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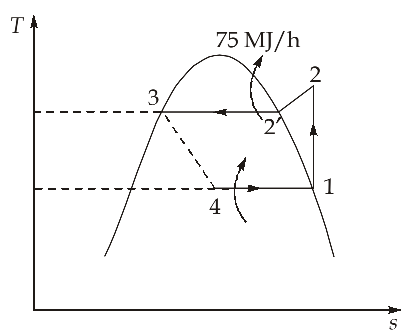
Detailed Solutions

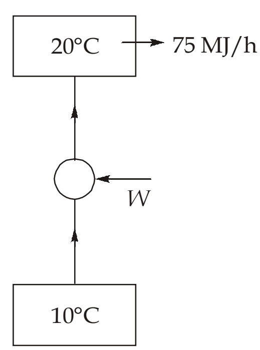
ESE-2019 Mains Test Series

Mechanical Engineering Test No : 12

Section-A

Q.1 (a) Solution:





In process 1–2 (isentropic process),

$$s_1 = s_2 \quad \text{...(1)}$$

$$s_1 = s_{g@320\text{ kPa}},\ s_2 = s_{g@800\text{ kPa}} + c_{p,v} \ln\left(\frac{T_2}{T_2'}\right)$$

$$T_2' = 31.317 + 273 = 304.317\text{ K}$$

from eq. (1),

$$1.72577 = 1.7140 + 1.0748 \ln\left(\frac{T_2}{304.317}\right)$$

$$T_2 = 307.667\text{K}$$

Now, enthalpy at state 2,

$$h_2 = h_g + c_{p,v}(T_2 - T_2')$$

$$= 414.7936 + 1.0748(307.667 - 304.317)$$

$$h_2 = 418.394 \text{ kJ/kg}$$

Now,

Heat rejected from house = $\dot{m} \times (h_2 - h_3)$ ...(2)

$$h_3 = h_{f@800 \text{ kPa}}$$

from eq. (2),

$$\frac{75\times10^3}{3600} = \dot{m}\times(418.394 - 243.6307)$$

$$\dot{m} = 0.1192 \text{ kg/s}$$

Now, power input to compressor,

$$P_{in} = \dot{m}\times(h_2 - h_1)$$

$$= 0.1192 \times (418.394 - 400.038)$$

$$P_{in} = 2.188 \text{ kW}$$

Now, Power saved = $\frac{75\times10^3}{3600} - 2.188$

$$P_{saved} = 18.645 \text{ kW}$$

Q.1 (b) Solution:

Since no water spills from the container, the air volume remains constant, that is,

$$\pi \times 10^2 \times 2 = \frac{1}{2}\pi R^2 \times 12$$

where we have used the fact that the volume of a paraboloid of revolution is one-half that of a circular cylinder with the same height and radius. This gives the value

$$R = 5.77 \text{ cm}$$

Using eq. with $r_2 = R$, we have

$$\frac{\omega^2 \times 0.0577^2}{2} = 9.81 \times 0.12$$

$$\therefore \quad \omega = 26.6 \text{ rad/s}$$

To find the pressure at point A, we simply calculate the pressure difference between A and O. Using Eq. with $r_2 = r_A = 0.1$ m, $r_1 = r_o = 0$, and $p_1 = p_o = 0$, there results

$$p_A = \frac{\rho\omega^2}{2}\left(r_A^2 - r_o^2\right)$$

$$= \frac{1000\times26.6^2}{2}\times0.1^2 = 3540 \text{ Pa or } 3.54 \text{ kPa}$$

The pressure at B can be found by applying Eq. to points A and B. This equation simplifies to

$$p_B - p_A = -\rho g(z_B - z_A)$$

Hence,

$$p_B = (3540) - (1000 \times 9.81 \times 0.12) = 2362.8 \text{ Pa or } 2.368 \text{ kPa}$$

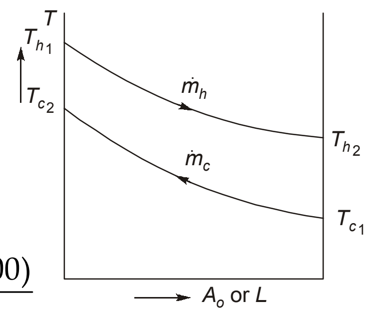
Q.1 (c) Solution:

By energy balance of the heat exchanger

$$Q = \dot{m}_h c_h (T_{h_1} - T_{h_2}) = \dot{m}_c c_c (T_{c_2} - T_{c_1})$$

$$= 2 \times 2330 \times (420 - 380)$$

$$\therefore \quad = 186{,}400 \text{ W}$$



To find T_{c2},

$$186{,}400 = 1 \times 4174\,(T_{c2} - 300)$$

$$\therefore \quad T_{c2} = 344.657 \text{ K}$$

$$\Delta T_m = \frac{(420 - 344.657) - (380 - 300)}{\ln \dfrac{420 - 344.657}{380 - 300}}$$

$$= \frac{75.343 - 80}{\ln \dfrac{75.343}{80}} = \mathbf{77.648\ K}$$

Now,

$$Q = U_0 A_0 \Delta T_m$$

$$186{,}400 = U_0 \times 3.33 \times 77.48$$

$$\therefore \quad U_0 = 720.894 \text{ W/m}^2\text{K}$$

Reduction in U_0 due to fouling = (930 – 720.894)/930 = 0.2248 or 22.48%

Q.1 (d) Solution:

Steam turbine power = mass of steam rate × (Adiabatic enthalpy drop),

$$P = 1.5 \times (3100 - 2700) = 600 \text{ kW}$$

Rate of heat addition to absorption system

$$= q_G = 1.5(2700 - 417.5) = 3423.75 \text{ kW}$$

Refrigeration capacity of vapour compression system

$$q_{E(\text{compression})} = P(\text{COP}) = 600 \times 3.8 = 2280 \text{ kW}$$

Refrigeration capacity of vapour absorption system

$$q_{E(\text{absorption})} = \text{Heat addition} \times (\text{COP})_{\text{absorption}}$$
$$= 3423.75 \times 0.65 = 2225.4375 \text{ kW}$$

Total refrigeration capacity

$$= q_{E(\text{total})} = q_{E(\text{comp})} + q_{E(\text{absorption})}$$
$$= 2280 + 2225.4375 = 4505.4375 \text{ kW}$$

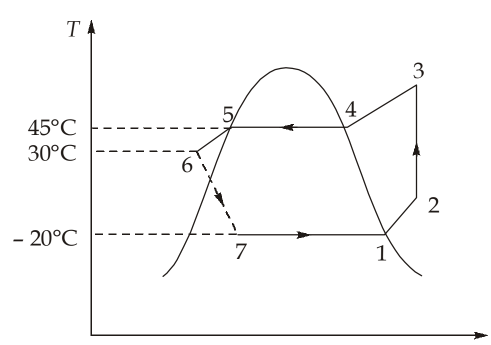
$$(\text{COP})_{\text{combined}} = \frac{q_{E(\text{total})}}{\text{Total heat supplied}} = \frac{4505.4375}{1.5(3100-417.5)} = 1.1197$$

Q.1 (e) Solution:

1. The maximum torque will be reduced because the b_{mep} is lower. Diesel engines can only operate smoke-free with air/fuel ratios weak of stoichiometric, say $\phi < 0.8$. The volumetric efficiency of the diesel engine will be lower than the spark ignition engine, because there is no evaporative cooling caused by fuel vaporization. There will thus be less fuel trapped in each cylinder, and although the efficiency of fuel conversion is greater in a diesel engine (say 40% instead of 30%) the maximum bmep in a diesel engine will be lower (say 7.5 bar, compared with a bmep of 10 bar for a spark ignition engine).
2. The maximum power will be lower because the torque is lower, and the maximum speed is lower and the power is the product of speed and torque.
3. The speed at which maximum power occurs is lower in the diesel engine than the spark ignition engine, because of the nature of diesel engine combustion. With spark ignition engines a flammable air fuel mixture is formed prior to ignition, whilst with diesel engines, the fuel has to be injected, then ignited, with the major part of combustion being diffusion controlled.

Q.2 (a) Solution:

i. Mass flow rate:



Refrigeration capacity = $\dot{m} \times RE$

$$9 \times 3.5 = \dot{m} \times (h_1 - h_7) \quad \text{...(i)}$$

$$h_1 = h_{g@-20°C}$$
$$h_1 = 178.7 \text{ kJ/kg}$$
$$h_7 = h_6$$
$$h_6 = h_5 - c_{pl}(T_5 - T_6)$$
$$= 79.7 - 1.02(45 - 30) \quad \{h_5 = h_{f@45°C}\}$$
$$= 64.4 \text{ kJ/kg}$$

from equation (i),

$$9 \times 3.5 = \dot{m} \times (178.7 - 64.4)$$

$$\dot{m} = 0.276 \text{ kg/s}$$

ii. Compression work:

$$W_c = \dot{m} \times (h_3 - h_2) \quad \text{...(ii)}$$

Now for h_2, from energy balance,

$$(h_2 - h_1) = (h_5 - h_6)$$
$$h_2 = h_1 + (h_5 - h_6)$$
$$h_2 = 178.7 + (79.7 - 64.4)$$
$$h_2 = 194 \text{ kJ/kg}$$

Process 2 -3,

$$s_2 = s_3 \quad \text{...(iii)}$$

$$s_2 = s_{g@-20°C} + c_{p,g} \ln\left(\frac{T_2}{T_1}\right)$$

For T_2,

$$(h_5 - h_6) = (h_2 - h_1)$$
$$c_{pl}(T_5 - T_6) = c_{p,g}(T_2 - T_1)$$
$$1.02 \times (45 - 30) = 0.61 \times (T_2 + 20)$$
$$T_2 = 5.0829°C = 5.082 + 273$$
$$T_2 = 278.082 \text{ K}$$

From eq. (iii),

$$s_{g@-20°C} + c_{p,g} \ln\left(\frac{T_2}{T_1}\right) = s_{g@45°C} + c_{p,g} \ln\left(\frac{T_3}{T_4}\right)$$

$$0.7088 + 0.61 \ln\left(\frac{278.082}{253}\right) = 0.6812 + 0.755 \ln\left(\frac{T_3}{318}\right)$$

$$T_3 = 356.017 \text{ K}$$

Now for h_3,

$$h_3 = h_{g@45°C} + c_{p,g}(T_3 - T_4)$$

$$= 204.9 + 0.755(356.017 - 318)$$

$$h_3 = 233.603 \text{ kJ/kg}$$

from eq. (ii),

$$W_c = 0.276(233.603 - 194)$$

$$W_c = 10.93 \text{ kW}$$

(iii) Heat rejected in condenser,

$$Q_c = \dot{m}(h_3 - h_5)$$

$$= 0.276(233.603 - 79.7)$$

$$Q_c = 42.477 \text{ kW}$$

$$\text{COP} = \frac{RE}{W_{net}} = \frac{h_1 - h_7}{h_3 - h_2} = \frac{178.7 - 64.4}{233.603 - 194}$$

$$\text{COP} = 2.866$$

Q.2 (b) Solution:

As we know

$$a = u\frac{\partial V}{\partial x} + v\frac{\partial V}{\partial y} + w\frac{\partial V}{\partial z} + \frac{\partial V}{\partial t}$$

$$= 20y^2(-20y\,\hat{j}) - 20xy(40y\,\hat{i} - 20x\,\hat{j})$$

$$= -800xy^2\,\hat{i} - 400(y^3 - x^2y)\,\hat{j}$$

where we have used $u = 20y^2$ and $v = -20xy$, as given by the velocity vector. All particles passing through the point (1, – 1, 2) have the acceleration

$$a = -800\left(1\times(-1)^2\,\hat{i}\right) - 400\left((-1)^3 - 1^2\times(-1)\right)\hat{j}$$

$$a = -800\hat{i} \text{ m/s}^2$$

The angular velocity has two zero components:

$$\omega_x = \frac{1}{2}\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right) = 0,\ \omega_y = \frac{1}{2}\left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right) = 0$$

The non-zero z-component is, at the point (1, – 1, 2),

$$\omega_z = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) = \frac{1}{2}(-20y - 40y) = 30 \text{ rad/s}$$

The vorticity vector is twice the angular velocity vector:

$$\omega = 2\omega_z\hat{k} = 60\hat{k} \text{ rad/s}$$

The non-zero rate-of-strain components are

$$\in_{xy} = \frac{1}{2}\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right) = \frac{1}{2}(-20y+40y) = -10 \text{ rad/s}$$

$$\in_{yy} = \frac{\partial v}{\partial y} = -20x = -20 \text{ rad/s}$$

All other rate of strain components are zero.

Q.2 (c) Solution:

The compression ratio is in the range 6 to 10 so this is a petrol engine.

1. Naturally aspirated engine (NA)

$$\text{BP} = \frac{P_m\times V_s\times N}{60\times 2} = \frac{10\times 100\times 0.004\times 5000}{120} = 166.667 \text{ kW}$$

$$\text{IP} = \frac{BP}{\eta_m} = \frac{166.667}{0.92} = 181.16 \text{ kW}$$

$$\text{Air-std efficiency, } \eta = 1-\frac{1}{r^{\gamma-1}} = 1-\frac{1}{8^{0.4}} = 0.5647$$

Given efficiency ratio = 0.55

So indicated thermal efficiency,

$$\eta_{I\,th} = 0.55 \times 0.5647 = 0.3106$$

We know,

$$\eta_{I\,th} = \frac{IP}{\dot{m}_f\times CV}$$

$$\text{mass flow rate of fuel, } \dot{m}_f = \frac{181.16\times 3600}{0.3106\times 43000} = 48.83 \text{ kg/h}$$

If the test duration is 't' hours then specific mass is given by

$$\left(\frac{\text{mass}}{IP}\right)_{NA} = \frac{258+48.83t}{181.16} = 1.424 + 0.2695t \quad \text{...(1)}$$

2. Super charged engine(s):

$$\text{BP} = \frac{P_m\times V_s N}{60\times 2} = \frac{15\times 100\times 0.004\times 5000}{120} = 250 \text{ kW}$$

$$\text{IP} = \frac{BP}{\eta_m} = \frac{250}{0.92} = 271.74 \text{ kW}$$

$$\text{air-std efficiency, } \eta = 1-\frac{1}{r^{\gamma-1}}$$

$$\eta = 1-\frac{1}{6^{0.4}} = 0.5116$$

Given efficiency ratio = 0.55

$$\therefore \quad \eta_{\text{Ith}} = 0.55 \times 0.5116 = 0.28138$$

$$\eta_{\text{Ith}} = \frac{IP}{\dot{m}_f \times CV}$$

$$\dot{m}_f = \frac{IP \times 3600}{\eta_{Ith} \times CV} = \frac{271.74 \times 3600}{0.28138 \times 43000}$$

$$\dot{m}_f = 80.85 \text{ kg/h}$$

If the test duration is 't' hours then specific mass is given by

$$\left(\frac{\text{Mass}}{IP}\right)_s = \frac{270 + 80.85t}{271.74} = 0.9936 + 0.2975t \quad \text{...(2)}$$

For the given condition,

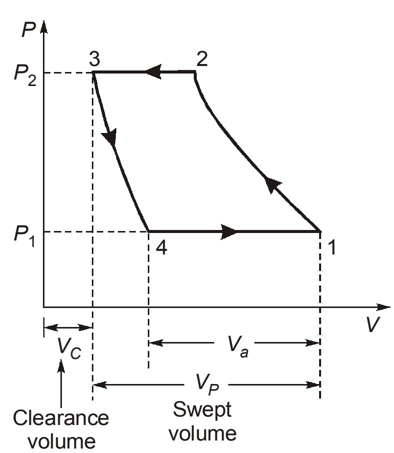
$$1.424 + 0.2695t = 0.9936 + 0.2975t$$

$$t = 15.37 \text{ h}$$

Q.3 (a) Solution:

(i) Consider a reciprocating compressor with swept volume V_p and volumetric efficiency η_V operating on cycle as shown in P-V diagram.

$$\text{Mass flow rate, } \dot{m} = \frac{\dot{V}_P}{v_1} \cdot \eta_V$$



Hence, we see for a given compressor swept volume V_p and suction vapour specific volume v_1,

$$\dot{m} \propto \eta_V$$

Also, $$\eta_V = 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/\gamma}$$

Hence, as pressure ratio $\left(\frac{P_2}{P_1}\right)$ is increased, volumetric efficiency decreases.

$\therefore$ So, $\left(\frac{P_2}{P_1}\right)$ increases, η_V decreases and mass flow rate also decreases.

(ii) A comparison between automatic expansion valve and thermostatic expansion valve can be drawn as follows:

Automatic expansion valve

1. Maintains constant evaporator pressure (and temperature)
2. When load increases, evaporator pressure rises to control due to which mass flow rate decreases.
3. When load decreases, mass flow rate increases.

Thermostatic expansion valve

1. Maintains constant degree of superheat in the evaporator
2. When load increases, degree of superheat increases and hence mass flow rate increase.
3. When load decreases, mass flow decreases.

   Hence, thermostatic expansion valve is a better choice for a throttling device since it regulates mass flow rate as per varying load and hence ensure effective utilization of evaporator.

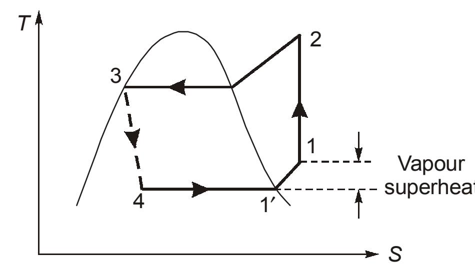
(iii) COP of refrigeration system $= \dfrac{\text{Heat extracted in evaporator}}{\text{Work input}}$

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

$$COP_C = \frac{T_0}{T_C - T_0}$$

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

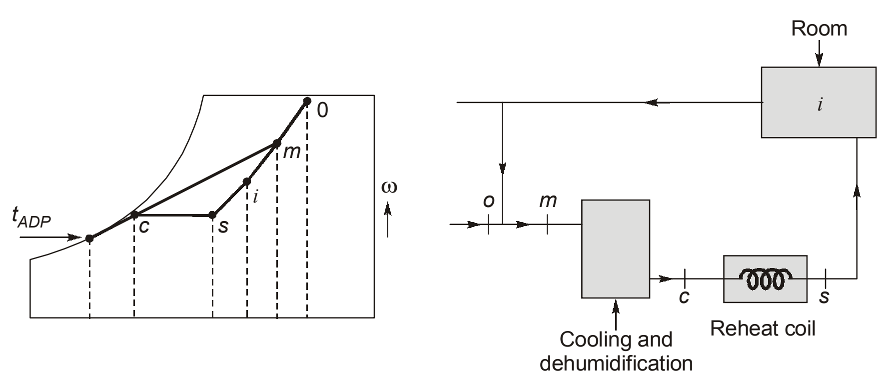
COP of refrigeration system is higher for a water condenser, as compared to an air condenser.

(iv) 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 transferred 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.

(v) 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 oil ADP. This leads to very low evaporator pressure and reduces COP of the system and increases costs.

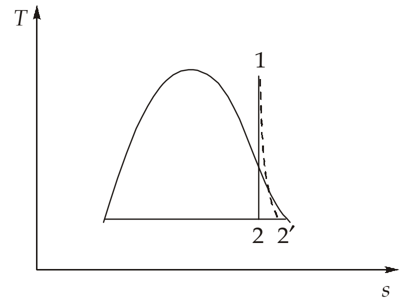
Hence, to increase ADP of cooling coil to manageable temperatures a reheat coil is introduced. Consider the setup as shown below and corresponding psychrometric chart.



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 reheat coil to achieve delivery temperature t_s.

Q.3 (b) Solution:

Assume process is isentropic,



$$s_1 = s_2$$
$$6.9589 = 0.6493 + x(8.1502 - 0.6493)$$
$$x = 0.8412$$

Enthalpy at exit,

$$h_2 = h_f + x(h_{fg})$$
$$h_2 = 191.83 + 0.8412(2584.7 - 191.83)$$
$$h_2 = 2204.712 \text{ kJ/kg}$$

Now if process is not isentropic,

$$\eta_{isen} = \frac{h_1 - h_{2'}}{h_1 - h_2}$$
$$h_{2'} = h_1 - \eta_{isen}(h_1 - h_2)$$
$$= 3633.7 - 0.8(3633.7 - 2204.712)$$
$$h_{2'} = 2490.51 \text{ kJ/kg}$$

Now, quality of steam at actual exit,

$$h_{2'} = h_f + x'(h_{fg})$$
$$2490.51 = 191.83 + x'(2584.7 - 191.83)$$
$$x' = 0.9606$$

Entropy at actual exit,

$$s_{2'} = s_f + s'(s_{fg})$$
$$= 0.6493 + 0.9606(8.1502 - 0.6493)$$
$$s_{2'} = 7.8546 \text{ kJ/kgK}$$

1. Exergy of incoming steam,

$$\phi_1 = (h_i - h_o) - T_o(s_i - s_o)$$
$$= (3633.7 - 125.79) - 303(6.9589 - 0.4369)$$
$$\phi_1 = 1531.74 \text{ kJ/kg}$$

2. Exergy of outgoing steam,

$$\phi_{2'} = (h_{2'} - h_o) - T_o(s_{2'} - s_o)$$

$$= (2490.51 - 125.79) - 303(7.8546 - 0.4369)$$

$$\phi_{2'} = 117.157 \text{ kJ/kg}$$

3. The work output of the turbine,

$$W = h_1 - h_{2'} \qquad \{\text{Neglecting change in kE and P.E.}\}$$

$$W = 3633.7 - 2490.51$$

$$W = 1143.19 \text{ kJ/kg}$$

4. Exergy loss in the process,

$$I = \phi_1 - \phi_{2'} - W$$

$$= 1531.74 - 117.157 - 1143.19$$

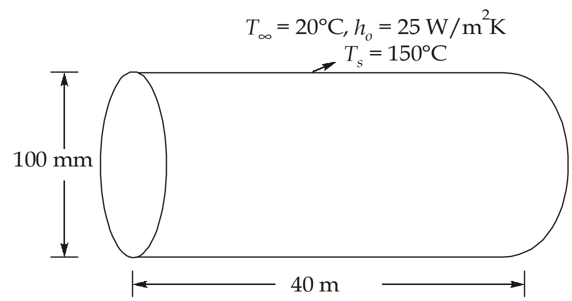
$$I = 271.393 \text{ kJ/kg}$$

Q.3 (c) Solution:

Ambient temperature, $T_\infty = 20°C$

Surface temperature of pipe, $T_s = 150°C$

Heat transfer coefficient, $h_o = 25 \text{ W/m}^2\text{K}$



So, rate of heat loss,

$$q = h_o A_o (T_s - T_\infty) = h_o(\pi d_o L)(T_s - T_\infty)$$

$$= 25 \times \left(\pi \times \frac{100}{1000} \times 40\right) \times (150 - 20) = 40840.7 \text{ W}$$

Total heat lost in one year,

$$Q = q \times (365 \times 24 \times 3600)$$

$$= 40840.7 \times 365 \times 24 \times 3600 = 1.288 \times 10^{12} \text{ J/yr}$$

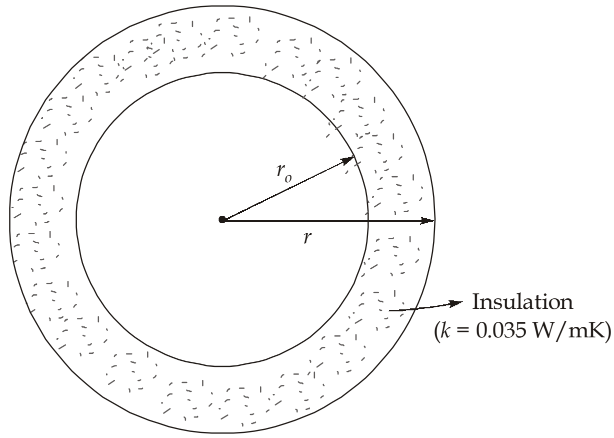
$$Q = \frac{1.288 \times 10^{12}}{1000 \times 3600} = 357764.57 \text{ kWh/yr}$$

Energy input to the natural gas furnace,

$$E = \frac{Q}{\eta_{\text{furnace}}} = \frac{357764.57}{0.8} = 447205.7 \text{ kWh/yr}$$

Annual cost of energy lost = price of gas × energy input to furnace

$$= 2 \times 447205.7 = \text{Rs.}894411.428$$



3. 90% reduction in heat transfer rate,

So, $q' = (1 - 0.9)q = 0.1q$

$$q' = 0.1 \times 40840.7 = 4084.07 \text{ W}$$

$$= \frac{T_o - T_\infty}{\frac{\ln(r/r_o)}{2\pi kL} + \frac{1}{h_o \pi d_o L}} = \frac{150-20}{\frac{\ln(r/50)}{2\pi \times 0.035 \times 40} + \frac{1}{25 \times \pi \times 0.1 \times 40}}$$

$$0.11368 \ln(r/50) + 3.1831 \times 10^{-3} = 0.031826$$

$$\ln\left(\frac{r}{50}\right) = 0.25196$$

$$r = 64.327 \text{ mm}$$

Thickness of insulation = $r - r_o$ = 64.327 - 50 = 14.327 mm

Q.4 (a) Solution:

Given: Time, t = 30 minutes, Cylinder diameter, d = 18 cm, Stroke, L = 24 cm, Speed, N = 300 rpm or Total number of revolutions = 9000 (in 30 min), Indicated mean effective pressure, P_{in} = 5 bar, Total number of explosions = 4425, Total gas consumption = 2.4 m^3, Calorific value, CV = 19000 kJ/m^3, ρ_{gas} = 1.275 kg/m^3, Air consumption = 32.1 m^3, ρ_{air} = 1.29 kg/m^3, $T_{exhaust}$ = 350°, $c_{p,\,gas}$ = 1 kJ/kg-K, Mass of cooling water, m_w = 120 kg, $(\Delta T)_w$ = 30°C, Net load, M = 38 kg

Effective drum diameter (d_m) = 1 m

Room temperature, T_0 = 27°C

$$\text{Brake Power, } BP = \left(\frac{W\pi d_m N}{60000}\right) = \frac{(38 \times 9.81)\pi \times 1 \times 300}{60000} = 5.852 \text{ kW} = 5.852 \text{ kJ/s}$$

Heat equivalent of BP = 5.852 × 60 kJ/min = 351.12 kJ/min

$$\text{Indicated Power, } IP = \frac{P_{im} \times \left(\frac{\pi}{4} D^2 L\right) \times \left(\frac{\text{No. of explosions}}{t(\text{in minutes})}\right)}{60 \times 1000}$$

$$= \frac{(5 \times 10^5) \times \frac{\pi}{4} \times 0.18^2 \times 0.24 \times \left(\frac{4425}{30}\right)}{60 \times 10^3}$$

$$= 7.50 \text{ kW}$$

$$\text{Heat lost to cooling medium, } \dot{Q}_c = \dot{m}_w \times c_w \times (\Delta T)_w = \left(\frac{120}{30}\right) \times 4.18 \times 30 = 501.6 \text{ kJ/min}$$

$$\text{Heat supplied} = \frac{2.4}{30} \times 19000 = 1520 \text{ kJ/min}$$

$$\text{Mass of air used} = \frac{(32.1 \times 1.29)}{30} = 1.38 \text{ kg/min}$$

$$\rho_{gas} = 1.275 \text{ kg/m}^3$$

$$\dot{m}_{gas} = \left(\frac{1.275 \times 2.4}{30}\right) = 0.102 \text{ kg/min}$$

$$\text{Total mass of exhaust gas, } \dot{m}_{exh} = \dot{m}_a + \dot{m}_{gas} = 1.38 + 0.102 = 1.482 \text{ kg/min}$$

$$\text{Heat lost to exhaust gas} = \dot{m}_{exh} \times c_{p.gas} \times (T_{exhaust} - T_0)$$

$$= 1.482 \times 1 \times (350 - 27) = 478.686 \text{ kJ/min}$$

Heat lost by radiation (unaccounted losses)

$$= 1520 - [351.12 + 501.6 + 478.686] = 188.594 \text{ kJ/min}$$

$$\text{Mechanical efficiency, } \eta_m = \frac{BP}{IP} \times 100 = \frac{5.852}{7.50} \times 100 = 78\%$$

$$\text{Indicated thermal efficiency, } \eta_{ith} = \frac{IP}{\text{Heat supplied}} = \frac{7.50}{(1520/60)} = 29.6\%$$

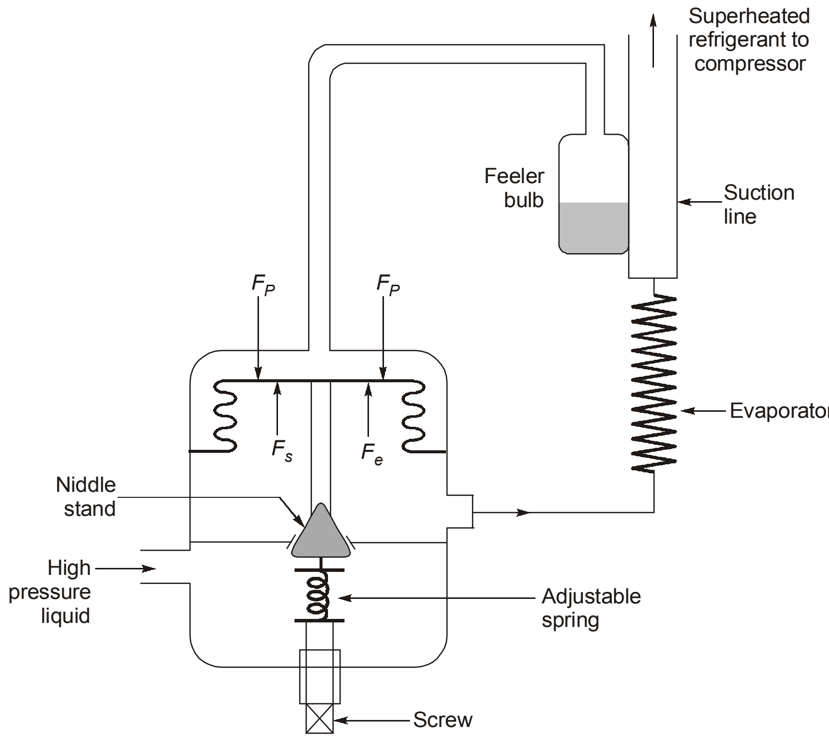
$$\text{Brake thermal efficiency, } \eta_{bth} = \frac{BP}{\text{Heat supplied}} = \frac{351.12}{1520} \times 100 = 23.1\%$$

Heat input (per minute)	(kJ)	Heat expenditure (per minute)	(kJ)	Percentage
Heat supplied	1520	1. Heat equivalent to BP	351.12	23.1%
		2. Heat cost to cooling medium	501.6	33%
		3. Heat lost to exhaust gas	478.686	31.49%
		4. Unaccounted losses	188.594	12.40%
		Total	1520	100%

Q.4 (b) Solution:

(i)

Thermostatic expansion valve: A thermostatic expansion valve is used to maintain a constant degree of superheat at the exit of evaporator, hence it is most effective for dry evaporators in preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. The schematic diagram of the valve is given as below:



It consists of a feeler bulb that is attached to the evaporator exit tube so that it senses the temperature at the exit of evaporator. The feeler bulb and the narrow tube contains some fluid that is called power fluid. The power fluid may be same as the refrigerant or it may be different. In case if it is different from the refrigerant then the TEV is called TEV with cross charge. Let P_p is the pressure of power fluid, P_e is the saturation pressure corresponding to evaporator exit temperature and evaporator temperature is T_e then the purpose of TEV is to maintain a temperature $(T_e + \Delta T_s)$ at evaporator exit where ΔT_s is the degree of superheat. Feeler bulb senses the temperature $(T_e + \Delta T_s)$ and its pressure P_p is saturation pressure at this temperature. So force exerted on the top area A_b of bellows:

$$F_p = P_p A_b \quad \text{...(i)}$$

Force exerted by evaporator pressure from bottom side of bellows:

This is called external equalizer if the evaporator is large and has significant pressure drop otherwise it is known as TEV with internal equalizer.

The difference of forces F_p and F_e is exerted on the top of the middle which controls the opening of orifice and is equal to spring force F_s i.e.

$$F_s = (P_p - P_e)A_b$$

Also
$$\Delta T_s \propto (P_p - P_e)A_b$$

As the compressor starts, P_e decreases so a positive spring force is applied on middle which opens the orifice and refrigerant flow starts.

(ii)

S.No.	Aspect	VARS	VCRS
1.	Quality of Energy	Low grade energy sources are more than capable of running VAR system. These sources can be waste heat from furnaces, exhaust steam etc. Solar power can also be used to run it.	VCR system needs high grade energy. It needs electrical or mechanical energy to run compressor which is essential part of VCR system
2.	Moving Parts in the system	The only moving part of VAR system is pump	In this moving part is compressor which is operated by electric motor.
3.	Effect of evaporator pressure	Very little effect is seen in refrigeration capacity with lowering evaporator pressure.	Performance reduces with varying load.
4.	Evaporator exit	In VAR system, if the liquid refrigerant leaves the evaporator, the refrigeration effect is reduced but the system functions well.	Liquid refrigerant at exit of evaporator is not desirable in this because it may damage the compressor. So in VCR system refrigerant at exit of evaporator is in superheated state.
5.	COP	The COP of VAR system is poor.	The COP of VCR system is very good
6.	Workability at varying load	Load variation does not have any effect on VAR system	At partial load VCR system does not work well and its performance is very poor
7.	Lowest Temperature	When water is used as refrigerant then the temperature attained is above 0°C	–150°C or even lower temperature can be achieved by the cascading system.
8.	Capacity	Capacity above 1000 TR is easily achievable	It is difficult to achieve capacity above 1000 TR with single compression system
9.	Noise and Vibration	Noise and vibration are minimised	High noise and vibration
10.	Maintenance	Extremely low maintenance is required	Usual maintenance is required.

Q.4 (c) Solution:

Temperature recorded by thermo couple, T_t = 580 K

Wall temperature, T_w = 300 K

Let surface area of thermocouple be A_t and surface area of shield be A_s.

Net radiation heat transfer rate per unit area between thermocouple and the wall.

$$\dot{q}_{\text{rad}} = \frac{\sigma\left(T_t^4 - T_w^4\right)}{\frac{1-\varepsilon_t}{\varepsilon_t A_t} + 2\left(\frac{1-\varepsilon_s}{\varepsilon_s A_s}\right) + \frac{1}{A_t F_{ts}} + \frac{1}{A_s F_{sw}} + \frac{1-\varepsilon_w}{\varepsilon_w A_w}}$$

$$= \frac{\sigma\left(T_t^4 - T_w^4\right)A_t}{\left(\frac{1}{\varepsilon_t} - 1\right) + \frac{2A_t}{A_s}\left(\frac{1}{\varepsilon_s} - 1\right) + 1 + \frac{A_t}{A_s}}$$

$$q = \frac{\sigma\left(T_t^4 - T_w^4\right)A_t}{\frac{1}{\varepsilon_t} + \frac{A_t}{A_s}\left(\frac{2}{\varepsilon_s} - 1\right)}$$

$$\dot{q}_{\text{rad}} = \frac{\sigma\left(T_t^4 - T_w^4\right)}{\left(\frac{1}{\varepsilon_t}\right) + \left(\frac{2}{\varepsilon_s} - 1\right)} \quad \text{(Because } A_t \simeq A_s\text{)}$$

$$= \frac{5.67 \times 10^{-8}\left(580^4 - 300^4\right)}{\left(\frac{1}{0.8}\right) + \left(\frac{2}{0.2} - 1\right)} = 581.189 \text{ W/m}^2$$

Heat transfer balance,

Heat convected to thermocouple = Heat radiated to wall

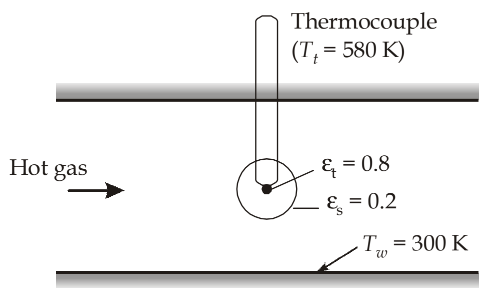
$$h(T_G - T_t) = 581.189$$

$$\Rightarrow \quad 120(T_G - 580) = 581.189$$

$$T_G = 584.843 \text{ K}$$

So, actual temperature of gas, $T_G = 584.843$ K

2. No radiation shield is used,



Net radiation heat transfer rate per unit area between thermocouple and the wall,

$$q_{\text{rad}} = \varepsilon_t \sigma\left(T_t^4 - T_w^4\right)$$

$$= 0.8 \times 5.67 \times 10^{-8} \times \left(T_t^4 - 300^4\right)$$

Convection heat transfer rate,

$$q_{\text{conv}} = h(T_G - T_t) = 120(584.843 - T_t)$$

Heat balance:

$$0.8 \times 5.67 \times 10^{-8} \times \left(T_t^4 - 300^4\right) = 120(584.843 - T_t)$$

$$4.536 \times 10^{-8} T_t^4 + 120T_t - 70548.576 = 0$$

On solving, $T_t = 552.645$ K

Section B

Q.5 (a) Solution:

Theoretical air required per kg fuel,

$$W_{Th} = 11.5C + 34.5\left(H - \frac{O}{8}\right) + 4.3S$$

$$= 11.5 \times 0.78 + 34.5\left(0.03 - \frac{0.03}{8}\right) + 4.3 \times 0.01$$

$$= 9.918625 \text{ kg air/kg fuel}$$

Actual air required, $W_A = 1.3 \times 9.918625$

$$= 12.8942125 \text{ kg air/kg fuel}$$

We know that, specific volume of air at room temperature,

$$\nu_{air} = \frac{RT}{P} = \frac{0.287 \times 303}{101.325} = 0.85824 \text{ m}^3/\text{kg}$$

$$\text{FD fan motor capacity} = \frac{W_A \times \dot{m}_f \times \nu_{air} \times \Delta P}{\eta_{fan}}$$

$$= \frac{12.8942125 \times 10 \times 10^3 \times 0.85824 \times 180 \times 10^{-3} \times 9.81 \times 1000}{3600 \times 0.6}$$

$$= 90467.24 \text{ Watt} = 90.467 \text{ kW}$$

Q.5 (b) Solution:

Centrifugal:

(i) The flow is radial

(ii) The pressure ratio per stage is high about 5:1. Thus the unit is compact. In supersonic compressors, the pressure ratio per stage is about 10 but at the cost of efficiency. Operation is not so difficult and risky.

(iii) The isentropic efficiency is about 82%.

(iv) Centrifugal compressors have a wide range of operation between surging and chocking limit. The head capacity curve is flat. The part load performance is better.

(v) Centrifugal compressors have a larger frontal area and thus produces more drag force for the same mass flow rate and pressure ratio if used in aviation,

(vi) When working with the contaminating fluids, the accumulation of deposits on the surface of flow passage do not adversely affect the performance.

(vii) It needs low starting torque

(viii) Its construction is simple, rigid and relatively cheap. At high altitude it is less sensitive to icing trouble

(xi) Multistaging is slightly difficult and upto 400 bar delivery pressure is possible

(x) It is used in blowing engines in steel mills, low pressure refrigeration, big central air conditioning plants, fertilizer industry, supercharging, gas pumping in long distance pipe line, petrochemical industries. Previously, it was used in jet engines, small air craft and gas turbines.

Axial:

(i) The flow is axial i.e., parallel to direction of the axis of machine.

(ii) The pressure ratio per stage is low, about 1.2:1. This is due to absence of centrifugal action. To achieve the pressure ratio equal that of per stage centrifugal, 10 to 12 stage are required. Thus, the unit is less compact and less rugged. The pressure ratio per stage in supersonic compressor is about 10 but the efficiency drops rapidly. The compression is achieved due to shock wave and the operation is risky.

(iii) The isentropic efficiency is about 88%.

(iv) Axial compressors have a narrow range of operation between surging and chocking limit. The part load performance is poor.

(v) Axial flow compressors have a small frontal area for the same mass flow rate and pressure ratio than that of centrifugal. This makes the axial flow compressors more suitable for jet engines. It is invariably used in aviation and power gas turbines.

(vi) The accumulation of deposits on the surface of flow passages affect adversely the performance of axial flow compressors.

(vii) It needs high starting torque.

(viii) Its construction is complex and costly. It is sensitive to icing troubles at high altitude.

(xi) It is most suitable for multistaging and upto 35 bar delivery pressure in a single casing is possible.

(x) Due to higher efficiency and smaller frontal area, axial flow compressors are mostly used in jet engines. In power plant gas turbines, axial compressors are invariably used.

Q.5 (c) Solution:

For safe and efficient operations, boilers are equipped with boiler mountings and accessories.

Boiler mountings: Boiler mountings are those machine components which are mounted over the body of the boiler itself for the safety of the boiler and for complete control of the process of steam generation. These mountings form an integral part of the boiler. Following are mountings that should be fitted on the boiler:

1. Two safety valves.
2. Two water level indicators.
3. Pressure gauge.

4. Fusible plug.
5. Steam stop valve.
6. Feed check valve.
7. Blow off cock.
8. Inspector test gauge.
9. Man and mud hole

Among the above mounting, safety valve, water level indicator and fusible plug are called safety fittings and the remaining are the control fittings.

Boiler accessories: These are those machine components which are installed either inside or outside the boiler to increase the efficiency of the plant and or to help in the proper working of the plant. The following accessories are generally used in the boiler.

1. Air preheater
2. Economiser
3. Superheater
4. Feed pump
5. Steam trap
6. Steam separator
7. Pressure reducing valve

Q.5 (d) Solution:

Given : Scale ratio (L_r) = 10, P_m = 1.84 kW, H_m = 5 m, N_m = 480 rpm, H_p = 40 m

$$\frac{H_m}{D_m^2 N_m^2} = \frac{H_p}{D_p^2 N_p^2}$$

$$N_p^2 = \left(\frac{H_p}{H_m}\right)\left(\frac{D_m}{D_p}\right)^2 \times N_m^2$$

$$N_p = \sqrt{\frac{H_p}{H_m}}\left(\frac{D_m}{D_p}\right) \times N_m$$

$$N_p = \sqrt{\frac{40}{5}} \times \left(\frac{1}{10}\right) \times 480 = 135.76 \text{ rpm}$$

$$\frac{P_m}{D_m^2 H_m^{3/2}} = \frac{P_p}{D_p^2 H_p^{3/2}}$$

$$P_p = 1.84 \times \left(\frac{10}{1}\right)^2 \times \left(\frac{40}{5}\right)^{3/2} = 4163.4 \text{ kW}$$

$$(N_s)_m = \frac{N_m\sqrt{P_m}}{H_m^{5/4}} = \frac{480\sqrt{1.84}}{(5)^{5/4}} = 87.08 \text{ units}$$

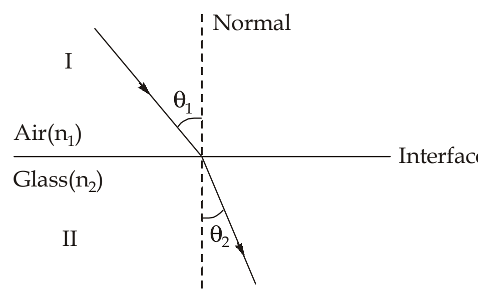
$$(N_s)_p = \frac{N_p\sqrt{P_p}}{H_p^{5/4}} = \frac{135.76\sqrt{4163.4}}{(40)^{5/4}} = 87.08 \text{ units}$$

Thus, $(N_s)_m = (N_s)_p$

We can observe that specific speeds are same for model and prototype. This is the speed of a member of the same homologous series as the actual turbine, so reduced in size as to generate unit power under a unit head of the fluid.

Q.5 (e) Solution:

According to Snell's law,



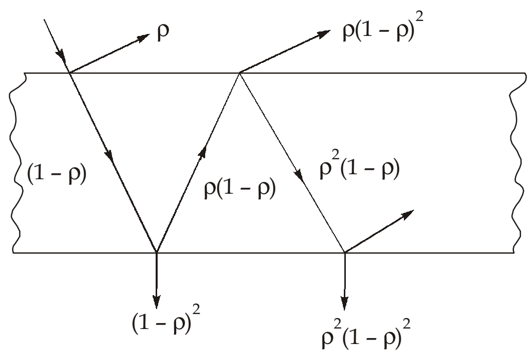
$$\frac{\sin\theta_1}{\sin\theta_2} = \frac{n_2}{n_1}$$

$$\theta_2 = \sin^{-1}\left(\frac{n_1}{n_2}\times\sin\theta_1\right)$$

where, $\rho = \frac{1}{2}(\rho_1+\rho_2)$

$$\rho_1 = \frac{\sin^2(\theta_2-\theta_1)}{\sin^2(\theta_2+\theta_1)} \text{ and } \rho_2 = \frac{\tan^2(\theta_2-\theta_1)}{\tan^2(\theta_2+\theta_1)}$$

Similarly, $\tau = \frac{1}{2}(\tau_1+\tau_2)$



$$\tau = (1-\rho)^2 + \rho^2(1-\rho)^2 + \rho^4(1-\rho)^2 + \ldots\ldots\infty$$

$$\tau = \frac{(1-\rho)^2}{1-\rho^2} = \frac{1-\rho}{1+\rho}$$

for $$\tau_1 = \frac{1-\rho_1}{1+(2M-1)\rho_1}$$

$$\tau = \frac{1}{2}(\tau_1 + \tau_2)$$

For numerical data;

Incidence angle, θ_1 = 45°

refractive index of air, n_1 = 1

refractive index of glass, n_2 = 1.5

Number of covers, M = 2

$$\theta_1 = 45°, \frac{n_1}{n_2} = \frac{1}{1.5}, M = 2$$

$$\theta_2 = \sin^{-1}\left(\sin\theta_1 \times \frac{n_1}{n_2}\right) = 28.125$$

$$\rho_1 = 0.092, \rho_2 = 0.008$$

Then, $$\tau_1 = 0.712$$

$$\tau_2 = 0.968$$

$$\tau = \frac{1}{2}(\tau_1 + \tau_2) = 0.84$$

Q.6 (a) Solution:

We know that

$$\frac{n-1}{n} = \left(\frac{\gamma-1}{\gamma}\right)\eta_p$$

where η_p is polytropic efficiency of turbine,

$$\frac{T_{02}}{T_{01}} = \left(\frac{p_{02}}{p_{01}}\right)^{\frac{n-1}{n}} = \left(\frac{p_{02}}{p_{01}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)0.82}$$

$$T_{02} = T_{01} \times \left(\frac{0.7}{7}\right)^{\left(\frac{1.3-1}{1.3}\right)0.82}$$

$$T_{0.2} = 773 \times 0.6479 = 500 \text{ K}$$

The specific volume v_{02} corresponding to stagnation conditions at outlet is determined with the superheated steam relation $pv = 0.231(h - 1943)$ and the perfect gas law $pv = RT$.

$$p_{01}v_{01} = 0.231(h_{01} - 1943) = RT_{01}$$
$$p_{02}v_{02} = 0.231(h_{02} - 1943) = RT_{02}$$

$\therefore$
$$\frac{T_{02}}{T_{01}} = \frac{h_{02}-1943}{h_{01}-1943}$$

$$\frac{500}{773} = \frac{h_{02}-1943}{3410-1943}$$

$\therefore$
$$h_{02} = 2891.9 \text{ kJ/kg}$$

So,
$$p_{02}v_{02} = 0.231(2891.9 - 1943)$$

$$v_{02} = \frac{219.1959}{700} = 0.3131 \text{ m}^3/\text{kg}$$

ii. The reheat factor is defined as,

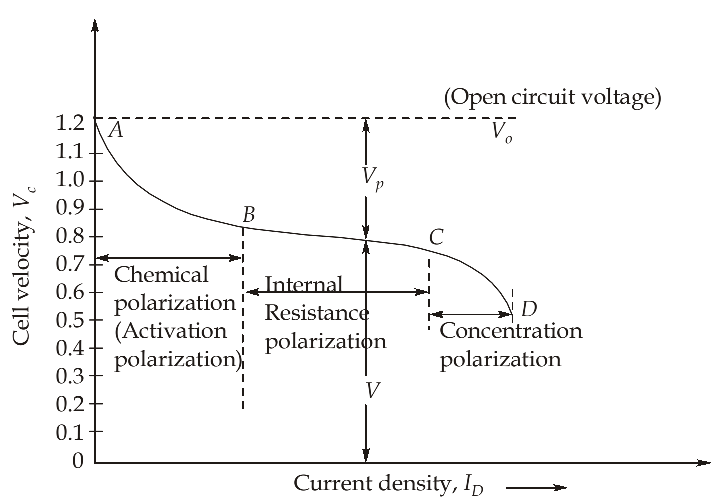
$$\eta_{is} = \eta_p \times R.F$$

$$R.F = \frac{\eta_{is}}{\eta_p}$$

$$\eta_{is} = \frac{T_{01}-T_{02}}{T_{01}-T_{02s}} = \frac{\left(1-\frac{T_{02}}{T_{01}}\right)}{\left(1-\frac{T_{02s}}{T_{01}}\right)} = \frac{\left(1-\frac{500}{773}\right)}{\left(1-\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}\right)} = 0.8568$$

$\therefore$
$$R.F = \frac{0.8568}{0.82} = 1.045$$

Q.6 (b) Solution:



V-I characteristics of fuel cell (Polarization curve)

The theoretical emf(E) of a fuel cell can be calculated from the change in Gibbs free energy (ΔG) during the reaction. The actual value of cell output voltage, V obtained during on load condition is less than E. The difference between actual and theoretical voltage is known as polarization, V_p. The effect of polarization is to reduce the voltage and thereby efficiency of the cell from its maximum value.

Voltage regulation is poor for small and large values of output current. Therefore, in practice the operating point is fixed in the range BC of the characteristics curve where voltage regulation is best and the output voltage is around 0.6 to 0.8 Volt.

At no load, the terminal voltage is equal to the theoretical open circuit voltage. As the cell is loaded, voltage and hence efficiency drops significantly.

The departure of output voltage from ideal emf is mainly due to the following reasons.

i. Activation polarization (chemical polarisation): This is related to activation energy barrier for the electron transfer process at the electrode. Certain minimum activation energy is required to be supplied so that sufficient number of electrons is emitted. At low current densities significant numbers of electrons are not emitted. This energy is supplied by the output of the cell, resulting in potential loss. It can be reduced by an effective electrochemical catalyst and also by increasing the operating temperature.

ii. Internal resistance polarisation: When a fuel cell operates, the ions liberated at one electrode move to the other electrode through the electrolyte causing flow of current in the external circuit. The internal resistance is the total of electrode resistance, the contact resistance between electrode and electrolyte and the electrolyte resistance. It is represented by the curve part BC.

It can be reduced by decreasing teh electrode size, coating the electrodes with a good electric conductor, increasing the electrolyte concentration and reducing the distance between the two electrodes.

iii. Concentration polarisation: This type of polarisation tends to limit the current drawn from the cell. This is related to mass transport within the cell and may further be subdivided into two parts.

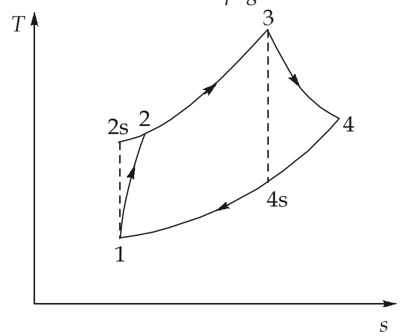
(a) Electrolyte polarisation: It is due to slow diffusion in the electrolyte causing change in concentration at the electrode. This effect can be reduced by increasing the electrolyte concentration or by stirring/circulating the electrolyte.

(b) Gas-side polarisation: It is caused due to slow diffusion of reactants through porous electrode to the site of reaction or slow diffusion of product away from the reaction site. Increasing the operating temperature also reduces this effect.

Note : All the losses in the fuel cell are reduced as operating temperature is increased. Therefore, in practice, a fuel cell is usually operated at the higher end of its operating temperature range.

Q.6 (c) Solution:

Given, T_1 = 300 K, η_c = 70.42%, η_T = 71%, $(c_p)_g$ = 1.147 kJ/kgK.



$$\text{Work ratio} = \frac{W_T - W_C}{W_T}$$

$$0.0544 = 1 - \frac{W_C}{W_T}$$

$$\frac{c_p(T_2 - T_1)}{c_p(T_3 - T_4)} = 1 - 0.0544$$

$$\frac{(T_{2s} - T_1)}{\eta_c \times (T_3 - T_{4s}) \times \eta_T} = 0.9456$$

$$\frac{T_1}{T_3} \frac{\left(\frac{T_{2s}}{T_1} - 1\right)}{\left(1 - \frac{T_{4s}}{T_3}\right) \times \eta_c \times \eta_T} = 0.9456$$

Let, pressure ratio is r_p ratio of maximum temperature to minimum temperature is t.

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}} = C$$

$$\frac{T_3}{T_{4s}} = (r_p)^{\frac{\gamma-1}{\gamma}} = C$$

$$\frac{T_3}{T_1} = t$$

$$\frac{(C-1)}{\left(1 - \frac{1}{c}\right) t \times \eta_c \times \eta_T} = 0.9456$$

$$\left(\frac{C}{t}\right) = 0.9456 \times 0.7042 \times 0.71 = 0.472783$$

$$C = (0.472383)t \quad ...(1)$$

Heat added, $Q = c_p(T_3 - T_2)$

We know that, $T_2 = T_1 + \frac{(T_{2s} - T_1)}{\eta_c} = T_1 + \frac{T_1}{\eta_c}(c-1)$

Now, $Q = (c_p)_g(T_3 - T_2)$

$$476.354 = 1.147\left[T_3 - \left(T_1 + \frac{T_1 C}{\eta_c} - \frac{T_1}{\eta_c}\right)\right]$$

$$415.304 = T_1\left[\frac{T_3}{T_1} - \left(1 + \frac{0.472783t}{0.7042} - \frac{1}{0.7042}\right)\right]$$

$$415.304 = 300\left[t - (1 + 0.67137t - 1.42)\right]$$

$$1.38434 = \left(\frac{1.38434 - 0.42}{0.32863}\right)$$

$$t = 2.9344$$

Now, $c = (0.472783)t$

$$c = (0.472783) \times 2.9344$$

$$c = 1.38733$$

$$(r_p)^{\frac{\gamma-1}{\gamma}} = 1.38733$$

$$(r_p) = (1.38733)^{1.4/1.4-1}$$

Pressure ratio, $r_p = 3.145$

Q.7 (a) Solution:

Density of air, $\rho = 1.226 \text{ kg/m}^3$

Rotor diameter, $D = 60$ m

We know that,

Velocity, $u \propto H^\alpha$

$$\frac{u_2}{u_1} = \left(\frac{H_2}{H_1}\right)^\alpha$$

$$u_2 = 8 \times \left(\frac{80}{10}\right)^{0.13}$$

Upstream velocity, $u_2 = 10.483$ m/s

1. Power available in the wind, $P = \frac{1}{2}\rho A u_2^3 = \frac{1}{2} \times 1.226 \times \frac{\pi}{4} \times D^2 \times (10.483)^3$

$$= \frac{1}{2} \times 1.226 \times \frac{\pi}{4} \times (60)^2 \times (10.483)^3$$

$$= 1996.685 \times 10^3 \text{ Watt}$$

$$P_{ava.} = 1996.685 \text{ kW}$$

2. Power extracted from the turbine, $P_{ex.} = \frac{1}{2}\dot{m}\left[u_2^2 - v_1^2\right]$

where, u_2 = Upstream velocity
v_1 = Downstream velocity
v = Velocity while passing through the rotor

$$v = \frac{u_2 + v_1}{2}$$

Power extracted from the turbine, $P_{ex.} = \frac{1}{2}\rho A \times v \times \left[u_2^2 - v_1^2\right]$

$$= \frac{1}{2} \times 1.226 \times \frac{\pi}{4} \times 60^2 \times \left[\frac{u_2 + v_1}{2}\right] \times \left[u_2^2 - v_1^2\right]$$

where, $v_1 = \frac{u_2}{2}$

$$P_{ex.} = \frac{1}{2} \times 1.226 \times \frac{\pi}{4} \times 60^2 \times \left[\frac{u_2 + 0.5u_2}{2}\right] \times \left[u_2^2 - (0.5u_2)^2\right]$$

$$= 0.5 \times 1.226 \times \frac{\pi}{4} \times 60^2 \times \left(\frac{3}{4}u_2\right) \times \left(\frac{3}{4}u_2^2\right)$$

$$= 0.5 \times 1.226 \times \frac{\pi}{4} \times 60^2 \times \left(\frac{9}{16}\right) \times (10.483)^3$$

$$= 1123.135 \times 10^3 \text{ Watt}$$

$$P_{ex.} = 1123.135 \text{ kW}$$

3. Axial force on the turbine, $F_{axial} = \dot{m}(u_2 - v_1)$

$$= \rho A \times v \times (u_2 - v_1) = \rho A \times \frac{(u_2 + v_1)}{2} \times (u_2 - v_1)$$

$$= \frac{1}{2} \times 1.226 \times \frac{\pi}{4} \times 60^2 \times \left[u_2^2 - (0.5u_2)^2\right]$$

$$= \frac{1}{2}\times 1.226\times\frac{\pi}{4}\times 60^2\times\frac{3}{4}\times(10.483)^2 = 142.851 \times 10^3 \text{ N}$$

$$F_{axial} = 142.851 \text{ kN}$$

4. For maximum power,

p. $$v_1 = \frac{u_2}{3},\ v = \frac{2}{3}u_2$$

$$F_{axial} = \dot{m}(u_2 - v_1) = \rho A\times v\times(u_2 - v_1)$$

$$= 1.226\times\frac{\pi}{4}\times 60^2\times\frac{2}{3}u_2\left(u_2 - \frac{u_2}{3}\right)$$

$$= 1.226\times\frac{\pi}{4}\times 60^2\times\left(\frac{4}{9}\right)\times(10.483)^2 = 169.305 \times 10^3 \text{ N}$$

$$F_{axial} = 169.305 \text{ kN}$$

q. When the wind speed at the turbine is reduced to zero ($v = 0$), in this condition no power is extracted. This state is known as stall state of blades.

$$F = \dot{m}(u_2 - v_1) = \rho A v(u_2 - v_1)$$

$$F = 0 \text{ Newton} \qquad \{\text{As, } v = 0 \text{ in case of stalling}\}$$

5. Blade tip ratio, $$\lambda = \frac{\text{Blade outer tip speed}}{\text{Upstream velocity}} = \frac{R\times\omega}{u_2}$$

$$9 = \frac{30\times\omega}{10.483}$$

$$\omega = 3.1449 \text{ rad/s}$$

$$\frac{2\pi N}{60} = 3.1449$$

$\Rightarrow$ $$N = \frac{60\times 3.1449}{2\pi} = 30.03 \text{ rpm}$$

Q.7 (b) Solution:

(i)

A two-stage, single-acting, reciprocating air compressor with perfect intercooling

$$\dot{m}_a = 2 \text{ kg/s} \qquad p_1 = 100 \text{ kPa}$$

$$T_1 = 35°\text{C} = 308 \text{ K} \qquad p_3 = 9 \text{ bar}$$

$$n = 1.3$$

Assumptions:

- For air, R = 0.287 kJ/kgK and c_p = 1 kJ/kgK.
- No effect of valve opening and closing on induction and delivery processes.

Analysis

For perfect intercooling

or $$p_2 = \sqrt{p_1 \times p_3} = \sqrt{1 \times 9} = 3 \text{ bar}$$

or $$\frac{p_2}{p_1} = 3$$

Power required to drive the compressor:

The minimum power input for two-stage compression with perfect intercooling.

$$IP_{\text{multi}} = 2\times\left(\frac{n}{n-1}\right)\dot{m}_a\ RT_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

$$= 2\times\left(\frac{1.3}{1.3-1}\right)\times 2\times 0.287\times 308\times\left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{2\times 1.3}} - 1\right]$$

$$= 442.1 \text{ kW}$$

Percentage saving in work of comparison with single-stage compression:

Power input with single stage compression from 1 bar to 9 bar.

$$IP_{\text{single}} = \frac{n}{n-1}\dot{m}_a RT_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{1.3-1}\times 2\times 0.287\times 308\times\left[\left(\frac{9}{1}\right)^{\frac{1.3-1}{1.3}} - 1\right] = 505.92 \text{ kW}$$

Saving in power due to multistage compression

$$= IP_{\text{single}} - IP_{\text{multi}} = 505.92 - 442.1 = 63.82 \text{ kW}$$

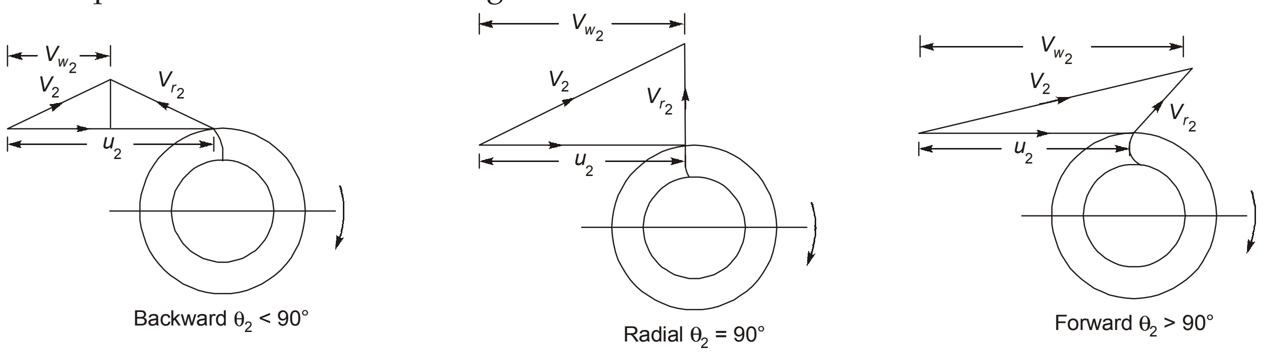
Percent saving $$= \frac{IP_{\text{multi}} - IP_{\text{single}}}{IP_{\text{single}}}\times 100 = \frac{63.82}{505.92}\times 100 = 12.61\ \%$$

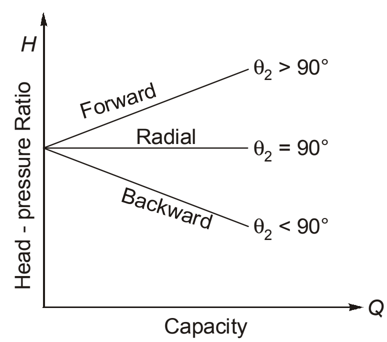
(ii)

For backward curved vanes, $\theta_2 < 90°$ and cot θ_2 is positive. It is apparent that an increase in mass flow rate, the Euler head H goes on falling. In other words the head-capacity characteristic has a negative slope.

For radial vanes, $\theta_2 = 90°$ and $\cot\theta_2 = 0$. Thus the head remains constant with variation in mass flow rate. For forward curved vanes, $\theta_2 > 90°$ and $\cot\theta_2$ is negative. Consequently with increase in mass flow rate, the Euler head would rise and the head capacity characteristic will have a positive slope.

The velocity diagrams and the theoretical head-capacity variation for different type of impeller vanes are shown in figure.





Characteristics of backward-curved, radial and forward-curved vanes

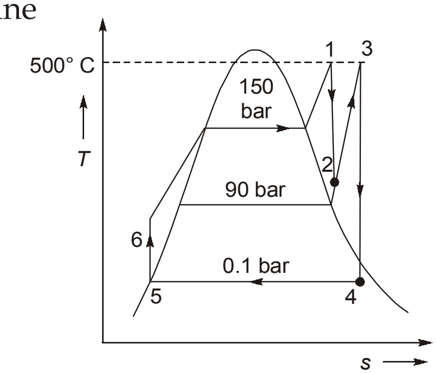
- It may be seen from figure that for backward-curved vanes the tangential component V_{w2} is much reduced and consequently for a given impeller speed, the impeller will have a low energy transfer (equation $\dot{m}V_{w_2}U_2$).
- In the case of forward curved vanes, V_{w2} is increased and consequently the energy transfer for forward curved vanes is maximum. However the absolute velocity at impeller outlet (V_2) is also increased. The high value of V_2 is not desirable as its conversion into static pressure cannot be very efficiently carried out in the diffuser section. In the diffusion process, there is always a tendency for the air to break away from the walls of the diverging passages. If the diffusion is too rapid, i.e., it is carried out in a small diffuser section, the air may reverse its direction and flow back in the direction of pressure gradient. The reversal results in the formation of eddies and turbulence which cause conversion of some of kinetic energy into heat rather than useful pressure energy.

- Normally backward vanes with θ_2 between 20 – 25° are employed except in the case where high head is the major consideration. Sometimes compromise is made between the low energy transfer (backward curved vanes) and high outlet velocity (forward curved vanes) by using radial vanes. Moreover the radial vanes can be manufactured easily and are free from complex bending stresses.

Q.7 (c) Solution:

Rankine cycle with reheat, superheated steam

(i) Steam expands to 90 bar in the first stage of the turbine



Given data:

$$h_1 = 3310.6 \text{ kJ/kg}$$
$$s_1 = 6.3487 \text{ kJ/kgK}$$

For condenser,

$$h_f = 191.83 \text{ kJ/kg}$$
$$h_g = 2584.8 \text{ kJ/kg}$$
$$s_f = s_5 = 0.6493 \text{ kJ/kgK}$$
$$s_g = 8.1511 \text{ kJ/kgK}$$
$$h_3 = 3389.2 \text{ kJ/kg}$$
$$s_3 = s_4 = 6.6728 \text{ kJ/kgK}$$
$$s_3 = s_{500}$$
$$s_1 = s_2 = 6.3487 \text{ kJ/kgK}$$

Using interpolation for finding T_2 and h_2

$$\frac{T_2 - T_{500}}{s_2 - s_{500}} = \frac{T_{500} - T_{400}}{s_{500} - s_{400}}$$

$$\Rightarrow \quad \frac{T_2 - 500}{6.3487 - 6.6728} = \frac{500 - 400}{6.6728 - 6.3067}$$

From table, $s_{400} = 6.3067$ kJ/kgK

$$T_2 = 411.47°\text{C}$$

Enthalpy at T_2, is h_2

$$\frac{T_2 - T_{500}}{h_2 - h_{500}} = \frac{T_{500} - T_{400}}{h_{500} - h_{400}}$$

From table

$$h_{500} = 3389.2 \text{ kJ/kg}$$
$$h_{400} = 3125.5 \text{ kJ/kg}$$

$$\Rightarrow \quad \frac{411.47 - 500}{h_2 - 3389.2} = \frac{500 - 400}{3389.2 - 3125.5}$$

$$\Rightarrow \quad h_2 = 3155.75 \text{ kJ/kg}$$

Assumption

- Expansion in turbine is isentropic

$$s_4 = s_5 + xs_{fg} = s_f + xs_{fg}$$
$$\Rightarrow \quad 6.6728 = 0.6493 + x(8.1511 - 0.6493)$$
$$x = 0.803$$
$$h_4 = h_f + xh_{fg} = 191.83 + 0.803\,(2584.8 - 191.83)$$
$$h_4 = 2113.385 \text{ kJ/kg}$$
$$h_5 = h_6 = h_f = 191.83 \text{ kJ/kg}$$

[∵ Pump work is neglected]

$$\eta_{th} = \frac{W_{net}}{Q_{input}} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_6) + (h_3 - h_2)}$$

$$= \frac{(3310.6 - 3155.75) + (3389.2 - 2113.385)}{(3310.6 - 191.83) + (3389.2 - 3155.75)} = 0.4268$$

$$\eta_{th} = 42.68\%$$

Q.8 (a) Solution:

Given: $a = 0.25$, $b = 0.58$, $\phi = 28.6°$

Day of the year, $n = 17$

$$\text{Declination angle, } \delta = 23.45° \times \left[\sin\left\{\frac{360}{365}(284 + n)\right\}^{\circ}\right] \quad \text{(By Copper's relation)}$$

$$= 23.45° \times \sin\left\{\frac{360(284 + 17)}{365}\right\}^{\circ}$$

Declination angle, $\delta = -20.917°$

Hour angle at sunrise may be calculated as,

$$\omega_s = \cos^{-1}[-\tan\phi \times \tan\delta]$$

$$= \cos^{-1}\left[-\tan(28.6)^\circ \times \tan(-20.917)^\circ\right]$$

$$\omega_s = 77.97^\circ$$

Average day length on a clear sky day:

$$L_m = \frac{2\times\omega_s}{15} = \frac{2\times 77.97}{15} = 10.396 \text{ hours}$$

Now,
$$I_n = I_{sc}\left\{1+0.033\cos\left(\frac{360\times n}{365}\right)\right\}$$

$$= 1367\left\{1+0.033\cos\left(\frac{360\times 17}{365}\right)\right\}$$

$$I_n = 1410.193 \text{ W/m}^2$$

Now,
$$\bar{H}_o = \int_{t_{SR}}^{t_{ss}} \mathrm{I_n}\cos\theta dt$$

$$= I_n \times \frac{24}{\pi}\int_o^{\omega_s}[\sin\phi\,\sin\delta + \cos\phi\,\cos\delta\,\cos\omega]\,d\omega$$

$$= 3600\times 1410.193\times\frac{24}{\pi}[\omega_s \times \sin\phi\,\sin\delta + \cos\phi\times\cos\delta\sin\omega_s]_0^{77.97^\circ}$$

$$= 3600\times 1410.193\times\frac{24}{\pi}\left[\left(77.97\times\frac{\pi}{180}\right)\times\sin 28.6^\circ\sin(-20.197)^\circ + \cos 28.6^\circ\times\cos(-20.917)^\circ\sin 77.97^\circ\right]$$

$$= 3600 \times 6135.753 \text{ J/m}^2\text{-day}$$

$$= \left(6135.753\times\frac{3600}{1000}\right)\text{ kJ/m}^2\text{-day}$$

$$\bar{H}_o = 22088.71 \text{ kJ/m}^2\text{-day}$$

$$\frac{\bar{H}_g}{\bar{H}_o} = a + b\left(\frac{L_a}{L_m}\right)$$

$$\bar{H}_g = 22088.71\left[0.25 + 0.58\times\left(\frac{7}{10.396}\right)\right]$$

$$\bar{H}_g = 14148.588 \text{ kJ/m}^2\text{-day}$$

Monthly average of the daily global radiation is 14148.588 kJ/m^2-day

Q.8 (b) Solution:

Given: r_p = 4, T_3 = 100 K, T_1 = 300 K, c_p = 1 kJ/kgK, c_v = 0.717 kJ/kgK

Now,
$$\gamma = \frac{c_p}{c_v} = \frac{1}{0.717} = 1.3947$$

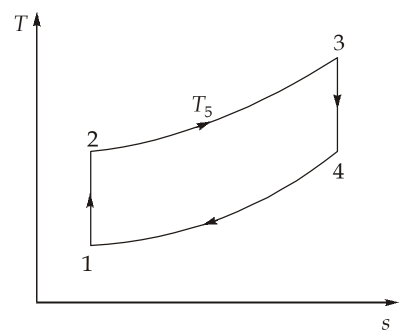
$$t = \frac{T_3}{T_1} = \frac{1000}{300} = 3.333$$

$$c = (r_p)^{(\gamma-1)/\gamma} = (4)^{0.3947/1.3947} = 1.4804$$

1. Net work output, $W_N = c_p(T_3 - T_4) - c_p(T_2 - T_1)$

$$= c_p\left[T_3\left(1-\frac{T_4}{T_3}\right)-T_1\left(\frac{T_2}{T_1}-1\right)\right]$$

$$= c_p\left[1000\left(1-\frac{T_4}{T_3}\right)-300\left(\frac{T_2}{T_1}-1\right)\right]$$



We know that,

$$\frac{T_3}{T_4} = (r_p)^{(\gamma-1)/\gamma} = 1.4804$$

$$\frac{T_2}{T_1} = (r_p)^{(\gamma-1)/\gamma} = 1.4804$$

$$W_N = 1\times\left[1000\left(1-\frac{1}{1.4804}\right)-300\,(1.4804-1)\right]$$

$$W_N = 180.3869 \text{ kJ/kg}$$

$$\eta = 1-\frac{1}{(r_p)^{(\gamma-1)/\gamma}} = 1-\frac{1}{1.4804} = 0.3245 = 32.45\%$$

2. With heat exchanger or for perfect regeneration:

Net work output will remain same.

$$W_N = 180.3869 \text{ kJ/kg}$$

$$\eta = \frac{W_{net}}{Q_s} = \frac{180.3869}{c_p(T_3 - T_5)}$$

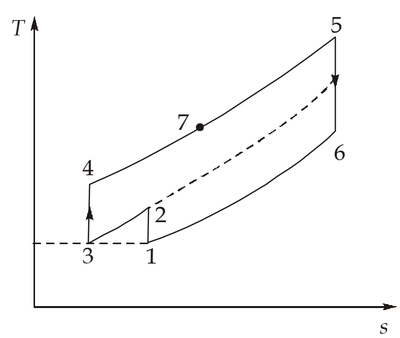
For ideal regeneration, $T_5 = T_4$

$$\eta = \frac{180.3869}{c_p(T_3 - T_4)} = \frac{180.3869}{1 \times T_3\left(1 - \frac{T_4}{T_3}\right)}$$

$$\eta = \frac{180.3869}{1000 \times \left(1 - \frac{1}{1.4804}\right)} = 0.55588 = 55.588\%$$

3. Basic cycle with two stage intercooled compressor:

For perfect inter cooling:



$$\frac{P_5}{P_6} = 4 = r_p,\ T_5 = 1000 \text{ K},\ T_1 = 300\text{K}$$

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \sqrt{r_p}, \frac{T_5}{T_6} = C = 1.4804$$

$$\frac{T_2}{T_1} = \frac{T_4}{T_3} = \sqrt{C} = \sqrt{1.4804}$$

Where, $C = (r_p)^{(\gamma-1)/\gamma}$

$$W_{net} = c_p[(T_5 - T_6) - (T_4 - T_3) - (T_2 - T_1)]$$

$\therefore$ $T_2 = T_4$ and $T_3 = T_1$ (For perfect intercooling)

$$W_{net} = 1 \times \left[T_5\left(1 - \frac{T_6}{T_5}\right) - 2(T_2 - T_1)\right]$$

$$= \left[1000\times\left(1-\frac{1}{1.4804}\right)-2T_1\left(\frac{T_2}{T_1}-1\right)\right]$$

$$= \frac{1000\times 0.4804}{1.4804}-2\times 300\left(\sqrt{1.4804}-1\right)$$

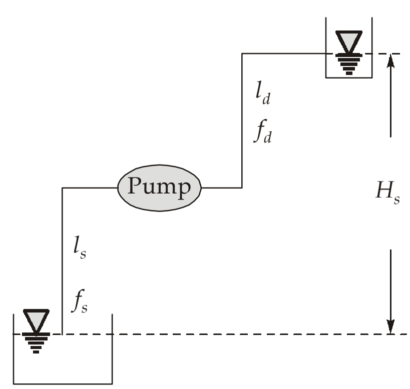
$$W_{net} = 194.476 \text{ kJ/kg}$$

$$\text{Efficinecy, } \eta = \frac{W_{net}}{Q_s} = \frac{194.476}{c_p(T_5-T_4)}$$

$$= \frac{194.476}{1\times\left[1000-\left(\sqrt{1.4804}\right)300\right]} \qquad \{T_4 = \sqrt{c}\,T_1\}$$

$$\eta = 0.306268 = 30.63\%$$

Q.8 (c) Solution:



System Requirement (S.R)

$$S.R = H_s + \frac{f_s l_s Q^2}{12.1 d_s^5} + \frac{f_d l_d Q^2}{12.1\, d_d^5}$$

$$S.R = (150-100) + \frac{0.025\times 50Q^2}{12.1\times(0.3)^5} + \frac{0.02\times 900Q^2}{12.1\times(0.2)^5}$$

$$S.R = 50 + 42.512Q^2 + 4648.76Q^2$$

$$S.R = 50 + 4691.272Q^2 \quad ...(i)$$

Given: Head discharge relationship as

$$H_p = 80 - 7000Q^2 \quad ...(ii)$$

Equating (i) and (ii),

$$80 - 7000Q^2 = 50 + 4691.272Q^2$$

$$11691.272Q^2 = 30$$

$$Q = 0.05065 \text{ m}^3/\text{s}$$

$$H_p = 80 - 7000 \times (0.05065 \text{ m}^3/\text{s}) = 62.04 \text{ m}$$

$$P = \frac{\rho Q_g H_p}{10^3} = \frac{10^3 \times 0.05065 \times 9.81 \times 62.04}{10^3} = 30.82 \text{ kW}$$

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