T_a - T_b}{T_b - T_c}\right)\frac{T_c}{T_a}$$

$$\frac{T_a}{T_c} = \frac{T_a - T_b}{T_b - T_c}$$

$$\frac{573}{253} = \frac{573 - T_b}{T_b - 253}$$

$$2.265 = \frac{573 - T_b}{T_b - 253}$$

$\Rightarrow$ $$T_b = 351.01 \text{ K} \approx 78.01°\text{C}$$

(iii)

$$\eta_{\text{engine}} = \frac{T_a - T_b}{T_a} = \frac{573 - 351.01}{573} = 0.3874 = 38.74\%$$

$$(\text{COP})_{\text{ref}} = \frac{T_c}{T_b - T_c} = \frac{253}{351.01 - 253} = 2.581$$

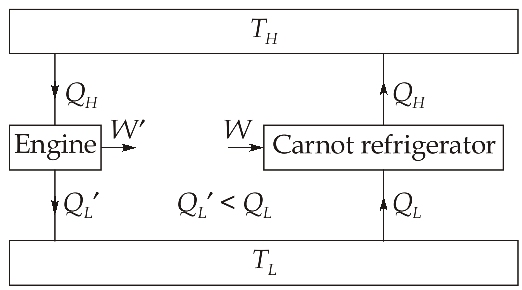
5. (c)

(i)

Assume that an engine exists, operating between two reservoirs, that has an efficiency greater than that of a Carnot engine; also assume that a Carnot engine operates as a refrigerator between the same two reservoirs, as shown in figure (a).

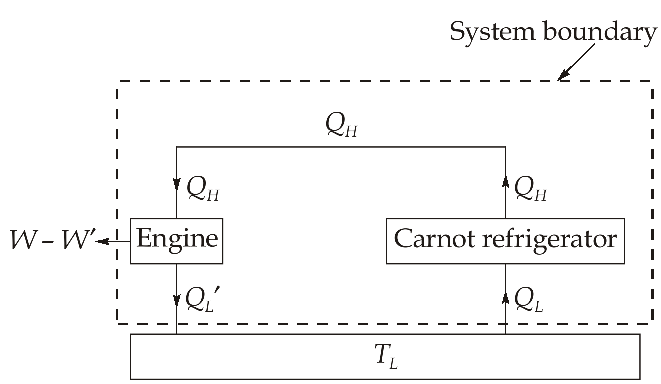
$$W' = Q_H - Q_L'; \qquad W = Q_H - Q_L \qquad [\because W' > W]$$

Let the heat transferred from the high temperature reservoir to the engine be equal to the heat rejected by the refrigerator then the work produced by the engine will be greater than the work required by the refrigerator (i.e. $Q_L' < Q_L$)



(a)

Since the efficiency of the engine is greater than Carnot engine, the system can be organized as shown in figure (b).



(b)

The engine drives the refrigerator using the heat rejected from the refrigerator. But there is some net work ($W' - W$) that leaves the system.

The net result is the conservation of energy from a single reservoir into work, a violation of the second law. Thus, the Carnot engine is the most efficient engine operating between two given thermal reservoir.

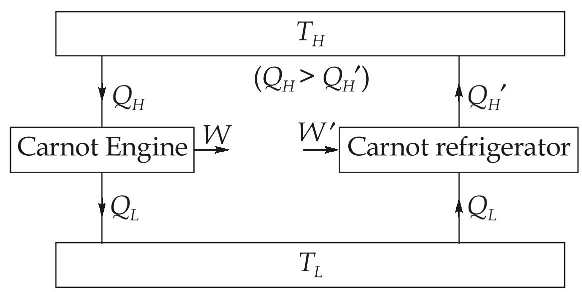
(ii)

Suppose that a Carnot engine drives a Carnot refrigerator as shown in (a).

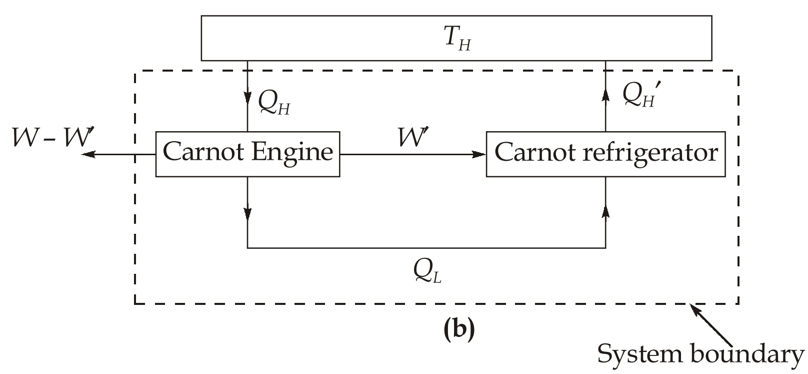
Let the heat rejected by the engine equal to heat required by the refrigerator.

$$Q_H - Q_L = W \; ; \qquad Q'_H - Q_L = W' \qquad [\therefore W > W']$$

Suppose the working fluid in the engine results in Q_H being Q'_H then W would be greater than W' (a consequence of the first law) and we would have the equivalent system as shown in figure (b).



(a)



(b)

The net result is a heat transfer $(Q_H - Q'_H)$ from a single reservoir and production of work, which is a clear violation of the second law. Thus, the efficiency of a Carnot engine is not dependent on the working substance.

5. (d)

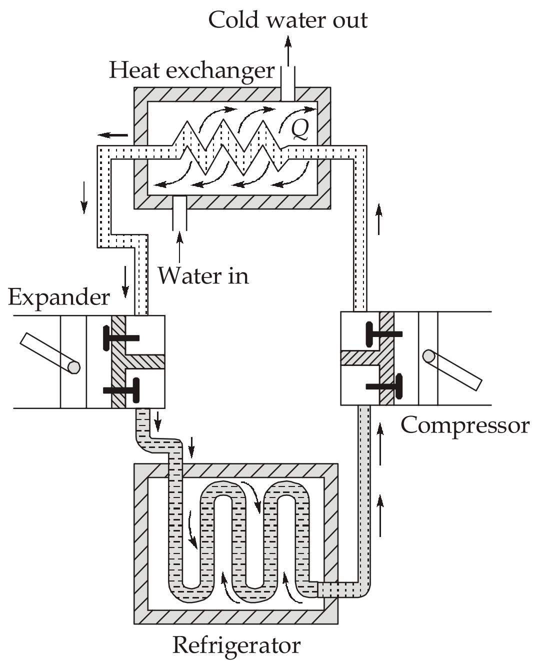
Refrigeration by expansion of air

According to the first law of thermodynamics applied to closed system,

$$Q = W + \Delta U$$

For reversible adiabatic process, $Q = 0$.

Therefore, $W = -\Delta U$



Air Refrigerator

This means that there is reduction in temperature. The temperature of the gas can be reduced by an adiabatic expansion of the gas. This principle was used in the Bell-Coleman air-refrigeration system.

The effect of expansion for producing colds can be explained by the following example.

Assuming the temperature of the atmosphere is 27°C and if air is compressed isentropically with the pressure ratio 5 then the final temperature of the compressed air T_2 is given by

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{0.286} = 300(5)^{0.286} = 475\text{ K}$$

This air with high temperature and pressure is cooled in the heat exchanger, then the final temperature will be 27°C under ideal condition and pressure will be 5 bar considering the pressure loss in heat exchanger is zero and if this air expands isentropically until the pressure falls to atmospheric pressure then the final temperature,

$$T_3 = \frac{300}{5^{0.286}} = 189.5\text{ K}$$

which is below atmospheric temperature (300 K).

This principle is universally used for producing low temperature in all air-refrigeration systems.

Refrigeration by Throttling of the gas

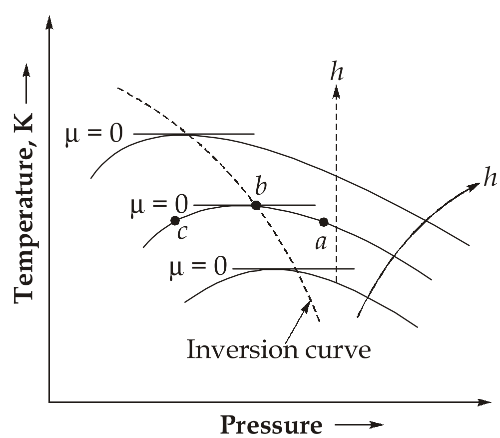
As we know that adiabatic throttling process is a constant enthalpy process, and as the enthalpy is only a function of temperature and the temperature of prefect gas remains constant after and before throttling. However with actual gases the temperature of the gas after throttling may either increase, decrease or remain constant.

The term which indicates the magnitude and sign of the change in temperature is called the Joule Thomson Coefficient and it is given by

$$\mu = \left(\frac{\partial T}{\partial p}\right)_h$$

which is the change in the temperature with respect to the pressure during constant enthalpy process.

The pressure versus temperature lines are drawn as shown in figure, taking enthalpy as a parameter.



Constant enthalpy lines for actual gas

In throttling process, the pressure of the gas decreases so that motion along the curve is towards the left. The graph shows that, if throttling occurs from the point *a* to *b*, the temperature of the gas increases, and at the point *b*; the temperature is maximum, and the Joule Thomson coefficient is zero. This point is known as "Inversion point" of the gas. If throttling occurs from the point *b* to *c*, then the temperature of the gas drops.

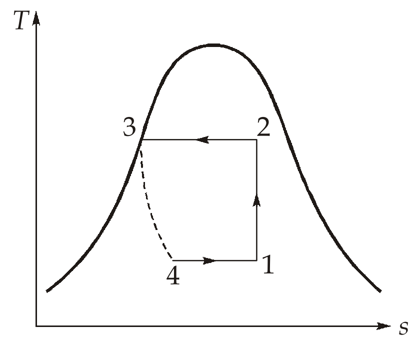
This is obvious from figure, that for reducing the temperature of the gas after throttling, the value of μ must be positive. The amount of the temperature drop depends upon Joule Thomson coefficient, the pressure drop, and the original state of the gas. In any event, the resulting temperature will be too high for refrigerating purposes unless the original temperature is relatively low.

Use of positive values of Joule Thomson coefficient are made in the liquification of gases such as air, nitrogen and oxygen.

5. (e)

The cycle is represented on T-s diagram. The given data is

$T_1 = -5°C, x_1 = 0.6, T_2 = 25°C, m = \frac{10}{60} = 0.333 \text{ kg/s}$



$$h_1 = h_{f1} + x_1 h_{fg1}$$
$$= -7.53 + 0.6 \times 245.8 = 139.95 \text{ kJ/kg}$$
$$h_4 = h_3 = h_{f3} = 81.25 \text{ kJ/kg,}$$

The compression process 1-2 is isentropic

$$s_{f2} + x_2 \frac{h_{fg2}}{T_2} = s_{f1} + x_1 \frac{h_{fg1}}{T_1}$$

$$0.2513 + x_2 \frac{121.6}{298} = -0.0419 + 0.6 \times \frac{245.8}{268}$$

$$x_2 = 0.63$$
$$h_2 = h_{f2} + x_2 h_{fg2} = 81.25 + 0.63 \times 121.6$$
$$= 157.86 \text{ kJ/kg}$$

Refrigeration effect required per kg of ice formed

$$= C_{pw} \times T_w + h_{fgice}$$
$$= 4.2 \times 10 + 335 = 377.0 \text{ kJ/kg}$$

Refrigeration effect produced by plant per minute

$$= m(h_1 - h_4) = 10 \times (139.95 - 81.25)$$
$$= 587 \text{ kJ/min}$$

∴ Refrigeration effect per day $= 587 \times 60 \times 24$

$$= 845280 \text{ kJ/day}$$

So, Ice formed per day $= \frac{845280}{377} = 2242.12 \text{ kg/day}$

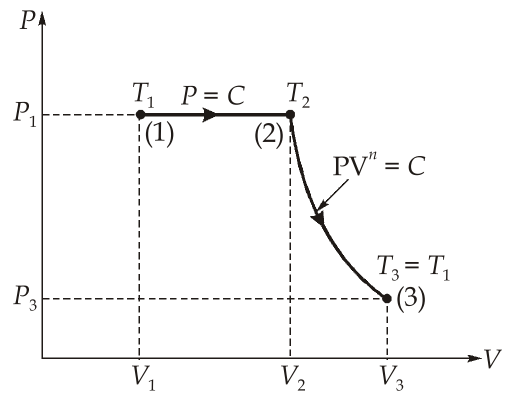
If plant efficiency is 60%, ice formed per day

$$= 2242.12 \times 0.6 = 1345.27 \text{ kg}$$

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{139.95 - 81.25}{157.86 - 139.95} = 3.278$$

6. (a)

(i)



Since $(\Delta S)_{2\text{-}1} = (\Delta S)_{3\text{-}2}$

For process 1-2, $(\Delta S)_{2\text{-}1} = C_P \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$

and $P_1 = P_2$ $\quad (\because P = C)$

$\therefore$ $(\Delta S)_{2\text{-}1} = C_P \ln\frac{T_2}{T_1}$ ... (i)

For process 3-2, $(\Delta S)_{3\text{-}2} = C_P \ln\frac{T_3}{T_2} - R\ln\frac{P_3}{P_2}$

and $T_3 = T_1$

$\therefore$ $(\Delta S)_{3\text{-}2} = C_P \ln\frac{T_1}{T_2} - R\ln\frac{P_3}{P_2}$ ... (ii)

and $\frac{P_3}{P_2} = \left(\frac{T_3}{T_2}\right)^{\frac{n}{n-1}} = \left(\frac{T_1}{T_2}\right)^{\frac{n}{n-1}}$

Putting $\frac{P_3}{P_2}$ in equation (ii), we get

$$(\Delta S)_{3\text{-}2} = C_P \ln\frac{T_1}{T_2} - R\left(\frac{n}{n-1}\right)\ln\left(\frac{T_1}{T_2}\right)$$

$$= \ln\frac{T_1}{T_2}\left(C_P - \frac{Rn}{n-1}\right)$$

$$= \ln\frac{T_1}{T_2}\left(\frac{\gamma R}{\gamma-1} - \frac{nR}{n-1}\right) \qquad \left[\because C_P = \frac{\gamma R}{\gamma-1}\right]$$

$$(\Delta S)_{3\text{-}2} = \ln\frac{T_1}{T_2}(R)\left(\frac{\gamma}{\gamma-1} - \frac{n}{n-1}\right) \qquad \text{... (iii)}$$

Now,$(\Delta S)_{2\text{-}1} = (\Delta S)_{3\text{-}2}$, putting values from equation (i) and (ii), we get

$$C_P \ln\frac{T_2}{T_1} = R\ln\frac{T_1}{T_2}\left(\frac{\gamma}{\gamma-1} - \frac{n}{n-1}\right)$$

$$\Rightarrow \quad \frac{\gamma R}{\gamma-1}\ln\frac{T_2}{T_1} = R\ln\frac{T_1}{T_2}\left(\frac{\gamma}{\gamma-1} - \frac{n}{n-1}\right)$$

$$\Rightarrow \quad \frac{\gamma}{\gamma-1} = \frac{n}{n-1} - \frac{\gamma}{\gamma-1}$$

$$\Rightarrow \quad \frac{2\gamma}{\gamma-1} = \frac{n}{n-1}$$

$$\Rightarrow \quad \frac{2\gamma}{\gamma-1} - 1 = \frac{n}{n-1} - 1$$

$$\Rightarrow \quad \frac{2\gamma-\gamma+1}{\gamma-1} = \frac{n-n+1}{n-1}$$

$$\Rightarrow \quad \frac{\gamma+1}{\gamma-1} = \frac{1}{n-1}$$

$$\Rightarrow \quad n - 1 = \frac{\gamma-1}{\gamma+1}$$

$$n = \frac{\gamma-1+\gamma+1}{\gamma+1} = \frac{2\gamma}{\gamma+1}$$

$$\Rightarrow \quad \frac{n}{2} = \frac{\gamma}{\gamma+1}$$

(ii)

Given: P_1 = 480 kPa, P_2 = ?, T_1 = 330 K, T_2 = 260 K, V_1 = 1.05 m^3, V_2 = 0.03 m^3

Mixture of CH_4 and C_2H_4 is an ideal gas mixture with $x_{CH_4} = x_{C_2H_4}$ = 0.5.

Now, we know,

$$R_{\text{mixture}} = \sum x_i R_i$$

$$= \left(\frac{8.314}{M_{CH_4}}\right)x_{CH_4} + \left(\frac{8.314}{M_{C_2H_4}}\right)x_{C_2H_4}$$

$$= 0.5196 \times 0.5 + 0.29692 \times 0.5$$

$$= 0.40826 \text{ kJ/kgK}$$

and,

$$C_{V,\text{mix}} = \sum x_i C_{Vi} = 0.5\left(C_{V_{CH_4}}\right) + 0.5\left(C_{V_{C_2H_4}}\right)$$

$$= 0.5(1.736) + 0.5(1.252)$$

$$= 1.494 \text{ kJ/kgK}$$

Now, $P_1V_1 = m(R_{\text{mixture}})T_1$

$\Rightarrow$ $480 \times 1.05 = m(0.4086) \times 330$

$\Rightarrow$ $m = 3.7378$ kg

So, $P_2V_2 = m(R_{\text{mixture}})T_2$

$$P_2 = \frac{3.7378 \times 0.40826 \times 260}{0.03} = 13225.28 \text{ kPa}$$

For a polytropic process, $P_1V_1^n = P_2V_2^n$

$\Rightarrow$ $480(1.05)^n = 13225.28(0.03)^n$

$\Rightarrow$ $$\left(\frac{1.05}{0.03}\right)^n = \frac{13225.28}{480}$$

$\Rightarrow$ $$n\ln\left(\frac{1.05}{0.03}\right) = \ln\left(\frac{13225.28}{480}\right)$$

$$n = 0.933$$

Work done for a polytropic process,

$$W = \frac{P_1V_1 - P_2V_2}{n-1} = \frac{480(1.05) - 13225.28(0.03)}{0.933 - 1}$$

$$= -1600.62 \text{ kJ}$$

and, Heat transfer, $Q = \Delta U + W = (U_2 - U_1) + W$

$$= mC_{V,\text{mix}}(T_2 - T_1) + W$$

$$Q = 3.7378 \times 1.494\,(260 - 330) - 1600.62$$

$$Q = -1991.52 \text{ kJ}$$

Change of entropy, $s_2 - s_1 = C_{V,\text{mix}} \ln\frac{T_2}{T_1} + R_{mix}\ln\frac{V_2}{V_1}$

$$= 1.494\ln\left(\frac{260}{330}\right) + 0.40826\ln\left(\frac{0.03}{1.05}\right)$$

$$= -0.3562 - 1.5151 = -1.80771 \text{ kJ/kgK}$$

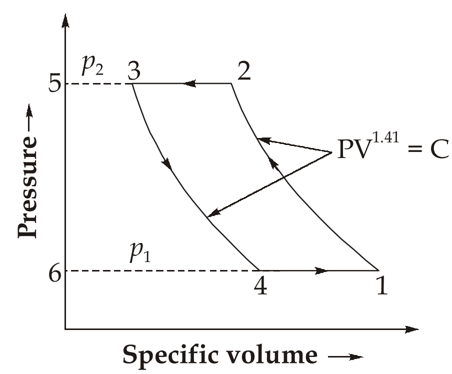
and $s_2 - s_1 = m(s_2 - s_1)$

$$= 3.7378(-1.80771) = -6.7569 \text{ kJ/K}$$

6. (b)

Given: $T_1 = 10°C = 283$ K, $T_3 = 25°C = 298$ K, $P_2 = 5$ bar, $P_1 = 1$ bar, $c_p = 1.009$ kJ/kgK, $\gamma = 1.41$

(i)



We have $\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{1}\right)^{\frac{1.41-1}{1.41}} = 1.5968$

Therefore, $T_2 = 1.5968 \times 283 = 451.89$ K

$$T_4 = \frac{298}{1.5968} = 186.62 \text{ K}$$

Therefore, RE/kg of air $= C_p(T_1 - T_4)$

$$\text{RE} = 1.009(283 - 186.62)$$

$$= 97.25 \text{ kJ/kg of air}$$

Heat received from cooling water/kg of air

$$= C_p(T_2 - T_3)$$

$$= 1.009(451.89 - 298) = 155.28 \text{ kJ/kg of air}$$

The net work done/kg of air, $w = 155.28 - 97.24$

$$= 58.04 \text{ kJ/kg of air}$$

Thus, Theoretical COP $= \dfrac{RE}{w} = \dfrac{97.24}{58.04} = 1.676$

(ii) Since the processes of expansion and compression are not adiabatic, there is heat transfer during the processes. Thus, work done in this case may be found out as follows. Let n_1 be the index of compression and n_2 be the index of expansion.

Net work done/kg of air = Work of compressor – Work of motor

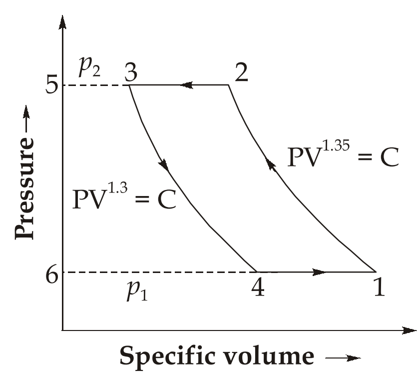
$$= \frac{n_1}{n_1 - 1} R(T_2 - T_1) - \frac{n_2}{n_2 - 1} R(T_3 - T_4)$$

We have, $T_1 = 283$ K, $T_3 = 298$ K

Also, $\dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n_1 - 1}{n_1}}$ for compression

$\Rightarrow$ $\dfrac{T_2}{T_1} = (5)^{\frac{1.35-1}{1.35}} = 5^{0.2592} = 1.5178$

$T_2 = 429.53$ K



Also, $\dfrac{T_3}{T_4} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n_2 - 1}{n_2}}$ for expansion

$$\frac{T_3}{T_4 = (5)^{(1.3-1)/1.3}} = 205.55 \text{ K}$$

Thus, RE/kg of air $= C_p(T_1 - T_4) = 1.009(283 - 205.55)$

$$= 78.15 \text{ kJ/kg}$$

Work done/kg of air, $w = \frac{n_1}{n_1 - 1}R(T_2 - T_1) - \frac{n_2}{n_2 - 1}R(T_3 - T_4)$

$$= R\left[\frac{1.35}{0.35}\times(429.53 - 283) - \frac{1.3}{0.3}\times(298 - 205.55)\right]$$

$$= R[565.187 - 400.617] = 47.23 \text{ kJ/kg of air}$$

Therefore, $$\text{COP} = \frac{RE}{w} = \frac{78.15}{47.23} = 1.655$$

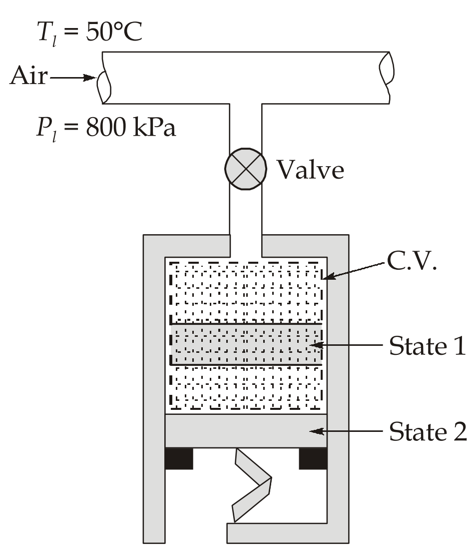
Comment on result: The refrigeration effect/kg of air, net work done per kg of air and COP of isentropic process is higher than that of actual process occuring in second case. As in the case of actual system which degrade the performance of system under consideration.

6. (c)

Given: T_∞ = 10°C = 283 K, A_P = 0.1 m^2, V_{stop} = 50L = 50 × 10^{-3} m^3, $C_{P\,air}$ = 1.005 kJ/kgK, R = 0.287 kJ/kgK, k = 100 kN/m, V_1 = 20L = 20 × 10^{-3} m^3, P_1 = 200 kPa, T_1 = 10°C = 283 K, P_l = 800 kPa, T_l = 50°C = 323 K, P_2 = 800 kPa, T_2 = 80°C = 353 K

Assumption: Air as an ideal gas

So, $$m_1 = \frac{P_1V_1}{RT_1} = \frac{200\times20\times10^{-3}}{0.287\times283} = 0.0492 \text{ kg}$$



At state (1), $$P_1A_P = F_S + P_{atm}A_P$$

$$P_1A_P = kx_1 + P_{atm}A_P \quad \text{... (i)}$$

When piston reaches the stoppers,

$$P_{stop}A_P = kx_2 + P_{atm}A_P \quad \text{... (ii)}$$

From equation (i) and (ii),

$$\Rightarrow \quad P_1A_P - P_{stop}A_P = k(x_1 - x_2)$$

$$\Rightarrow \quad (P_{stop})A_P = P_1A_P + k(x_2 - x_1)$$

$$(P_{stop})A_P = P_1A_P + k\left(\frac{V_{stop}}{A_P} - \frac{V_1}{A_P}\right)$$

$$P_{stop} = P_1 + \frac{k}{A_P^2}\left(V_{stop} - V_1\right)$$

$$= 200 + \frac{100}{(0.1)^2}\left(50 \times 10^{-3} - 20 \times 10^{-3}\right)$$

$$= 500 \text{ kPa}$$

Since, $P_2 > P_{stop}$, piston will hit the stops.

Now, $\quad V_2 = 50 \times 10^{-3} = V_S$

So, $\quad m_2 = \frac{P_2V_2}{RT_2} = \frac{800 \times 50 \times 10^{-3}}{0.287 \times 353} = 0.3948 \text{ kg}$

Now, from mass conservation,

$$\left(\frac{dm}{dt}\right)_{CV} = \dot{m}_i - \cancel{\dot{m}_e}^{\,o} = \dot{m}_i$$

and, $\quad m_i = m_2 - m_1 \quad \text{... (iii)}$

From energy conservation,

$$\left(\frac{dU}{dt}\right)_{CV} = \dot{m}_ih_i + \dot{Q} - \cancel{\dot{m}_e}^{\,o}h_e - \dot{W}_{CV}$$

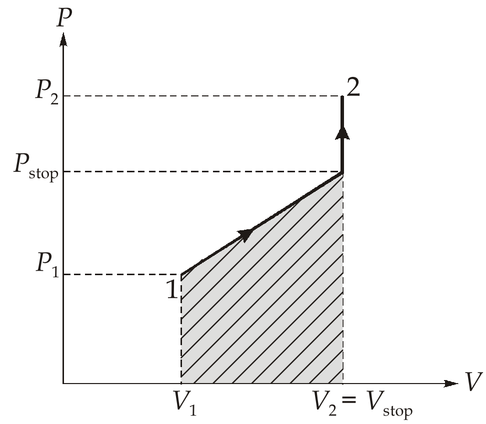
$$\Rightarrow \quad m_2u_2 - m_1u_1 = m_ih_i + Q - W_{CV}$$

$$Q = m_2u_2 - m_1u_1 - m_ih_i + W_{CV}$$

From equation (iii), we get,

$$Q = m_2C_VT_2 - m_1C_VT_1 - (m_2 - m_1)C_PT_i + W_{CV} \quad \text{... (iv)}$$

Now, $\quad W_{CV} = \int PdV$



$$W_{CV} = \frac{(P_1 + P_{stop})}{2}(V_{stop} - V_1)$$

$$= \left(\frac{200+500}{2}\right)(50\times10^{-3} - 20\times10^{-3})$$

$$W_{CV} = 10.5 \text{ kJ}$$

From equation (iv), we have

$Q = 0.3948\,(1.005 - 0.287)(353) - 0.0492(1.005 - 0.287)\,(283) - (0.3948 - 0.0492)(1.005)(323) + 10.5$

$Q = 100.0636 - 9.9971 - 112.1869 + 10.50$

$Q = -11.62$ kJ

Now, entropy change associated with process,

$$(\Delta S)_{net} = m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{CV}}{T_\infty}$$

$$= m_2 s_2 - m_1 s_1 - (m_2 - m_1)s_i - \frac{Q_{CV}}{T_\infty}$$

$$= m_2(s_2 - s_i) - m_1(s_1 - s_i) - \frac{Q_{CV}}{T_\infty} \qquad \dots \text{(v)}$$

and $$s_2 - s_i = C_P \ln\left(\frac{T_2}{T_i}\right) - R\ln\left(\frac{P_2}{P_i}\right)$$

$$= 1.005\ln\left(\frac{353}{323}\right) - 0.287\ln\left(\frac{800}{800}\right) = 0.08925 \text{ kJ/kgK}$$

Also, $$s_1 - s_i = C_P \ln\left(\frac{T_1}{T_i}\right) - R\ln\left(\frac{P_1}{P_i}\right)$$

$$= 1.005\ln\left(\frac{283}{323}\right) - 0.287\ln\left(\frac{200}{800}\right) = 0.265 \text{ kJ/kgK}$$

Now, from equation (v) we get,

$$(\Delta S)_{net} = 0.3948(0.8925) - 0.0492(0.265) - \frac{(-11.62)}{283}$$
$$= 0.35235 - 0.01304 + 0.04113 = 0.380 \text{ kJ/K}$$

7. (a)

(i)

We know,
$$(\text{COP})_{HP} = \frac{\dot{Q}_{H2}}{\dot{W}} = \frac{\dot{Q}_{H2}}{\dot{Q}_{H2} - \dot{Q}_{L2}} = \frac{T_{room}}{T_{room} - T_{amb}}$$

and,
$$\dot{W} = \eta_{HE}\dot{Q}_{H1} = \left(1 - \frac{T_{room}}{T_H}\right)\dot{Q}_{H1}$$

Also,
$$\dot{W} = \frac{\dot{Q}_{H2}}{(COP)_{HP}} = \left(\frac{T_{room} - T_{amb}}{T_{room}}\right)\dot{Q}_{H2}$$

So,
$$\dot{Q}_{H2} = \left(\frac{T_{room}}{T_{room} - T_{amb}}\right)\left(1 - \frac{T_{room}}{T_H}\right)\dot{Q}_{H1} \quad \text{... (i)}$$

and,
$$\dot{Q}_{L1} = \dot{Q}_{H1} - \dot{W} = \dot{Q}_{H1} - \left(1 - \frac{T_{room}}{T_H}\right)\dot{Q}_{H1}$$
$$= \dot{Q}_{H1}\left(1 - 1 + \frac{T_{room}}{T_H}\right) = \frac{T_{room}}{T_H}(\dot{Q}_{H1}) \quad \text{... (ii)}$$

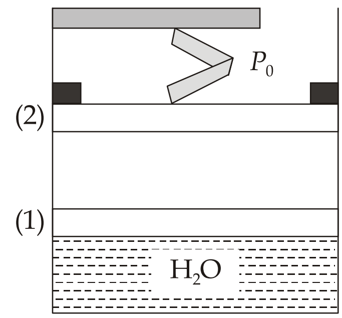
Now, from equation (i) and (ii)

$$\frac{\dot{Q}_{L1} + \dot{Q}_{H2}}{\dot{Q}_{H1}} = \frac{\left(\frac{T_{room}}{T_H}\right)\dot{Q}_{H1} + \left(\frac{T_{room}}{T_{room} - T_{amb}}\right)\left(1 - \frac{T_{room}}{T_H}\right)\dot{Q}_{H1}}{\dot{Q}_{H1}}$$

$$\frac{\dot{Q}_{L1} + \dot{Q}_{H2}}{\dot{Q}_{H1}} = \left(\frac{T_{room}}{T_H}\right) + \left(\frac{T_{room}}{T_H}\right)\left(\frac{T_H - T_{room}}{T_{room} - T_{amb}}\right)$$
$$= \frac{T_{room}}{T_H}\left(1 + \frac{T_H - T_{room}}{T_{room} - T_{amb}}\right)$$

$$\frac{\dot{Q}_{L1} + \dot{Q}_{H2}}{\dot{Q}_{H1}} = \frac{T_{room}}{T_H}\left(\frac{T_H - T_{amb}}{T_{room} - T_{amb}}\right)$$

(ii)



Given: m_w = 1 kg (Saturated), P_0 = 100 kPa, P_1 = 100 kPa, V_1 = 0.2 m^3, V_2 = 1 m^3

Now, when piston just touches the stoppers, i.e. state (2)

$$T_2 = 400°\text{C}$$

and $$v_2 = \frac{V_2}{m_w} = \frac{1}{1} = 1\ \text{m}^3/\text{kg}$$

From table of superheated water and specific volume, at given temperature and specific volume, the pressure P_2 will be between 3.0 to 3.5 bar.

So, by inter-polation

$$\frac{1 - 0.8835}{1.031 - 0.8835} = \frac{P - 3.5}{3 - 3.5}$$

$\Rightarrow$ $$P = 3.105\ \text{bar} = 310.5\ \text{kPa}$$

Since, $$P_{stop} < P_3\ (0.45\ \text{MPa})$$

So, process 2-3 will be constant volume heat addition process,

We have, $$P_3 = 0.45\ \text{MPa} = 450\ \text{kPa}$$

$$v_2 = v_3 = 1\ \text{m}^3/\text{kg}$$

Again from table, we need to apply interpolation between specific volume and temperature at given pressure of 450 kPa (or 4.5 bar).

So, $$\frac{1.1 - 1}{1.1 - 0.9971} = \frac{800 - T_3}{800 - 700}$$

$$T_3 = 702.8°\text{C}$$

Final temperature after heat addition will be T_3 = 702.8°C

Now, work done during process, $W = {}_1W_2 + {}_2W_3$

$$= \frac{1}{2}(P_1 + P_2)(V_2 - V_1) + 0 \qquad [\because \text{isochoric}]$$

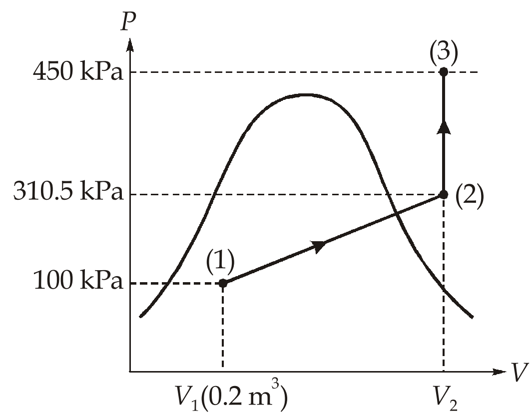
$$= \frac{1}{2}(100 + 450)(1 - 0.2)$$

$$W = 220 \text{ kJ}$$

and, since specific volume at (1)

$$v_1 = \frac{v_1}{m_w} = \frac{0.2}{1} = 0.2 \text{ m}^3/\text{kg} \;\; (< V_g \text{ at } 100 \text{ kPa})$$

So, state 1, will be in wet region.



7. (b)

Theoretical piston displacement/min

$$V_P = \frac{\pi}{4} \times \frac{D^2 L N \times n}{10^6}$$

where, V_P = Piston displacement in m^3/min, N = Revolution per min, n = Number of cylinders, D = Cylinder bore in cm, L = Stroke of piston in cm

$$V_P = \frac{\pi \times 5.65 \times 5.65 \times 5 \times 1450 \times 2}{4 \times 10^6}$$

$$= 0.3635 \text{ m}^3/\text{min}$$

(i) Density of Freon-12 saturated vapour at –10°C $= \frac{1}{v_g} = \frac{1}{0.07702}$ kg/m^3

$\therefore$ Refrigerant mass circulated at –10°C suction $= \frac{0.3635}{0.07702} = 4.72$ kg/min

Enthalpy of Freon-12 saturated vapour at –10°C = 347.96 kJ/kg

Enthalpy of Freon-12 saturated liquid at 40°C = 239.29 kJ/kg

Thus, Refrigerating effect/kg = 347.96 – 239.29

$$= 108.67 \text{ kJ/kg}$$

Theoretical R.C. at –10°C suction of compressor

$$= 108.67 \times 4.72 = 512.92 \text{ kJ/min}$$

$$= \frac{512.92}{211} \text{ ton} = 2.431 \text{ ton}$$

(ii) Density of Freon-12 saturated vapour at +10°C = $\frac{1}{v_g} = \frac{1}{0.04119}$ kg/m^3

The actual vapour at +10°C sucked in = 0.3635 m^3/min.

Therefore mass of refrigerant circulated at 10°C suction

$$= 0.3635 \times \frac{1}{0.04119} = 8.825 \text{ kg/min}$$

Enthalpy of Freon-12 saturated vapour at +10°C = 356.79 kJ/kg

Enthalpy of Freon-12 saturated liquid at +40°C = 239.29 kJ/kg

Therefore refrigerant effect/kg = 356.79 – 239.29 = 117.5 kJ/kg

Theoretical RC of compressor at 10°C = 117.5 × 8.825

$$= 1036.93 \text{ kJ/kg}$$

$$= \frac{1036.93}{211} = 4.914 \text{ ton}$$

Comment: The volume of vapour sucked in is the same in both the cases, but the refrigerant mass circulation by the same compressor has increased from 4.72 kg/min at –10°C to 8.82 kg/min at 10°C. The increase is due to increased density. The percentage increase in refrigerant mass circulation

$$= \frac{8.825 - 4.72}{4.72} \times 100 = 86.97\%$$

But the RC has increased from 2.431 ton to 4.914 ton for the suction temperature rise from –10°C to 10°C. This increase as a percentage is

$$= \frac{4.914 - 2.431}{2.431} = 102.14\%$$

It may be noted that the additional gain of (102.14 – 86.97)% = (15.17%) in the capacity results from an increase in RE per kg of refrigerant.

(c) Saturated discharge temperature = 40°C

Absolute pressure corresponding to 40°C saturation temperature = 9.5944 bar

Absolute pressure corresponding to –10°C saturation temperature = 2.1928 bar

Thus, pressure ratio of compressor = $\frac{9.5944}{2.1928} = 4.37541$

Also, $\eta_{vol} = 1 + C - C\left(\frac{P_d}{P_s}\right)^{1/n} = 1 + 0.04 - 0.04(4.37541)^{1/1.13}$

$= 0.89232 = 89.232\%$

Thus the refrigeration capacity gets modified as

$RC = 2.431 \times 0.89232 = 2.17$ ton

Absolute pressure corresponding to 40°C saturation temperature = 9.5944 bar

Absolute pressure corresponding to 10°C saturation temperature = 4.2356 bar

Thus pressure ratio of compressor = $\frac{9.5944}{4.2356} = 2.26518$

Also, $\eta_{vol} = 1 + 0.04 - 0.04(2.26518)^{1/1.13}$

$= 0.95753 = 95.753\%$

Therefore, $RC = 4.914 \times 0.95753 = 4.7053$ ton

Thus the percentage increase in RC is further increased by improvement in volumetric efficiency and now total percentage increase is

$$\frac{4.7053 - 2.17}{2.17} \times 100 = 116.83\%$$

7. (c)

(i)

Using the first law in the form,

$$Q - W = \Delta E$$

Applied to the process 1 → 2, we have

$$a - 100 = 100$$

$\Rightarrow$ $a = 200$ kJ

Applied to process 3 → 1, we get,

$$100 - d = -200$$

$$d = 300 \text{ kJ}$$

Now, $\oint dW = \oint dQ$

$\Rightarrow$ $W_{12} + W_{23} + W_{31} = Q_{12} + Q_{23} + Q_{31}$

$\Rightarrow$ $100 - 50 + d = a + b + 100$

$$\Rightarrow \qquad 100 - 50 + 300 = 200 + b + 100$$

$$\Rightarrow \qquad b = 50 \text{ kJ}$$

and for a cycle $\sum \Delta E = 0$, we have

$$100 + c - 200 = 0$$

$$\Rightarrow \qquad c = 100 \text{ kJ}$$

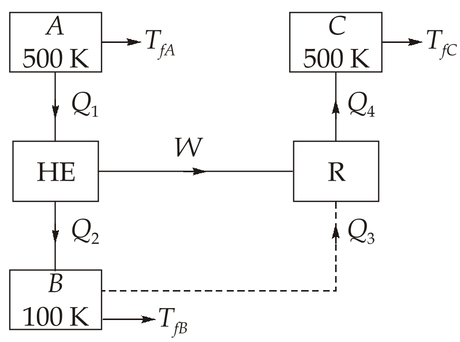
(ii)

Let the identical bodies A, B and C having the same heat capacity C_p be respectively at 500 K, 100 K and 500 K.

Let us operate a heat engine and a refrigerator as shown in the figure.

Let T_{fA}, T_{fB} and T_{fC} are the final temperature of bodies A, B and C respectively.

Now, feasible heat engine and refrigeration arrangement:



$$(\Delta S)_A = C \ln \frac{T_{fA}}{500}$$

$$(\Delta S)_B = C \ln \frac{T_{fB}}{100}$$

$$(\Delta S)_C = C \ln \frac{T_{fC}}{500}$$

$$(\Delta S)_{HE} = 0 \text{ (reversible)}$$

$$(\Delta S)_{ref} = 0 \text{ (reversible)}$$

Since, $(\Delta S)_{univ} \geq 0$

$$\left[C \ln \frac{T_{fA}}{500} + C \ln \frac{T_{fB}}{100} + C \ln \frac{T_{fC}}{500} \right] \geq 0$$

$$\Rightarrow \qquad C \ln \frac{T_{fA} T_{fB} T_{fC}}{25000000} \geq 0$$

Let, $T_{fA} = T_{fB}$

$\Rightarrow$ $$C\ln\frac{T_{fA}^2 T_{fC}}{25000000} \geq 0$$

For minimum value of T_{fA},

$$C\ln\frac{T_{fA}^2 T_{fC}}{25000000} = 0 = \ln 1$$

$\Rightarrow$ $$T_{fA}^2 T_{fC} = 25000000 \quad \text{... (i)}$$

Now, $Q_1 = C(500 - T_{fA})$

$Q_2 = C(T_{fB} - 100)$

$Q_4 = C(T_{fC} - 500)$

We have, Q_1 = Heat removed from body A = Heat discharged to bodies B and C.

$\Rightarrow$ $Q_1 = Q_2 + Q_4$ ($\because$ No work and heat supplied from outside)

$$C(500 - T_{fA}) = C(T_{fB} - 100) + C(T_{fC} - 500)$$

$$500 - T_{fA} = T_{fB} - 100 + T_{fC} - 500$$

$\because$ $T_{fA} = T_{fB}$

$\Rightarrow$ $$T_{fC} = 1100 - 2T_{fA} \quad \text{... (ii)}$$

Putting value of T_{fC} from equation (ii) in equation (i) we get,

$$T_{fA}^2\left(1100 - 2T_{fA}\right) = 25000000$$

$\Rightarrow$ $$2T_{fA}^3 - 1100T_{fA}^2 + 25000000 = 0$$

$\Rightarrow$ $T_{fA} = -135.07$ or 185.07 or 500

Here, 500 and –135.07 are discarded ($\because$ not feasible)

So, $T_{fA} = 185.07$ K

So, From equation (ii), $T_{fC} = 1100 - 2\,(185.07) = 729.86$ K

So, highest temperature can be raised by operation of heat engine or refrigerators is 729.86 K (of body C which was earlier at 500 K)

8. (a)

Given: $V_{storage} = 10\text{ m}^3$, $P_{storage} = 2\text{ MPa} = 2000\text{ kPa}$, $T_{storage} = 20°\text{C} = 293\text{ K}$, $P_{ambient} = P_0 = 100\text{ kPa}$,

$D_1 = 1$ m, $P = P_0 + C\left(1 - \dfrac{D_1}{D}\right)\dfrac{D_1}{D}$ (When, $D > D_1$), $D_f = 5$ m, $P_f = 500$ kPa, $T_f = 20°\text{C} = 293$ K

At the end of the process we have

$$D = D_f = 5 \text{ m}$$

and $$P = P_f = 500 \text{ kPa}$$

So, $$P = P_0 + C\left(1 - \frac{D_1}{D}\right)\frac{D_1}{D} \quad \text{... (i)}$$

Equation (i) becomes,

$$500 = 100 + C\left(1 - \frac{1}{5}\right)\frac{1}{5}$$

$\Rightarrow$ $$C = 2500$$

Let $x = \frac{D}{D_1}$, so equation (i) becomes

$$P = P_0 + C\left(1 - x^{-1}\right)x^{-1}$$

and $$\frac{dP}{dD} = 0$$

$$\frac{dP}{dD} = C\left(-x^{-2} - (-2)x^{-3}\right)\left(\frac{1}{D_1}\right) = 0$$

$\Rightarrow$ $$\frac{dP}{dD} = -x^{-2} + 2x^{-3} = 0$$

$\Rightarrow$ $$x = 2 \Rightarrow \frac{D}{D_1} = 2$$

So, maximum pressure inside the balloon at any time during this inflation process will occur when $D = 2D_1$, i.e. $D = 2(1) = 2$ m

and $$P_{max} = P_0 + 2500\left(1 - \frac{1}{2}\right)\frac{1}{2}$$

$$= 100 + 625 = 725 \text{ kPa}$$

Assuming, Helium as ideal gas

Now, Mass of helium inside, when pressure is maximum,

$$(m_{He})_{balloon} = \frac{P_{max} \times V}{R_{He}T}$$

$$V = \frac{\pi}{6}D^3 = \frac{\pi}{6}(2)^3 = 4.188 \text{ m}^3$$

$$(m_{He})_{balloon} = \frac{725 \times 4.188}{2.077 \times (273+20)} = 4.99 \text{ kg}$$

Now, mass of helium in tank at initial condition is,

$$(m_{storage})_{initial} = \frac{P_{storage} \times V_{storage}}{R_{He} \times T_{storage}} = \frac{2000 \times 10}{2.077 \times (273+20)} = 32.86 \text{ kg}$$

and mass after filling balloon to maximum pressure is,

$$(m_{storage})_{final} = (m_{storage})_{critical} - (m_{He})_{balloon}$$
$$= 32.86 - 4.99 = 27.87 \text{ kg}$$

Now, pressure after filling balloon to maximum pressure is,

$$(P_{storage})_{final} (V_{storage})_{final} = (m_{storage})_{final} R_{He} (T_{storage})_{final}$$
$$(P_{storage})_{final} = \frac{27.87 \times 2.077 \times 293}{10} = 1696.28 \text{ kPa}$$

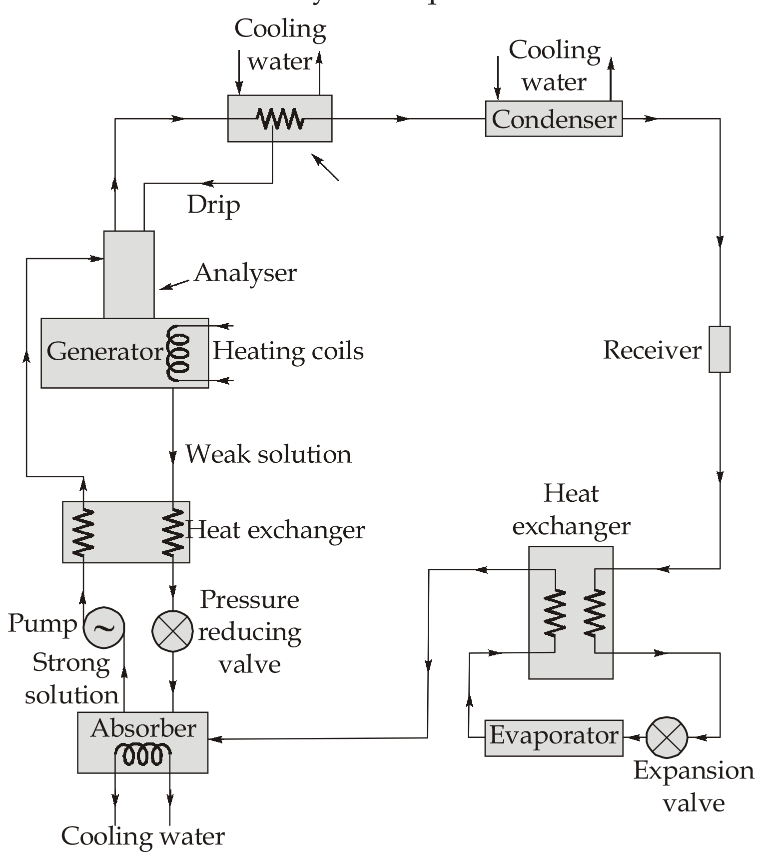
8. (b)

The simple absorption system is not very economical. In order to make the system more practical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in Figure. These accessories help to improve the performance and working of the plant, as discussed below:

1. **Analyser.** When ammonia is vaporised in the generator, some water is also vaporised and will flow into the condenser along with the ammonia vapours in the simple system. If these unwanted water particles are not removed before entering into the condenser, they will enter into the expansion valve where they freeze and choke the pipeline. In order to remove these unwanted particles flowing to the condenser, an analyser is used. The analyser may be built as an integral part of the generator or made as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier are introduced at the top of the analyser and flow downward over the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator. The vapour is cooled and most of the water vapour condenses, so that mainly ammonia vapour (approximately 99%) leaves the top of the analyser. Since the aqua is heated by the vapour, less external heat is required in the generator.
2. **Rectifier:** In case the water vapours are not completely removed in the analyser, a closed type vapour cooler called rectifier (also known as dehydrator) is used. It is generally water cooled and may be of the double pipe, shell and coil or shell and tube type. Its function is to cool further the ammonia vapours leaving the analyser

so that the remaining water vapours are condensed. Thus, only dry or anhydrous ammonia vapours flow to the condenser. The condensate from the rectifier is returned to the top of the analyser by a drip returned pipe.

3. **Heat exchangers:** The heat exchanger provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to die generator and the amount of cooling required for the absorber. Thus the economy of the plant increases.



The heat exchanger provided between the condenser and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub-cooled by the low temperature ammonia vapour from the evaporator as shown in figure. This sub-cooled liquid is now passed to the expansion valve and then to the evaporator.

In this system, the net refrigerating effect is the heat absorbed by the refrigerant in the evaporator. The total energy supplied to the system is the sum of work done by the pump and the heat supplied in the generator. Therefore, the coefficient of performance of the system is given by

$$\text{COP} = \frac{\text{Heat absorbed in evaporator}}{\text{Work done by pump+Heat supplied in generator}}$$

$$\text{COP} = \frac{T_e}{T_g}\left[\frac{T_g - T_c}{T_c - T_e}\right]$$

T_g = 150 + 273 = 423 K, T_c = 30 + 273 = 303 K, T_e = –20 + 273 = 253 K

$$\text{COP} = \frac{253}{423}\left[\frac{423-303}{303-253}\right] = 1.435$$

Now,

$$T_g = 200 + 273 = 473 \text{ K}$$

$$T_c = 30 + 273 = 303 \text{ K}$$

$$T_e = -40 + 273 = 233 \text{ K}$$

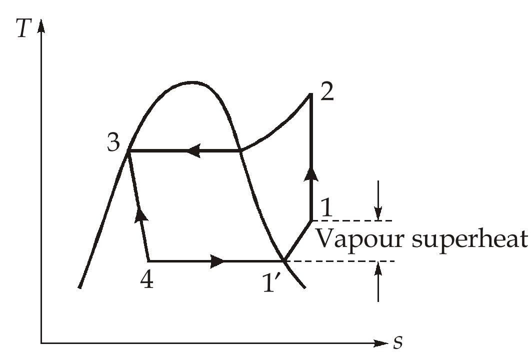
$$\text{COP} = \frac{233}{473}\left[\frac{473-303}{303-233}\right] = 1.2$$

$\therefore$ Percentage decrease in COP = $\dfrac{1.435 - 1.2}{1.435} \times 100\% = 16.4\%$

8. (c)

(i)

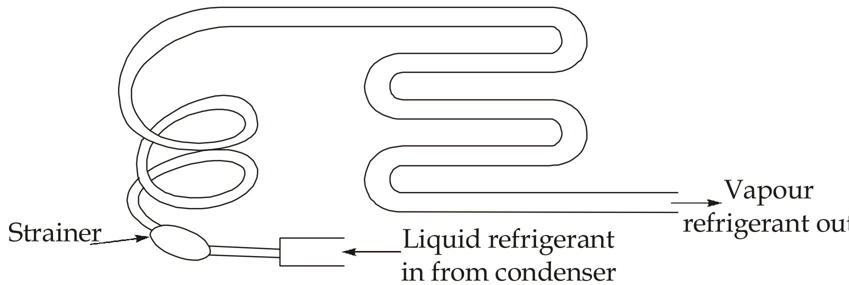
In a hermetically sealed compressor, both motor and compressor are housed in the same unit and have the same shaft.



In this case heat from motor winding is continuously heating to suction vapour which passes over it in the refrigeration circuit. This raises the temperature of suction vapour as motor cooling load also becomes a part of the overall cooling load of refrigeration system. Hence, suction vapour which enters at a saturated condition to the compressor gets heated by motor winding and always becomes super heated.

(ii)

Capillary tube is used as expansion device in small capacity hermetic sealed refrigeration units such as in domestic refrigeration, water coolers, room air conditioners and freezers. It is a copper tube of small internal diameter and of varying length depending upon the application. It is installed in the liquid line between the condenser and the evaporator. A fine mesh screen is provided at the inlet of the tube in order to protect it from contaminants.



Capillary tube

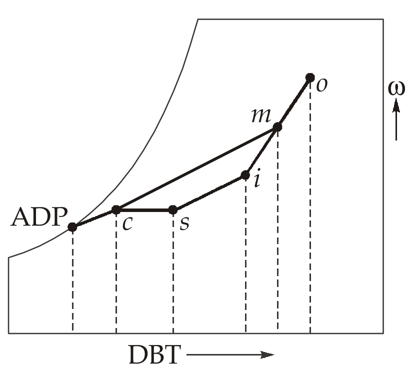
Advantages of using capillary tube in refrigeration system:

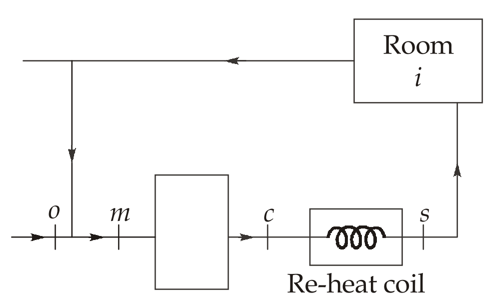
1. The cost of capillary tube is less than all the other form of expansion devices.
2. When the compressor stops, the refrigerant continues to flow into the evaporator and equalizes the pressure between the high side and low side of the system. This considerably decreases the starting load on the compressor. Thus a low starting torque motor (low cost motor) can be used to drive compressor.
3. Since the refrigerant charge in a capillary tube system is critical, therefore no receiver is necessary.

(iii)

When room sensible heating factor is low or latent heat factor is high, e.g. in cases of high humidity/high dehumidification needs or for high internal latent heat loads, then simple air conditioning system has very low coil ADP. This leads to very low evaporator pressure and reduces COP of the system and increases costs.

Hence, to increase ADP of cooling coil to managable temperature a reheat coil is introduced. Consider the setup as shown below and corresponding psychrometric chart.





Room air at state i mixes with ventilation air from surroundings at o and then enters cooling and dehumidification coil at state m. Exit happens at c post which it enters re-heat coil to achieve delivery temperature t_s.

(iv)

$$\text{COP of refrigeration system} = \frac{\text{Heat extracted in evaporator}}{\text{Work input}}$$

$$= \frac{\text{Heat extracted in evaporator}}{\text{Heat rejected in condensor} - \text{Heat extracted in evaporator}}$$

$$\text{COP}_C = \frac{T_0}{T_C - T_0}$$

For a water cooled condenser, coefficient of heat transfer is higher due to larger specific heat of water and higher thermal conductivity than air. Hence heat rejection takes place at a lower temperature for water as compared to air since

$$Q = UA\Delta T_{lm}$$

Since U is higher ΔT_{lm} is lower.

Since, heat rejection for water takes place at a lower temperature COP of refrigeration system is higher for a water condenser, as compared to an air condenser.

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