DETAILS EXPLANATIONS

ME: Paper-1 (Paper-3) [Full Syllabus]

[PART : A]

1. The triple point is merely the point of intersection of sublimation and vapourisation curves.

2. "It states that of all engines operating between a given constant temperature source and a given constant temperature sink, none has a higher efficiency than a reversible engine"

3. It is the temperature to which air must be cooled at constant pressure in order to cause condensation of any of its water vapour. It is equal to steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (t_{dp})

4. It is the temperature at which the water or ice can saturate air by evaporating adiabatically into it. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) (t_{wb})

5. The by-pass factor (BF) for the process is defined as the ratio of the difference between the mean surface temperature of the coil and leaving air temperature to the difference between the mean surface temperature and the entering air temperature.

6. When no incident radiation is transmitted through the body, it is called an 'opaque body'.

For the opaque body $\tau = 0$, and equation becomes

$$\alpha + \rho = 1$$

Solids generally do not transmit unless the material is of very thin section. Metals absorb radiation within a fraction of a micrometer, and insulators within a fraction of millimeter. Glasses and liquids are, therefore, generally considered as opaque.

7. The intensity of radiation (I) is defined as the rate of energy leaving a surface in a given direction per unit solid angle per unit area of the emitting surface normal to the mean direction in space.

8. The law states that the total emissive power E_θ from a radiating plane surface in any direction is directly proportional to the cosine of the angle of emission. The angle of emission θ is the angle subtended by the normal to the radiating surface and the direction vector of emission of the receiving surface. If En be the total emissive power of the radiating surface in the direction of its normal, then

$$E_\theta = En \cos\theta$$

The above equation is true only for diffuse radiation surface. The radiation emanating from a point on a surface is termed diffused if the intensity, I is constant. This law is also known as Lambert's law of diffuse radiation.

9. The momentum equation for compressible fluids is similar to the one for incompressible fluids. This is because in momentum equation the change in momentum flux is equated to force required to cause this change.

$$\text{Momentum flux} = \text{mass flux} \times \text{velocity} = \rho AV \times V$$

But the mass flux i.e., $\rho AV = \text{constant}$

10. Given, $T_2 = 305$ K

$T_1 = 260$ K

C.O.P. of a refrigerating machine

We know that C.O.P. of a refrigerating machine,

$$(\text{C.O.P.})_R = \frac{T_1}{T_2 - T_1} = \frac{260}{305 - 260} = 5.78$$

11. It is the ratio of heat transfer coefficient to the flow of heat per unit temperature rise due to the velocity of the fluid.

$$S_t = \frac{h}{\rho U_{C_p}}$$

or,

$$S_t = \frac{\frac{hL}{K}}{\left[\frac{\rho UL}{\mu}\right]\left[\frac{\mu c_p}{K}\right]} = \frac{Nu}{Re \times Pr}$$

Thus, Stanton number may also be defined as the ratio of Nusselt number and the product of Reynolds number and Prandtl number.

It is worth doling that Stanton number can be used only in correlating forced convection data (since the expression contains the velocity U).

12. The skin friction coefficient (C_f) is defined as the ratio of shear stress τ_0 at the plate to the dynamic head $\frac{1}{2}\rho U^2$ caused by free stream velocity. Thus the local skin friction coefficient C_μ at any value of x is.

$$C_{fx} = \frac{\tau_0}{\frac{1}{2}\rho U^2} = \frac{\mu\left(\frac{\partial u}{\partial y}\right)_{y=0}}{\frac{1}{2}\rho U^2}$$

13. The heat exchanger effectiveness (ε) is defined as the ratio of actual heat transfer to the maximum possible heat transfer. Thus,

$$\varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} = \frac{Q}{Q_{max}}$$

14. In solids, the heat is conducted by the following two mechanisms :

(i) By lattice vibration (the faster moving molecules or atoms in the hottest part of a body transfer heat by impacts some of their energy to adjacent molecules).

(ii) By transport of free electrons (Free electrons provide an energy flux in the direction of decreasing temperatue – For metals, especially good electrical conductors, the electrtonic mechanism is responsible for the major portion of the heat flux except at low temperature.)

15. The economizer provides and regulates enriched mixture required at full throttle operation for speeds above the cruising range.

16. Wet bulb depression of a moist air is defined as the difference between its dry and wet bulb temperatures. It is zero for saturated air.

17. For cyclic processes, the heat interaction is equal to work transfer. Hence, for the given case

$$85 - 90 + 0 = 0 - 20 + W$$

$$W = 15 \text{ kJ}$$

18. The resulting entropy change is given by

$$DS = \frac{5 \times 3600}{20 + 273}$$

$$= 61.43 \text{ kJ/K}$$

19. For quasi-static isothermal processes

$$W = p_1 v_1 \ln\frac{v_2}{v_1} = 0.8 \times 10^6 \times 0.015 \ln\left(\frac{0.030}{0.015}\right) = 8.317 \text{ kW}$$

20. Given, that

$$p_1 = 1 \text{ bar}$$
$$p_2 = 30 \text{ bar}$$
$$\rho = 990 \text{ kg/m}^3$$

Isentropic work done by the pump is

$$w_p = v(p_2 - p_1) = \frac{1}{\rho}(p_2 - p_1) = 2.92929 \text{ kJ/kg}$$

[PART : B]

21. The horizontal force = γ Ah where A is the projected area.

Considering unit width,

Horizontal force = 9810 × 2 × 4 × 1 = 78480 N, to the right

It acts at the centre of pressure of the projected area

i.e., at = 1.333 m from the bottom (i.e., $\frac{1}{3} \times 4$)

Vertical force = the weight to the liquid displaced

$$= \pi \times 4^2 \times 1 \times \frac{9810}{4} = 123276 \text{ N, upwards.}$$

It acts at $\frac{4r}{3\pi}$ = 1.698 m, from the hinge.

Resultant force = $(1232762 + 784802)^{0.5}$ = 146137 N

Angle is determined by $\tan\theta = \frac{78480}{123276} = 0.6366$

∴ $\theta = 32.48°$

where θ is the angle with vertical.

To check for the resultant to pass through the centre the sum of moment about O should be zero.

78480 × (4 – 1.3333) – 123276 × 1.698 = 42.

Compared to the values the difference is small and these can be assumed to the equal. Hence the resultant passes through the centre of the circle.

22. Total force on the gate = γ Ah

h = centre of gravity of the semicircular surface $\frac{2D}{3\pi}$

$$= 2 \times \frac{3}{3} \times \pi = 0.6366 \text{ m}$$

$$\text{Total force} = 1000 \times 9.81 \times \left(\pi \times \frac{3^2}{4} \times 2\right) \times 0.6366$$
$$= 22072 \text{ N} = 22.072 \text{ kN}$$

$$I_{base} = \pi\frac{D^4}{128} \text{ (about the diameter)}$$

$$\text{Depth centre of pressure} = \frac{I_{base}}{Ah} = \left(\pi \times \frac{3^4}{128}\right)\left(2 \times \frac{4}{\pi} \times 3^2\right)\left(\frac{1}{0.6366}\right) = 0.8836 \text{ m}$$

23. In the Lagrangian method a single particle is followed over the flow field, the co-ordinate system following the particle. The flow description is particle based and not space based. A moving coordinate system has to be used. This is equivalent to the observer moving with the particle to study the flow of the particle. This method is more involved mathematically and is used mainly in special cases.

In the Eularian method, the description of flow is on fixed coordinate system based and the description of the velocity etc. are with reference to location and time i.e., V = V (x, y, z, t) and not with reference to a particular particle. Such an analysis provides a picture of various parameters at all locations in the flow field at different instants of time. This method provides an easier visualisation of the flow field and is popularly used in fluid flow studies. However the final description of a given flow will be the same by both the methods.

24. When fluids flow over surfaces, the molecules near the surface are brought to rest due to the viscosity of the fluid. The adjacent layers also slow down, but to a lower and lower extent. This slowing down is found limited to a thin layer near the surface. The fluid beyond this layer is not affected by the presence of the surface. The fluid layer near the surface in which there is a general slowing down is defined as boundary layer. The velocity of flow in this layer increases from zero at the surface to free stream velocity at the edge of the boundary layer. The development of the boundary layer in flow over a flat plate and the velocity distribution in the layer are shown in figure.

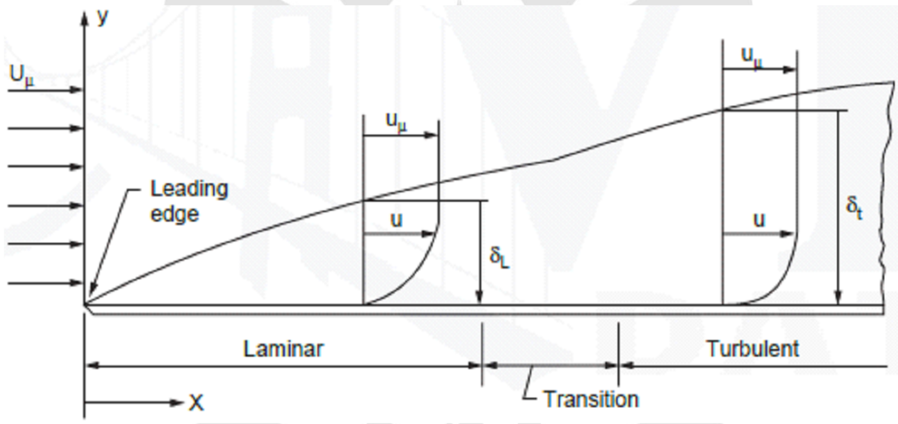


Figure : Boundary Layer Development (flat-plate)

Pressure drop in fluid flow is to overcome the viscous shear force which depends on the velocity gradient at the surface. Velocity gradient exists only in the boundary layer. The study thus involves mainly the study of the boundary layer. The boundary conditions are

(i) At the wall surface, (zero thickness) the velocity is zero.

(ii) At full thickness the velocity equals the free stream velocity.

(iii) The velocity gradient is zero at the full thickness.

Use of the concept is that the main analysis can be limited to this layer.

25. The official code defines the head on the pump as the difference in total energy heads at the suction and delivery flanges. This head is defined as manometric head.

The total energy at suction inlet (expressed as head of fluid)

$$\frac{P_s}{\gamma}+\frac{V_s^2}{2g}+Z_s$$

where Z = The height of suction gauge from datum

The total energy at the delivery of the pump

$$= \frac{P_d}{\gamma} + \frac{V_d^2}{2g} + Z_d$$

Z_2 = The height of delivery gauge from datum

The difference in total energy is defined as H_m

$$= \left(\frac{P_d}{\gamma} - \frac{P_s}{\gamma}\right) + \frac{V_d^2 - V_s^2}{2g} + (Z_d - Z_s)$$

From equation $$\frac{P_d}{\gamma} - \frac{P_s}{\gamma} = H_e + \frac{V_s^2}{2g}$$

Substituting $$H_m = H_e + \frac{V_d^2}{2g} + (Z_d - Z_s)$$

As $(Z_d - Z_s)$ is small and $\frac{V_d^2}{2g}$ is also small as the gauges are fixed as close as possible.

$\therefore$ H = Static head + all losses.

26. Crude oil : Specific gravity = 0.8

$\therefore$ Density of oil, $\rho = 0.8 \times 1000 = 800 \text{ kg/m}^3$

Bulk modulus, $K = 1.5 \text{ GN/m}^2$

Mercury : Bulk modulus, $K = 27 \text{ GN/m}^2$

Density of mercury, $\rho = 13.6 \times 1000 = 13600 \text{ kg/m}^3$

Sonic velocity, C_{oil}, C_{Hg}:

Sonic velocity is given by the relation :

$$C = \sqrt{\frac{K}{\rho}}$$

$$C_{oil} = \sqrt{\frac{1.5 \times 10^9}{800}} = 1369.3 \text{ m/s}$$

$$C_{Hg} = \sqrt{\frac{27 \times 10^9}{13600}} = 1409 \text{ m/s}$$

27. The output of a gas turbine can be amply improved by expanding the gases in two stages with a reheater between the two as shown in figure.

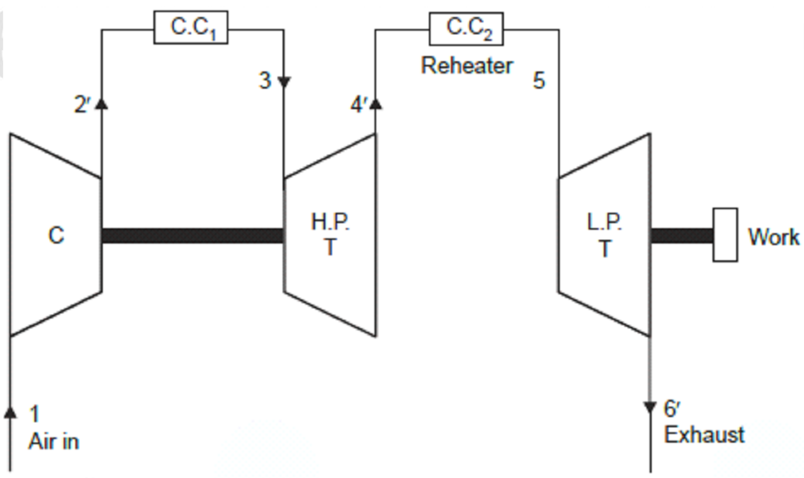


Figure : Gast Turbine with reheater

The H.P. turbine drives the compressor and the L.P. turbine provides the useful power output. The corresponding T-s diagram is shown in figure.

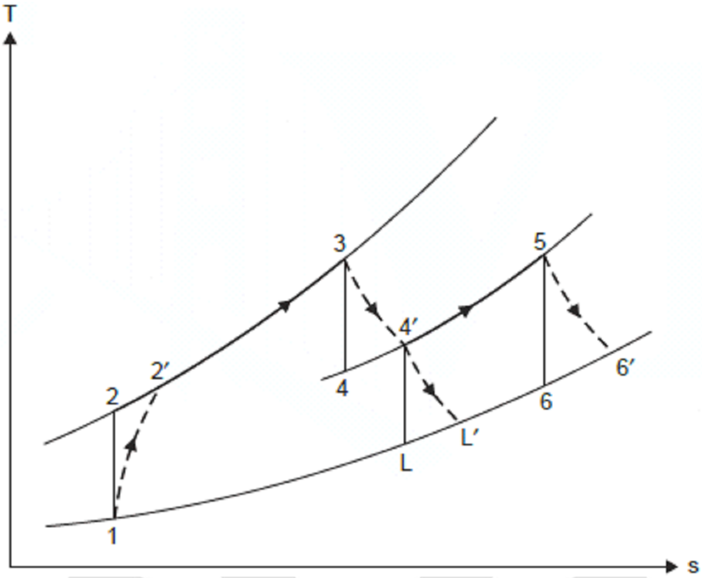


Figure : T-s Diagram for the unit

The line 4′-L′ represents the expansion in the L.P. turbine if reheating is not employed.

Neglecting mechanical losses the work output of the H.P. turbine must be exactly equal to the work input required for the compressor.

i.e., $$c_{pa}(T_2' - T_1) = c_{pg}(T_3 - T_4')$$

The work output (net output) of L.P. turbine is given by,

$$\text{Net work output (with reheating)} = c_{pg}(T_5 - T_6')$$

and $$\text{Net work output (without reheating)} = c_{pg}(T_4' - T_L')$$

Since the pressure lines diverge to the right on T-s diagram it can be seen that the temperature difference $(T_5 - T_6')$ is always greater than $(T_4' - T_L')$, so that reheating increases the net work output.

Although net work is increased by reheating the heat to be supplied is also increased, and the net effect can be to reduce the thermal efficiency

$$\text{Heat supplied} = c_{pg}(T_3 - T_2') + c_{pg}(T_5 - T_4')$$

28.
- The heating process in the boiler tends to become reversible.
- The thermal stresses set up in the boiler are minimised. This is due to the fact that temperature ranges in the boiler are reduced.
- The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.
- Heat rate is reduced.
- The blade height is less due to the reduced amount of steam passed through the low pressure stages.
- Due to many extractions there is an improvement in the turbine drainage and it reduces erosion due to moisture.
- A small size condenser is required.

29. In a given combustion process, that takes place adiabatically and with no work or changes in kinetic or potential energy involved, the temperature of the products is referred to as the 'adiabatic flame temperature'. With the assumptions of no work and no changes in kinetic or potential energy, this is the maximum temperature that can be achieved for the given reactants because any heat transfer from the reacting substances and any incomplete combustion would tend to lower the temperature of the products.

The following points are worthnoting :

- The maximum temperature achieved through adiabatic complete combustion varies with the type of reaction and per cent of theoretical air supplied. An increase in the air-fuel ratio will effect a decrease in the maximum temperature.
- For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flame temperature that can be achieved is with a 'stoichiometric' mixture.
- The adiabatic flame temperature can be controlled by the amount of excess air that is used. This is important, for example, in gas turbines, where the maximum permissible temperature is determined by metallurgical considerations in the turbine, and close control of the temperature of the products is essential.

30. If air is passed through a humidifier which has heated water sprays instead of simply recirculated spray, the air is humidified and may be heated, cooled or unchanged in temperature. In such a process the air increases in specific humidity and the enthalpy, and the dry bulb temperature will increase or decrease according to the initial temperature of the air and that of the spray. If sufficient water is supplied relative to the mass flow of air, the air will approach saturation at water temperature. Examples of such processes are shown on figure.

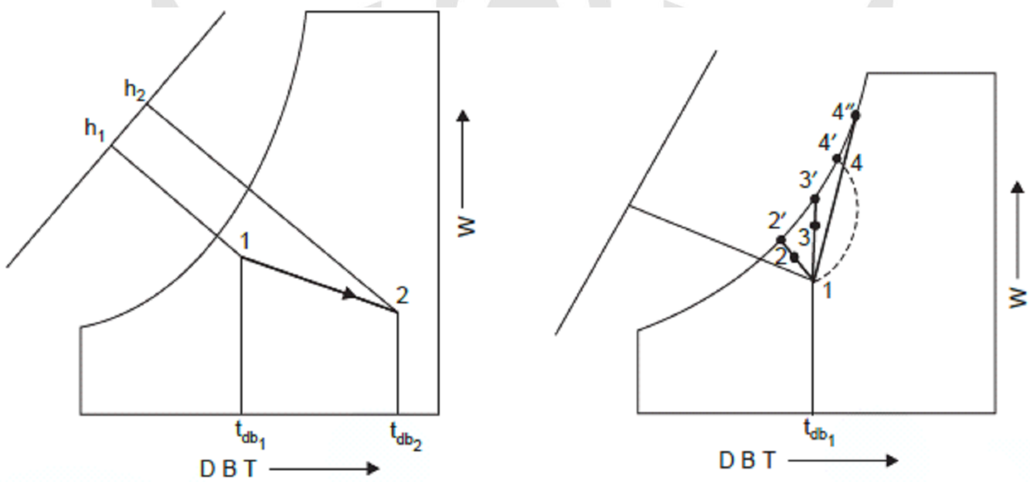


Figure : Heating and dehumidification or Heating and Humidification

Process 1 – 2 : It denotes the cases in which the temperature of the heated spray water is less than the air DBT.

Process 1 – 3 : It denotes the cases in which the temperature is equal to the air DBT.

Process 1 – 4 : It denotes the cases in which a spray temperature is greater than air DBT.

As in the case of adiabatic saturation, the degree to which the process approaches saturation can be expressed in terms of the by-pass factor or a saturating efficiency.

If the water rate relative to the air quantity is smaller, the water temperature will drop significantly during the process. The resultant process will be a curved line such as the dashed 1 – 4 where 4 represents the leaving water temperature.

Note : It is possible to accomplish heating and humidification by evaporation from an open pan of heated water, or by direct injection of heated water or steam. The latter is more common. The process line for it is of little value because the process is essentially an instantaneous mixing of steam and the air. The final state point of the air can be found, however by making a humidity and enthalpy balance for the process. The solution of such a problem usually involves cut-and-try procedure.

31. The work done in a non-flow reversible system (per unit mass) is given by :

$$W = Q - (u_0 - u_1)$$
$$= T.ds - (u_0 - u_1)$$
$$= T(s_0 - s_1) - (u_0 - u_1)$$

i.e., $$W = (u_1 - Ts_1) - (u_0 - Ts_0) \ldots$$

The term $(u - T_s)$ is known as Helmholtz function. This gives maximum possible output when the heat Q is transferred at constant temperature and is the case with a very large source.

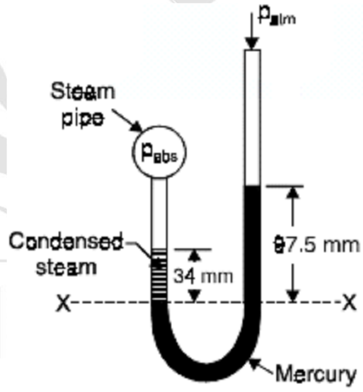
If work against atmosphere is equal to $p_0(v_0 - v_1)$, then the maximum work available,

$$W_{max} = W - \text{Work against atmosphere}$$
$$= W - p_0(v_0 - v_1)$$
$$= (u_1 - Ts_1) - (u_0 - Ts_0) - p_0(v_0 - V_1)$$
$$= (u_1 + p_0 v_1 - Ts_1) - (u_0 + p_0 v_0 - Ts_0)$$
$$= (h_1 - Ts_1) - (h_0 - Ts_0)$$

i.e., $$Wmax = g_1 - g_0$$

where $g = h - T.s$ is known as Gibb's function or free energy function.

32. Equating the pressure in mm of Hg on both arms above the line XX, shown in figure.



We get $$p_{abs} + p_{water} = p_{Hg} + p_{atm}$$

Now, $$p_{water} = \frac{34}{13.6} = 2.5 \text{ mm of Hg}$$

$\therefore$ $$p_{abs} + 2.5 = 97.5 + 760$$
$$p_{abs} = 97.5 + 760 - 2.5 = 855 \text{ mm of Hg}$$
$$= 855 \times p_{hg} \times g \times 10^{-5} \text{ bar}$$
$$= \frac{855}{1000}(m) \times (13.6 \times 1000) \text{ kg/m}^3 \times 9.81 \times 10^{-5}$$
$$= 1.1407 \text{ bar}$$

[PART : C]

33. Inlet swirl is assumed as zero. Total head against the pump is

$$40 + 6 + 2 + 8 = 56 \text{ m}$$

$$u_2 = \pi \times 0.5 \times \frac{1200}{60} = 31.42 \text{ m/s}$$

$$h_m = \frac{gH}{u_2 V_{u2}} = 0.8$$

$$\therefore \quad \frac{9.81 \times 56}{3142 \times V_{u2}} = 0.8$$

Solving $V_{u2} = 21.36$ m/s

To calculate V_f, the velocity triangle is used.

$$\tan\beta_2 = \frac{V_{f2}}{u_2 - V_{u2}}$$

$$\therefore \quad V_{f2} = \tan 30(31.42 - 21.86) = 5.252 \text{ m/s}$$

Flow rate $= \pi D_2 b_2 V_{f2} = \pi \times 0.5 \times 0.15 \times 5.52 = 0.13006 \text{ m}^3/\text{s}$

$$\therefore \quad \text{Power} = \frac{0.13006 \times 10^3 \times 9.81 \times 56}{0.75 \times 10^3} = 95.3 \text{ kW}$$

Considering the water level and the suction level as 1 and 2

$$\frac{p_1}{\gamma} + 0 + 0 = \frac{p_2}{\gamma} + Z + \frac{V_2^2}{2g} + \text{losses}$$

$$10 = \frac{p_2}{\gamma} + 6 + \frac{5.52^2}{2 \times 9.81} + 2, \text{ solving,}$$

$$\frac{p_2}{\gamma} = 0.447 \text{ m absolut (vacuum)}$$

Consider suction side and delivery side, as 2 and 3

$$\frac{p_2}{\gamma} + \frac{V_2^2}{2g} + \frac{u_2 V_{u2}}{g} = \frac{p_3}{\gamma} + \frac{V_3^2}{2g}$$

$$V_3 = \sqrt{21.86^2 + 5.52^2} = 22.55 \text{ m/s}$$

$$\frac{p_3}{\gamma} = 0.447 + \frac{5.52^2}{2 \times 9.81} + \frac{3142 \times 21.86}{9.81} - \frac{22.55^2}{2 \times 9.81}$$

$$= 40.1 \text{ m absolute}$$

34. For a given pump, diameter, blade angles and physical parameters remain the same. Hence, we can derive the following relations. (similar to unit quantities).

$$Q = AV_f \text{ (A is constant)}$$

$$\therefore \quad Q \propto V_f$$

$$V_f \propto u \text{ and } u \propto N$$

$$\therefore \quad Q \propto N \text{ or } \left(\frac{Q}{N} = \text{constant}\right)$$

$$\therefore \quad \frac{Q_2}{Q_1} = \frac{N_2}{N_1} \quad \text{....(1)}$$

For centrifugal pump, $H \propto u^2 \propto N^2$

$$\therefore \quad \frac{H}{N^2} = \text{constant}$$

$$\therefore \quad \frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2 \quad \text{...(2)}$$

$$\therefore \quad \frac{p_2}{p_1} = \left(\frac{N_2}{N_1}\right)^3 \quad \text{...(3)}$$

Using the equation (1), (2) and (3)

$$p_1 = \frac{1000 \times 0.4 \times 9.81 \times 16}{1000 \times 0.82} = 76.57 \text{ kW}$$

$$Q_2 = Q_1 . \frac{N_2}{N_1} = 0.4 \times \frac{1450}{950} = 0.61 \text{ m}^3/\text{s}$$

$$H_2 = H_1 . \left(\frac{N_2}{N_1}\right)^2 = 16 \times \left(\frac{1450}{950}\right)^2 = 37.27 \text{ m}$$

$$P_2 = 76.57 \times \left(\frac{1450}{950}\right)^3 = 272 \text{ kW}$$

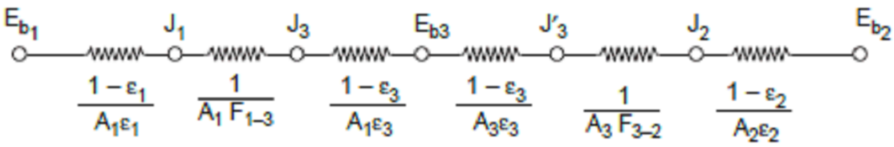
Check: $$P_2 = \frac{1000 \times 0.61 \times 37.27 \times 9.81}{1000 \times 0.82} = 272 \text{ kW}$$

35. Given : $\varepsilon_1 = 0.3$; $\varepsilon_2 = 0.8$; $\varepsilon_3 = 0.04$

Consider all resistances (surface resistances and space resistances) per unit surface area. For steady state heat flow,

$$\frac{E_{b_1} - E_{b_3}}{\left(\frac{1-\varepsilon_1}{\varepsilon_1}\right) + 1 + \left(\frac{1-\varepsilon_3}{\varepsilon_3}\right)} = \frac{E_{b_3} - E_{b_2}}{\left(\frac{1-\varepsilon_3}{\varepsilon_3}\right) + 1 + \left(\frac{1-\varepsilon_2}{\varepsilon_2}\right)}$$

[$\because A_1 = A_2 = A_3 = 1\text{m}^2$ and $F_{1-3}, F_{3-2} = 1$]



or $$\frac{\sigma(T_1^4 - T_3^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_3} - 1} = \frac{\sigma(T_3^4 - T_2^4)}{\frac{1}{\varepsilon_3} + \frac{1}{\varepsilon_2} - 1}$$

or $$\frac{T_1^4 - T_3^4}{\frac{1}{0.3} + \frac{1}{0.04} - 1} = \frac{T_3^4 - T_2^4}{\frac{1}{0.04} + \frac{1}{0.8} - 1}$$

or $$\frac{T_1^4 - T_3^4}{27.33} = \frac{T_3^4 - T_2^4}{25.25}$$

or $$T_1^4 - T_3^4 = \frac{27.33}{25.25}\left(T_3^4 - T_2^4\right) = 1.08\left(T_3^4 - T_2^4\right) = 1.08T_3^4 - 1.08T_2^4$$

or $$2.08T_3^4 = T_1^4 + 1.08T_2^4$$

or $$T_3^4 = \frac{1}{2.08}\left(T_1^4 + 1.08T_2^4\right) = 0.48\left(T_1^4 + 1.08T_2^4\right) \quad ...(1)$$

Q_{12} (Heat flow without shield)

$$= \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{0.3} + \frac{1}{0.8} - 1} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{3.58} \quad ...(2)$$

Q_{13} (Heat flow with shield)

$$= \frac{\sigma\left(T_1^4 - T_3^4\right)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_3} - 1} = \frac{\sigma\left(T_1^4 - T_3^4\right)}{\frac{1}{0.3} + \frac{1}{0.4} - 1} = \frac{\sigma\left(T_1^4 - T_3^4\right)}{27.33} \quad ...(3)$$

$\therefore$ Percentage reduction in heat flow due to shield

$$= \frac{Q_{12} - Q_{13}}{Q_{12}} = 1 - \frac{Q_{13}}{Q_{12}} = 1 - \frac{\frac{\sigma\left(T_1^4 - T_3^4\right)}{27.33}}{\frac{\sigma\left(T_1^4 - T_2^4\right)}{3.58}}$$

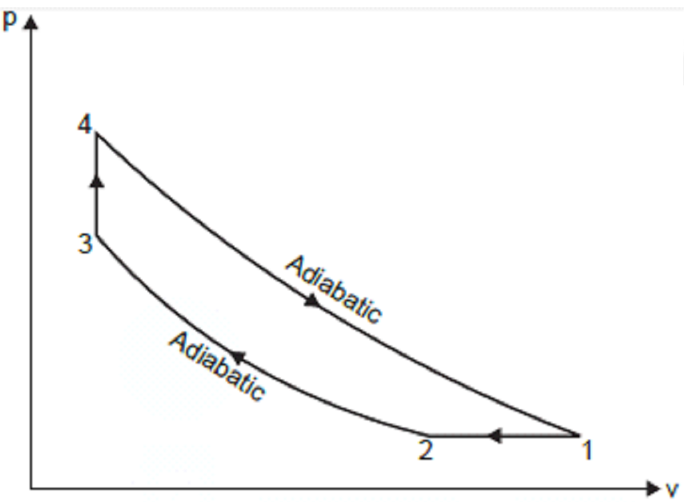
$$= 1 - \frac{3.58}{27.33}\left[\frac{T_1^4 - T_3^4}{T_1^4 - T_2^4}\right] = 1 - 0.131\left[\frac{T_1^4 - 0.48\left(T_1^4 + 1.08T_2^4\right)}{T_1^4 - T_2^4}\right]$$

$$= 1 - 0.131\left[\frac{T_1^4 - 0.48\left(T_1^4 + 0.52T_2^4\right)}{T_1^4 - T_2^4}\right]$$

$$= 1 - 0.131\left[\frac{0.52\left(T_1^4 - T_2^4\right)}{\left(T_1^4 - T_2^4\right)}\right]$$

$= 1 - 0.131 \times 0.52 = 0.932$ or 93.2%

36. This cycle consists of two adiabatics, a constant volume and a constant pressure process. p-V diagram of this cycle is shown in figure. It consists of the following four operations :



(i) 1 – 2 → Heat rejection at constant pressure

(ii) 2 – 3 → Adiabatic compression

(iii) 3 – 4 → Addition of heat at constant volume

(iv) 4 – 1 → Adiabatic expansion

Considering 1 kg of air

$$\text{Compression ratio} = \frac{v_2}{v_3} = a$$

$$\text{Expansion ratio} = \frac{v_1}{v_4} = r$$

$$\text{Heat supplied at constant volume} = c_v(T_4 - T_3)$$

$$\text{Heat rejected} = c_v(T_1 - T_2)$$

$$\text{Work done} = \text{Heat supplied} - \text{Heat rejected}$$

$$= c_v(T_4 - T_3) - c_v(T_1 - T_2)$$

$$\eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_4 - T_3) - c_p(T_1 - T_2)}{c_v(T_4 - T_3)}$$

$$= 1 - \gamma.\frac{(T_1 - T_2)}{(T_4 - T_3)} \quad ...(1)$$

During adaiabatic compression 2 – 3

$$\frac{T_3}{T_2} = \left(\frac{v_2}{v_3}\right)^{\gamma-1} = (\alpha)^{\gamma-1}$$

or $$T_3 = T_2(\alpha)^{\gamma-1} \quad ...(2)$$

During constant pressure operation 1 – 2,

$$\frac{v_1}{T_1} = \frac{v_2}{T_2}$$

or $$\frac{T_2}{T_1} = \frac{v_2}{v_1} = \frac{\alpha}{r} \qquad \left(\frac{v_2}{v_1} = \frac{v_2}{v_3} \times \frac{v_3}{v_1} = \frac{v_2}{v_3} \times \frac{v_4}{v_1} = \frac{\alpha}{r}\right) \quad ...(3)$$

During adiabatic expansion 4 – 1,

$$\frac{T_4}{T_1} = \left(\frac{v_1}{v_4}\right)^{\gamma-1} = (r)^{\gamma-1}$$

$$T_1 = \frac{T_4}{(r)^{\gamma-1}} \quad ...(4)$$

Putting the value of T_1 in equation (3), we get

$$T_2 = \frac{T_4}{(r)^{\gamma-1}} \cdot \frac{\alpha}{r} = \frac{\alpha T_4}{r^\gamma} \quad ...(5)$$

Substituting the value of T_2 in equation (2), we get

$$T_3 = \frac{\alpha T_4}{r^\gamma}(\alpha)^{\gamma-1} = \left(\frac{\alpha}{r}\right)^\gamma . T_4$$

Finally putting the values of T_1, T_2 and T_3 in equation (1), we get

$$\eta = 1 - g\left(\frac{\frac{T_4}{r^{\gamma-1}} - \frac{\alpha T_4}{(r)^{\gamma}}}{T_4 - \left(\frac{\alpha}{r}\right)^{\gamma}.T_4}\right) = 1 - \gamma\left(\frac{r-\alpha}{r^{\gamma}-\alpha^{\gamma}}\right)$$

Hence, air standard efficiency = $1 - \gamma\left(\frac{r-\alpha}{r^{\gamma}-\alpha^{\gamma}}\right)$

36. Given :

$Q = 10$ TR

$p_1 = 0.9$ bar

$T_1 = 10^{o}C = 10 + 273 = 283$ K bar

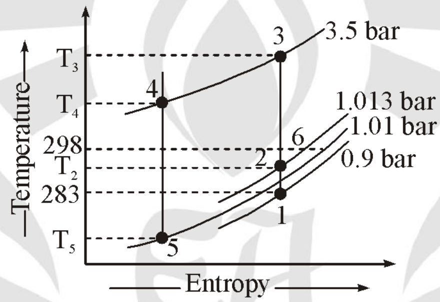
$p_5 = p_6 = 1.01$ bar

$T_6 = 25^{o}C = 25 + 273 = 298$ K

$p_3 = 3.5$ bar

(i) Power required to take the load of cooling in the cabin :

First of all, let us find the mass of air (m_a) required for the refrigeration purpose. Since the compressions and expansions are isentropic, therefore the various processes on the T-s diagram are as shown in figure.



Let, T_2 = Temperature of air at the end of ramming or entering the main compressor.

T_3 = Temperature of air leaving the main compressor after isentropic compression.

T_4 = Temperature of air leaving the heat exchanger.

and T_5 = Temperature of air leaving the cooling turbine.

We know that $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.013}{0.9}\right)^{\frac{1.4-1}{1.4}} = (1.125)^{0.286} = 1.034$

$\therefore$ $T_2 = T_1 \times 1.034 = 283 \times 1.034 = 292.6$ K

Similarly, $\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.5}{1.013}\right)^{\frac{1.4-1}{1.4}} = (3.45)^{0.286} = 1.425$

$\therefore$ $T_3 = T_2 \times 1.425 = 292.6 \times 1.425 = 417 \text{ K} = 144^{o}C$

Since the temperature of air is reduced by 50°C in the heat exchanger, therefore temperature f air leaving the heat exchanger.

$T_4 = 144 - 50 = 94^oC = 367$ K

We know that $\frac{T_5}{T_4} = \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.01}{3.5}\right)^{\frac{1.4-1}{1.4}} = (0.288)^{0.286} = 0.7$

$\therefore$ $T_5 = T_4 \times 0.7 = 367 \times 0.7 = 257$ K

We know that mass of air required for the refrigeration purpose,

$$m_d = \frac{210Q}{c_p(T_6 - T_5)} = \frac{210 \times 10}{1(298 - 257)} = 51.2 \text{ kg/min}$$

(Taking c_p for air = 1 kJ/kg K)

$\therefore$ Power required to take the load of cooling in the cabin,

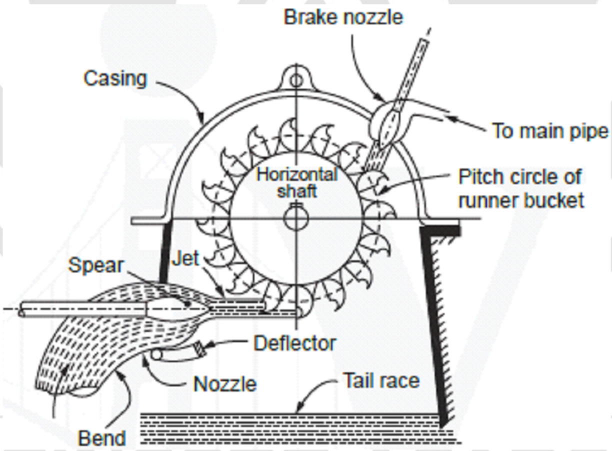
$$p = \frac{m_a c_p(T_3 - T_2)}{60} = \frac{51.2 \times (417 - 292.6)}{60} = 106 \text{ kW}$$

(ii) C.O.P. of the system :

We know that C.O.P. of the system

$$= \frac{210Q}{P \times 60} = \frac{210 \times 10}{106 \times 60} = 0.33$$

38. This is the only type used in high head power plants. This type of turbine was developed and patented by L.A. Pelton in 1889 and all the type of turbines are called by his name to honour him. A sectional view of a horizontal axis Pelton turbine is shown in figure.



The main components are

- The runner with the (vanes) buckets fixed on the periphery of the same
- The nozzle assembly with control spear and deflector
- Brake nozzle
- The casing.

The rotor or runner consists of a circular disc, fixed on suitable shaft, made of cast or forged steel. Buckets are fixed on the periphery of the disc. The spacing of the buckets is decided by the runner diameter and jet diameter and is generally more than 15 in number. These buckets in small sizes may be cast integral with the runner. In larger sizes it is bolted to the runner disc.

The buckets are also made of special materials and the surfaces are well polished. A view of a bucket is shown in figure with relative dimensions indicated in the figure. Originally spherical buckets were used and pelton modified the buckets to the present shape.

It is formed in the shape of two half ellipsoids with a splilter connecting the two. A cut is made in the lip to facilitate all the water in the jet to usefully impinge on the buckets. This avoids interference of the incoming bucket on the jet impinging on the previous bucket. Equations are available to calculate the number of buckets on a wheel. The number of buckets.

$$Z = \left(\frac{D}{2d}\right) + 15$$

Where, D = The runner diameter

and d = The jet diameter.

The nozzle and controlling spear and deflector assembly

The head is generally constant and the jet velocity is thus constant. A fixed ratio between the jet velocity and runner peripheral velocity is to be maintained for best efficiency. The nozzle is designed to satisfy the need. But the load on the turbine will often fluctuate and some times sudden changes in load can take place due to electrical circuit tripping. The velocity of the jet should not be changed to meet the load fluctuation due to frequency requirements. The quantity of water flow only should be changed to meet the load fluctuation. A governor moves to and fro a suitably shaped spear placed inside the nozzle assembly in order to change the flow rate at the same time maintaining a compact circular jet.

When load drops suddenly, the water flow should not be stopped suddenly. Such a sudden action will cause a high pressure wave in the penstock pipes that may cause damage to the system. To avoid this a deflector as shown in figure 14.6.3 is used to suddenly play out and deflect the jet so that the jet bypasses the buckets. Meanwhile the spear will move at the safe rate and close the nozzle and stop the flow. The deflector will than move to the initial position.

Even when the flow is cut off, it will take a long time for the runner to come to rest due to the high inertia. To avoid this a braking jet is used which directs a jet in the opposite direction and stops the rotation. The spear assembly with the deflector is shown in figure. Some other methods like auxiliary waste nozzle and tilting nozzle are also used for speed regulation. The first wastes water and the second is mechanically complex. In side the casing the pressure is atmospheric and hence no need to design the casing for pressure. It mainly serves the purpose of providing a cover and deflecting the water downwards. The casing is cast in two halves for case of assembly. The casing also supports the bearing and as such should be sturdy enough to take up the load.

When the condition is such that the specific speed indicates more than one jet, a vertical shaft system will be adopted. In this case the shaft is vertical and a horizontal nozzle ring with several nozzle is used. The jets in this case should not interfere with each other.

Generally the turbine directly drives the generator. The speed of the turbine is governed by the frequency of AC. Power used in the region. The product of the pairs of poles used in the generator and the speed in rps gives the number of cycles per second. Steam turbines operate at 3000 rpm or 50 rps in the areas where the AC frequency is 50 cycles per second. Hydraulic turbines handle heavier fluid and hence cannot run at such speeds. In many cases the speed in the range to 500 rpm. As the water flows out on both sides equally axial thrust is minimal and heavy thrust bearing is not required.

39. The PI terms of interest are the head coefficient, power coefficient scale and $\frac{Q}{\omega D^3}$ called flow coefficient ($\omega \propto N$). Considering flow coefficient, denoting the larger machines as 1 and the smaller as 2,

$$\frac{Q_1}{\omega_1 D_1^3} = \frac{Q_2}{\omega_2 D_2^3}$$

$$\therefore \qquad Q_2 = Q_1 \frac{\omega_2}{\omega_1}\frac{D_2^3}{D_1^3} = 200 \times \frac{1450}{950}\left(\frac{20}{30}\right)^3 = 90.45 \ l/s$$

Considering head coefficient, (g being common)

$$\frac{gh_1}{\omega_1^2 D_1^2} = \frac{gh_2}{\omega_2^2 D_2^2}$$

$$\therefore \qquad h_2 = h_1\left(\frac{\omega_2}{\omega_1}\right)^2\left(\frac{D_2}{D_1}\right)^2$$

$$\therefore \qquad h_2 = 25 \times \left[\frac{1450}{95}\right]^2\left(\frac{20}{30}\right)^2 = 25.885 \text{ m}$$

Consider power coefficient

$$\frac{P_1}{\rho_1\omega_1^3 D_1^5} = \frac{P_1}{\rho_1\omega_1^3 D_1^5} \text{ as } \rho_1 = \rho_2$$

$$\frac{P_2}{P_1} = \left(\frac{\omega_2}{\omega_1}\right)^3\left(\frac{D_2}{D_1}\right)^5 = \left(\frac{1450}{950}\right)^3\left(\frac{20}{30}\right)^5 = 0.468$$

As efficiencies should be the same,

$$Q_1\rho_1 h_1 = Q_2\rho_2 h_2, \text{ with } \rho_1 = \rho_2$$

$$0.200 \times 25 = 0.09045 \times \frac{25.885}{0.468}$$

$$5.00 = 5.00 \text{ (checks)}$$

$$\text{Specific speed} = N\frac{\sqrt{Q}}{H^{3/4}} = 1450\frac{\sqrt{0.09045}}{25.885^{3/4}} = 38 \text{ (demensional)}$$

$$\text{For larger pump, specific speed} = 950\frac{\sqrt{0.2}}{25^{3/4}} = 38 \text{ checks.}$$

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