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# ESE-2021
## (MAINS)

**QUESTIONS WITH DETAILED SOLUTIONS**

# MECHANICAL ENGINEERING

## PAPER-I

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# MECHANICAL ENGINEERING

# ESE _MAINS_2021_PAPER – I

# Questions with Detailed Solutions

## SUBJECT WISE WEIGHTAGE

| S.No. | NAME OF THE SUBJECT | Marks |
|---|---|---|
| 1 | Fluid Mechanics | 52 |
| 2 | Turbomachinery (HM + Thermal) | 72 |
| 3 | Thermodynamics | 32 |
| 4 | IC Engines | 52 |
| 5 | Refrigeration and Air conditioning | 34 |
| 6 | Power Plant Engineering | 84 |
| 7 | Heat Transfer | 82 |
| 8 | Renewable Sources of Energy | 72 |

## SECTION – A

**01(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 M)**

**Sol:** **Laminar sublayer:** When a turbulent boundary layer is formed, a very thin region inside the boundary layer near the surface experiences laminar flow. This region of the flow is known as laminar sublayer.

The thickness of this layer is given by $\delta'$

$$\delta' = \frac{11.6\nu}{V^*}$$

The velocity profile inside this layer is assumed to be linear.

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

$$\Rightarrow u = \frac{2U}{\delta}y - \frac{U}{\delta^2}y^2$$

$$\frac{du}{dy} = \frac{2U}{\delta} - \frac{2U \times y}{\delta^2}$$

At y = 0, $\frac{du}{dy} = \frac{2U}{\delta} > 0$

→ The flow is attached.

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

$\Rightarrow u = -\frac{2U}{\delta} \times y + \frac{U}{\delta^2} \times y^2$

$\frac{du}{dy} = -\frac{2U}{\delta} + \frac{2U \times y}{\delta^2}$

At y = 0, $\frac{du}{dy} = -\frac{2U}{\delta} < 0$

→ The flow is separated.

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

$\frac{du}{dy} = \frac{3U \times y}{\delta^2} + \frac{3U \times y^2}{2\delta^3}$

At y = 0, $\frac{du}{dy} = 0$

→ The flow is at the verge of separation.

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**01(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 the three 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 is irreversible, which is reversible and which is impossible:**

**(i) 210 kJ/cycle rejected**

**(ii) 105 kJ/cycle rejected**

**(iii) 315 kJ/cycle rejected** **(12 M)**

**Sol: Clausius Inequality**

It represents mathematical expression to the second law of thermodynamics.

$$\oint \frac{\delta Q}{T} \le 0$$

It is valid for all the processes.

$\oint \frac{\delta Q}{T} = 0$ (For Reversible cycle)

$\oint \frac{\delta Q}{T} < 0$ (For Irreversible cycle)

$\oint \frac{\delta Q}{T} > 0$ (For impossible cycle) (as it violates $2^{nd}$ law of thermodynamics).

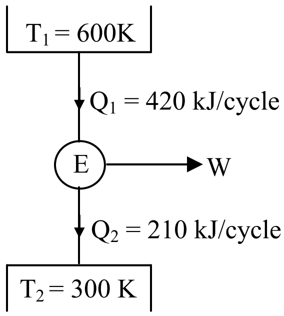
Given data,

Source temperature = 327°C = 600 K

Sink temperature = 27°C = 300 K

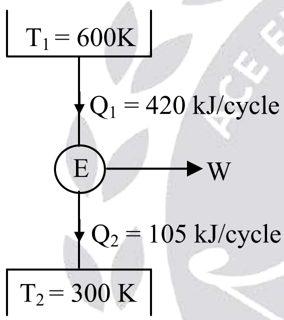
Heat received from source 327°C ($Q_1$) = 420 kJ/cycle

(i) For heat rejection ($Q_2$) of 210 kJ/cycle



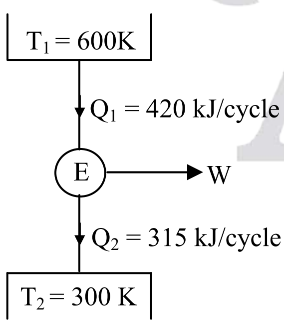

$$\oint \frac{\partial Q}{T} = \frac{Q_1}{T_1} - \frac{Q_2}{T_2} \Rightarrow \frac{420}{600} - \frac{210}{300} = 0 \quad \rightarrow \text{Reversible}$$

(ii) For heat rejection ($Q_2$) of 105 kJ/cycle




$$\oint \frac{\partial Q}{T} = \frac{Q_1}{T_1} - \frac{Q_2}{T_2} = \frac{420}{600} - \frac{105}{300} = 0.35 \rightarrow \text{Impossible}$$

(iii) For heat rejection ($Q_2$) of 315 kJ/cycle




$$\oint \frac{\partial Q}{T} = \frac{Q_1}{T_1} - \frac{Q_2}{T_2} = \frac{420}{600} - \frac{315}{300} = -0.35 \rightarrow \text{Irreversible}$$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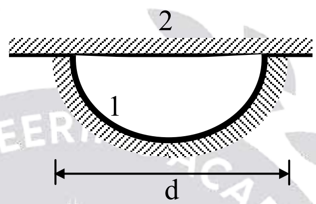
**01(c). Deduce an expression for the shape factor of a hemispherical cavity within itself.**

**(12 M)**

**Sol:** Let, amount radiation energy emitted from surface $A_1$ is $Q_1$, then amount of energy fall on surface '2' is $Q_{12}$. Then shape factor of surface 1 with respective to '2' is denoted as $F_{1-2} = \frac{Q_{1-2}}{Q_1}$

Similarly, $F_{2-1} = \frac{Q_{2-1}}{Q_2}$




$A_2 = \pi R^2$

$A_1 = 2\pi R^2$

$F_{12} + F_{11} = 1$ ……… (1)

$F_{21} + F_{22} = 1$ …….. (2)

$F_{21} = 1$ $\quad (\because F_{22} = 0)$

From (1) and (2)

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

From reciprocity theorem $A_1F_{12} = A_2F_{21}$

$$F_{12} = \frac{A_2}{A_1}F_{21} = \frac{\pi R^2}{2\pi R^2} \times 1$$

$$F_{12} = \frac{1}{2}$$

$$F_{11} = 1 - \frac{1}{2} = 0.5$$

$\therefore$ Thus half of the radiation from hemispherical cavity falls on itself and the remaining half is intercepted by the plane closing surface.

**01(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 M)**

**Sol:** Sometimes actual discharge delivered by the reciprocating pump is more than theoretical discharge. Hence the slip which is defined as difference between theoretical and actual discharge (slip = $Q_{th} - Q_{actual}$) becomes negative.

Negative slip is present if acceleration head in suction pipe is high enough to open the delivery valve before completion of suction stroke.

Given Data:

$H_s = 4$ m, $H_{atm} = 10.3$ m

$L_s = 6.5$ m, $H_v = 2.6$ m

$D_s = 100$ mm, $f = 0.024$

$L = 0.45$ m, $D_p = 150$ mm

For double acting pump the discharge is given by,

$$Q = \frac{ALN}{30} = \frac{\pi}{4} \times D_p^2 L \frac{N}{30}$$

$$Q = \frac{\pi D_p^2 LN}{120} \quad \ldots.(1)$$

The condition for separation is, the pressure head at the beginning of suction stroke becomes equal to vapour pressure head.

i.e., $H_{atm} - H_s - h_{fsa} = H_v$

$$H_{atm} - H_s - \frac{fL_sQ^2}{12.1D_s^5} = H_V$$

$$10.3 - 4 - \frac{0.024 \times 6.5}{12.1 \times 0.1^5} \times \left(\frac{\pi \times 0.15^2 \times 0.45N}{120}\right)^2 = 2.6$$

$\Rightarrow N = 202.1$ rpm

**01(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 M)**

**Sol:** $C_xH_y + a\,(O_2 + 3.76\,N_2) \rightarrow 12\,CO_2 + 0.8\,CO + 3.1\,O_2 + 84.1\,N_2 + b\,H_2O$

**Carbon balance :**

$x = 12 + 0.8 = 12.8$

**Nitrogen balance :**

$7.52\,a = 84.1 \times 2$

$$a = \frac{84.1 \times 2}{7.52} = 22.367$$

**Oxygen balance :**

$2a = 12 \times 2 + 0.8 \times 1 + 3.1 \times 2 + b$

$= 24.8 + 6.2 + b$

$2 \times 22.367 = 31 + b$

$\Rightarrow b = 2 \times 22.367 - 31 = 13.73$

**Hydrogen balance :**

$y = 2b = 2 \times 13.73 = 27.47$

Composition of fuel = $C_x\, H_y$

$= C_{12.8} \times H_{27.47}$

$$\text{Actual air fuel ratio} = \frac{\text{Mass of air}}{\text{Mass of fuel}}$$

$$(AFR)_{actual} = = \frac{22.37 \times 4.76 \times 28.97}{12 \times 12.8 + 27.47 \times 1} = \frac{3084.76}{181.07} = 17.03$$

For stochiometric combustion products are $CO_2$, $N_2$ and $H_2O$ vapor

$C_{12.8}\, H_{27.47} + a\,(O_2 + 3.76\, N_2) \rightarrow b\, CO_2 + c\, H_2O + d\, N_2$

Carbon balance: $b = 12.8$

Hydrogen balance: $2c = 27.47 \Rightarrow c = 13.735$

**Oxygen balance :**

$2a = 2b + c$

$= 2 \times 12.8 + 13.735 = 39.335$

$$a = \frac{39.335}{2} = 19.6675 \approx 19.67$$

**Nitrogen balance:**

$2d = 7.52\, a$

$d = 3.76\, a = 3.76 \times 19.67 = 72.18$

$$(AFR)_{cc} = \frac{\text{Mass of air}}{\text{Mass of fuel}} = \frac{19.67 \times 4.76 \times 28.97}{12.8 \times 12 + 27.47 \times 1} = \frac{2712.44}{181.07} = 14.98$$

$$\text{Percentage of excess air} = \frac{(AFR)_{actual} - (AFR)_{cc}}{(AFR)_{cc}} \times 100$$

$$= \frac{17.03 - 14.98}{14.98} \times 100 = 13.68\ \%$$

**02(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 M)**

**Sol:** No. of cylinders, x = 4

length, $L = 0.1$ m

Diameter = d = 0.1 m

L = d for square engine

Speed = N = 4000 rpm

Compression ratio, $r_k = 7$

As compression ratio is 7 it is a SI engine.

Mean effective pressure ($P_m$) = 8.5 bar,

Relative efficiency = 60 %

bsfc = 0.25 kg/kWhr

Air standard efficiency,

$$\eta_{air} = 1 - \left(\frac{1}{r_k}\right)^{\gamma-1}$$

$$\eta_{air} = 1 - \left(\frac{1}{7}\right)^{1.4-1} = 0.5408$$

$$\eta_{rel} = \frac{\text{Brake thermal efficiency}}{\eta_{air\,std}}$$

$$0.6 = \frac{\text{Brake thermal efficiency}}{0.5408}$$

Brake thermal efficiency = 0.3245

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$$BP\,(kW) = \frac{P_m\,L\,A\,N\,x}{120}$$

$$= \frac{850 \times 0.1 \times \frac{\pi}{4} \times (0.1)^2 \times 4000 \times 4}{120}$$

$= 88.995$ kW

$$\text{Brake Thermal efficiency} = \frac{BP(kW) \times 3600}{\dot{m}_f\,(kg/hr) \times CV(kJ/kg)}$$

$$= \frac{3600}{bsfc \times CV}$$

$$CV = \frac{3600}{\text{Br. Thermal efficiency} \times bsfc}$$

$$= \frac{3600}{0.3245 \times 0.25} = 44375.9630 \text{ kJ/kg}$$

$$bsfc = \frac{\dot{m}_f\,(kg/hr)}{BP(kW)}$$

$\dot{m}_f\,(kg/hr) = bsfc \times BP$

$= 0.25 \times 88.995 = 22.2487$ kg/hr

No. of sparks in one minute per cylinder $= \frac{N}{2} = \frac{4000}{2} = 2000$

No. of sparks per minute per cylinder = No. of thermodynamic cycles per minute per cylinder = 2000 per minute

No. of thermodynamic cycles per second per cylinder

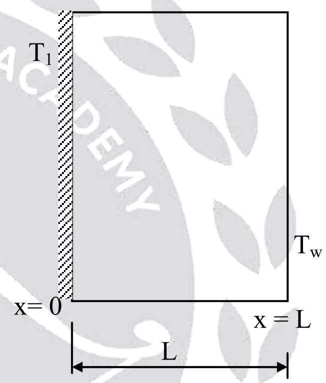
$$= \frac{\text{No. of thermodynamic cycles per cylinder}}{60}$$

$$= \frac{2000}{60} = 33.33$$

**02(b). An infinite slab of thickness "L" (m) is having thermal conductivity "K" (W/mK). It is generating heat at a uniform rate of "$\dot{q}$" (W/m$^3$). 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 M)**

**Sol: Assumptions:**

- One dimensional heat flow
- Steady state
- Uniform internal heat generation
- Material homogenous and isotropic
- Thermal conductivity value is constant
- Surfaces are isothermal




L = Thickness of the plane wall

$\dot{q}$ = internally generated heat

K = Thermal conductivity

**Heat conduction equation:**

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

T = f(x)

$$\frac{d^2T}{dx^2}=-\frac{\dot{q}}{K}$$

$$\int\frac{d^2T}{dx^2}=\int-\frac{\dot{q}}{K}$$

$$\frac{dT}{dx}=-\frac{\dot{q}\,x}{K}+C_1$$

$$x=0,\quad \frac{dT}{dx}=0\,,\ C_1=0$$

$$\frac{dT}{dx} = -\frac{\dot{q}x}{K}$$

$$\int \frac{dT}{dx} = \int -\frac{\dot{q}\,x}{K}$$

$$T = -\frac{\dot{q}\,x^2}{2K} + C_2 \quad \text{---- (1)}$$

$x = L,\ T = T_w$

$$T_w = -\frac{\dot{q}\,L^2}{2K} + C_2$$

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

Put $C_2$ in eq. (1)

$$T = -\frac{\dot{q}\,x^2}{2K} + T_w + \frac{qL^2}{2K}$$

$$T = T_w + \frac{\dot{q}}{2K}\left(L^2 - x^2\right)$$

$$T = T_w + \frac{\dot{q}L^2}{2K}\left(1 - \left(\frac{x}{L}\right)^2\right) \quad \text{----- (2)}$$

For location of maximum temperature,

$$\frac{dT}{dx} = 0$$

$$\frac{dT}{dx} = -\frac{\dot{q}x}{K}$$

$x = 0 \rightarrow$ Location of maximum temperature.

As the surface is insulated, the maximum temperature present at insulated surface hence maximum temperature present at $x = 0$, $T = T_{max}$

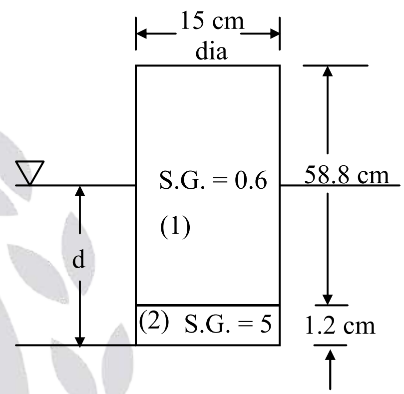
If we put $x = 0$ in eq. (2), we get $T_{max}$

$$T_{max} = T_w + \frac{\dot{q}L^2}{2K}$$

**02(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 M)**

**02.(c)**

**Sol:** Given data:




Cylinder dia = 15 cm = 0.15 m

Total length = 60 cm = 0.6 m

Cylinder has two parts.

Let the part at base be (2) and the rest be (1)

$(S.G)_2 = 5$ & $(S.G)_1 = 0.6$

Weight of the cylinder = Weight of volume of liquid displaced.

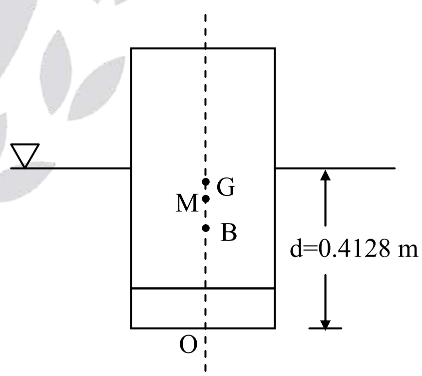
$$0.6\times10^3\times g\times A_1\times0.588+5\times10^3\times g\times A_2\times0.012=10^3\times g\times A\times d$$

where $A_1 = A_2 = A$

d = Depth of immersion

Thus, $d = 0.6 \times 0.588 + 5 \times 0.012$

$= 0.4128$ m




So, $\overline{OB}=\dfrac{d}{2}=0.2064\,\text{m}$

Let $W_2$ be the weight of part (2). So,

$$W_2=5\times10^3\times9.81\times\frac{\pi}{4}\times0.15^2\times0.012$$

$= 10.4014$ N

And the weight of part (1),

$$W_1=0.6\times10^3\times9.81\times\frac{\pi}{4}\times0.15^2\times0.588$$

$= 61.16$ N

For finding out the distance of centre of gravity from the base, we can write

$$W_1 \times \left(\frac{0.588}{2} + 0.012\right) + W_2 \times \frac{0.012}{2} = (W_1 + W_2)\overline{OG}$$

Or, $61.16 \times 0.306 + 10.4014 \times 0.006 = 71.5614 \times \overline{OG}$

$\Rightarrow$ $\overline{OG} = 0.2624$ m

So, $\overline{BG} = \overline{OG} - \overline{OB}$

$= 0.2624 - 0.2064$

$= 0.056$ M

Now, $\overline{BM} = \dfrac{I}{\forall} = \dfrac{\pi}{64} \times \dfrac{0.15^4}{\frac{\pi}{4} \times 0.15^2 \times d}$

$$= \frac{0.15^2}{16 \times 0.4128} = 0.0034 \text{ m}$$

Therefore, metacentric height,

$\overline{GM} = \overline{BM} - \overline{BG}$

$= 0.0034 - 0.056$

$= -0.0526$ m

Since, $\overline{GM}$ is negative, the given solid cylinder CAN NOT float vertically in water.

**03(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**

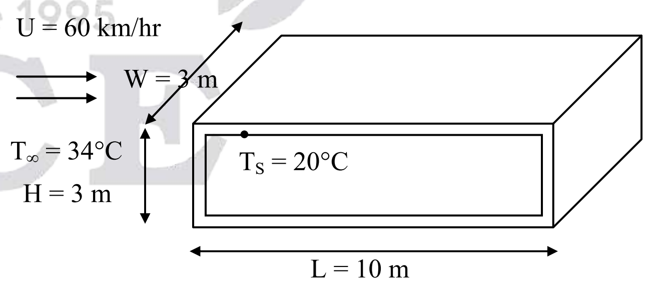
**Pr = 0.708**

**For turbulent flow** $\overline{Nu_L} = 0.036\, Re_L^{0.8}\, Pr^{0.33}$. **(20 M)**

**Sol:** Given data

Assume flow is parallel to 10m long side

$$Re_L = \frac{UL}{\nu} = \frac{\frac{60\times1000}{3600}\times10}{1.569\times10^{-5}} = 10.62\times10^6$$




$Re_L > 5\times10^5$ flow is turbulent

$$\overline{Nu_L} = 0.036\, Re_L^{0.8}\, Pr^{0.33}$$

$$\frac{\overline{h}L}{K} = 0.036\, Re_L^{0.8}\, pr^{0.33}$$

$$\frac{\overline{h}\times10}{0.02624} = 0.036(10.62\times10^6)^{0.8}(0.708)^{0.33}$$

$$\overline{h} = 35.217\ W/m^2K$$

**Heat Gained from roof and side walls**

In the question it is not mentioned, how many side we need to consider and Nu equation given for flow along the length side therefore assume that heat transfer from the front and back of the truck Neglected

$$A_s = 2[LW + HL]$$

$$A_s = 2[10\times 3 + 3\times 10] = 120\ m^2$$

$$Q = hA_s\ \Delta T$$

$$Q = hA_s(T_\infty - T_s)$$

$$Q = 35.217\times 120\times(34-20)$$

$$Q = 59.16\ kW$$

Capacity of air conditioning unit in TR

1 tonne of refrigeration = 3.517 kW

$$Q = 59.16\ kW = \frac{59.16}{3.517} = 16.82\ TR$$

Q = 16.82 TR

**Power required**

$$COP = \frac{\text{Desired output}}{\text{work input}}$$

$$4 = \frac{\text{cooling capacity}}{\text{Power required}}$$

$$4 = \frac{59.16}{\text{Power required}}$$

Power required = 14.79 kW

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**03(b)(i).**

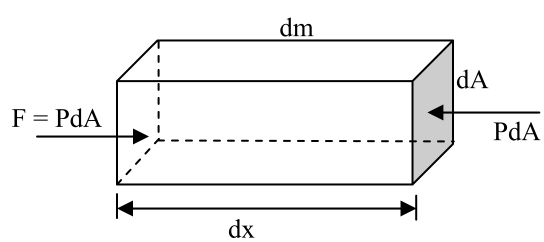
**(I) For a given thermodynamic system while considering control volume approach, what is the significance of flow work? Is flow work a path function or point function? (4 M)**

**Sol: Flow work:**

- The work involved in crossing the fluid element across the control surface, either to enter or leave the control volume is known as flow work.
- Flow work is the work done by a fluid to move against pressure.
- In simple words, flow work is the work required to maintain the flow through a control volume.
- Flow work is also known as flow energy, convected energy or transport energy.
- In fluid mechanics flow work is also known as pressure energy.

**Expression of flow work :**

Volume (dV) = dA dx




Specific volume (ν) = $\dfrac{dV}{dm}$

Flow work ($W_f$) = $\vec{F}\ \vec{dx}$

$W_f = F \times dx \times \cos\theta$

$W_f = F \times dx \times \cos 0°$

$F = PdA$

$W_f = PdA \times dx \times 1$

$W_f = PdA\ dx$

$W_f = PdV$

Specific flow work ($w_f$)

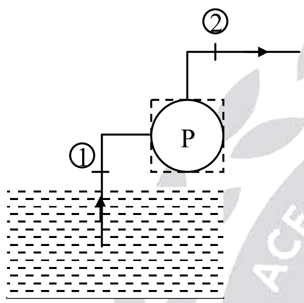
$$w_f = \frac{\text{Flow work}\left(W_f\right)}{\text{Mass (dm)}}$$

**03(b)(i).**

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

**(6 M)**

**Sol: Pump:**




Liquid ($H_2O$)

S.F.E.E:

$$\frac{V_1^2}{2} + gz_1 + h_1 + q = \frac{V_2^2}{2} + gz_2 + h_2 + W_{C.V}$$

where: $[W_{C.V} = W_{pump}]$

$$-W_P = h_1 - h_2$$

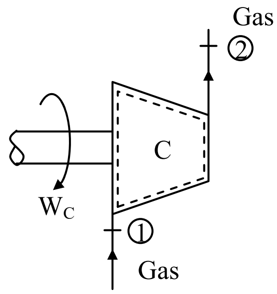
Generally:

(i) d (K.E) = 0

(ii) d(P.E) = 0

(iii) q = 0

**Compressor:**




S.F.E.E:

$$\frac{V_1^2}{2} + gz_1 + h_1 + q = \frac{V_2^2}{2} + gz_2 + h_2 + W_{C.V}$$

where:

$$[W_{C.V} = W_{compressor}]$$

$$-W_C = h_1 - h_2$$

$$W_C = h_2 - h_1$$

Generally:

(i) d (K.E) = 0

(ii) d(P.E) = 0

(iii) q = 0

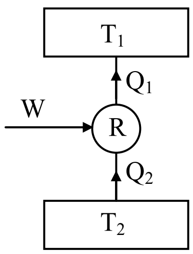
**Inferences:**

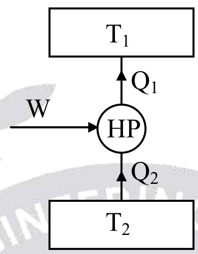
- Pump and compressors are steady flow devices since fluid enters and leaves continuously.
- Pumps are used for incompressible fluids like water.
- Compressors are used for compressible fluids like gases.
- The work input equation is same for compressor and pump.

**03(b)(ii).**

**(I). Show that COP of a heat pump is greater than COP of a refrigerator. (4 M)**

**Sol:**




$[W_R = Q_1 - Q_2]$

$\left[COP_R = \frac{Q_2}{W}\right]$

$\left[COP_R = \frac{Q_2}{Q_1 - Q_2}\right]$

$[W_{H.P} = Q_1 - Q_2]$

$\left[COP_{H.P} = \frac{Q_1}{W}\right]$

$\left[COP_{H.P} = \frac{Q_1}{Q_1 - Q_2}\right]$

$$COP_{H.P} - COP_R = \frac{Q_1}{Q_1 - Q_2} - \frac{Q_2}{Q_1 - Q_2}$$

$$COP_{H.P} - COP_R = 1$$

$$COP_{H.P} = 1 + COP_R$$

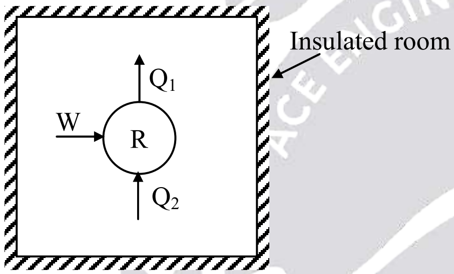
$$\therefore \quad COP_{H.P} > COP_R$$

**03(b)(ii).**

**(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?**

**(6 M)**

**Sol:**




First law: Refrigerator operates on cycle

$$\oint Q = \oint W$$

$$Q_2 - Q_1 = -W$$

$$W = Q_1 - Q_2$$

where:

$Q_1$ = Heat rejection

$Q_2$ = Heat absorption

→ When refrigerator is operated with open door, the room temperature goes on increasing since heat rejection is greater than heat absorption.

→ From first law $[Q_1 > Q_2]$

**03(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$^4$**

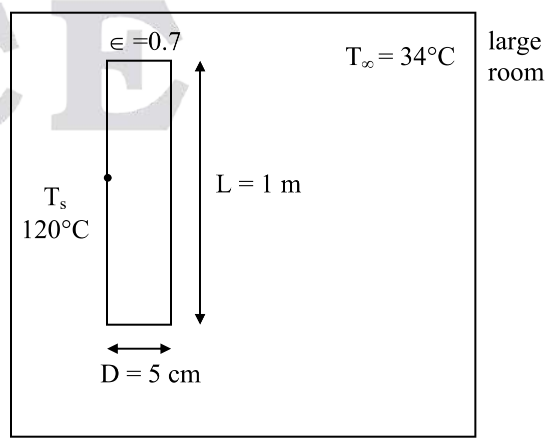
**Use correlation,** $\overline{Nu_L} = 0.1[Gr_L\ Pr]^{0.33}$

**Neglect heat loss from the ends.** **(20 M)**

**Sol:** Given data

$T_s = 120°C = 393\text{ K}$ ; $T_\infty = 34°C = 307\text{ K}$

$$T_{avg} = \frac{Ts + T\infty}{2} = 350K$$




β = Coefficient of volume expansion

$$\beta = \frac{1}{T_{avg}} = \frac{1}{350}$$

$$Gr_L = \frac{g\beta\Delta TL^3}{\nu^2}$$

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$$Gr_L = \frac{9.81 \times \frac{1}{350} \times (120-34)(1)^3}{\left(2.076 \times 10^{-5}\right)^2}$$

$$Gr_L = 5.6 \times 10^9$$

$$Gr_L \,.\, Pr = 5.6 \times 10^9 \times 0.697$$

$$Gr_L \; Pr = 3.9 \times 10^9$$

$$\overline{Nu} = 0.1(Gr_L\, pr)^{0.33}$$

$$\overline{Nu} = \frac{\bar{h}L}{k} = 0.1\left(3.9 \times 10^9\right)^{0.33}$$

$$\bar{h} = \frac{0.1 \times 0.03}{1}\left(3.9 \times 10^9\right)^{0.33}$$

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

$$Q_{conv} = \bar{h} A_s \Delta T$$

$$Q_{conv} = \bar{h} \pi D L \Delta T$$

$$Q_{conv} = 4.386 \times \pi \times 5 \times 10^{-2} \times 1 \times (120-34)$$

$$Q_{conv} = 59.25 W$$

$$Q_{rad} = \in A\sigma\left(T_s{}^4 - T_\infty{}^4\right)$$

$$Q_{rad} = 0.7 \times \pi \times 5 \times 10^{-2} \times 1 \times 5.67 \times 10^{-8}\left(393^4 - 307^4\right)$$

$$Q_{rad} = 93.34 W$$

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

$$Q_{total} = 152.59 \text{ W}$$

$$\%\text{convection} = \frac{Q_{conv}}{Q_{total}} = \frac{59.25}{152.59} = 38.82\%$$

$$\%\text{ radiation} = \frac{Q_{rad}}{Q_{total}} = \frac{93.34}{152.59} = 61.11\%$$

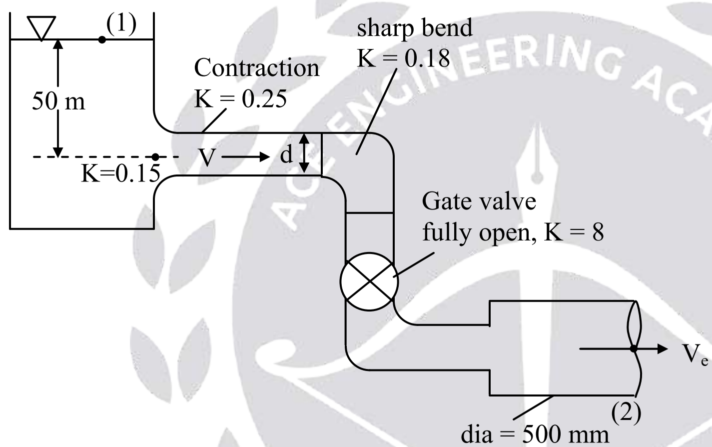
Radiation heat loss contribution is 61%. So radiation can not be neglected.

Hence, it is not correct to neglect radiation

**04(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 M)**

**Sol:**




Let the velocity of water flowing through the contraction, bend and gate valve be V and that through the pipe with sudden expansion (assumed to take place at the end of pipe) be $V_e$.

Applying energy equation between sections (1) and (2), we can write:

$$\frac{P_1}{\gamma_w}+\frac{V_1^2}{2g}+Z_1=\frac{P_2}{\gamma_w}+\frac{V_e^2}{2g}+Z_2+K_{ent}\frac{V^2}{2g}+K_{cont}\frac{V^2}{2g}+K_{bend}\frac{V^2}{2g}+K_{valve}\frac{V^2}{2g}+\frac{(V-V_e)^2}{2g}+\frac{fLV^2}{2gd}$$

where, $P_1 = P_2 = P_{atm}$

$V_1 = 0$; $Z_1 = 50$ m; $Z_2 = 0$,

$K_{ent} = 0.15$; $K_{cont} = 0.25$; $K_{bend} = 0.18$, $K_{valve} = 8$;

$f = 0.048$ ; $L = 500$ m; $d = 0.4$ m

Substituting all the given values, we have,

$$0+0+50=0+\frac{V_e^2}{2g}+0+(0.15+0.25+0.18+8)\frac{V^2}{2g}+\frac{V^2}{2g}\left[1-\frac{V_e}{V}\right]^2+\frac{0.048\times 500\times V^2}{2g\times 0.4} \quad \text{..........} (1)$$

From equation of continuity,

$$\frac{\pi}{4}\times 0.4^2\times V=\frac{\pi}{4}\times 0.5^2\times V_e$$

$$V_e=\left(\frac{0.4}{0.5}\right)^2\times V \;\; = 0.64\ V$$

or, $\frac{V_e}{V}=0.64$

Thus, equation (1) becomes,

$$50=(0.64)^2\frac{V^2}{2g}+8.58\frac{V^2}{2g}+\frac{V^2}{2g}(1-0.64)^2+\frac{V^2}{2g}$$

$$=\frac{V^2}{2g}\left[0.64^2+8.58+0.36^2+60\right]$$

$$=69.1192\times\frac{V^2}{2g}$$

$$V^2=\frac{50\times 2\times 9.81}{69.1192}$$

$\Rightarrow$ V = 3.7673 m/s

Thus, the discharge is

$$Q=\frac{\pi}{4}\times 0.4^2\times 3.7673$$

Q = 0.4734 m$^3$/s

**04(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,** $C=\dfrac{C_{min}}{C_{max}}$ **.** **(20 M)**

**Sol:** Given data,

$T_{h1}$ = inlet temperature of hot fluid

$T_{c1}$ = inlet temperature of cold fluid

$T_{h2}$ = outlet temperature of hot fluid

$T_{c2}$ = outlet temp of cold fluid

Exit temp of fluid ($T_{h2}$ = ? $T_{c2}$ = ?),

For counter flow arrangement

**Procedure:**

$$\dot{m}_h c_h (T_{h1}-T_{h2}) = \dot{m}_c c_c (T_{c_2}-T_{c_1})$$

$$\dot{m}_h c_h (120-80) = \dot{m}_c c_c (50-20)$$

$$40\ \dot{m}_h c_h = 30\ \dot{m}_c\ c_c$$

$$\frac{4}{3}\dot{m}_h c_h = \dot{m}_c c_c$$

$$\dot{m}_h c_h < \dot{m}_c c_c$$

$$\dot{m}_h c_h = c_{min}$$

$$\dot{m}_c c_c = c_{max}$$

Heat capacity ratio, $(C) = \dfrac{C_{min}}{C_{max}}$

$$C = \frac{C_{min}}{C_{max}} = \frac{3}{4} = 0.75$$

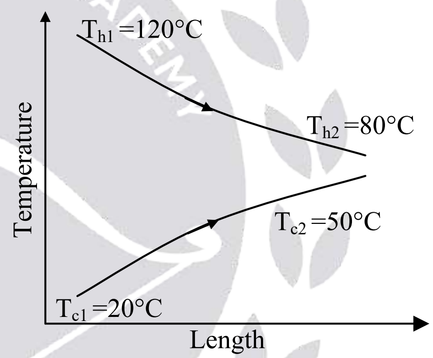
C = 0.75

$$NTU = \frac{UA}{C_{min}}$$

**Parallel flow :**

$\in$ = effectiveness

$$\in = \frac{Q_{act}}{Q_{max}}$$




$$\in = \frac{\dot{m}_h c_h (T_{h1} - T_{h2})}{C_{min}(T_{h1} - T_{c_1})}$$

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

$$\in = \frac{1 - e^{-NTU(1+C)}}{1 + C}$$

$$\in (1 + C) = 1 - e^{-NTU(1+C)}$$

$$e^{-NTU(1+C)} = 1 - \in (1 + C)$$

$$-NTU(1 + C) = \ell n[1 - \in (1 + C)]$$

$$-NTU = \frac{\ell n[1 - \in (1 + C)]}{1 + C}$$

$$-NTU = \frac{\ell n[1 - \in (1 + C)]}{1 + C}$$

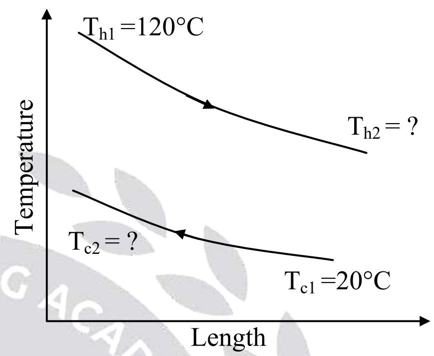
$$-NTU = \ell n\left[\frac{1 - 0.4(1 + 0.75)}{1 + 0.75}\right]$$

NTU = 0.687

**Counter flow :**

Assuming the NTU value is same in parallel flow & counter flow

$$\in = \frac{1-e^{-NTU(1-C)}}{1-ce^{-NTU(1-C)}} = \frac{1-e^{-0.687(1-0.75)}}{1-0.75e^{-0.687(1-0.75)}}$$




$$\in = \frac{1-0.842}{1-(0.75\times 0.842)} = 0.4287$$

$$\in = 0.4287$$

$$\in = \frac{Q_{act}}{Q_{max}}$$

$$\in = \frac{\dot{m}_h c_h (T_{h1} - T_{h2})}{C_{min}(T_{h1} - T_{c_1})}$$

$$0.4287 = \frac{120 - Th_2}{120 - 20}$$

$$T_{h2} = 77.13°C$$

$$\in = \frac{Q_{act}}{Q_{max}} = \frac{\dot{m}_c c_c (T_{c_2} - T_{c_1})}{C_{min}(T_{h1} - T_{C_1})}$$

$$\in = \frac{C_{max}(T_{c_2} - T_{c_1})}{C_{min}(T_{h1} - T_{c_1})}$$

$$0.4287 = \frac{T_{c_2} - 20}{0.75(120-20)}$$

$$T_{c_2} = 52.15°C$$

**04(c). Explain variation is 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 M)**

**Sol:** Variation in specific heat of gases and its influence on engines.

All gases except monatomic gases show an increase in specific heat with temperature.

$$c_p = a_1 + k_1T$$

$$c_v = b_1 + k_1T$$

Range of 300 K – 1500 K

$a_1, b_1, k_1$ are constants.

$$R = c_p - c_v = a_1 - b_1$$

Above 1500 K

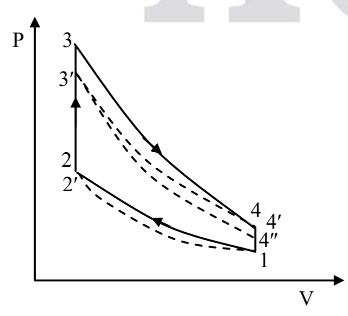
$$c_p = a_1 + k_1T + k_2T^2$$

$$c_v = b_1 + k_1T + k_2T^2$$

As temperature is raised larger fractions of heat would be required to produce motion of atoms within the molecule. As temperature is the result of motion of molecules as a whole energy which goes into moving the atoms does not contribute to proportional temperature rise.

More heat is required to raise the temperature of unit mass through one degree at higher levels.

$\frac{c_p}{c_v} = \gamma$ value decreases with rise in temperature, $(c_p - c_v)$ is constant with rise in temperature.



1-2-3-4 cycle with constant specific heats.

1-2′-3′-4′ cycle with variable specific heats.

1-2-3′-4″ cycle with constant specific heats from point 3′.

1-2: with constant specific heats, $T_2 = T_1(r_k)^{\gamma-1}$

1-2′: with variable specific heats, $T_2' = T_1(r_k)^{\gamma-1}$ with variable specific heats.

As k < r, $T_2' < T_2$

With constant specific heats.

$$T_3 = T_2 + \frac{Q}{c_v} \quad \text{(Q is heat supplied)}$$

With variable specific heats.

$$T_3' = T_2' + \frac{Q}{c_{v1}} \quad (c_{v1} > c_v)$$

Hence $T_3' < T_3$ as $T_2' < T_2$ and $c_{v1} > c_v$ (Q is heat supplied)

3′ - 4″: With constant specific heats after heat supply

$$T_4'' = \frac{T_3'}{(r_k)^{\gamma-1}}$$

3′ - 4′: With variable specific heats after heat supply

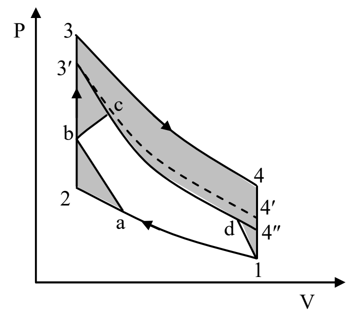
$$T_4' = \frac{T_3'}{(r_k)^{\gamma-1}} \quad (k < \gamma)$$

Hence, $T_4' > T_4''$

Hence due to variation of

- Specific heat temperature and pressure at end of compression is low.
- Temperature and pressure at end of heat supply is less.
- Temperature and pressure at beginning of exhaust increases
- Heat rejection also increases
- Thermal efficiency decreases

**Fuel air ratio vs Actual cycle :**




1-2-3-4 → Fuel air cycle

1-a-b-c-d-4″ → Actual cycle

1-2-3′-4′ → Isentropic expansion through C

Heat addition is assumed to be instantaneous from 2-3, but it happens over a period of time. Crank shaft usually turns 30° – 40° between initiation of spark and end of combustion. There will be a time loss during this period and is called time loss factor.

Shaded portion between a–2–3′–c and a–b–c represents time loss factor which is 6%.

<u>Heat loss factor,</u>

2 – 3 represents instantantaneous combustion without time loss

3 – 4 expansion fuel air cycle ideal expansion

c - 4″ is actual expansion

Shaded portion 3′ – 3 – 4 – 4″ – 3′ represents heat loss during expansion. From cylinder gases to cylinder walls and cylinder head into water jacket or cooling fins, heat loss is 12 %.

Exhaust blow down loss d–4″–1.

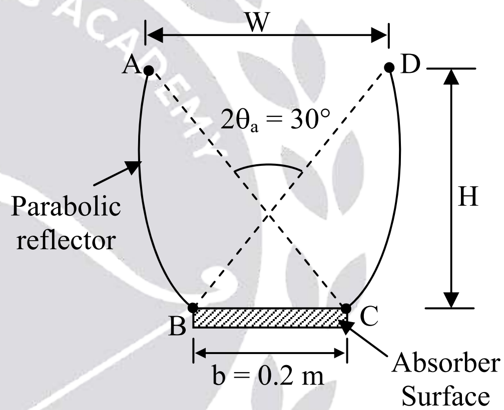
Shaded area represents exhaust blow down loss.

- Cylinder pressure at end of exhaust stroke is 7 bar depending on compression ratio employed.
- If exhaust valve is opened at BDC, the piston has to do work against high cylinder pressures during early part of exhaust stroke.
- If exhaust valve is opened too early a part of expansion stroke is lost.
- Best compromise is to open exhaust valve 40° to 70° before BDC, thereby reducing cylinder pressure to 3.5 bar before exhaust stroke begins.
- Exhaust blow down loss is around 2 %.

## SECTION – B

**05(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 M)**

**Sol:** Consider a compound parabolic collector (CPC) as shown :




$L = 2$ m, $2\theta_a = 30°$, $b = 0.2$ m

(i) Concentration ratio :

$$C = \frac{1}{\sin\theta_a} = \frac{1}{\sin(15°)}$$

$C = 3.86$

(ii) $W = C \times b$

$W = 3.86 \times 0.2 = 0.7727$ m

Aperture width, W = 77.27 cm

(iii) Height (H) :

$$\frac{H}{W} = \left(\frac{1+C}{2}\right) \times \cos\theta_a$$

$$\frac{H}{77.27} = \left(\frac{1+3.86}{2}\right) \times \cos(15°)$$

$H = 181.36$ cm

(iv) Surface area of concentrator :

$$\frac{A_{con}}{WL} = 1 + C$$

$A_{conc} = 0.7727 \times 2 \times (1 + 3.86)$

$A_{conc} = 7.5\ m^2$

**05(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?**

**(6 M)**

**Sol:** *Multi valve engines have mainly 3 advantages:*

1. It increases the coverage of valves over the combustion chamber, allowing faster breathing thus enhance power at high revolutions.
2. It allows the spark plug to be positioned in the centre of combustion chamber, enabling quicker flame propagation, more even and more efficient burning.
3. Using more but smaller valves instead of two large valves means lower mass for each valve. This prevents the valves "float" from its designed position at very high revolutions, thus enabling the engine to revolve higher and make more power as a result. Also, thus allows the use of lighter springs and reduces the force on linkages. Lighter values can be opened and closed faster.

Due to the above advantages, in modern engines multiple intake and multiple exhaust valves are used.

The exhaust valves are generally thicker when the stem meets the valve. Some high end exhaust valves are sodium filled to help transfer heat from the valve upto the stem to be transferred into the head through the valve guide. Some exhaust valves have vanes or blades around the stem just above the seat. The intake valves are usually larger and are thinner near the stem and valve. Some are necked down to further enhance airflow between the valve guide touching the valve and the valve flaring out to become large to seal the port at the valve seat.

Inlet valves are larger than exhaust valves.

**05(b).**

**(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 M)**

**Sol:** There are two major characteristics of piston shapes:

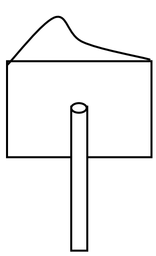
→ Profile

→ Ovality

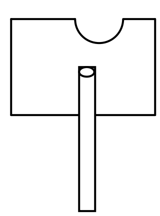
These characteristics determine, how the piston will wear over time and how well the piston can perform.

**Profile:** If we roll a piston across a flat surface, we will notice that it does not roll in a straight line. As aluminum conducts so much heat, pistons are designed with a taper – the top of the piston, near the crown, is a smaller diameter than the bottom of the piston, near the skirt. The skirt of the piston actually designed with what is called a barrel shape. This is because temperatures near the dome of the piston vary from the temperatures at the skirt of the piston, resulting in different levels of expansion. The tapered shape allows the piston to expand as heat is applied, so the piston does not bind in the cylinder bore. The design challenge then becomes calculating the degree of taper.

To identify a piston of two stroke engine from the piston of a four stroke engine through visual inspection with some capacity we need to check the top surface of piston. In two stroke engine the top of the piston usually has a projection to deflect the fresh air to sweep upto the top of the cylinder before flowing to the exhaust ports. While in four stroke engine the top of the piston has projection as shown in the figure. The figure below shows the difference:



*Piston of two stroke engine*

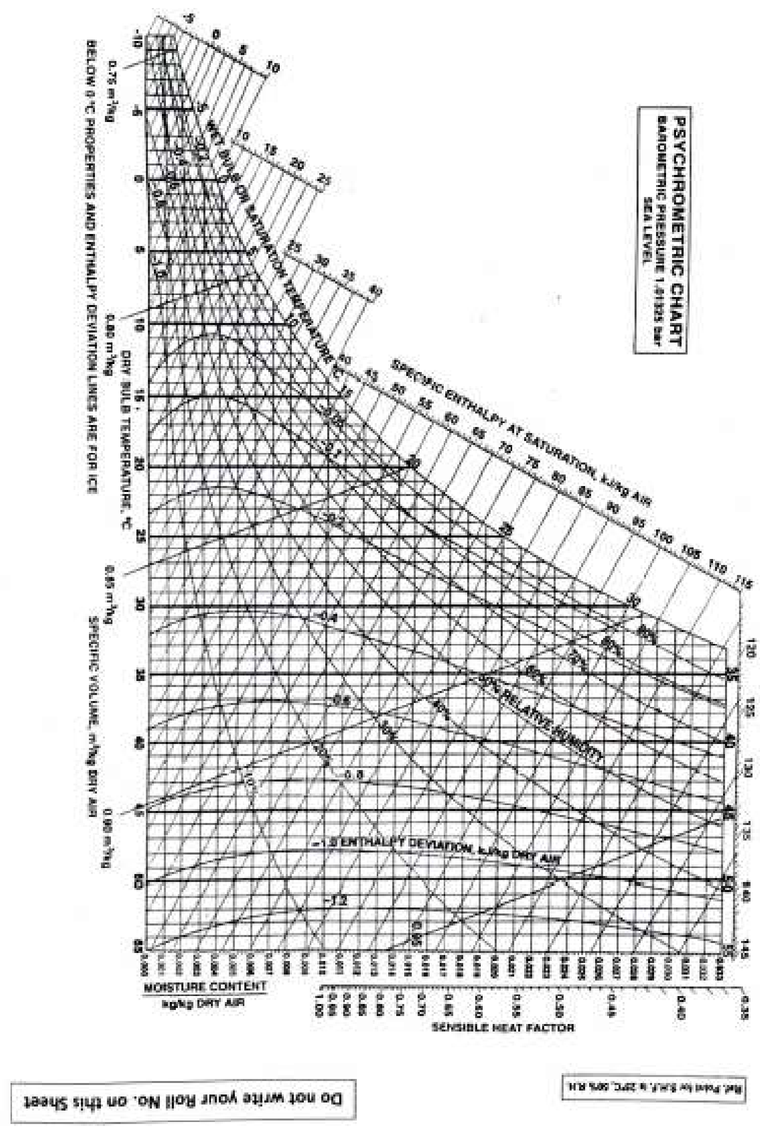


*Piston of four stroke engine*

**05(c). In an air-conditioning unit, 10 m³/min of air from atmospheric condition of DBT 40°C and Relative humidity 70% is adiabatically mixed with 90 m³/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 skeleton Psychrometric chart.**

**[Psychrometric chart is attached.]** **(12 M)**



**Sol: Given:**

(1) Atmospheric condition: $t_1 = 40°C$, $\phi_1 = 70$ %, $\dot{V}_1 = 10^3 / \text{min}$

(2) Recirculation air: $t_2 = 25°$, $wbt_2 = 20°C$, $\dot{V}_2 = 90^3\ m^3/\text{min}$

**Procedure:**

(i) Mark point (1) and point (2) on pyschrometric chart

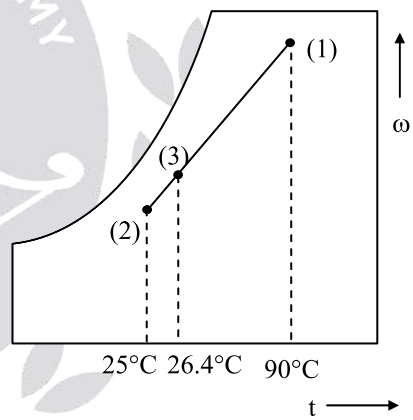
(ii) Join this points by line

(iii) Find value of temperature of mixture by using formula

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

(iv) Mark point (3) on chart and find properties of mixture.

**From Chart:**




Point (1) : $h_1 = 124$ kJ/kg, $\omega_1 = 0.0325$ kg/kg d.a

$v_1 = 0.93\ m^3/kg$

Point (2) : $h_2 = 57.5$ kJ/kg, $\omega_2 = 0.0125$ kg/kg d.a

$v_2 = 0.86\ m^3/kg$

Now, $\dot{m}_1 = \dfrac{\dot{V}_1}{v_1} = \dfrac{10/60}{0.93} = 0.1792\ \text{kg/s}$

$\dot{m}_2 = \dfrac{\dot{V}_2}{v_2} = \dfrac{90/60}{0.86} = 1.7441\ \text{kg/s}$

Now, $t_3 = \dfrac{\dot{m}_1 t_1 + \dot{m}_2 t_2}{\dot{m}_1 + \dot{m}_2} = \dfrac{0.1792 \times 40 + 1.7441 \times 25}{0.1792 + 1.7441} = 26.3975$ °C

$t_3 = 26.4°C$

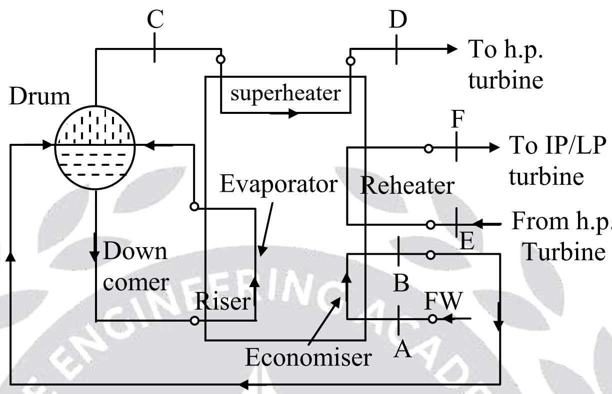
Properties of mixture,

$t_3 = 26.4°C$, $h_3 = 63.7$ kJ/kg d.a

$wbt_3 = 22.8°C$, $\omega_3 = 0.0143$ kg/kg d.a, $\phi_3 = 66$ %

**05(d). With the help of diagram, show the placing of evaporator, superheater, reheater and economizer in the boiler. Also justify the placement at specific location. (12 M)**

**Sol:**




**Superheater and reheater :**

- It is a heat exchanger in which products of heat of combustion are utilized to dry the wet steam and to make it superheated by increasing its temperature. During superheating of the steam, pressure remains constant, and its volume and temperature increase. A super-heater consists of set of small-diameter U tubes in which steam flows and takes up the heat from hot flue gases.
- The smaller diameter tubes have lower pressure stresses and can withstand better. The tube material should be carefully selected, because the tubes are subjected to high temperature, pressure and thermal stresses. The maximum steam temperature at the superheater exit is about $540°C$, the superheaters and re-heaters, which are operating at this temperature, are made of special high-strength alloy steels, which have high strength and corrosion resistance.

**Economiser :**

- An economizer is a heat exchanger used for heating the feed water before it enters the boilers. The economizer recovers some of waste heat of hot flue gases going to the chimney thus it helps in improving the boiler efficiency. It is placed in the path of flue gases at the rear end of the boiler just before the air preheater.

**Evaporator :**

- Evaporator is placed before steam drum.

**05(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 M)**

**Sol:** Effect of changes in various parameters on COP

$$COP = \frac{RE}{W_{in}}$$

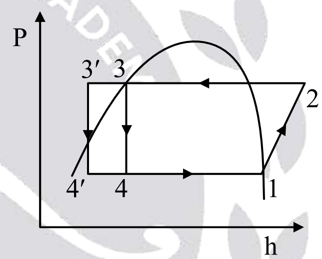
**(i) Subcooling of the liquid in condenser:**

The effect of subcooling is as shown,

$RE = h_1 - h_4$

$W_{in} = h_2 - h_1$

$RE' = h_1 - h_4'$




Cycle 1-2-3-4: ideal vapour compustion cycle

1–2–3–4: Vapour compression with subcooling

Due to subcooling of liquid in condenser, refrigeration effect will get increased, there will not be any change in work input. So COP of the vapour compression cycle will increase.

**(ii) Decrease of evaporator temperature :**

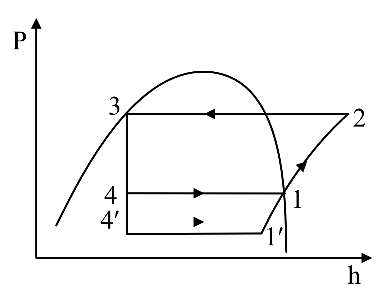
The effect of decrease in evaporator temperature is as shown :

$RE = h_1 - h_4$

$W_{in} = h_2 - h_1$

$RE' = h_1' - h_4'$

$W_{in}' = h_2 - h_1'$



1–2–3–4: ideal vapour compression cycle.

1′–2–3–4′: vapour compression with decrease in evaporator temperature.

Due to reduction in evaporator temperature the refrigeration effect will get reduced and the work input will get increased. So the COP of the cycle will get reduced.

**(iii) Wet compression:**

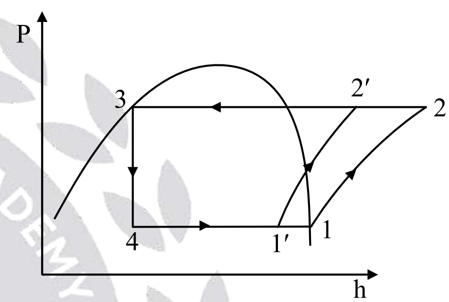
The effect of wet compression will be as shown.

$RE = h_1 - h_4$

$W_{in} = h_2 - h_1$

$RE' = h_1' - h_4$

$W_{in}' = h_2' - h_1'$




1–2–3–4 : Ideal vapour compression cycle

1′–2′–3–4 : Vapour compression with wet compression,

Due to wet compression, the refrigeration effect will get reduced. Due to wet refrigerant, liquid particles wash out lubricant and it increases wear. The COP of cycle decreases.

**06(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 $\beta'$ = (180 – blade angle)**

**A double overhang Pelton wheel unit is coupled to a generator producing 30,000 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 and specific 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 $\rho_{water}$ = 1000 kg/m$^3$.** **(20 M)**

**Sol: Work done and efficiencies of Pelton wheel:**

**a) Velocity of whirl ($V_w$):**

The components of absolute velocities

$V_{w1} = V_1$

$V_{w2} = V_{r2}\cos\phi° - U_2$

are the velocity of whirls at inlet and outlet. These are computed for force exerted and power developed.

**b) Force exerted by the fluid on the buckets:**

$F = \dot{m}[V_{w1} \pm V_{w2}]$

$= \rho Q[V_{w1} \pm V_{w2}]$

$\phi° < 90°$,

$F = \rho Q[V_{w1} - (-V_{w2})]$

$= \rho Q[V_{w1} - V_{r2}\text{Cos}\phi° - U]$

$= \rho Q[(V_{w1} - U) + V_{r2}\text{Cos}\phi°]$

$= \rho Q\,[V_{r1} - K.V_{r1}\text{Cos}\phi°]$

$= \rho Q\ V_{r1}[1 + K\text{Cos}\phi°]$

$\therefore F = \rho Q\,[V_1 - U]\,[1 + K\ \text{Cos}\phi°]$

**c) Work done/Sec (or) Power developed by buckets (runner): ( $\phi° < 90°$)**

**i)** Power developed by the wheel, (P)

$= \rho Q\,(V_{w1} + V_{w2})U$

$= \rho Q\,[V_1 - U]\,[1 + K\cos\phi°]\,U$

**ii)** Power developed per unit mass/sec

$= (V_1 - U)\,(1 + K\text{Cos}\phi°)U$

**iii)** Power developed/unit weight of water / sec

$$= \frac{(V_1 - U)(1 + K\cos\phi^{o})}{g}U$$

**iv)** Hydraulic Efficiency (Pelton Wheel)

$$= \frac{\text{Shaft power}}{\text{kinetic energy supplied / sec}}$$

Kinetic Energy/unit mass $= \dfrac{V_1^2}{2}$

$\therefore$ Hydraulic Efficiency $= \dfrac{2(V_1 - U)(1 + K\cos\phi^{O})U}{V_1^2}$

**v)** Conditions for Maximum Efficiency:

(Where $\phi°$= outlet blade angle)

$$K = 1,\ U = \frac{V_1}{2}$$

$\therefore$ Maximum Hydraulic Efficiency, $\eta_h = \dfrac{(1 + \cos\phi^{o})}{2}$

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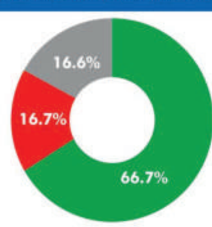
### TEST WISE STATISTICS:

**TIME**

Test Duration : **3 Hours**
Your Time : **2Hrs 45Min 23Sec**

**ANSWERS EVALUTION**




Right | Wrong | Un Attempted

**RESULTS**

OBTAINED MARKS

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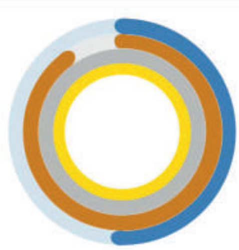
**AIR 380**

* Total Attempted Students ( till now ) : 11250

### QUESTION WISE STATISTICS:

Time Usage


Performance



Top - 10 | Top - 100
Top - 50 | Avg. Performance of all

Difficulty Level


* Difficulty level is a relative value calculated based on overall students performance.

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**vi)** Hydraulic Efficiency with respect to unit weight of water passing through the wheel

$$= \frac{2(V_1 - U)(1 + K\cos\phi)}{gH}$$

$$= \frac{2(V_1 - U)(1 + K\cos\phi)u}{gH}$$

$$= \frac{2\left(V_1 - \frac{V_1}{2}\right)(1 + K\cos\phi)\frac{V_1}{2}}{gH}$$

$$= \frac{2 \times \frac{V_1^2}{4}(1 + K\cos\phi)}{gH} = C_v{}^2\frac{(1 + K\cos\phi)}{2}$$

Given data,

$P = 30000$ kW, $H = 300$ m,

$d = ?$, $N = ?$, $N_s = ?$

$\eta_{generator} = 0.93$, $\eta_{overall} = 0.85$

$C_v = 0.97$ , $K_u = 0.46$

$f_{generator} = 50$ Hz

Pair of poles = 16

$$\frac{D}{d} = 12$$

$\rho_{water} = 1000$ kg/m$^3$

(i) Discharge,

Power generated = $P_{gen} = \eta_g \times \eta_{overall} \times w \times Q \times H$

$$\eta_g \times \eta_{overall} = \frac{P_{gen}}{wQH}$$

$$0.93 \times 0.85 = \frac{(3000)}{9810 \times Q \times 300}$$

$$Q = 12.895 \text{ m}^3/\text{sec}$$

As Pelton wheel is double overhang,

Number of jets = n = 2

$$Q = AV \times n$$

$$Q = \frac{\pi}{4} \times d^2 \times C_v \sqrt{2gH} \times n$$

$$12.8952 = \frac{\pi}{4} \times d^2 \times 0.97 \times \sqrt{2 \times 9.81 \times 300} \times 2$$

$\Rightarrow$ d = 0.3321 m

(ii) $\frac{D}{d} = 12$

D = 12 × 0.3321 = 3.9856 m

(iii) Synchronous speed,

$$N = \frac{f \times 60}{p} = \frac{50 \times 60}{16} = 187.5 \text{ rpm}$$

(iv) $N_s = \frac{N\sqrt{P}}{H^{5/4}}$

$$K_u = \frac{u}{\sqrt{2gH}}$$

$$u = K_u \sqrt{2gH} = 0.46 \times \sqrt{2 \times 9.81 \times 300} = 35.2913 \text{ m/s}$$

$$u = \frac{\pi DN}{60} = 35.2913$$

$$N = \frac{60 \times 35.2913}{\pi \times 3.9856} = 169.11 \text{ rpm}$$

P = mechanical power per wheel $= \frac{P_{gen}}{\eta_g \times 2} = \frac{30000}{0.93 \times 2} = 16129.032$ kW

$$N_s = \frac{169.11 \times \sqrt{16129.032}}{(300)^{5/4}} = 17.2017$$

**06(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. (8 M)**

**Sol: The sources of air in the condenser are due to the following :**

- Leakage through packing glands and joints.
- Leakage through condenser accessories, such as, atmospheric relief valve, etc.,
- Air associated with exhaust steam may also liberate at low pressure.
- In the jet condenser, the dissolved air in the cooling water liberates at low pressures.

**The effects of presence of air in a condenser are :**

- The pressure in the condenser is increased, this reduces the work done by the engine or turbine.
- Partial pressure of steam and temperature are reduced. The steam tables tell us that at lower pressure, the latent heat of steam is more. In order to remove this greater quantity of heat, more cooling water has to be supplied and, thus undercooling of the condensate is likely to be more severe resulting in lower overall efficiency.
- The presence of air reduces the rate of condensation of steam since the abstraction of heat by the circulating cooling water is partly from the steam and partly from the air.
- The rate of heat transfer from the vapor is reduced due to poor thermal conductivity of air. Thus, the surface of the tubes has to be increased for a given condenser duty.
- An air extraction pump is needed to remove air still some quantity of steam escapes with the air even after shielding to the air extraction section. This reduces the amount of condensate. Moreover, the condensate is undercooled, with the result that more heat has to be supplied to the feed water in the boiler.

**06(b)(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**

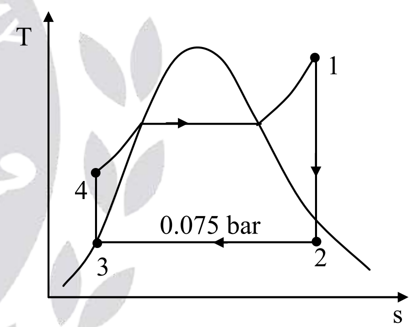
**$h_{fg}$, specific enthalpy of conversion from saturated liquid to dry vapour = 2418.6 kJ/kg**

**$v_f$, 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/kg K, R = 0.287 kJ/kg K (12 M)**

**Sol:** $\dot{m}_s = 650\ \text{T/hr} = \dfrac{650\times10^3}{3600} = 180.55\ \text{kg/s}$




$x_2 = 0.88$,

$h_2 = h_3 + x_2 \times h_{fg} = 146.7 + 0.88 \times 2418.6 = 2275.1\ \text{kJ/kg}$

$v_2 = v_3 + x_2\ v_{fg} = 0.001006 + 0.88 \times 25.22 = 22.194\ \text{m}^3/\text{kg}$

By energy balance :

$$\dot{m}_s \times (h_2 - h_3) = \dot{m}_w \times c_{pw} \times \Delta T_w$$

$$\dot{m}_w = \frac{180.55\times(2275.1-146.7)}{4.187\times(33-23)} = 9178\ \text{kg/s} \quad \text{(Cooling water flow rate)}$$

Now, $P_{air} = P_c - P_{sat}$

$= 0.075 - P_{sat}(35°C)$ [Using steam table]

$= 0.075 - 0.05629$

$P_{air} = 0.01871$ bar

Using : $P_{air} \times \dot{V}_{air} = \dot{m}_a \times RT$

$(0.0187 \times 100) \times (180.55 \times 22.194) = \dot{m}_a \times 0.287 \times 308$

$\Rightarrow \quad \dot{m}_a = 84.81\ \text{kg/s}$

**06(c). What do you understand by biomass gasification? How are gasifiers classified? Describe with a schematic diagram the working of a downdraft gasifer. (20 M)**

**Sol: Gasification of biomass:**

Gasification is the process of conversion of solid fuel into a gaseous fuel by a thermo-chemical method without leaving any solid carbonaceous residue. Gasifier is the equipment which converts biomass into producer gas. The in feed raw materials for gasifier are: wood chips, saw dust, wood sticks, rice husk, coconut shells, etc.

The gas obtained from wood gasification typically consist of following composition:

| | |
|---|---|
| CO | 18-22% |
| $H_2$ | 13-19% |
| Methane | 1-5% |
| HC | 0.2-0.4% |
| $N_2$ | 45-55% |
| Water vapor | 4% |

The gas obtained can be used for generation of motive power either in dual fuel engines or in diesel engines.

**Various types of gasifiers are:**

a) Down drought b) Up drought c) Cross drought

**a) Down drought gasifier-**

In this gasifier, air enters at the combustion zone and gas produced leaves near the bottom of the gasifier.

The flow of air travel is opposite to the flame front.

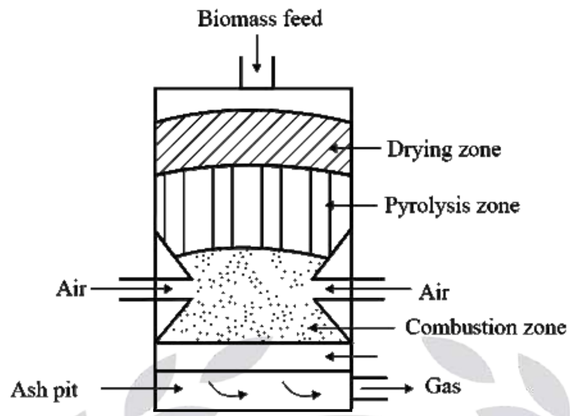



***Fig: Gasification process in downdraft gasifier***

Gasification involves the partial combustion and reduction operations of biomass.

The composition of gas production depends upon the degree of equilibrium among various reactions.

Following reactions occur during gasification

$C + O_2 \rightarrow CO_2 + 393.8$ MJ/kg.mol (Combustion)

$C+H_2O \rightarrow CO + H_2 -131.4$ MJ/kg.mol (water gas)

$CO + H_2O \rightarrow CO_2 + H_2 + 41.2$ MJ/kg.mol (water shift reaction)

$C + CO2 \rightarrow 2CO - 172.6$ MJ/kg.mol

$C + 2H_2O \rightarrow CO_2 + 2H_2 - 78.7$ MJ/kg.mol

$C + 2H_2 \rightarrow CH_4 + 75$ MJ/kg.mol (methane reaction)

Down draft gasifier has following zones:

1) **Drying zone:**

In this biomass entry zone, moisture content of biomass are removed.

Temperature is up to 120°C.

**2) Pyrolysis zone:**

The heat from combustion zone is transferred upwards by radiation, conduction and convection causing wood chips to pyrolyze and loose 70-80% of their weight.

The temperature ranging from 350°C to 600°C.

Pyrolysis converts organic waste to char, tar and oils and gas.

**3) Oxidation / combustion zone:**

In oxidation zone, the oxygen in air stream blast reacts with the carbon in the fuel to reduce carbon to form hydrogen and carbon monoxide.

The oxidation zone has highest temperature ranging from 1000°C to 1200°C.

**4) Reduction zone:**

In reduction zone, the $CO_2$ coming from oxidation zone is reduced.

Temperature of zone is 700°C to 1000°C. The regenerative heating due to transfer of heat from hot gas to the biomass moving downwards also increases its residence time in high temperature zone. The raw gases forced by the throat to pass through a high temperature zone and most of the unburnt tars are cracked into gaseous hydrocarbons. Therefore, it produces relatively a clean gas.

**07(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 desired 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/ kg K, $C_v$ = 0.71 kJ/kg K. (20 M)**

**Sol:**

**(I)** Speed of compressor,

$$\eta_{vol} = 1 - C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

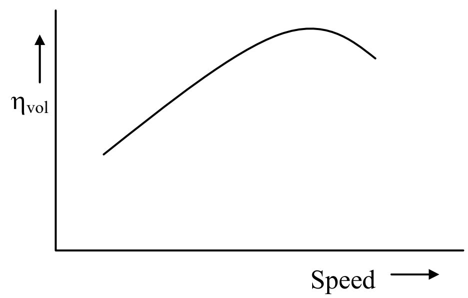
C = Clearance ratio,

$P_1$ = Suction pressure,

$P_2$ = Delivery pressure

n = Polytropic index of compression

- Higher speeds head to higher volumetric efficiency because of higher speeds give higher vaccum at port and consequent larger air flow rate. Further increase in engine speed leads to maximum value of $\eta_{vol}$. Volumetric efficiency increases with increase of engine speed upto certain limit and then decreases.




- As the pressure ratio increases the volumetric efficiency decreases and vice versa.
- If pressure ratio increases due to throttling on suction side the pressure ratio increases and volumetric efficiency decreases.
- If pressure ratio decreases due to throttling on delivery side the pressure ratio decreases and volumetric efficiency increases.

**(II)** Suction pressure ($P_1$) = 1 bar

Suction Temperature ($T_1$) = 25°C = 398 K

Delivery Pressure ($P_d$) = 160 bar

Polytropic index (n) = 1.25

Temperature after inter cooling ($T_3$) = 30°C = 303 K

Maximum temperature ($T_2$) = 125°C = 298 K

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

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

$$= \left(\frac{398}{298}\right)^{\frac{1.25}{1.25-1}} \times 1$$

$$= 4.249 \text{ bar} \simeq 4.25 \text{ bar}$$

Pressure ratio ($r_P$) = $\frac{P_2}{P_1} = \frac{4.25}{1} = 4.25$

After first stage cooled to 30°C and followed by compression to 125°C and repeated to reach delivery pressure 160 bar. (Assuming perfect intercooling)

Overall pressure ratio $\left(r_{P_o}\right) = \frac{160}{1} = 160$

Maximum pressure ratio in second state

$$r_P = \frac{P_4}{P_3} = \left(\frac{T_2}{T_3}\right)^{\frac{n}{n-1}} = \left(\frac{398}{303}\right)^{\frac{1.25}{1.25-1}} = 3.91$$

which is same for remaining stages

$\therefore$ For k + 1 total stage

$$r_{P_1}.r_P^k = r_{P_o}$$

$$\Rightarrow \quad k = \frac{\ell n\left(\frac{160}{4.25}\right)}{\ell n(3.91)} = 2.66$$

i) No. of stages = k + 1 = 2.66 + 1 = 3.66 $\simeq$ 4 stages

$\therefore$ No. of intercoolers = 3

Pressure ratio in each of next three stages = $\left(\frac{160}{4.25}\right)^{1/3}$ = 3.3515

ii) Work input in the first compressor per kg of air

$$= \frac{n}{n-1} RT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

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

= 143.51 kJ/kg

Work input in the remaining stages

$$= \frac{3n}{n-1} RT_3 \left[ \left( \frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{3 \times 1.25}{1.25 - 1} \times 0.287 \times 303 \left[ (3.3515)^{\frac{1.25-1}{1.25}} - 1 \right]$$

= 356.94 kJ/kg

Total work input = Work input in first stage + Work input in the remaining stages

= 143.51 + 356.94

= 500.45 kJ/kg

iii) Heat rejected in intercoolers = No. of intercoolers × m $c_P$ ($T_2 - T_3$)

= 3 × (0.71 + 0.287) × (398 – 303)

= 284.145 kJ/kg

**07(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 M)**

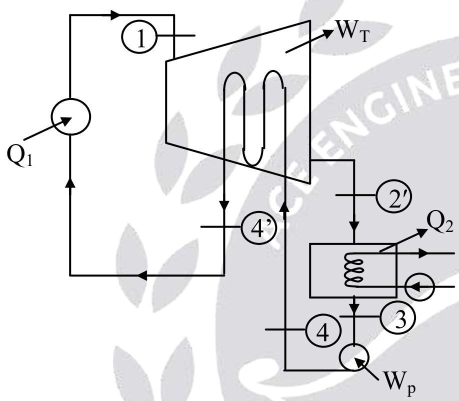
**Sol: Regenerative feed heating:**

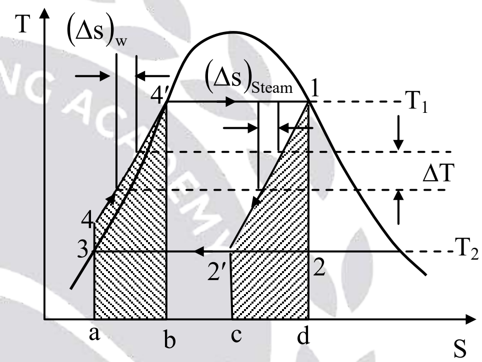
The aim of regenerative feed heating cycle is to supply the feedwater to boiler at a temperature much higher than the temperature of condensate in condenser by heating the feed water in one or more feed heaters with the help of steam bled off during expansion from steam turbine.

**Ideal Regenerative Cycle:**

- The mean temperature of heat addition can also be increased by decreasing the amount of heat added at low temperatures.
- A considerable part of the total heat supplied is in the liquid phase when heating up water from 4 to 4′, at a temperature lower than $T_1$.

- For maximum efficiency, all heat should be supplied at $T_1$, and feed water should enter the boiler at state 4′.
- This may be accomplished in what is known as an ideal regenerative cycle.
- Let us assume that this is a reversible heat transfer, i.e., at each point the temperature of the vapour is only infinitesimally higher than the temperature of the liquid. The process 1-2′ in figure thus represents reversible expansion of steam in the turbine with reversible heat rejection.




Heat supplied to the boiler = Heat recovered from steam

$$(\Delta T)_{water} = -(\Delta T)_{steam}$$

$$\Delta s_{(water)} = -\Delta s_{(steam)}$$

$$(s_{4'} - s_4) = (s_1 - s_{2'})$$

$$(s_1 - s_{4'}) = (s_{2'} - s_3) \quad (\because s_3 = s_4)$$

Turbine work $W_T = (h_1 - h_{2'})$

- Then the slopes of lines 1-2′ and 4′-3 in the figure will be identical at every temperature and the lines will be identical in contour.

- Areas 4-4′-b-a-4 and 2′-1-d-c-2′ are not only equal but congruous. Therefore, all the heat added from an external source (Q1) is at the constant temperature T1, and all the heat rejected (Q2) is at the constant temperature $T_2$, both being reversible.

  Then $Q_1 = h_1 - h_{4'} = T_1\,(s_1 - s_{4'})$

  $Q_2 = h_{2'} - h_3 = T_2\,(s_{2'} - s_3)$

  Since $s_{4'} - s_3 = s_1 - s_{2'}$

  $\Rightarrow \quad s_{1'} - s_{4'} = s_{2'} - s_3$

$$(\eta_{th})_{Regen} = 1 - \frac{Q_2}{Q_1} = \left[1 - \left(\frac{T_2(s_{2'} - s_3)}{T_1(s_1 - s_{4'})}\right)\right]$$

$$(\eta_{th})_{Ideal\ Regenerative} = \left[1 - \frac{T_2}{T_1}\right]$$

$$\therefore (\eta_{th})_{Ideal\ Regenerative} = (\eta_{th})_{Carnot}$$

- The efficiency of the ideal regenerative cycle is thus equal to the Carnot cycle efficiency.

  Writing the steady flow energy equation for the turbine

$$h_1 - W_T - h_{2'} + h_4 - h_{4'} = 0$$

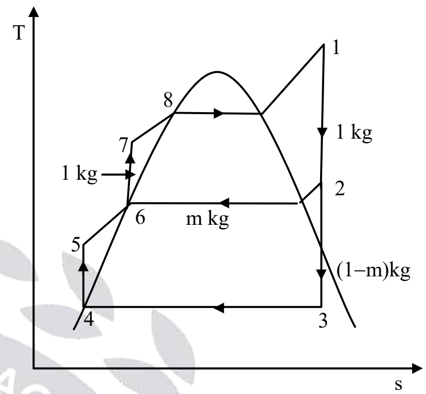
$$W_T = (h_1 - h_{2'}) - (h_{4'} - h_4)$$

- The pump work remains the same as in the Rankine cycle, i.e. $W_p = h_4 - h_3$
- The net work output of the ideal regenerative cycle is thus less, and hence its steam rate will be more, although it is more efficient, when compared with the Rankine cycle.

- However, the cycle is not practicable for the following reasons:
  (i) Reversible heat transfer cannot be obtained in finite time.
  (ii) Heat exchanger in the turbine is mechanically impracticable
  (iii) The moisture content of the steam in the turbine will be high.

Let 'm' kg of steam is extracted from turbine for each kg of steam entering.

Energy balance equation for feed water :




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

$$m = \frac{h_6 - h_5}{h_2 - h_5} \qquad \text{------ (1)}$$

Thermal efficiency of cycle :

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

$$= 1 - \frac{(h_2 - h_6)(h_3 - h_4)}{(h_2 - h_4)(h_1 - h_6)} \quad - \text{ ----- (2)}$$

Assuming that turbine expansion line follows a path on diagram such that

$$(h - h_f) = \text{constant} = \beta$$

$\therefore \quad h_1 - h_8 = h_2 - h_6 = h_3 - h_4 = \beta = \text{constant}$

Let, $h_8 - h_4 = \alpha$ and $\gamma$ = enthalpy rise of water in feed water

From eq. (2):

$$\eta = 1 - \frac{\beta^2}{(\beta + \gamma)(\alpha + \beta + \gamma)} \qquad \text{----- (3)}$$

$\alpha$, $\beta$ are fixed and $\gamma$ is variable.

For maximum efficiency,

$\frac{d\eta}{d\gamma} = 0$ gives optimum degree of regeneration.

$\therefore \quad \gamma = \frac{\alpha}{2}$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AIR 2nd ME JAY CHAVDA
AIR 2nd PI GOWTHAM GUDIMELLA
AIR 2nd EC M POOJASREE
AIR 2nd ES ANUSH VERMA
AIR 3rd ME MUNISH KUMAR
AIR 3rd ME NUKULA VISWA TEJA
AIR 3rd PI REHAN ANWAR DESAI
AIR 3rd EC MANOJ KUMAR
AIR 3rd CE SHASHIKANT KUMAR
AIR 3rd EE JAYMAL KHUNTI
AIR 4th PI ROHIT SONI
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AIR 4th IN DIVAKAR CH.
AIR 4th IN SAURABH JAISWAL
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AIR 7th ME VATSAL PANCHAL
AIR 7th PI SACHIN DUBOLIYA
AIR 7th PI ATULYA JYOTHI
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AIR 8th EC ANKUR LAL MEENA
AIR 8th IN KUNAL SAURAV
AIR 8th CE PRANSHU JANGID
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AIR 8th EE HEMANT JINDAL
AIR 9th EC ABHISHEK SINGH
AIR 9th EC ALEESHA ROSE
AIR 9th EC SAI VAMSI DOSAPATI
AIR 9th IN RAMESH KUMAR
AIR 9th CS NIRANJAN NITIN DHOOT
AIR 9th XE RACHIT KUMAR
AIR 10th ME DEEPESH AGARWAL
AIR 10th PI VYOM SHARMA
AIR 10th IN PRAGYA KAUSHIK
AIR 10th CE GOVIND PRASAD B
AIR 10th XE D KUMAR PATIL
AIR 10th XE R SHIVAJI NALE
and many more...
Total 57 Ranks in Top 10
ME : 10
PI : 09
CE : 08
EE : 4
EC : 09
CS : 03
IN : 08
XE : 06

**07(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. (10 M)**

**Sol:** Consider a horizontal axis wind turbine with 'n' number of blades. The blade must take position of next blade without must wind pass from passage between two adjacent blades while rotating.

Time take by blade to move to next position = time taken by distributed wind to pass

$$\frac{2\pi}{n\times\omega}=\frac{t}{V_i} \quad (t = \text{zone thickness})$$

$$t=\frac{R}{2}$$

$$\therefore \quad \frac{2\pi}{n\times\omega}=\frac{R}{2\times V_i}$$

For optimum number of blades :

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

**Significance of optimal tip speed ratio:**

- Electricity generation requires high rotational speed. Hence, $\lambda \rightarrow 6$ to $9$
- For water pumping application, high torque is required. Hence, $\lambda \rightarrow 1.5$ to $2$

**07(c)(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 M)**

**Sol:** For optimum tip speed ratio,

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

i.e., $$\frac{\omega R}{V_i} = \frac{4\pi}{n}$$

$$\omega = \frac{4\pi V_i}{nR}$$

$$= \frac{4\pi \times 20}{3 \times 40} = \frac{2\pi}{3}\,\text{rad/s}$$

$$N = \frac{60 \times \omega}{2\pi} = \frac{60 \times \frac{2\pi}{3}}{2\pi}$$

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

**08(a). Prove that the efficiency corresponding to the maximum work done in a Brayton cycle is given by the relation**

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

**where t is the ratio of the maximum and minimum temperatures of the cycle.**

**An ideal open-cycle gas turbine plant using air operates in 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 $C_V = 0.717\ kJ/kg\ K$, 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 M)**

**Sol:** Condition for maximum work output of Brayton cycle is $T_2 = T_4 = \sqrt{T_1T_3}$

Pressure ratio ($r_p$) = $\left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$

$$\Rightarrow (r_p)^{\frac{\gamma-1}{\gamma}} = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2}}$$

Brayton cycle efficiency ($\eta_B$) = $1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}}} = 1 - \frac{1}{(t)^{\frac{1}{2}}} = 1 - \frac{1}{\sqrt{t}}$

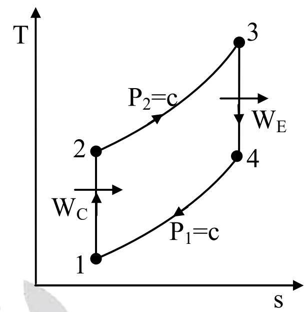
Overall pressure ratio = 4 = $\frac{P_2}{P_1}$

Minimum temperature ($T_1$) = 300 K

Maximum temperature ($T_3$) = 1200 K

$c_p = 1$ kJ/kgK

$c_v = 0.717$ kJ/kgK




$$\gamma = \frac{c_p}{c_w} = 1.394 \approx 1.4$$

Process (1) – (2): isentropic

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 300(4)^{\frac{0.4}{1.4}} = 373 \text{ K}$$

Process (3) – (4): isentropic

$$T_4 = \frac{T_3}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{1200}{(4)^{\frac{0.4}{1.4}}} = 807.54 \text{ K}$$

$Q_S = c_p (T_3 - T_2) = 1 \times (1200 - 373) = 827$ kJ/kg

$Q_R = c_p (T_4 - T_1) = 1 \times (807.54 - 300) = 507.54$ kJ/kg

Specific work output $= Q_S - Q_R$

$= 827 - 507.54 = 319.46$ kJ/kg

$$\text{Thermal efficiency } (\eta_{th}) = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

$$= 1 - \frac{1}{(4)^{\frac{0.4}{1.4}}} = 32.7\ \%$$

**(i) Basic cycle with regenerator (Ideal regenerator)**

$$\eta_{th} = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}}$$

$$= 1 - \frac{300}{1200} \times (4)^{\frac{0.4}{1.4}} = 0.6285 = 62.85\ \%$$

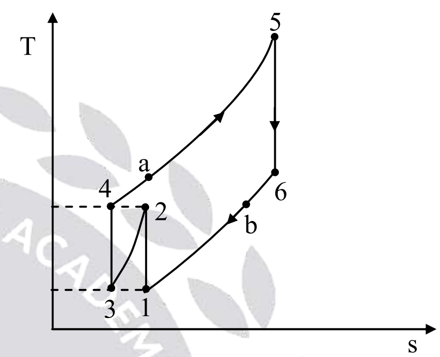
$W_{net} = Q_s - Q_R = 319.46$ kJ/kg

**(ii) If two stage compression is happened for ideal intercooling,**

$$T_2 = T_1(2)^{\frac{\gamma-1}{\gamma}}$$

$$= 300(2)^{\frac{0.4}{1.4}} = 365.7\,K$$

$$\frac{P_2}{P_1} = \left(\frac{P_d}{P_s}\right)^{\frac{1}{2}} = (4)^{\frac{1}{2}} = 2$$




Compression work = 2 $c_p$ ($T_2 - T_1$)

= 2 × 1 × (365.7 – 300)

= 131.4 kJ/kg

Turbine work ($W_T$) = $c_p$ ($T_5 - T_6$)

= 1 (1200 – 807.54)

= 392.4 kJ/kg

Specific net work output

$W_{net} = W_T - W_C$

= 392.4 – 131.4

= 261.06 kJ/kg

Assuming perfect regeneration,

$T_a = T_6$

Heat supplied = $Q_s = c_p (T_5 - T_a)$ = 1× (1200 – 807.54) = 392.4 kJ/kg

$$\text{Efficiency} = \eta = \frac{\text{Net work}}{\text{Heat supplied}}$$

$$= \frac{W_{net}}{Q_s} = \frac{261.04}{392.4}$$

= 0.6652 = 66.52 %

**08(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 M)**

**Sol:** Given:

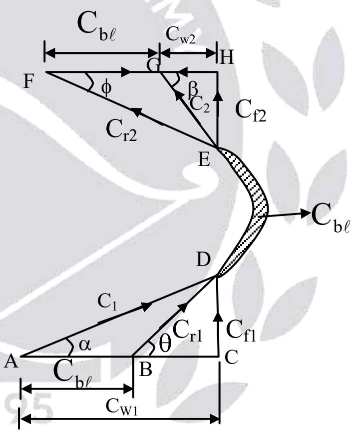
Symmetrical blades.

Steam entry velocity, $C_1 = 800$ m/s

Blade angle at inlet, $\alpha = 22°$

Blade velocity, $C_{b\ell} = 300$ m/s

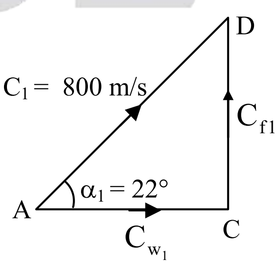
Mass flow rate $(\dot{m}) = 2$ kg/s




Consider triangle ACD

$$C_{f_1} = C_1 \sin\alpha$$

$$= 800 \sin(22°) = 299.685 \simeq 300 \text{ m/s}$$



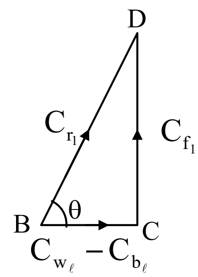

$$C_{w_1} = C_1 \cos\alpha$$

$$= 800 \cos 22°$$

$$= 741.75 \text{ m/s}$$

Consider triangle BCD

$$\tan\theta = \frac{C_{f_1}}{C_{w_1} - C_{b\ell}}$$

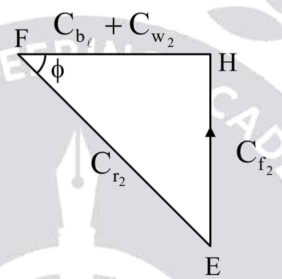



$$\theta = \tan^{-1}\left(\frac{C_{f_1}}{C_{w_1} - C_{b\ell}}\right)$$

$$= \tan^{-1}\left(\frac{300}{741.75 - 300}\right)$$

$$\theta = 34.18°$$

$$C_{r_1} = \sqrt{(C_{f_1})^2 + (C_{w_1} - C_{b\ell})^2}$$




$$= \sqrt{(300)^2 + (741.75 - 300)^2}$$

$= 533.99$ m/s

$= 534$ m/s

$C_{r_1} = C_{r_2}$ (Considering no friction)

$\theta = \phi$ ($\because$ Symmetrical blades)

$= 34.18°$

Consider triangle EHF

$$\cos\phi = \frac{C_{b\ell} + C_{w_2}}{C_{r_2}}$$

$$C_{w_2} = C_{r_2}\cos\phi - C_{b\ell}$$

$= \{534 \cos (34.18°)\} - 300$

$= 141.766$ m/s

$$\sin\phi = \frac{C_{f_2}}{C_{r_2}} \quad \Rightarrow \quad C_{f_2} = C_{r_2}\sin\phi$$

$= 534 \times \sin (34.18°)$

$= 300$ m/s

(i) Blade angles

$$\theta = 34.18°$$

ii) Tangential thrust

$$(F_x) = \dot{m}\left(C_{w_1} + C_{w_2}\right)$$

$$= 2(741.75 + 141.766)$$

$$= 1767 \text{ N}$$

(iii) Diagram power

$$(P) = F_x \times C_{b\ell} = 1767 \times 300 = 530.1 \text{ kW}$$

(iv) Diagram efficiency ($\eta_{diag}$) $= \dfrac{\dot{m}\left(C_{w_1} + C_{w_2}\right)C_{b\ell}}{\frac{1}{2}\dot{m}C_1^2}$

$$= \frac{530.1 \times 10^3}{\frac{1}{2} \times 2 \times 800^2} = 82.82\ \%$$

(v) Axial thrust ($F_y$) $= \dot{m}\left(C_{f_1} - C_{f_2}\right) = 2(300 - 300) = 0$

**08(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? (6 M)**

**Sol:**

- India is in Northern hemisphere. Most of the solar energy need for India is during winter period and during this period declination angle for Sun-Earth geometry is negative.
- Hence, solar rays are incident from south direction in India. This makes necessary to incline solar panels towards South to gain more heat energy.
- In the absence of sun tracking mechanism, the solar panels in Northern hemisphere should face South and panels in Southern hemisphere should face North.
- The solar panels are tilted equal to angle of latitude of that location. [$\beta = \phi$]

**08(c)(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 modules requirement in the array. Assume global radiations incident normally to the panel as 1000 W/m². (14 M)**

**Sol:** P = 1 hp = 746 W

Power required to supply to motor to obtain 1 hp shaft power:

$$P_1 = \frac{746}{0.8} = 932.5\text{ W} \qquad \text{------ (1)}$$

Power produced by PV panel :

$P_2 = q \times (n \times m) \times \text{Area} \times \eta$ [where, n = number of modules]

$= 1000 \times (n \times 9 \times 4) \times 0.125^2 \times 0.15$

$P_2 = 84.375 \times n$ ------ (2)

Power balance equation :

$P_1 = P_2$

$932.5 = 84.375 \times n$

n = number of modules = 11.05

$n \approx 11$ modulus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AIR 2 PARTH BATRA E&T
AIR 2 PRASHANT SINGH CE
AIR 3 SURAJ KUMAR S ME
AIR 3 VIKASH SHANKAR EE
AIR 3 RAHUL NAREDI E&T
AIR 3 KULDEEP JANGRA CE
AIR 4 SHUBHAM B ME
AIR 4 ANUPAM S EE
AIR 4 SHUBHAM E&T
AIR 4 ANISH BAGGA CE
AIR 5 KAMLESH PARWAR ME
AIR 5 MANOJ KUMAR E&T
AIR 5 PAVITRA GOYAL CE
AIR 6 MD ZUHAIB ME
AIR 6 VISHWA SIMHAA EE
AIR 6 SAURAV KUMAR S E&T
AIR 7 PRATEEK S ME
AIR 7 DINESH KUMAR S EE
AIR 7 RAGHAV PURWAR E&T
AIR 8 V SAIKRISHNA REDDY ME
AIR 8 GAGAN GHUNAWAT EE
AIR 8 RAM KRISHNA E&T
AIR 9 GANESH KUMAR A ME
AIR 9 AKSHAY KUMAR T EE
AIR 9 CHHAVI JAIN E&T
AIR 9 ARPIT JAIN CE
AIR 10 HEMABH TRIVEDI ME
AIR 10 RAJAT DIXIT EE
AIR 10 L KUMARI JAISWAL E&T
AIR 10 AMIT SHARMA CE
and many more...
TOTAL 36 RANKS IN TOP 10
ME 10
EE 09
E&T 10
CE 07