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ESE 2021

Main Exam Detailed Solutions

Mechanical Engineering

PAPER-I

EXAM DATE : 21-11-2021 | 9:00 AM to 12:00 PM

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ANALYSIS

Mechanical Engineering Paper-I
ESE 2021 Main Examination

Sl.	Subjects	Marks
1.	Fluid Mechanics	52
2.	Thermodynamics	34
3.	Heat Transfer	92
4.	IC Engines	52
5.	Refrigeration and Air-Conditioning	34
6.	Turbo Machinery	72
7.	Power Plant	72
7.	Renewable Sources of Energy	72
		Total 480

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Section-A

Q.1 (a) What is Laminar sublayer? For the velocity profiles given below, state whether the boundary layer has separated or is on the verge of separation or will remain attached with the surface:

(i) $\frac{u}{U} = 2\left(\frac{y}{\delta}\right) - \left(\frac{y}{\delta}\right)^2$

(ii) $\frac{u}{U} = -2\left(\frac{y}{\delta}\right) + \left(\frac{y}{\delta}\right)^2$

(iii) $\frac{u}{U} = \frac{3}{2}\left(\frac{y}{\delta}\right)^2 + \frac{1}{2}\left(\frac{y}{\delta}\right)^3$

The symbols have their usual meaning.

[12 marks : 2021]

Solution:

Laminar Sublayer:

- It appears when the plate is very smooth.
- Since, the flow in the laminar sublayer is laminar so the parabolic velocity distribution holds good. But, this velocity distribution can be assumed to be linear because the thickness (δ') of laminar sublayer is very small.

$$\delta' = \frac{11.6\nu}{V_*} = \frac{11.6\nu}{\sqrt{\frac{\tau_0}{\rho}}}$$

where, $V_* = \sqrt{\frac{\tau_0}{\rho}}$ is known as shear or friction velocity.

(i)
$$\frac{u}{U} = 2\left(\frac{y}{\delta}\right) - \left(\frac{y}{\delta}\right)^2$$

$$u = \left[2\left(\frac{y}{\delta}\right) - \left(\frac{y}{\delta}\right)^2\right]U$$

Now,
$$\left(\frac{\partial u}{\partial y}\right)_{y=0} = \left[\frac{2}{\delta} - 2\left(\frac{y}{\delta}\right)_{y=0}\right]U = \left(\frac{2}{\delta} - 0\right)u = \frac{2u}{\delta} = +ve$$

Hence, flow is not separating and will remain attach with surface.

(ii)
$$\frac{u}{U} = -2\left(\frac{y}{\delta}\right) + \left(\frac{y}{\delta}\right)^2$$

$$u = U\left[-2\left(\frac{y}{\delta}\right) + \left(\frac{y}{\delta}\right)^2\right]$$

Now,
$$\left(\frac{\partial u}{\partial y}\right)_{y=0} = U\left[-\frac{2}{\delta} + \frac{2y}{\delta^2}\right]_{y=0} = \frac{-2U}{\delta} = -ve$$

Hence, flow has separated.

(iii)
$$\frac{u}{U} = \frac{3}{2}\left(\frac{y}{\delta}\right)^2 + \frac{1}{2}\left(\frac{y}{\delta}\right)^3$$
$$u = U\left[\frac{3}{2}\left(\frac{y}{\delta}\right)^2 + \frac{1}{2}\left(\frac{y}{\delta}\right)^3\right]$$
$$\frac{\partial u}{\partial y} = U\left[\frac{3}{2}\times 2\left(\frac{y}{\delta^2}\right) + \frac{1}{2}\times 3\left(\frac{y^2}{\delta^3}\right)\right]$$
$$\left(\frac{\partial u}{\partial y}\right)_{y=0} = U\left[3\times\left(\frac{0}{\delta^2}\right) + \frac{3}{2}\left(\frac{0}{\delta^3}\right)\right]$$
$$\left(\frac{\partial u}{\partial y}\right)_{y=0} = 0$$

Hence, flow is on the verge of separation.

End of Solution

Q.1 (b) A heat engine receives reversibly 420 kJ/cycle of heat from a source at 327°C and rejects heat reversibly to a sink at 27°C. There are no other heat transfers. For each of there hypothetical amounts of heat rejected in (i), (ii) and (iii) below, compute the cyclic integral of $\oint \frac{dQ}{T}$. From these results, show which case irreversible, which is reversible and which is impossible:

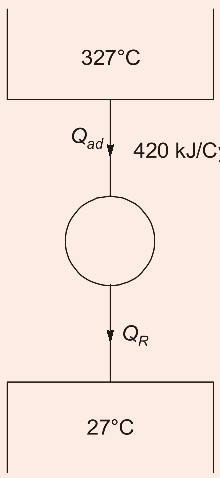
(i) 210 kJ/cycle rejected

(ii) 105 kJ/cycle rejected

(iii) 315 kJ/cycle rejected

[12 marks : 2021]

Solution:



Given: $Q_{ad} = 420$ kJ/cycle

$$\oint \frac{dQ}{T} = \frac{Q_{ad}}{600} - \frac{Q_R}{300} \quad \text{..(A)}$$

(i) If $Q_R = 210$ kJ/Cycle

From (A) $$\oint \frac{dQ}{T} = \frac{420}{600} - \frac{210}{300} = 0 \quad \text{(Reversible)}$$

(ii) If $Q_R = 105$ kJ/Cycle

From (A) $$\oint \frac{dQ}{T} = \frac{420}{600} - \frac{105}{300} = 0.35 \quad \text{(Impossible)}$$

(iii) If $Q_R = 315$ kJ/Cycle

$$\oint \frac{dQ}{T} = \frac{420}{600} - \frac{315}{300} = -0.35 \quad \text{(Irreversible)}$$

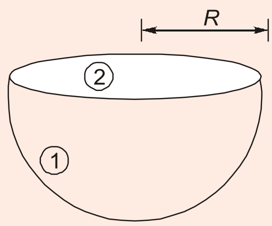
End of Solution

Q.1 (c) Deduce an expression for the shape factor of a hemispherical cavity within itself.
[12 marks : 2021]

Solution:

Hemispherical cavity $A_2 = \pi R^2$

$A_1 = 2\pi R^2$



Assumptions:

(i) Cavity forms enclosure.

(ii) Diffused surface

By conservation of energy for surface 1 and 2:

$$F_{11} + F_{12} = 1$$

$$F_{11} = 1 - F_{12} \quad \text{...(i)}$$

$$F_{22} + F_{21} = 1$$

$$F_{21} = 1 - F_{22}$$

$$F_{22} = 0 \text{ for flat surface}$$

$$F_{21} = 1$$

$$A_1F_{12} = A_2F_{21} \quad \text{(By reciprocity theorem)}$$

$$F_{12} = \frac{A_2}{A_1} \quad \text{...(ii)}$$

By (i) and (ii) equations

$$F_{11} = 1 - \frac{A_2}{A_1}$$

$$F_{11} = 1 - \frac{\pi R^2}{2\pi R^2}$$

$$F_{11} = 0.5$$

End of Solution

Q.1 (d) What is negative slip in a reciprocating pump?

The suction lift is 4 m, length of suction pipe 6.5 m, diameter of suction pipe 100 mm, diameter of piston 150 mm and length of stroke is 0.45 m. Assume simple harmonic motion, atmospheric pressure head as 10.3 m of water and separation occurs at 2.6 m of water absolute.

Determine the maximum speed at which a double acting reciprocating pump can be operated if fitted with an air vessel on the suction side close to the pump. Darcy's f = 0.024.

[12 marks : 2021]

Solution:

The slip in a reciprocating pump is defined as the difference between theoretical discharge and actual discharge.

When actual discharge becomes more than theoretical discharge then the slip becomes negative.

Negative slip occurs when delivery pipe is short and suction pipe is long and pump is running at high speed.

Given: For a double acting reciprocating pump,

$$h_s = 4 \text{ m}$$
$$L_s = 6.5 \text{ m}$$
$$d_s = 100 \text{ mm} = 0.1 \text{ m}$$
$$D = 150 \text{ mm} = 0.15 \text{ m}$$
$$L = 0.45 \text{ m}$$
$$H_{atm} = 10.3 \text{ m}$$
$$H_{vap.} = 2.6 \text{ m}$$
$$N_{max} = ?$$
$$F = 0.024$$
$$H_{vap.} = H_{atm} - (h_S + h_{as} + h_{fs})$$

Since the air vessel is installed very close to cylinder, then it can be assumed that in the entire suction pipe flow is steady.

Therefore $\quad H_{vap.} = H_{atm} - (h_S + h_{fs})$

($\because h_{as} = 0$)

$$h_{fs} = \left.\frac{FLV^2}{2gD}\right|_{\text{suction pipe}}$$

$\therefore$
$$Q = \frac{2ALN}{60} \text{ [Double acting pump]}$$

$$Q = \frac{2 \times \left[\frac{\pi}{4} \times 0.15^2\right] \times 0.45 \times N}{60}$$

$$Q = 2.65 \times 10^{-4} N \text{ m}^3/\text{sec}$$

$$V_s = \frac{Q}{A_s}$$

$$V_s = \frac{2.65 \times 10^{-4} \times N}{\left(\frac{\pi}{4} \times 0.1^2\right)}$$

$$V_s = 0.03375 \text{ N m/s}$$

$$H_{vs} = 10.3 - \left[4 + \frac{0.024 \times 6.5 \times (0.03375\text{N})^2}{2 \times 9.81 \times 0.1}\right]$$

$$2.6 = 10.3 - [4 + 9.056 \times 10^{-5} \text{ N}^2]$$

$$N_{max} = 202.12 \text{ rpm}$$

End of Solution

Q.1 (e) Flue gas analysis using Orsat apparatus provides the following data for combustion of an unknown hydrocarbon:

CO_2 = 12.0%
CO = 0.8%
O_2 = 3.1%
N_2 = 84.1%

Determine air-fuel ratio, fuel composition on mass basis, stoichiometric air-fuel ratio and percentage of excess air.

[12 marks : 2021]

Solution:

Orsat apparatus gives dry flue gas analysis on volume basis.

$$C_XH_Y + Z\left\{O_2 + \frac{79}{21}N_2\right\} \longrightarrow \frac{12n}{100}CO_2 + \frac{0.8n}{100}CO + \frac{3.1n}{100}CO_2 + \frac{84.1n}{100}N_2 + \frac{Y}{2}H_2O$$

where, total moles of dry exhaust is = 'n'

$$\Rightarrow \quad \text{Balancing Carbon, } X = \frac{12n}{100} + \frac{0.8n}{100}$$

$$\therefore \quad X = 0.128\,n$$

$$\Rightarrow \quad \text{Balancing Nitrogen, } Z \times \frac{79}{21} = \frac{84.1}{100}n$$

$$Z = 0.22356n$$

$$\Rightarrow \quad \text{Balancing Oxygen, } 2Z = 2 \times \frac{12n}{100} + \frac{0.8n}{100} + 2 \times \frac{3.1n}{100} + \frac{Y}{2}$$

$$2Z = 0.24n + 0.008n + 0.062n + \frac{Y}{2}$$

$$2 \times 0.22356n = 0.31n + \frac{Y}{2}$$

$$Y = 0.2742\,n$$

Now,

(i)
$$A/F = \frac{m_a}{m_f} = \frac{Z \times \left[32 + \frac{79}{21} \times 28\right]}{X \times 21 + Y \times 1} = \frac{0.22356n(32 + 105.33)}{0.128n \times 12 + 0.27424n}$$
$$= 16.9599 \approx 16.96$$

(ii) Fuel composition on mass basis:

$$\text{Percentage of Carbon} = \frac{12X}{12X + Y \times 1} = \frac{12 \times 0.128n}{12 \times 0.128n + 0.27424n}$$
$$= 0.8485 = 84.85\%$$
$$\text{Percentage of Hydrogen} = 100 - \text{Percentage of Carbon}$$
$$= 100 - 84.85 = 15.15\%$$

(iii) Stoichiometric chemical reaction:

$$C_XH_Y + Z\left\{O_2 + \frac{79}{21}N_2\right\} \longrightarrow CO_2 + \frac{Y}{H}H_2O + Z\frac{79}{21}N_2$$

Balancing Oxygen,
$$2Z' = 2 \times X + \frac{Y}{2}$$
$$Z' = X + \frac{Y}{4}$$
... (i)

Now, since carbon is 84.85% by mass

$$\therefore \quad \frac{12X}{12X + 1Y} = 0.8485$$
$$12X = 10.182X \times 0.8485Y$$
$$1.818X = 0.8485Y$$
$$Y = 2.1426X$$

$\therefore$ From (i)
$$Z' = X + \frac{Y}{4}$$
$$Z' = 1.5356X$$

$$\therefore \quad (A/F)_{\text{Stoichimetric}} = \left(\frac{m_a}{m_f}\right)_{\text{stoichiometric}} = \frac{Z'\left(32 + \frac{79}{21} \times 28\right)}{12X + 1Y}$$
$$= 1.5356 \times \left(\frac{137.33}{12X + 2.1426X}\right) = \frac{210.88X}{14.1426X} = 14.91$$
$$(A/F)_{\text{Stoichimetric}} = 14.91$$

(iv)
$$\%\text{ of excess air} = \left[\frac{(A/F)_{\text{actual}} - (A/F)_{\text{Stoichiometric}}}{(A/F)_{\text{Stoichiometric}}}\right] \times 100$$
$$= \frac{16.96 - 14.91}{14.91} \times 100 = 13.75\%$$

End of Solution

Q.2 (a) A four cylinder, four stroke square engine having a bore of 100 mm operating at 4000 rpm has a compression ratio 7.

If the relative efficiency is 60% when the specific fuel consumption is 250 gm/kWh, estimate (i) how many times the spark will trigger in one minute per cylinder, (ii) number of thermodynamic cycles per cylinder per second, (iii) calorific value of the fuel, and (iv) corresponding fuel consumption in kg/hr, given that the mean effective pressure is 8.5 bar.

[20 marks : 2021]

Solution:

Given for an SI engine,

$$\text{Number of cylinders} = 4$$

$$n = 4 \text{ (4 stroke)}$$

Square engine $\Rightarrow$ $D = L = 100$ mm

$$N = 4000 \text{ rpm}$$

$$r = 7$$

$$\eta_{rel} = 0.6$$

Assuming the specific fuel consumption and mean effective pressure to be at brake values,

$$\text{bsfc} = 250 \times 10^{-3} \text{ kg/kWh}, \; P_{mep} = 8.5 \text{ bar}$$

$$\text{BP} = \frac{P_{mep} \times LANK}{60 \times n}$$

$$\text{BP} = \frac{8.5 \times 100 \times 0.1 \times \frac{\pi}{4} \times (0.1)^2 \times 4000 \times 4}{60 \times 2} = 89.012 \text{ kW}$$

(i) No. of sparks in one minute per cylinder $= \dfrac{N}{2}$

[$\because$ 2 revolutions correspond to 1 cycle]

$$= \frac{4000}{2} = 2000$$

(ii) No. of thermodynamic cycles per cylinder per second

$$= \frac{N}{2} \times \frac{1}{60} = 33.33$$

(iii) Now,

$$\eta_{Otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(7)^{1.4-1}} = 0.541$$

$$\eta_{rel} = \frac{\eta_{bth}}{\eta_{Otto}}$$

$$0.6 = \frac{\eta_{bth}}{0.541}$$

$$\eta_{bth} = 0.3245$$

$$\Rightarrow \qquad \frac{BP}{Q_S} = 0.3245$$

$$\therefore \qquad Q_S = 274.3 \text{ kJ/sec}$$

Now,
$$\text{bsfc} = \frac{\dot{m}_f}{BP}$$

$$0.25 = \frac{\dot{m}_f}{89.012}$$

$$\dot{m}_f = 22.253 \text{ kg/hr}$$

Also,
$$Q_S = \dot{m}_f \times CV$$

$$274.3 = \frac{22.253}{3600} \times CV$$

$$\text{CV} = 44375.14 \text{ kJ/kg}$$

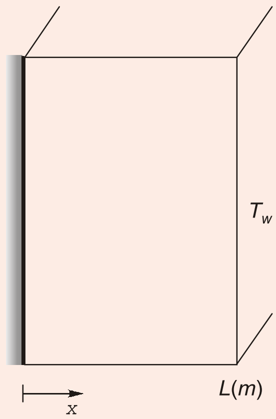
(iv)
$$\dot{m}_f = 22.253 \text{ kg/hr}$$

End of Solution

Q.2 (b) An infinite slab of thickness "L" (m) is having thermal conductivity "K" $\left(\frac{W}{mK}\right)$. It is generating heat at a uniform rate of "$\dot{q}$" $\left(\frac{\text{W}}{\text{m}^3}\right)$. One of the sides of the slab is perfectly insulated and the other side is maintained at a constant temperature of "T_W" (°C). Deduce an expression for the temperature distribution within the slab. Also find out the position of maximum temperature in the slab.

[20 marks : 2021]

Solution:



Assumptions:

(i) One dimensional heat conduction

(ii) Uniform heat generation slab

(iii) Steady state

(iv) Constant properties.

General heat conduction equation for slab:

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}+\frac{\dot{q}}{k} = \frac{1}{\alpha}\frac{\partial T}{\partial t}$$

$$\frac{\partial^2 T}{\partial x^2} = -\frac{\dot{q}}{k}$$

$$\frac{\partial T}{\partial x} = -\frac{\dot{q}}{k}x + C_1$$

$$T = -\frac{\dot{q}x^2}{2k} + C_1 x + C_2 \qquad \text{...(i)}$$

at $x = 0$, $\frac{\partial T}{\partial x} = 0$ (For insulation boundary condition.)

$\Rightarrow$ $C_1 = 0$

$$T = -\frac{\dot{q}x^2}{2k} + C_2$$

at $x = L$, $T = T_w$ (Constant wall temperature boundary condition.)

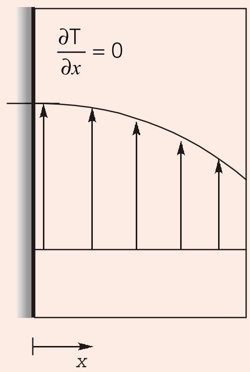
$$C_2 = T_w + \frac{\dot{q}L^2}{2k}$$

$$T = \frac{\dot{q}x^2}{2k} + T_w + \frac{\dot{q}L^2}{2k}$$

$$T - T_w = \frac{q}{2k}[L^2 - x^2] \qquad \text{...(ii)}$$

For maximum temperature condition, $\frac{\partial T}{\partial x} = 0$ and $\frac{\partial^2 T}{\partial x^2} < 0$

So, by eq. (ii)



$$\frac{\partial T}{\partial x} = \frac{\dot{q}x}{k}$$

$\frac{\partial T}{\partial x} = 0$ then $x = 0$

So, max. temperature is at insulation surface.

So, by equation (ii) at $x = 0$ T = T_{max}

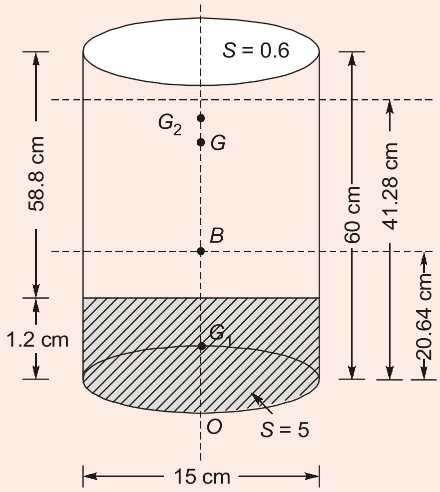
$$T_{max} - T_w = \frac{\dot{q}}{2k}L^2$$

End of Solution

Q.2 (c) A solid cylinder of 15 cm diameter and 60 cm length, consists of two parts made of different materials. The first part at base is 1.2 cm long and has specific gravity of 5.0. The other part of the cylinder is made of material having specific gravity of 0.6. Determine whether it can float vertically or not in water.

[20 marks : 2021]

Solution:



Let 'h' be the height of cylinder submerged in water.

Applying principle of floatation,

$$\text{Weight of cylinder} = F_B$$

$$\left[5000\times 9.81\times\frac{\pi}{4}\times(0.15)^2\times 0.012\right]+\left[600\times 9.81\times\frac{\pi}{4}\times(0.15)^2\times 0.588\right]$$

$$= 10^3\times 9.81\times\frac{\pi}{4}\times(0.15)^2\times h$$

$$[60+352.8] = 1000\,h$$

$$h = 41.28 \text{ cm}$$

$$OB = \frac{h}{2} = 20.64 \text{ cm}$$

Calculation of OG:

Let W_1 be the weight of upper part and W_2 be the weight of lower part of cylinder,

$$(W_1 + W_2)OG = (W_1 \times OG_1) + (W_2 \times OG_2)$$

$$OG = \frac{\left[\left(60\times 0.6\times 10^{-2}\right)+\left[352.8\times\left(1.2+\frac{58.8}{2}\right)\times 10^{-2}\right]\right]}{(60+352.8)}$$

$$OG = \frac{0.36+107.95}{412.8} = 0.2623 \text{ m}$$

$$OG = 26.23 \text{ cm}$$

$$BG = 5.59 \text{ cm}$$

$$BM = \frac{I}{V} = \frac{\pi}{64} \times \frac{(0.15)^4 \times 4}{(0.15)^2 \times 0.4128} = 0.34066 \text{ cm}$$

$$GM = BM - BG = -ve$$

$\therefore$ Cylinder is not floating in vertical position.

End of Solution

Q.3 (a) An all glass body air-conditioned bus is having height of 3 m, width of 3 m and length of 10 m. Inside surfaces of the glass are maintained at 20°C. The bus is moving at a speed of 60 kmph. Atmospheric temperature is 34°C. Neglecting the conduction resistance of the glass and assuming walls and roof are perfectly flat, find the following:

(i) Heat gained by the bus from the roof and side walls. (Neglect Laminar Region).

(ii) Capacity of Air Conditioner Unit required in tonnes of refrigeration (TR) to remove the heat gained as given in (i).

(iii) Power required to run the air-conditioning unit if the COP is 4.

Take the properties of the air as given below:

Density = 1.1774 kg/m^3

Kinematic viscosity = 1.569 × 10^{-5} m^2/s

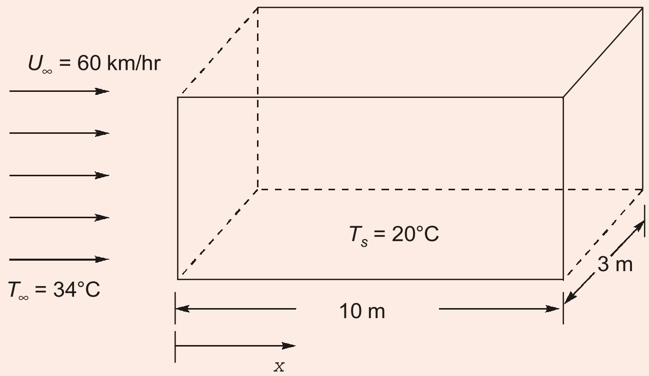
Thermal conductivity = 0.02624 W/mK

Pr = 0.708

For turbulent flow $\overline{Nu_L} = 0.036\ Re_L^{0.7} Pr^{0.33}$

[20 marks : 2021]

Solution:



Assumptions:

(i) Steady flow of air

(ii) Entire flow is turbulent flow

(iii) Constant wall temperature of bus surface.

(iv) Neglecting radiation heat transfer

Velocity of air w.r.t to space $U_{\infty} = \frac{60 \times 5}{18} = 16.67\,\text{m/s}$

$$\text{Re}_L = \frac{\rho U_{\infty} x}{\mu} = \frac{U_{\infty} x}{\nu} = \frac{16.67 \times 10}{1.569 \times 10^{-5}}$$

$$= 10.62 \times 10^6 > 5 \times 10^5$$

So, turbulent flow.

For turbulent flow:

$$\text{Nu} = 0.036\,\text{Re}_L^{0.8}\,\text{Pr}^{0.33} \qquad \text{(given)}$$

$$\frac{h \times 10}{0.02624} = 0.036 \times [10.62 \times 10^6]^{0.8}[0.708]^{0.33}$$

$$h = 35.21\ \text{W/m}^2\text{-K}$$

Total convective area along Boundary layer

$$A = (10 \times 3) \times 4 = 120\ \text{m}^2$$

Heat gained by bus, $q = hA\Delta T$

$$= 35.21 \times 120 \times [34 - 20]$$

$$= 59152.8\ \text{W} = 59.15\ \text{kW}$$

(ii) Tonnage of Refrigeration needed:

$$\text{Tonnage} = \frac{59.15}{3.51} = 16.85\,\text{Ton}$$

(iii) If COP = 4

$$\text{Power input} = \frac{\text{RE}}{\text{COP}}$$

$$= \frac{59.15\,\text{kW}}{4} = 14.78\,\text{kW}$$

End of Solution

Q.3 (b) (i) (I) For a given thermodynamic system while considering control volume approach, what is the significance of flow work? Is flow work a path function or point function?

(II) With a neat sketch apply steady flow energy equation to a system handling incompressible fluid (like pump) and a system handling compressible fluid (like compressor). Draw important inferences from the steady flow energy equations of pump and compressor.

[4 + 6 marks : 2021]

(ii) (I) Show that COP of a heat pump is greater than COP of a refrigerator.

(II) A housewife keeps the door of a refrigerator open in order to beat the heat of summer by closing the door and window of a kitchen. However the cooling effect wears out with the passage of time and she feels uncomfortable with the rise of temperature. Assume the room plaster is well insulated with no heat exchange to the surroundings. How will you evaluate this case in the context of first law of thermodynamics.

[4 + 6 marks : 2021]

Solution:

(i) (I) Flow Work:

- **Flow work:** It helps us to understand the total energy brought in by the fluid entering the control volume. So, the total energy brought in while fluid enters the control volume = Internal energy + Flow work done on it.

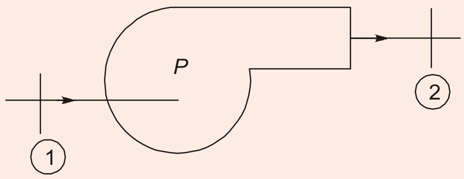
$$H = U + PV$$

- We consider the flow work as in the total energy brought in by the fluid. So, the total energy brought in by the fluid = $U + PV$. Thus flow work is a point function.

(i) (II)

SFEE for pump

For incompressible fluid



Applying SFEE,

$$u_1 + p_1v_1 + kE_1 + PE_1 + \frac{\delta Q}{dm} = u_2 + p_2v_2 + kE_2 + PE_2 + \frac{\delta W}{dm}$$

Assumptions:

- Incompressible fluid,

$$v_1 = v_2 = v$$

- No heat interaction

$$\frac{\delta Q}{dm} = 0$$

- No change in internal energy

$$u_1 = u_2$$

So,
$$p_1v + \frac{v_1^2}{2} + gz_1 = p_2v + \frac{v_2^2}{2} + gz_2 + \frac{\delta W}{dm}$$

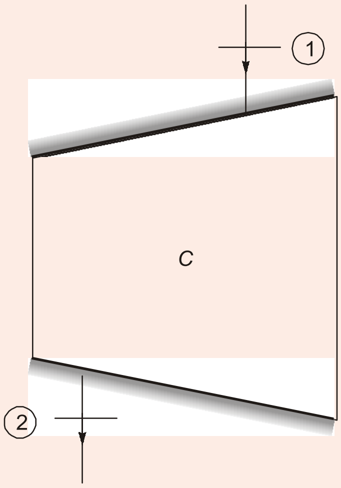
$$\frac{\delta W}{dm} = \underbrace{\left(\frac{P_1}{\rho} + \frac{v_1^2}{2} + gz_1\right)}_{\text{Bernaulli's equation}} - \left(\frac{P_2}{\rho} + \frac{v_2^2}{2} + gz_2\right)$$

$$W_{pump} = m\left[\Delta\left(\frac{P}{\rho} + \frac{v^2}{2} + gz\right)\right]$$

From eq. (i),

Since specific volume of incompressible fluid is same at both sections (1) and (2),

So, SFEE reduces to Bernoulli's equation in case of pump.
SFEE for compressor:
For compressible fluid:



$$u_1 + p_1v_1 + kE_1 + PE_1 + \frac{\delta Q}{dm} = u_2 + p_2v_2 + kE_2 + PE_2 + \frac{\delta W}{dm}$$

$$h = u + Pv$$

$$\Rightarrow \quad h_1 + kE_1 + PE_1 + \frac{\delta Q}{dm} = h_2 + kE_2 + PE_2 + \frac{\delta W}{dm}$$

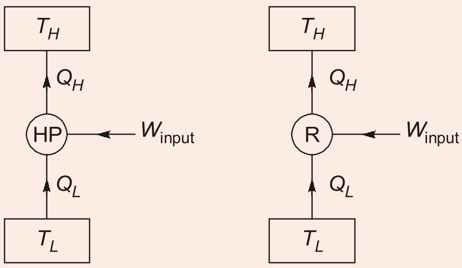
Neglecting kE and PE change, no heat transfer from compressor, equation reduces to

$$h_1 = h_2 + \frac{\delta W}{dm} \qquad \text{(for unit mass flow)}$$

$$W_{\text{compressor}} = h_1 - h_2 \qquad ...(2)$$

For compressible fluid, SFEE does not reduce to Bernoulli's equation as written in equation (2)
A comparison of equations (1) and (2) shows that the Bernoulli's equation is restricted to frictionless incompressible fluids and SFEE is valid for viscous compressible fluids as well.
Bernoulli's equation is special limiting case of the more general steady flow energy equation.

(ii) (I)



For refrigerator: Desired effect = Q_L

For heat pump: Desired effect = Q_H

$$\text{COP} = \frac{\text{Desired effect}}{\text{Work input}}$$

For same work input

COP $\propto$ desired effect (DE)

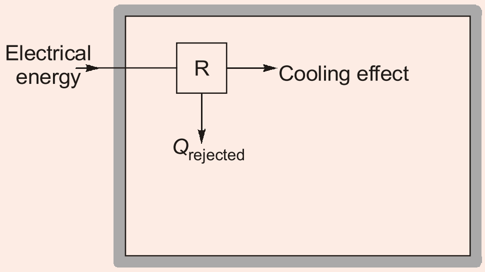
$$(\text{DE})_{\text{HP}} = Q_H = Q_L + W_{\text{input}}$$

$$(\text{DE})_{\text{Ref}} = Q_L$$

$$\Rightarrow \quad Q_H > Q_L$$

$$\therefore \quad (\text{COP})_{\text{HP}} > (\text{COP})_{\text{Ref}}$$

(II)



Insulated room

Considering the kitchen room with the refrigerator to be a thermodynamic system. Since it is insulated and all doors and windows are closed, the only heat interaction is in the form of electrical energy which is being given for operating the refrigerator.

To produce the cooling effect inside the refrigerator, this electrical energy is utilized and at the same time, the refrigerator must reject some heat to the kitchen (which acts as high temperature reservoir).

Thus, after some time, the housewife feels uncomfortable due to this heat which is rejected into the kitchen room as it causes a rise in temperature.

End of Solution

Q.3 (c) A vertical cylindrical rod of 1 m length is maintained at a temperature of 120°C. Diameter of the rod is 5 cm. It is exposed to a very large room having surrounding air and wall temperature at 34°C. It has surface emissivity of 0.7.

Find the following:

(i) Heat lost by the rod by convection.

(ii) Heat lost by the rod by radiation.

(iii) Total heat loss by the rod.

(iv) Percentage of convection and radiation heat loss.

(v) Is it correct to neglect the radiation heat loss for this situation?

Take the property values of air as given below:

Density = 0.998 kg/m^3

Kinematic viscosity = 2.076×10^{-5} m^2/s

Thermal conductivity = 0.03 W/mK

$Pr = 0.697$

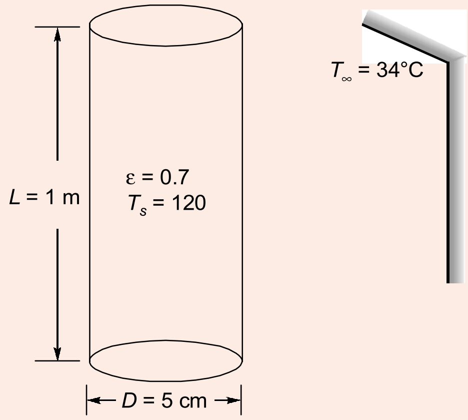
Stefan's Constant = 5.67×10^{-8} W/m^{2}K^4

Use correlation $\overline{Nu_L} = 0.1[Gr_L\ Pr]^{0.33}$

Neglect heat loss from the ends.

[20 marks : 2021]

Solution:



Given:

$$Pr = 0.697$$
$$\rho = 0.998 \text{ kg/m}^3$$
$$k = 0.03 \text{ W/mK}$$
$$\sigma = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$$

(i) For convection heat loss:

Assumption:

(i) Natural convection heat transfer.

(ii) Air and wall temperature of room are at same temperature.

Grashoff number:

$$Gr = \frac{g\beta L_c^3 \Delta T}{\nu^2}$$

$$\beta = \frac{1}{T_f} = \frac{1}{\frac{(120+34)}{2} + 273}$$

$$\beta = 0.002857$$

$$Gr = \frac{9.81 \times 0.002857 \times 1^3 \times (120-34)}{(2.076 \times 10^{-5})^2}$$

$$Gr = 5592719018$$

By correlation:

$$Nu = 0.1[Gr\ Pr]^{0.33}$$

$$\frac{h \times 1}{0.03} = 0.1[5592719018 \times 0.697]^{0.33}$$

$$h = 4.386 \text{ W/m}^2\text{-K}$$

$$q = hA\Delta T = h(\pi DL)\Delta T = 4.386(\pi \times 0.05 \times 1)(120-34)$$

$$q = 59.25 \text{ W}$$

(ii) For radiation heat transfer:

Assumption:

(i) Rod is a small body and room is a large enclosure.

(ii) Neglecting end effect of the rod.

$$q = \varepsilon\sigma A_s[T^4 - T_\infty{}^4]$$
$$= 0.7 \times 5.67 \times 10^{-8} \times (\pi DL)[(393)^4 - (307)^4]$$
$$q = 93.34 \text{ W}$$

(iii) Total heat loss:

$$q_{\text{total}} = q_{\text{conv.}} + q_{\text{rad}}$$
$$= 59.25 + 93.34 = 152.59 \text{ W}$$

(iv) Loss in percentage:

$$\text{Convection loss} = \frac{59.25}{152.59} \times 100 = 38.83\,\%$$

$$\text{Radiation loss} = \frac{93.34}{152.59} \times 100 = 61.17\,\%$$

(v) Radiation heat loss has high percentage so it can not be neglected .

End of Solution

Q.4 (a) A circular pipe of length 500 m and diameter 400 mm is connected with a reservoir at one end and to the atmosphere at the other end. The pipe has rounded entrance (K = 0.15), sudden contraction to 400 mm (K = 0.25), sharp bend (K = 0.18), gate valve full open (K = 8) and sudden expansion to 500 mm pipe. Assuming pipe friction loss coefficient as 0.012, determine discharge for head of 50 m at entrance. K is the head loss coefficient.

[20 marks : 2021]

Solution:

Given: $L = 500$ m, $d = 400$ mm

$$A_1V_1 = A_2V_2$$

$$\frac{\pi}{4} \times 400^2 \times V_1 = \frac{\pi}{4} \times 500^2 \times V_2$$

$$V_2 = \frac{400^2}{500^2} \times V_1 = 0.64V_1$$

$$50 = 0.15\frac{V_1^2}{2g} + 0.25\frac{V_1^2}{2g} + 0.18\frac{V_1^2}{2g} + 8\frac{V_1^2}{2g} + \frac{(V_1 - V_2)^2}{2g} + \frac{4 \times 0.012 \times 500}{400 \times 10^{-3}} \times \frac{V_1^2}{2g}$$

$$50 = [0.15 + 0.25 + 0.18 + 8 + 0.5904 + 60]\frac{V_1^2}{2g}$$

$$V_1 = 3.765 \text{ m/s}$$

$$Q = \frac{\pi}{4} \times (0.400)^2 \times 3.75 = 0.4732 \text{ m}^2/\text{s}$$

End of Solution

Q.4 (b) In a double pipe, parallel flow heat exchanger, hot fluid enters at 120°C and leaves at 80°C. Cold fluid enters at 20°C and leaves at 50°C. If inlet temperatures, overall heat transfer coefficient and flow rate of the fluids remain same, find the exit temperatures of the fluids if counter flow arrangement is used. Use effectiveness method.

Effectiveness of parallel flow heat exchanger = $\dfrac{1-e^{[-N(1+C)]}}{1+C}$

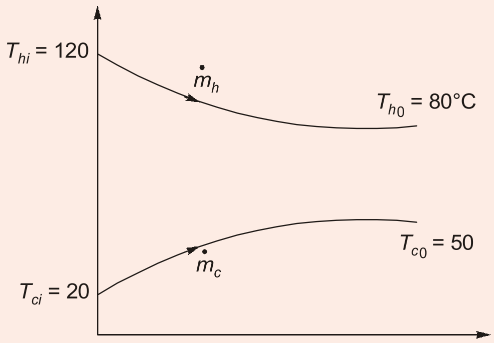
Effectiveness of counter flow heat exchanger = $\dfrac{1-e^{[-N(1-C)]}}{1-Ce^{[-N(1-C)]}}$

where N = NTU, C = C_{min}/C_{max}.

[20 marks : 2021]

Solution:

For parallel flow heat exchanger:



Heat capacity rate of hot fluid, $C_h = \dot{m}_h C_h$

Heat capacity rate of cold fluid, $C_C = \dot{m}_C C_c$

By energy balance eq.

$$\dot{m}_C C_h \Delta T_h = \dot{m}_C C_c \Delta T_c$$
$$C_h \Delta T_h = C_c \Delta T_c$$
$$C_h[120-80] = C_c[50-20]$$
$$C_h \times 40 = C_c \times 30$$
$$C_h < C_c$$

So, hot fluid is minimum heat capacity rate fluid.

$$\text{Capacity ratio} = \frac{C_{min}}{C_{max}} = \frac{C_h}{C_c} = \frac{30}{40} = \frac{3}{4} = 0.75$$

$$\text{Effectiveness of parallel flow, } \varepsilon_p = \frac{C_h \Delta T_h}{C_{min} \Delta T_{max}}$$

$$\varepsilon_p = \frac{\Delta T_h}{\Delta T_{max}} \qquad \{\because C_h = C_{min}]$$

$$\varepsilon_p = \frac{120-80}{120-20} = 0.4$$

For parallel flow:
$$\varepsilon_p = \frac{1-exp[-N(1+C)]}{1+C}$$

$$0.4 = \frac{1-e^{[-N(1+0.75)]}}{1+0.75}$$

Number of transfer units, N = 0.688

For counter flow:

NTU = 0.688 as all conditions are same.

$$\text{Effectiveness of counter flow} = \frac{1-e^{[-N(1-C)]}}{1-Ce^{[-N(1-C)]}}$$

$$\varepsilon_c = \frac{1-e^{[-0.688(1-0.75)]}}{1-0.75e^{[-0.688(1-0.75)]}}$$

$$\varepsilon_c = 0.4288$$

For hot fluid exit temperature, $\varepsilon_c = \dfrac{C_h \Delta T_h}{C_{min}\Delta T_{max}}$ {As $C_h = C_{min}$}

$$\varepsilon_c = \frac{\Delta T_h}{\Delta T_{max}}$$

$$0.4288 = \frac{(T_{h_i} - T_{ho})}{(120-20)}$$

$$T_{hi} - T_{ho} = 42.8$$
$$T_{ho} = 42.8$$
$$= 120 - 42.8 = 74.2$$
$$T_{ho} = 77.12°C$$

For cold fluid exit temperature

$$\varepsilon_c = \frac{C_c \Delta T_c}{C_{min}\Delta T_{max}} = \frac{C_{max}\Delta T_c}{C_{min}\Delta T_{max}}$$

$$\varepsilon_c = \frac{\Delta T_c}{C\Delta T_{max}}$$

$$0.428 = \frac{\Delta T_c}{0.75(120-20)}$$

$$\Delta T_c = 32.1$$
$$T_{c_0} - T_{c_i} = 32.1$$
$$T_{c_0} = 52.1°C$$

End of Solution

Q.4 (c) Explain variation in specific heat of gases and its influence on engine performance. Also explain how actual cycle differs from air fuel cycle. Explain exhaust blow-down loss for SI engine.

[20 marks : 2021]

Solution:

There are 2 types of specific heat of gases (C_P, C_V)
Specific heat value increases with increase in temperature.
On increasing temperature both value of specific heats increases with same amount as

$$C_P - C_V = R = \text{Constant}$$

With $T \uparrow \rightarrow C_P \uparrow$ and $C_V \uparrow \rightarrow \gamma = \frac{C_P}{C_V} \downarrow$

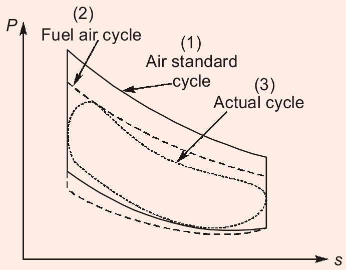
So, γ directly associated with engine efficiency $\Rightarrow$ with $\gamma \downarrow \rightarrow$ engine efficiency $\downarrow$.

Fuel air cycle: Theoretical cycle based on actual properties of the cylinder contents is called the fuel air cycle. The fuel air cycle take into consideration the following.

1. The actual composition of the cylinder contents.
2. The variation in the specific heat of the gases.
3. The dissociation effect.
4. The variation in the number of moles present in the cylinder as the pressure and temperature change

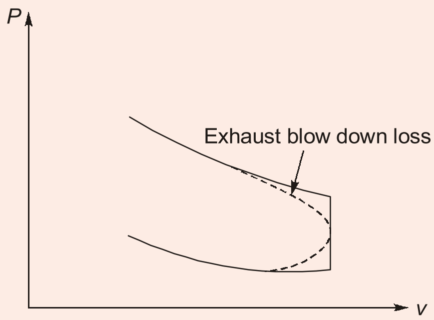
Actual cycle: The actual cycle experienced by internal combustion engines is an open cycle with changing composition, actual cycle efficiency is much lower than the air standard efficiency due to various losses occurring in the actual engine. Losses are:

1. Losses due to dissociation.
2. Time losses, effect of spark timing.
3. Internal combustion loss.
4. Direct heat loss.
5. Pumping losses
6. Friction losses
7. Effect of throttle opening



Figures shows (1) two constant volume cycles (heat added and rejected) air standard cycle (2) Fuel air cycle with variable specific heat and dissociation (3) actual cycle.

Exhaust blow down loss: The opening of exhaust valve before BDC reducing cylinder pressure, causing the roundness of the end of P-V diagram, as shown in figure, this means a reduction in the work done per cycle.



Effect of exhaust valve opening time

End of Solution

Section-B

Q.5 (a) A compound parabolic collector, 2 m long (L), has an acceptance angle ($2\theta_a$) of 30°. The absorber surface of the collector is flat and has a width (b) of 20 cm. Calculate the concentration ratio (C), the aperture width (W), the height (H), and the surface area (A_{con}) of the concentrator.

[12 marks : 2021]

Solution:

Given: Acceptance angel ($2\theta_a$) = 30° = θ_a = 15°

Absorber surface width (b) = 20 cm

L = 2 m

- $$\text{Concentration ratio, (C)} = \frac{1}{\sin(\theta_a)} = \frac{1}{\sin(15°)}$$

$$C = 3.863$$

- $$\text{Aperture width (W)} = C \times b = 3.863 \times 20 \text{ cm}$$

$$W = 77.26 \text{ cm}$$

- $$\frac{H}{W} = \frac{1}{2}\left[1+\frac{1}{\sin\theta_a}\right]\cos\theta_a = \frac{1}{2}\left[1+\frac{1}{\sin(15°)}\right]\cos(15°)$$

$$\Rightarrow \quad \frac{H}{77.26} = 2.3489$$

$$\Rightarrow \quad H = 181.482 \text{ cm}$$

- $$\frac{A_{cone}}{wL} = 1 + C$$

$$\Rightarrow \quad \frac{A_{cone}}{\left(\frac{77.26}{1000}\right)\times 2} = 1 + 3.863$$

$$\Rightarrow \quad A_{conc} = 0.7514 \text{ m}^2$$

End of Solution

Q.5 (b) In the context of engine components, answer the following:

(i) Why are there multiple intake and multiple exhaust valves nowadays in modern engines? How will you identify inlet valve from exhaust valve through visual inspection?

(ii) Why are pistons made tapered ? How will you identify a piston of a two-stroke engine from the piston of a four-stroke engine through visual inspection assuming same engine capacity?

[6 + 6 marks : 2021]

Solution:

(i) Multi valve engines have mainly 3 advantages. Firstly it increases the coverage of valves over the combustion chamber allowing faster breathing thus enhance power at high revolution. Secondly it allows the spark plug to be positioned in the center of combustion chamber, enabling quicker flame propagation more even and more efficient burning. Thirdly, using more but smaller valves instead of two large valve means lower mass for each valve. This prevents the valve "float" from its designed position at very high revolution thus enabling the engine to revolution higher and make more power as a result.

Inlet valves are larger while exhaust valves are smaller. It is so because over priority is at inlet to give cylinder area more to inlet valve so that more air flows into the engine which increases its volumetric efficiency but for exhaust valve, already exhaust air is at high pressure which can escape by itself from the cylinder. New design of exhaust valve have vanes or blades around the steam just have the seat. This is for rotation of the valve and for removal of carbon particles that might be accumulated on the valve seats and causing erosion. Intake valves do not have there blades.

(ii) Pistons are designed with a taper with the smallest diameter of the taper at the piston head. The taper shape compensates for thermal expansion and thermal growth. Thermal growth is the increase in size of a material when heated with little or no change back to original dimensions. The taper design allows the piston to move freely in the cylinder bore regardless of the heat applied to the piston head.

We can identify a piston of a two stroke engine by visualizing as it has a deflector on the top of it, which is used to deflect the fresh air-fuel mixture entering the cylinder exhaust towards the top of cylinder for minimizing the loss of fresh air fuel mixture during the scavenging process.

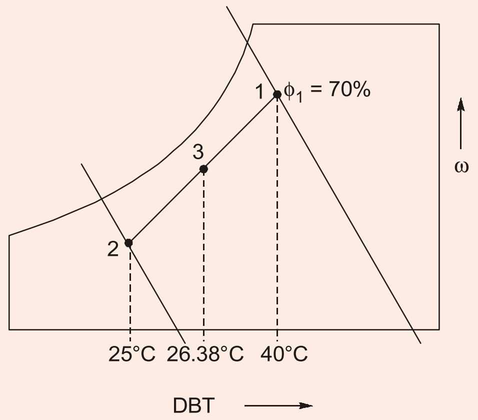
End of Solution

Q.5 (c) In an air-conditioning unit, 10 m^3/min of air from atmospheric condition of DBT 40°C and Relative humidity 70% is adiabatically mixed with 90 m^3/min of recirculation air coming from the air-conditioned chamber. Condition of the recirculation air is DBT 25°C and WBT 20°C. Find the Enthalpy, Specific humidity, Relative humidity and WBT of the air after mixing.

Also draw the process in a skelton Psychrometric chart. [Psychrometric chart is attached.

[12 marks : 2021]

Solution:



$$v = \frac{V}{\dot{m}}$$

$$v_1 = 0.94\ m^3/kg$$

$$v_2 = 0.86\ m^3/kg$$

$$\dot{V}_1 = 10\ m^3/kg$$

$$\dot{V}_2 = 90\ m^3/kg$$

$$\dot{m}_2 = 104.65\ kg/min = 1.7442\ kg/sec$$

$$\dot{m}_1 = 10.63\ kg/min = 0.177\ kg/sec$$

$$\dot{m}_1 t_1 + \dot{m}_2 t_2 = \dot{m}_3 t_3$$

$$1.7442 \times 25 + 0.177 \times 40 = (1.7442 + 0.177) t_3$$

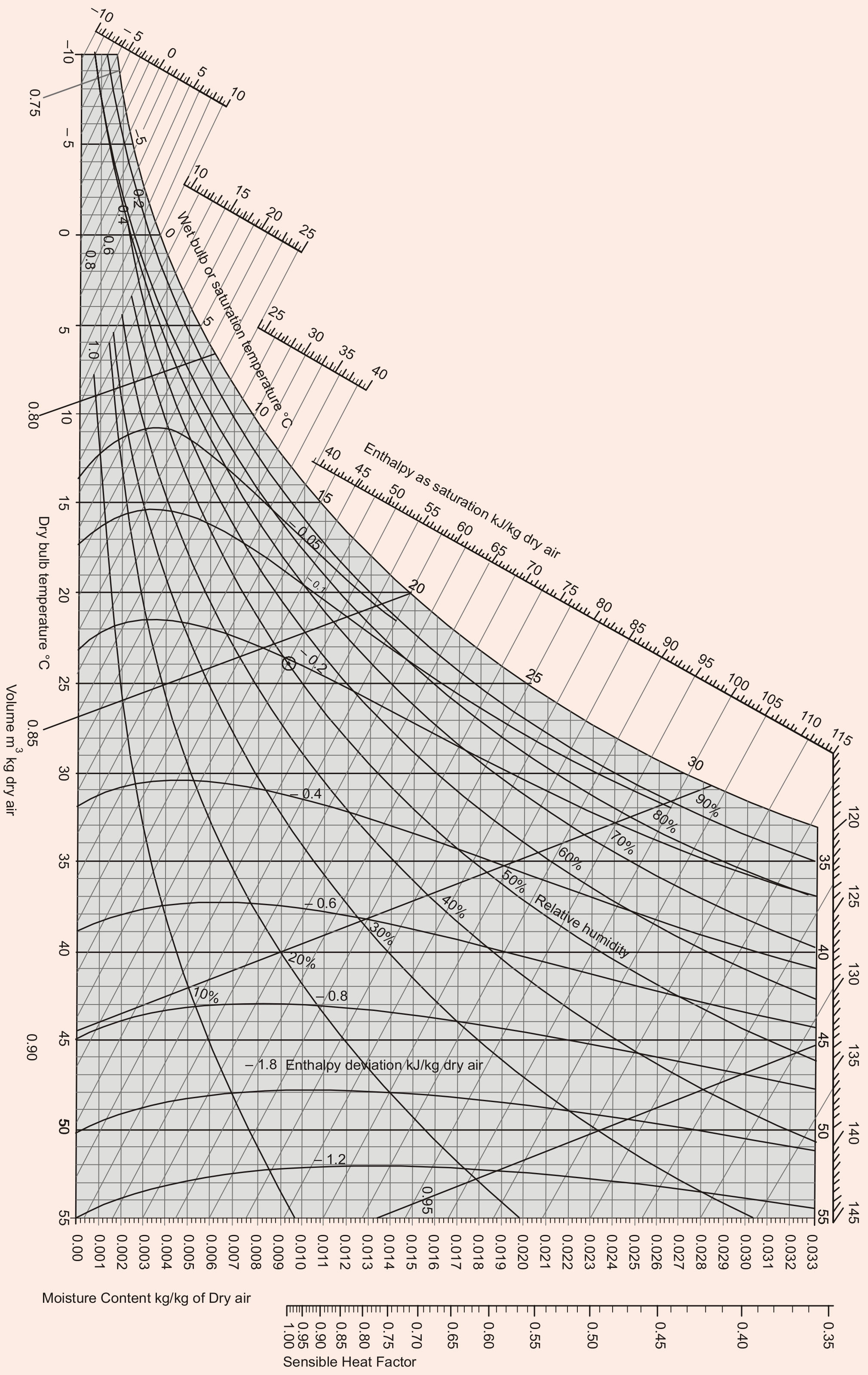
$$t_3 = 26.38°$$

$$h_3 = 64\ kJ/kg$$

$$\omega_3 = 0.0145\ kg/kg\ da$$

$$\phi_3 = 65\%$$

$$WBT_3 = 21.8°C$$



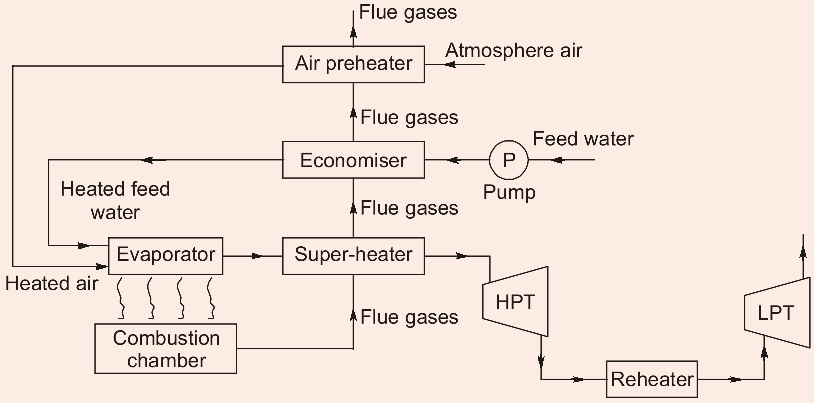
End of Solution

Q.5 (d) With the help of diagram, show the placing of evaporator, superheater, reheater and economiser in the boiler. Also justify the placement at specific location.

[12 marks : 2021]

Solution:

The placement of various elements of boiler and the reheater is as shown in the schematic diagram below:



Evaporator is placed near the combustion chamber where it receives hot combustion gases and the conversion of water into steam (generally, wet steam takes place).
Superheater is placed after the evaporator but before the high pressure turbine so that it can utilise the heat going away with hot gases by superheating the steam.
Then the steam is sent to high pressure turbine for expansion upto the reheater pressure. The reheater is placed between the high and low pressure turbines so that it can reheat the steam coming from the pressure turbine.
Economizer is a device which utilizes the heat from hot flue gases coming out of superheater to preheat the feed water. Due to this the heat be supplied in the evaporator reduces by some amount; increasing the efficiency at the same time. Thus, the most appropriate position where the economizer can be placed is after superheater.

End of Solution

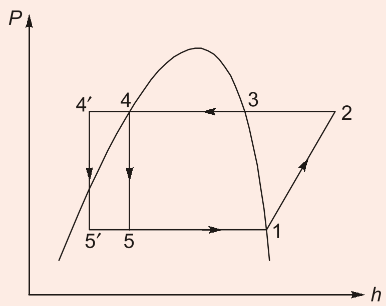
Q.5 (e) Explain the effect of the following on the COP of Vapour compression refrigeration cycle with suitable P-h diagram:

(i) Subcooling of the liquid in condenser.
(ii) Decrease of Evaporator temperature.
(iii) Wet Compression

[12 marks : 2021]

Solution:

(i) Effect of subcooling



It is process decreasing temperature at constant pressure below saturated liquid.

1 - 2 - 3 - 4 - 5

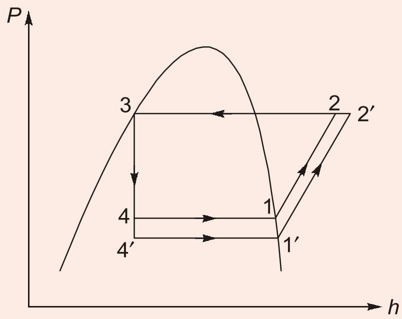
$$R_E = h_1 - h_5$$
$$W_{in} = h_2 - h_1$$
$$COP = \frac{R_E}{W_{in}}$$

1 - 2 - 3 - 4′ - 5′ (Subcooling)

$$R_E = h_1 - h_5' \uparrow$$
$$W_{in} = h_2 - h_1 \rightarrow \text{Constant}$$
$$\uparrow COP = \frac{R_E \uparrow}{W_{in}}$$

(ii) Decreases in Evaporator temperature,

1 - 2 - 3 - 4 - 1

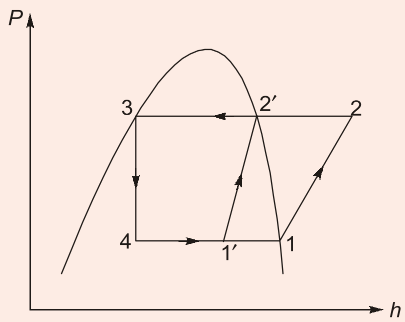


$$R_E = h_1 - h_4$$
$$W_{in} = h_2 - h_1$$
$$COP = \frac{R_E}{W_{in}}$$

1′ - 2′ - 3 - 4′ - 1′ (Decrease in T_E)

$$R_E = h_{1'} - h_{4'} \downarrow$$
$$W_{in} = h_{2'} - h_{1'} \uparrow$$
$$\downarrow COP = \frac{R_E \downarrow}{W_{in} \uparrow}$$
$$\frac{T_C}{T_{E_2}} \uparrow \Rightarrow \frac{P_C}{P_{E_2}} \uparrow \Rightarrow \eta_V \downarrow$$

(iii) Effect of wet compression: We compression means the entry point to the compression in wet region that is a mixture of liquid and vapour.



1 - 2 - 3 - 4

$$R_E = h_1 - h_4$$

1′ - 2′ - 3 - 4

$$R_E = h_1' - h_4 \downarrow$$

The liquid particle which is present in the mixture of refrigerant may wash away the lubricant and it increases the chance of wear and tear.

End of Solution

Q.6 (a) Show that a Pelton turbine with coefficient of velocity C_v and blade friction coefficient K can have a maximum hydraulic efficiency:

$$(\eta_H)_{max} = \frac{1}{2}C_v^2(1 + K\cos\beta')$$

where β' = (180° – blade angle)

A double overhang Pelton wheel unit is coupled to a generator producing 30000 kW under an effective head of 300 m at the base of the nozzle. Find the size of the jet, mean diameter of runner, synchronous speed of each wheel. Assume generator efficiency as 93% overall efficiency of turbine as 85%, coefficient of nozzle velocity as 0.97, speed ratio as 0.46, frequency of generator as 50 cycles per second, pair of poles as 16 and the jet ratio as 12.

Also take ρ_{water} = 1000 kg/m³.

[20 marks : 2021]

Solution:

We know that,

$$\eta_H = \frac{RP}{WP} = \frac{\rho Q(V_{w1} + V_{w2})u}{\rho g Q H}$$

$\therefore$

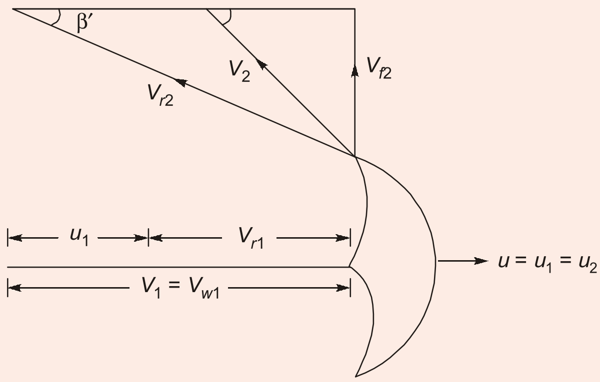
$$\eta_{nozzle} = \frac{\frac{1}{2}\dot{m}V_1^2}{\rho g Q H}$$

$$\rho g Q H = \frac{\frac{1}{2}\dot{m}V_1^2}{\eta_{nozzle}}$$

$$\eta_H = \frac{\rho Q(V_{w1} + V_{w2})u}{\dfrac{\frac{1}{2}\dot{m}V_1^2}{\eta_{nozzle}}} \quad [\therefore \text{ We know that } \eta_{nozzle} = C_V^2]$$

$$\eta_H = \frac{\rho Q(V_{w1} + V_{w2})u}{\dfrac{\dot{m}V_1^2}{2 \times C_V^2}} \quad [\because \dot{m} = \rho Q]$$

$$\eta_H = \frac{2C_V^2(V_{w1} + V_{w2})u}{V_1^2}$$



From velocity triangles,

$$V_{r1} = V_1 - u_1,\ V_{r2} = kV_{r1}$$
$$V_{w1} = V_1$$
$$V_{w2} = V_{r1}\cos\beta' - u_2 = k(V_1 - u_1)\cos\beta' - u_2$$

$$\eta_H = \frac{2C_V^2\left[V_1 + k(V_1 - u_1)\cos\beta' - u_2\right]y}{V_1^2}$$

$$\eta_H = \frac{2C_V^2(V_1 - u_1)(1 + k\cos\beta')u}{V_1^2} \qquad (\because u = u_1 = u_2)$$

$$\eta_H = f(V_1, u_1) \text{ only}$$

If V_1 = Constant,

$$\eta_H = f(u) \text{ only}$$

For maximum efficiency condition,

$$\frac{d\eta_H}{du} = 0$$

$$\frac{d}{du}\left[\frac{2C_V^2(V_1 - u_1)(1 + k\cos\beta')u}{V_1^2}\right] = 0$$

$$\frac{d}{du}\left[(V_1 - u_1)u\right] = 0$$

$$V_1 = 2u = 0$$

$$u = \frac{V_1}{2}$$

For

$$\eta_{max} = \frac{2C_V^2\left[V_1 - \frac{u_1}{2}\right]\times\left[1 + k\cos\beta'\right]\times\frac{V_1}{2}}{V_1^2}$$

$$\eta_{H/max} = \frac{C_V^2\left[1 + k\cos\beta'\right]}{2}$$

Double overhang wheel,

$$G_P = 30000 \text{ kW}$$
$$H = 300 \text{ m}$$
$$\eta_G = 93\%,\ \eta_o = 85\%$$
$$C_V = 0.97$$
$$k_{u1} = 0.46$$
$$f = 50 \text{ Hz}$$

16 Pair of poles,

No. of poles = 32

$$m = \frac{D}{d} = \text{Jet ratio} = 12$$

$\Rightarrow$ $$\eta_G = \frac{GP}{SP} \Rightarrow 0.93 = \frac{30000}{SP}$$

$$SP = 32258.06 \text{ kW}$$

Shaft power produced per wheel,

$$SP = \frac{32258.06}{2} = 16129.03 \text{ kW}$$

$$\eta_o = \frac{SP}{\rho g Q H}$$

$$Q = \frac{16129.03 \times 10^3}{1000 \times 9.81 \times 0.85 \times 300} = 6.447 \text{ m}^3/\text{sec}$$

$$Q = a_1 V_1 = \frac{\pi}{4} d^2 V_1$$

$$V_1 = C_V \sqrt{2gH} = 0.97\sqrt{2 \times 9.81 \times 300} = 74.42 \text{ m/s}$$

$$Q = 6.447 = \frac{\pi}{4} d^2 \times 74.42$$

$$d = 0.332 \text{ m}$$

Given that, $$m = \frac{D}{d} = 12$$

$$D = 12 \times 0.332 = 3.985 \text{ m}$$

$$k_u = 0.46 = \frac{u_1}{\sqrt{2gH}} = \frac{u_1}{\sqrt{2 \times 9.81 \times 300}}$$

$$u_1 = 35.29 \text{ m/s} = u_2$$

$$u_1 = \frac{\pi D N}{60} \Rightarrow 35.29 = \frac{\pi \times 3.985 \times N}{60}$$

$$N = 169.13 \text{ rpm}$$

Synchronous speed of runner $= f = \frac{PN}{120}$

$$50 = \frac{32 \times N}{120} \quad [\because P = 32]$$

$$N = 187.5 \text{ rpm}$$

Specific speed of each wheel, $N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{169.13\sqrt{16129.03}}{300^{5/4}}$

$$N_s = 17.19 \text{ (SI)}$$

End of Solution

Q.6 (b) (i) What are the major sources of air leakage in the condenser of a power plant? Write the effect of pressure of air on the performance of the plant. Also discuss working of air ejector.

(ii) In a typical power plant, steam at 35°C goes to the condenser. Steam flow is 650 T/hr. Moisture in steam at inlet of condenser is 12%. Condenser pressure is maintained at 0.075 bar. Cooling water enters at 23°C and leaves condenser at 33°C. Find rate of cooling water flow and rate of air leak in the condenser. Take the following data at 0.075 bar:

h_f, specific enthalpy of saturated water = 146.7 kJ/kg

h_{fg}, specific volume of conversion from saturated liquid to dry vapour = 2418.6 kJ/kg.

v_f, specific volume of saturated water = 0.001006 m^3/kg

V_{fg}, specific volume of conversion from saturated liquid to dry vapour = 25.22 m^3/kg

Specific heat of water = 4.187 kJ/kgK

R = 0.287 kJ/kgK

[8 + 12 marks : 2021]

Solution:

(i)

Sources of air in the condenser: The main sources of air leakage in the condenser are the following:

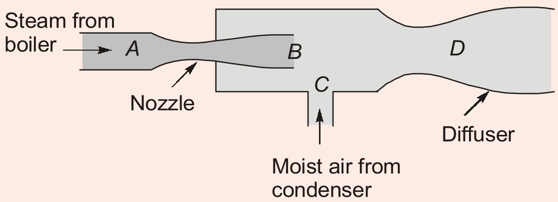
1. The ambient air leaks to the condenser chamber at the joints and glands which are internally under pressure lower than that of ambient. It can be reduced by taking utmost care while designing and making vacuum joints.
2. Another source of air is the dissolved air with feed water. The dissolved air in feed with steam into the boiler and it travels with steam into the condenser. Its quantity depends upon the quality of feed water.
3. In case of a jet condenser, some air comes in with the injected water in which it is dissolved.

Effects of Air Leakage into the Condenser: The presence of air in the condenser is a far serious concern. It affects the performance of the condenser to a great extent, thereby, the performance of steam power plant. The presence of air into the condenser puts the following effects:

1. The presence of air lowers the vacuum in the condenser. Thus the back pressure of the plant increases, and consequently, the work output of the turbine reduces.
2. The presence of air also lowers the partial pressure of steam and hence lower saturation temperature. The steam with lower saturation temperature has higher latent heat. A large quantity of cooling water is required to get the desired result in the condenser.
3. Air forms the film adjacent to the tube surface in the condenser. Air has very poor thermal conductivity. Hence, the rate of heat transfer from vapour to cooling medium is reduced.

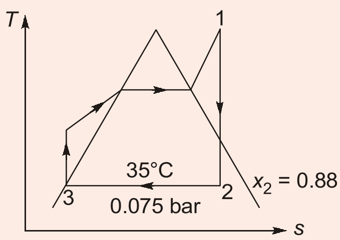
4. The presence of air in the condenser corrodes to the metal surfaces. Therefore, the life of condenser is reduced.

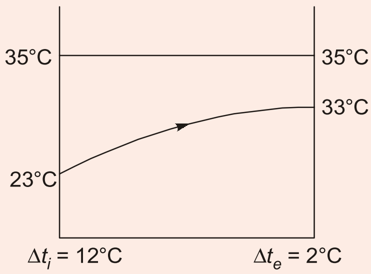
Air Ejector Working: It used main steam at a reduced pressure that enters a driving flow nozzle in the first stage ejector, from which it exists with high velocity and momentum but at a reduced pressure. This reduced pressure draws non-condensable from the condenser and as a result of momentum exchange, the gases are carried by steam jet. The combined flow of steam and gas is raw compressed and pressure increases in the diffuser of the first stage ejector and discharged into small intercondenser where steam is condensed.



Cooling is here done by main condenser and is part of feed water heating system resulting in increase of efficiency of the plant. The condensed steam is drained and returned to a low pressure part of cycle. The non-condensable and any remaining steam are then passed to the second stage ejector, where they are compressed also known as vent condenser. The steam if any get condensed and non-condensable (air) at higher pressure than atmospheric pressure is vented out.

(ii)





$$\dot{m}_s = 650 \text{ T/Hr}$$

0.075 Bar ⇒
$$h_f = 146.7 \text{ kJ/kg}$$
$$h_{fg} = 2418.6 \text{ kJ/kg}$$

$$(Q_{lost})_{steam} = (Q)_{H_2O} \qquad \left(Q_{H_2O} = \dot{m}_w C_P wW\Delta t\right)$$

$$\frac{650\times10^3}{3600}\times2418.6\times0.88 = \dot{m}_w\times4.187(33-23)$$

$$\dot{m}_w = 9178.13 \text{ kg/sec}$$

At 35°C;
$$P_{sat} = 0.05628 \text{ bar} \qquad \text{(From steam tables)}$$
$$P = P_{sat} + P_{air}$$
$$0.075 = 0.05628 + P_{air}$$

$$P_{air} = 0.01872 \text{ bar}$$
$$v_2 = v_f + x_2\, v_{fg}$$
$$v_2 = 0.001006 + 0.88 \times 25.22$$
$$v_2 = 22.19 \text{ m}^3\text{/kg}$$
$$\therefore \quad P_{air}V = \dot{m}_{air} R_{air} T$$
$$0.01872 \times 10^5 \times \frac{650 \times 10^3}{3600} \times 22.19 = \dot{m}_{air} \times 0.287 \times 10^3 \times 308$$
$$\dot{m}_{air} = 84.84 \text{ kg/sec}$$

End of Solution

Q.6 (c) What do you understand by biomass gasification? How are gasifiers classified? Describe with a schematic diagram the working of a downdraft gasifier.

[20 marks : 2021]

Solution:

The word gasification (or thermal gasification) implies converting solid fuel into a gaseous fuel by thermochemical method without leaving any solid carbonaceous residue. Gasification is an established technology, the first commercial application of which dates back to 1830. During World War II, biomass gasification systems appeared all over the world to power vehicles to keep the basic transport systems running. Gasifier is the equipment that converts biomass into producer gas. The most common raw materials used are wood chips and other wastes from wood industry, coconut shells and straw. Biomass that has high ash content such as rice husk can also be handled with some difficulty.

Gasification involves partial combustion (oxidation in restricted quantity of air/oxidant) and reduction operations of biomass. In a typical combustion process generally the oxygen surplus, there in a gasification process the fuel is surplus. The combustion products, mainly carbon dioxide, water vapour, nitrogen, carbon monoxide and hydrogen pass through the glowing layer of charcoal for the reduction process to occur. During this stage, both carbon dioxide and water vapour, oxidize the char to form CO, H_2 and CH_4. The following are the typical reactions, which occur during gasification:

$$C + O_2 \rightarrow CO_2 \quad \text{(Combustion, 240 kJ/mole)} \qquad ...(1)$$

The moisture available in the biomass is converted to steam and generally no extra moisture is required. Thus the products of combustion of pyrolysis gases results in CO_2 and H_2O (steam), which further react with char:

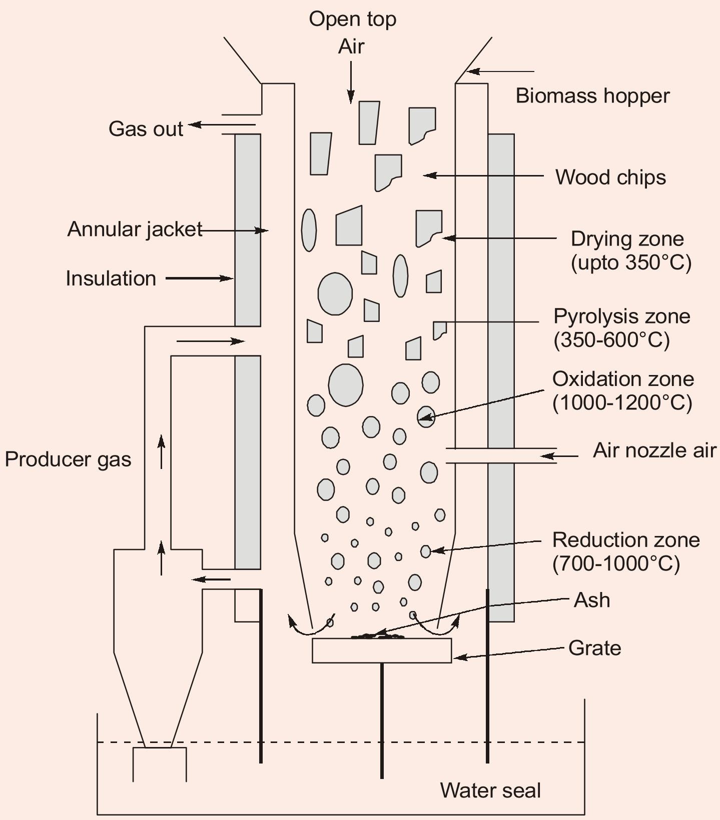
$$C + CO_2 \rightarrow 2CO \quad \text{(Boudourad reaction, –164.9 kJ/mole)} \qquad ...(2)$$
$$C + H_2O \rightarrow CO + H_2 \quad \text{(Water gas reaction, –122.6 kJ/mole)} \qquad ...(3)$$
$$CO + H_2O \rightarrow CO_2 + H_2 \quad \text{(Water shift reaction, 40.2 kJ/mole)} \qquad ...(4)$$
$$C + 2H_2 \rightarrow CH_4 \quad \text{(Methane reaction, 83.3 kJ/mole)} \qquad ...(5)$$

The composition of the gas produced depends on the degree of equilibrium attained among various reactions.

Gasifiers are broadly classified into (i) fixed-bed gasifier and (ii) fluidized-bed gasifier.

The fixed-bed gasifiers are further classified as (a) downdraft, (b) updraft, and (c) cross-draft types, depending upon the direction of air flow.

Downdraft Type: The downdraft type is best suited for a variety of biomass. Its design forces the raw products to pass through a high-temperature zone so that most of the unburnt pyrolysis products (especially tar) can be cracked into gaseous hydrocarbons, thus producing a relatively clean gas. A recently developed open-top downdraft-type gasifier is shown in figure.



Downdraft biomass gasification plant

In steady state operation, heat from the combustion zone, near, the air nozzle is transferred upwards by radiation, conduction and convection causing wood chips to pyrolyse and lose 70–80% of their weight. These pyrolysed gases burn with air to form CO, CO_2, H_2 and H_2O, thereby raising the temperature to 1000–1200°C. The product gases from the combustion zone further undergo reduction reaction with char to generate combustible products like CO, H_2 and CH_4. Generally abut 40 – 70% air is drawn through the open top depending on the pressure drop conditions due to the size of wood chips and gas-flow rate. This flow of air opposite to the flame front helps in maintaining homogeneous air/gas flow across the bed. Combining the open top with the air nozzle towards the bottom of the reactor helps in stabilizing the combustion zone by consuming the top. As a consequence, the high temperature zone spreads above the air nozzle by radiation and conduction, aided by air flow from the top. The tar thus is eliminated in the best possible way be creating a high-temperature oxidizing atmosphere in the reactor itself. The gas produced is withdrawn from an exit at the bottom and reintroduced in the annular jacket for heat recovery. The hot gas which enters the annular jacket around 500°C, transfers some heat to the wood chips inside, improving the thermal efficiency of the system in addition to drying the wood in this zone. The inner wall temperature reaches more than 350°C after a few hours of operation. This aspect enables the use of wood chips with moisture content as high as 25%. The regenerative heating due to the transfer of heat from hot gas to the biomass moving downwards also increases its residence time in the high-temperature zone. This leads to better tar cracking. The raw producer gas thus obtained can be used as such for thermal applications. However, for use in IC engines, further processing is required. Admission of hot gas into an engine results in loss of power and hence the gas has to be cooled. Raw gas contains varying amounts of dust (ash and char) particles, moisture and tar. Dust and tar are detrimental to the life of an engine. Hence, the gas has to be cooled and cleaned before admitting to the engine. The upper limit of allowable tar is about 50 mg/m^3 and that for particulate (size less than 10 μm) content is about 50 mg/m^3. The gas may be cooled t the ambient temperature by direct injection of cooling water from a spray tower. A sand-bed filter may be deployed to remove the particulate collected by the cooling water. Periodic washing of this sand bed is adequate to keep the operation smooth. For filtering of the gas, a sand-bed filter with specific particle size distribution is used. The filter is divided into coarse (sand particle size 0.5 to 2 mm) and fine sections (particle size 0.2 to 0.6 mm). The size of the filter area is so chosen that the gas velocities through the filter bed do not exceed 0.1 m/s. This low velocity coupled with a tortuous path causes the removal of a large part of the dust from the gas. Some part of the tar also gets deposited in the filter circuit, particularly when the moisture carried over from the cooler causes slight wetting of the sand bed.

End of Solution

Q.7 (a) What is meant by volumetric efficiency of a reciprocating compressor? How is it affected by

(i) Speed of the compressor

(ii) Throttling across valves, and

(iii) Delivery pressure?

It is described to compress air at 1 bar and 25°C and deliver it at 160 bar using multi-stage compression and intercoolers. The maximum temperature during compression must not exceed 125°C and cooling in the intercooler is done so as not to drop the temperature below 30°C. The law of compression followed is $PV^{1.25}$ = constant for all stages.

Calculate:

(i) Number of stages required,

(ii) Work input per kg of air, and

(iii) Heat rejected in the intercoolers.

Take R = 0.287 kJ/kgK

C_v = 0.71 kJ/kgK

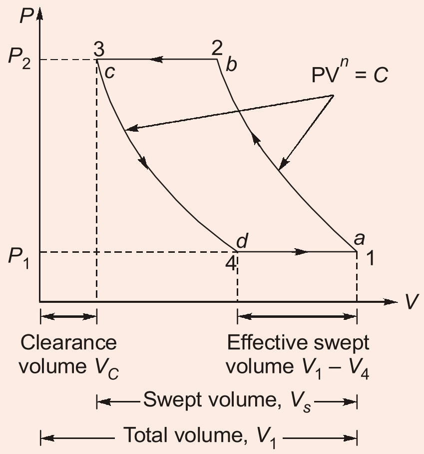
[20 marks : 2021]

Solution:

Volumetric efficiency: Volumetric efficiency of compressor is the measure of the deviation from volume handling capacity of compressor. Mathematically, the volumetric efficiency is given by the ratio of actual volume sucked and swept volume of cylinder.

It is the ratio of actual volume of the FAD at standard atmospheric condition in one delivery stroke (actual air intake) to the swept volume (Theoretical air intake) by the piston during the stroke.

$$\text{Volumetric efficiency} = \frac{\text{Actual volume}(V_a)}{\text{Theoretical volume}(V_s)} \times 100\%$$



(i) Speed of compressor:

$$\eta_V = \frac{V_{act}}{\frac{\pi}{4} \times D^2 \times L \times \frac{N}{60} \times K}$$

[K = Number of cylinders]

$$\text{with} \uparrow N \Rightarrow \eta_L \downarrow$$

(ii) Throttling across valve:

As the load is increases (full load condition), the throttle valve is opened allowing more air to pass into inlet manifold. During this throttling process lots of throttling losses takes place, and hence less air is pumped due to this loss. Hence volumetric efficiency is reduced.

(iii) Delivery pressure:

$$\eta_V = 1 + C - C\left(\frac{p_H}{p_L}\right)^{1/n}$$

$$p_H = \text{Delivery pressure}$$

$$\text{With } p_H \uparrow \Rightarrow \eta_L \downarrow$$

Suction pressure, P_s = 1 bar

Suction temperature, T_s = 25 + 273 = 298 K

Let P_1 and T_1 be the pressure and temperature after Ist stage

(i) Number of stages,

$$\frac{P_1}{P_s} = \left(\frac{T_1}{T_s}\right)^{n/n-1} = \left(\frac{398}{298}\right)^{\frac{1.25}{1.25-1}} = 4.25$$

$$P_1 = 4.25 \times 1 = 4.25 \text{ bar}$$

In multistage compression and intercoolers,

$$\frac{T_2}{T_1} = \frac{T_3}{T_2} = \frac{T_4}{T_3} \ldots\ldots\ldots \frac{T_m}{T_{m-1}} \qquad [\because (m+1) \text{ be the stages}]$$

$$\frac{T_2}{T_1} = \frac{1.25+273}{30+273} = 1.313$$

Now,
$$\left(\frac{P_n}{P_1}\right)^{1/m} = \left(\frac{T_2}{T_1}\right)^{\left(\frac{n}{n-1}\right)}$$

$$\left(\frac{160}{4.25}\right)^{1/m} = (1.3135)^{1.25/0.25}$$

$$\frac{1}{m}\ln\left(\frac{160}{4.25}\right) = \ln(3.91)$$

$$\frac{1}{m} = 0.3758 \; [m = \text{Number of stages after suction stage}]$$

$$m = 2.66$$

So, number of stages = $m+1 = 3.66 \approx 4$

(ii) Workdone per kg of air,

Pressure ratio in 1st stage = 4.25

Pressure ratio in 2nd and after stages

$$= \left(\frac{160}{4.25}\right)^{1/3} = 3.351$$

$$\text{Workdone /kg in 1st stage} = \frac{n}{n-1}RT_s\left[\left(\frac{P_1}{P_s}\right)^{n-1/n} - 1\right]$$

$$= \frac{1.25}{0.25} \times 0.287 \times 298 \times \left[(4.25)^{0.25/1.25} - 1\right]$$

$$= 143.51 \text{ kJ/kg}$$

$$\text{Work done in 2nd, 3rd and 4th stages} = 3 \times \frac{n}{n-1}RT_1\left[(3.351)^{0.25/1.25} - 1\right]$$

$$= 3\times\frac{1.25}{0.25}\times 0.287\times 303\times\left[(3.351)^{1/5}-1\right]$$

$$= 356.89 \text{ kJ/kg}$$

$$\text{Total work done} = 143.51 + 356.89 = 500.40 \text{ kJ/kg}$$

(iii) Heat rejected in intercoolers per kg of air

$$= 3 \times \text{Heat rejected in per intercooler}$$
$$= 3\times C_P \times [T_2 - T_1]$$
$$= 3\times [C_V + R][T_2 - T_1]$$
$$= 3\times [0.71 + 0.287]\,[398 - 303]$$
$$= 3\times 0.997\times 95$$
$$= 284.145 \text{ kJ/kg}$$

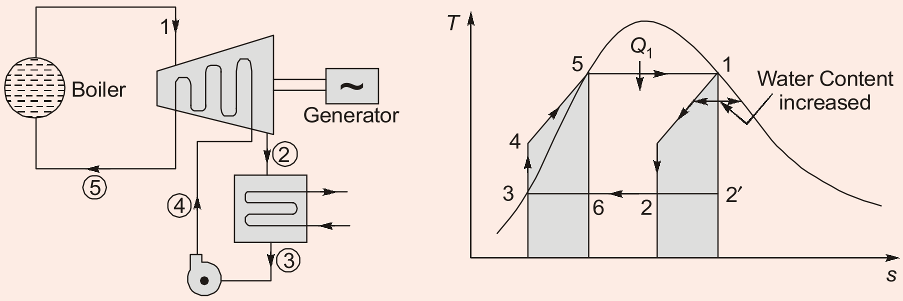
End of Solution

Q.7 (b) Explain why ideal regenerative feed water heating is not used in practice. Derive expression of optimum regeneration to get maximum efficiency with one regenerative feed water heater.

[20 marks : 2021]

Solution:

Ideal regeneration: The liquid from condenser is sent around the turbine, where it takes up heat from the steam flowing inside. The steam condition line deviates from the original expansion line. In original expansion, water content in steam increases as expansion proceeds towards the end. The expansion line is isentropic or vertical, but in ideal regeneration case, as the steam gives out heat to water flowing outside the turbine, condensing more steam and water contents in the steam, at every stage of expansion process, it becomes more than the original line as shown in figure .



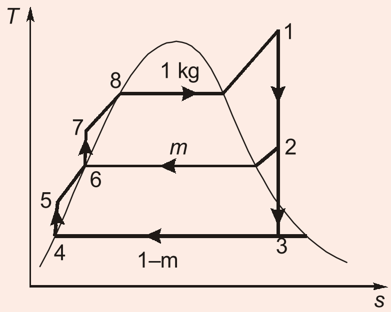
(a) (b)

However the cycle is not practicable because:

- reversible heat transfer cannot be realized in finite time.
- heat exchanger in the turbine is mechanically impracticable
- the moisture content of the steam in the turbine is high, which leads to excessive erosion of turbine blades.

Optimum regeneration with one feed water Heater: Complete carnotization of Rankine cycle is not possible with a finite number of heaters. If there is one feedwater heater used,

m kg of steam is extracted from the turbine of each kg of steam entering it to heat the feedwater from state 5 to state 6 so that by energy balance,



$$m(h_2 - h_6) = (1 - m)(h_6 - h_5)$$

or
$$m = \frac{h_6 - h_5}{h_2 - h_5} = \frac{h_6 - h_4}{h_2 - h_4}$$

Therefore, the thermal efficiency of the cycle is

$$\eta = 1 - \frac{(1-m)(h_3 - h_4)}{h_1 - h_6} = 1 - \frac{\left(1 - \dfrac{h_6 - h_4}{h_2 - h_4}\right)(h_3 - h_4)}{h_1 - h_6}$$

$$= 1 - \frac{(h_2 - h_6)(h_3 - h_4)}{(h_2 - h_4)(h_1 - h_6)} \qquad \text{...(a)}$$

It may be approximately assumed that the turbine expansion line follows a path on the diagram such that $(h - h_f)$ = constant = β, where h is the local enthalpy on the expansion line at a given pressure and h_f is the enthalpy of saturated water at that pressure. Therefore, as seen in figure (b),

$h_1 - h_8 = h_2 - h_6 = h_3 - h_4 = \beta$ = constant. Let the enthalpy rise of feedwater in the heater is γ, which is equal to $(h_6 - h_4)$.

Now, $\quad h_2 - h_4 = h_2 - h_6 + h_6 - h_4 = \beta + \gamma$

If total enthalpy rise of feedwater is equal to $\alpha = h_8 - h_4$, then

$$h_1 - h_6 = h_1 - h_8 + h_8 - h_4 + h_4 - h_6 = \beta + \alpha - \gamma$$

Therefore eq. (a) can be written in the form

$$\eta = 1 - \frac{\beta^2}{(\beta + \gamma)(\alpha + \beta - \gamma)} \qquad \text{...(b)}$$

Here, α and β are fixed and γ is variable. So, there is an optimum value of γ for which η is a maximum. On differentiation.

$$\frac{d\eta}{d\gamma} = b^2(a + b - g) - (b + g) = 0 \qquad \left(\therefore \gamma = \frac{\alpha}{2}\right) \qquad \text{...(c)}$$

The cycle efficiency is maximum when the total enthalpy rise of feedwater $(h_8 - h_4)$ from the condenser temperature to the boiler saturation temperature is divided equally between the feedwater heater and the economiser (i.e. $h_8 - h_6 = h_6 - h_4$) in a single bleed cycle. So, the temperature rise of feedwater in the heater is

$$\Delta t = \frac{1}{2}\left(t_{\text{boiler saturation}} - t_{\text{condenser}}\right)$$

and the corresponding cycle efficiency is

$$\eta = 1 - \frac{\beta^2}{\left(\beta + \frac{\alpha}{2}\right)\left(\alpha + \beta - \frac{\alpha}{2}\right)} = 1 - \frac{\beta^2}{\left(\beta + \frac{\alpha}{2}\right)^2} = \frac{\alpha^2 + 4\alpha\beta}{(\alpha + 2\beta)^2} \quad \text{...(d)}$$

For a non-regenerative cycle,

$$\eta_0 = 1 - \frac{h_3 - h_4}{h_1 - h_4}$$

Now, $h_3 - h_4 = \beta$ and $h_1 - h_4 = h_1 - h_8 + h_8 - h_4 = \beta + \alpha$

$$\therefore \quad \eta_0 = 1 - \frac{\beta}{\alpha + \beta} = \frac{\alpha}{\alpha + \beta} \quad \text{...(e)}$$

The efficiency gain due to regeneration

$$\Delta\eta = \eta - \eta_0 = \frac{\alpha_2 + 4\alpha\beta}{(\alpha + 2\beta)^2} - \frac{\alpha}{\alpha + \beta} = \frac{\alpha^2\beta}{(\alpha + \beta)(\alpha + 2\beta)^2} \quad \text{...(f)}$$

This is positive. This shows that the cycle efficiency has improved due to regeneration.

End of Solution

Q.7 (c) (i) What is the consideration while deciding number of blades for a horizontal axis wind turbine? State the significance of optimal tip-speed ratio and comment what will happen if the tip-speed ratio is very high or very low.

(ii) A 3-bladed rotor of horizontal axis wind turbine having blade length of 40 m is installed at a location where free wind velocity of 20 m/sec is available. What shall be the ideal rotor speed that can be maintained for optimal energy extraction?

[10 marks : 2021]

Solution:

(i) Consideration while deciding number of blades for HAWT depends on type of application.
For power generation: Speed of rotor required as high, so it should have low solidity and high TSR (Tip) speed ratio)
For water pumping: Speed of rotor required is low, so It should have high solidity and low TSR(λ)
TIP speed ratio (λ):
It is ratio of speed of tip of blade to speed of undisturbed wind

$$\lambda = \frac{R\omega}{V_o}$$

For constant wind speed (V_i): The power extracted by a turbine (P_t) will decrease if

Blades are so closer together, or rotating so rapidly, i.e. TSR is very high that a blade moves into the turbulence created by a preceding blade.

(ii) The blades are so far apart or rotating slowly (i.e. TSR is very low) that much of air passed through the cross-section of the device without interacting with the blades.

Tip speed ratio at optimum power extraction (λ_0):

$$\lambda_0 = \frac{4\pi}{n}$$

n → Number of blades

For 2-blades Turbine (HAWT): Optimum power extraction occurs at

$$\lambda_0 = 2p$$

Given that: 3-blade rotor

$$n = 3 \qquad R = 40 \text{ m} \qquad V_i = 20 \text{ m/s}$$

$$\lambda_0 = \frac{4\pi}{n}$$

$$\Rightarrow \quad \lambda_0 = \frac{4\times\pi}{3} = 4.188$$

$$\lambda_0 = \frac{R\times\omega}{v_i}$$

$$\Rightarrow \quad 4.188 = \frac{40\times\omega}{20}$$

$$\omega = 2.094 \text{ rad/s}$$

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

$$2.094 = \frac{2\pi\times N}{60}$$

$$\Rightarrow \quad N = 19.99$$

$$N = 20 \text{ rpm}$$

End of Solution

Q.8 (a) Prove that the efficiency corresponding to the maximum work done in a Brayton cycle is given by the ratio

$$\eta_{w\,max} = 1 - \frac{1}{\sqrt{t}}$$

where t is the ratio of the maximum to minimum temperature of the cycle.

An ideal open cycle gas turbine plant using air operates with an overall pressure ratio of 4 and between temperature limits of 300 K and 1200 K. Assuming the constant value of specific heat C_p =
1 kJ/kgK and C_v = 0.717 kJ/kgK, evaluate the specific work output and thermal efficiency for:

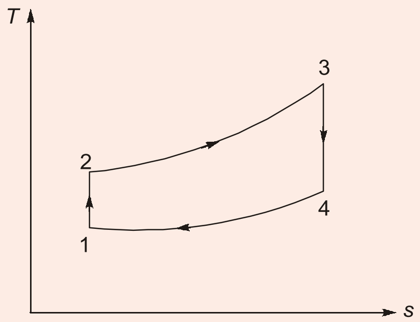
(i) basic cycle with regenerator (heat exchanger), and

(ii) basic cycle with regenerator (heat exchanger) and two-stage intercooled compressor.

Assume optimum stage pressure ratios, perfect intercooling and perfect regeneration.

[20 marks : 2021]

Solution:



$$\text{Work, } W = W_T - W_C$$
$$= (h_3 - h_4) - (h_2 - h_1)$$
$$= C_P[T_3 - T_4 - T_2 + T_1] \quad \text{... (i)}$$

Since,
$$T_2 T_4 = T_1 T_3$$
$$T_2 = \frac{T_1 T_3}{T_4}$$

$$\Rightarrow \quad W = C_P\left[T_3 - T_4 - \frac{T_1 T_3}{T_4} + T_1\right]$$

$$\frac{dW}{dT_4} = 0 \Rightarrow \quad T_4 = \sqrt{T_1 T_3}$$

$$\therefore \quad T_2 = T_4\sqrt{T_1 T_3}$$

$$W_{max} = C_P\left[\sqrt{T_3} - \sqrt{T_1}\right]^2$$

$(r_p)_{opt} \Rightarrow$ For maximum work output

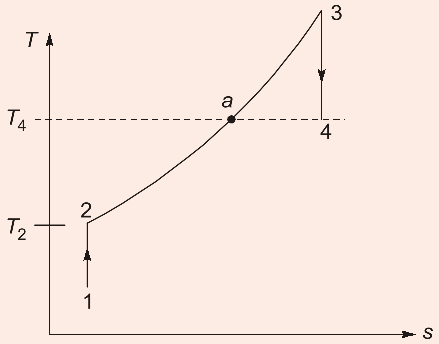
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{\sqrt{T_1 T_3}}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

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

$$\text{Maximum work output, } \eta_{max} = 1 - \frac{1}{\left[\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}\right]^{\frac{\gamma-1}{y}}} = 1 - \frac{1}{\sqrt{\frac{T_3}{T_1}}} = 1 - \sqrt{\frac{T_1}{T_3}}$$

$$\eta_{max} = 1 - \frac{1}{\sqrt{t}} \quad \text{Hence proved.}$$



T_1 = 300 K, T_3 =1200 K, C_P = 1 kJ/kgK, C_V = 0.717 kJ/kgK, r_p = 4, $r = \frac{C_P}{C_L} = 1.4$

Perfect regeneration,

$$\frac{T_2}{300} = (4)^{\frac{1.4-1}{1.4}} = \frac{1200}{T_4}$$

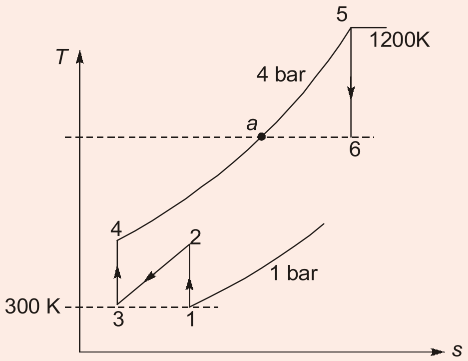
$$T_2 = 445.79 \text{ K}, \; T_4 = 807.54 \text{ K}$$

(i) Specific work output $= W_T - W_C$

$$C_P = (T_3 - T_4) - (T_2 - T_1)$$
$$= 1[(1200 - 807.54) - (445.79 - 300)]$$
$$= 246.67 \text{ kJ/kg}$$

$$\eta_{\text{perfect reg}} = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}} = 1 - \frac{300}{1200}(4)^{\frac{1.4-1}{1.4}} = 0.628$$

(ii)



T_5 = 1200 K, T_1 = T_3 = 300 K,

$$\frac{T_5}{T_6} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}}$$

$$T_6 = 807.54 \text{ K}$$

$$(T_2 = T_4) \Rightarrow \quad \frac{T_2}{300} = \left(\frac{2}{1}\right)^{\frac{1.4-1}{1.4}} \Rightarrow T_2 = 365.7 \text{ K}$$

$$\text{Specific work} = W_T - 2W_C$$
$$= C_P[(T_5 - T_6) - 2(T_2 - T_1)]$$
$$= 1[(1200 - 807.54) - 2(365.7 - 300)]$$
$$= 261.06 \text{ kJ/kg}$$

$$\eta = \frac{\text{Net work output}}{\text{Heat supplied}} \times 100\%$$

$$= \frac{261.06}{(1200\text{-}807.57)} \times 100 = 66.51\%$$

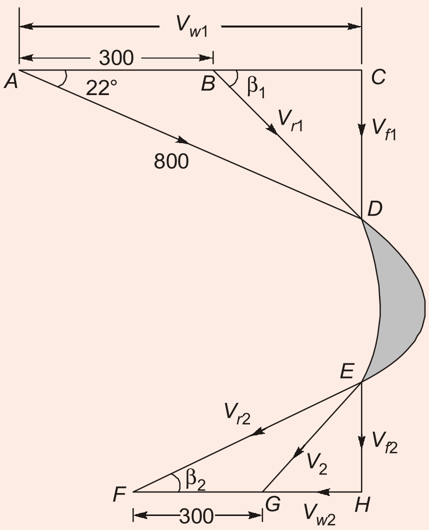
End of Solution

Q.8 (b) The velocity of steam entering in a simple impulse turbine is 800 m/s and nozzle angle is 22°. The mean peripheral velocity of blades is 300 m/s and blades are symmetrical. Calculate the following for steam flow of 2 kg/sec:

(i) Blade angles for entry without shock
(ii) Tangential thrust
(iii) Diagram power
(iv) Diagram efficiency
(v) Axial thrust

[20 marks : 2021]

Solution:



Given:

$$\dot{m} = 2 \text{ kg/sec}$$
$$\beta_1 = \beta_2$$

$$V_{w1} = 800\cos 22° = 741.74 \text{ m/sec}$$
$$V_{f1} = 800\sin 22° = 299.68 \text{ m/sec}$$
$$V_{r1} = \sqrt{(V_{w1} - 300)^2 + V_{F1}^2} = 533.79 \text{ m/sec}$$
$$V_{r1}\sin\beta_1 = V_{f1}$$
$$\Rightarrow \quad \beta_1 = 34.15°$$
$$\therefore \quad \beta_1 = \beta_2$$
$$V_{r2}\cos\beta_2 = 300 + V_{w2} \qquad [V_{r2} = V_{r1}]$$
$$V_{r2}\sin\beta_2 = V_{f2}$$
$$V_{w2} = 141.74 \text{ m/sec}$$
$$V_{f2} = 299.64 \text{ m/sec}$$
$$\text{Tangential thrust} = \dot{m}(V_{w1} + V_{w2}) = 1766.96 \text{ N}$$
$$\text{Axial thrust} = \dot{m}(V_{f1} - V_{f2}) = 0.08 \text{ N}$$
$$\text{Diagram power} = \frac{\dot{m}u(V_{w1} + V_{w2})}{1000} = 530 \text{ kW}$$
$$\eta = \frac{\dot{m}u(V_{w1} - V_{w2})}{\frac{1}{2}\dot{m}V_1^2} = 0.8282$$

End of Solution

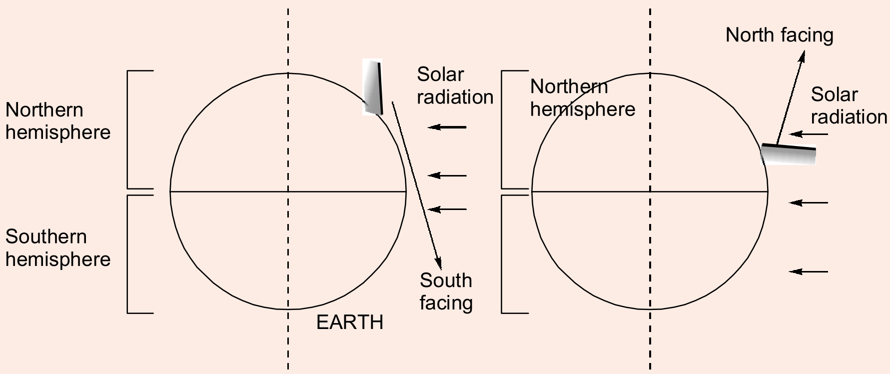
Q.8 (c) (i) Why are solar PV panels placed inclined due south in Indian context? What is the basis of deciding the slope of such solar panels?

(ii) A solar PV panel feeds a dc motor to produce 1 hp of power at shaft output. The motor efficiency is 80%. Each module has multicrystalline silicon solar cells arranged in 9 × 4 matrix. The cell is 125 mm × 125 mm and cell efficiency is 15%. Calculate the number of modulus requirement in the array. Assume global radiations incident normally to the panel as 1000 W/m^2.

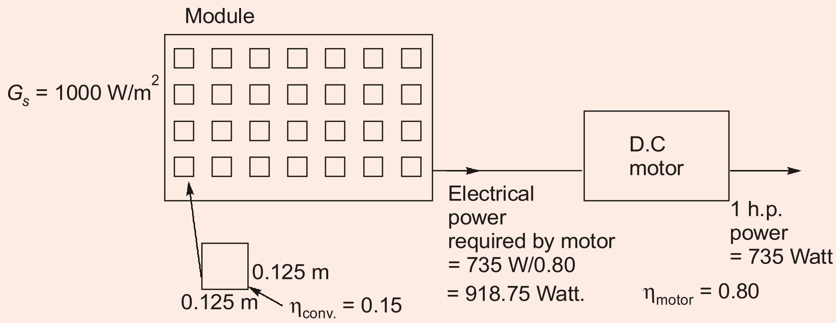
[6 + 14 marks : 2021]

Solution:

(i) As India lies completely in northern hemisphere, it must be installed south facing so that solar radiation falls close to perpendicularly on collector. If placed north facing then sun rays will fall obliquely and collector efficiency will decrease significantly.



(ii)



Let 'N' number of module is required .

Total cell area in one module(A) $= 9 \times 4 \times 0.125 \times 0.125\ m^2$

$$A = 0.5625\ m^2$$

Total energy output from solar array = $N \times 0.5625 \times 1000 \times \eta_{conv.}$

$$N \times 0.5625 \times 1000 \times 0.15 = \text{Total electrical power required by motor}$$

$$N \times 0.5625 \times 1000 \times 0.15 = 918.75$$

$$N = 10.88$$

$\Rightarrow$ $N = 11$ modulus are required.

End of Solution

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