

## DETAILS EXPLANATIONS

## ME: Paper-2 (Paper-2) [Full Syllabus]

## [PART : A]

1. *Homogeneous System :*

A system which consists of a single phase is termed as homogeneous system.

*Examples :* Mixture of air and water vapour, water plus nitric acid and octane plus heptane.

*Heterogeneous System :*

A system which consists of two or more phases is called a heterogeneous system.

*Examples :* Water plus steam, ice plus water and water plus oil.

2. There are certain quantities which cannot be located on a graph by a point but are given by the area or so, on that graph. In that case, the area on the graph, pertaining to the particular process, is a function of the path of the process. Such quantities are called path functions.

*Examples :* Heat, work etc.

Heat and work are inexact differentials. Their change cannot be written as difference between their end states.

3. Diameter of the piston  $d = 10 \text{ cm} (= 0.1 \text{ m})$   
Force applied on the piston,  $F = 1000 \text{ N}$

$$\begin{aligned} \text{Pressure on the piston, } p &= \frac{\text{Force}}{\text{Area}} = \frac{F}{A} = \frac{1000}{\pi/4 \times (0.1)^2} \\ &= 127307 \text{ N/m}^2 = 127.307 \text{ kN/m}^2 \end{aligned}$$

4. A pure substance is a system which is (i) homogeneous in composition, (ii) homogeneous in chemical aggregation, and (iii) invariable in chemical aggregation. — “Homogeneous in composition” means that the composition of each part of the system is the same as the composition of every other part. “Composition means the relative proportions of the chemical elements into which the sample can be analysed. It does not matter how these elements are combined.
5. It is the amount of heat required to convert water at a given temperature and pressure into steam at the same temperature and pressure. It is expressed by the symbol  $h_{ig}$  and its value is available from steam tables. The value of latent heat is not constant and varies according to pressure variation.
6. The first law of thermodynamics states the general principle of the conservation of energy. Energy is neither created nor destroyed, but only gets transformed from one form to another. There can be no machine which would continuously supply mechanical work without some form of energy disappearing simultaneously. Such a fictitious machine is called a perpetual motion machine of the first kind, or in brief, PMM 1. A PMM 1 is thus impossible

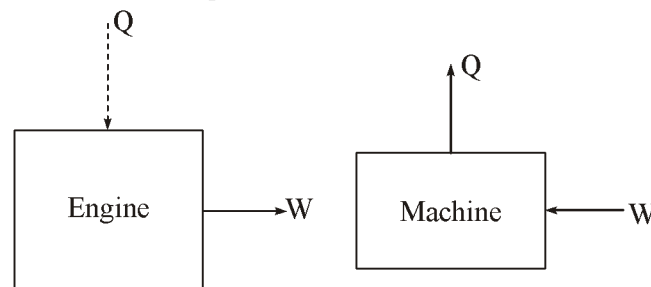


Figure : A PMM-1

Figure : The converse of PMM-1

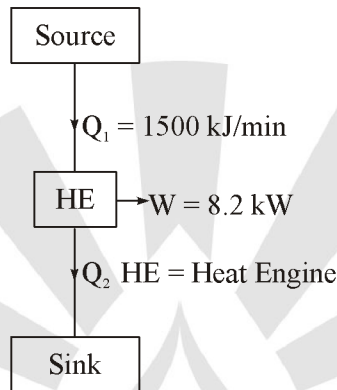
The converse of the above statement is also true, i.e., there can be no machine which would continuously consume work without some other form of energy appearing simultaneously

7. The following assumptions are made in the system analysis :

- (i) The mass flow through the system remains constant.
- (ii) Fluid is uniform in composition.
- (iii) The only interaction between the system and surroundings are work and heat.
- (iv) The state of fluid at any point remains constant with time.
- (v) In the analysis only potential, kinetic and flow energies are considered.

8. Heat received by the heat engine,  $Q_1 = 1500 \text{ kJ/min} = \frac{1500}{60} = 25 \text{ kJ/s}$

Work output,  $W = 8.2 \text{ kW} = 8.2 \text{ kJ/s}$



(i) Thermal efficiency,  $h_{th} = \frac{W}{Q_1} = \frac{8.2}{25} = 0.328 = 32.8\%$

Hence, thermal efficiency = 32.8%

(ii) Rate of heat rejection,

$$Q_2 = Q_1 - W = 25 - 8.2 = 16.8 \text{ kJ/s}$$

Hence, the rate of heat rejection = 16.8 kJ/s.

9.  $s = f(T, p)$

$$Tds = T \left( \frac{\partial s}{\partial T} \right)_p dT + T \left( \frac{\partial s}{\partial p} \right)_T dp \quad \dots(1)$$

Where,  $c_p = T \left( \frac{\partial s}{\partial T} \right)_p$

Also,  $\left( \frac{\partial s}{\partial p} \right)_T = - \left( \frac{\partial v}{\partial T} \right)_p \dots$  Maxwell relation

Substituting these in equation (1) we get

$$T_{ds} = c_p dT - T \left( \frac{\partial v}{\partial T} \right)_p dp$$

10. The relationship between the partial pressures of the constituents is expressed by Dalton's law, as follows :

- The pressure of a mixture of gases is equal to the sum of the partial pressures of the constituents.
- The partial pressure of each constituent is that pressure which the gas would exert if it occupied alone that volume occupied by the mixtures at the same temperature.

11. Stoichiometric (or chemically correct) mixture of air and fuel is one that contains just sufficient oxygen for complete combustion of the fuel.

- A weak mixture is one which has an excess of air.
- A rich mixture is one which has a deficiency of air.

The percentage of excess air is given as under :

$$\% \text{age excess air} = \frac{\text{Actual A/F ratio} - \text{Stoichiometric A/F ratio}}{\text{Stoichiometric A/F ratio}}$$

(where A and F denote air and fuel respectively)

12. The primary object of superheating steam and supplying it to the primemovers is to avoid too much wetness at the end of expansion. Use of inadequate degree of superheat in steam engines would cause greater condensation in the engine cylinder ; while in case of turbines the moisture content of steam would result in undue blade erosion. The maximum wetness in the final condition of steam that may be tolerated without any appreciable harm to the turbine blades is about 12 per cent. Broadly each 1 per cent of moisture in steam reduces the efficiency of that part of the turbine in which wet steam passes by 1 per cent to 1.5 per cent and in engines about 2 per cent.

13. The ratio of Thermal conductivity to Thermal capacity is known as thermal diffusivity.

$$\alpha = \frac{k}{\rho \cdot c} = \frac{\text{Thermal conductivity}}{\text{Thermal capacity}}$$

The quantity,  $\alpha = \frac{k}{\rho \cdot c}$  is known as thermal diffusivity.

The larger the value of  $\alpha$ , the faster will the heat diffuse through the material and its temperature will change with time. This will result either due to a high value of thermal conductivity  $k$  or a low value of heat capacity  $\rho \cdot c$ . A low value of heat capacity means the less amount of heat entering the element would be absorbed and used to raise its temperature and more would be available for onward transmission. Metals and gases have relatively high value of  $\alpha$  and their response to temperature changes is quite rapid. The non-metallic solids and liquids respond slowly to temperature changes because of their relatively small value of thermal diffusivity.

14. Temperature of the inner surface of the wall,  $t_1 = 60^\circ\text{C}$

Temperature of the outer surface of the wall,  $t_2 = 35^\circ\text{C}$

The thickness of the wall,  $L = 220 \text{ mm} = 0.22 \text{ m}$

Thermal conductivity of the brick,  $k = 0.51 \text{ W/m}^\circ\text{C}$

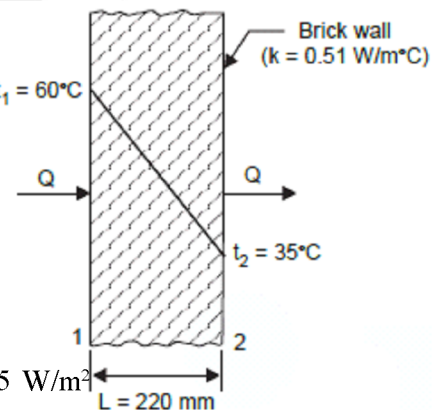
Rate of heat transfer per  $\text{m}^2$ ,  $q$  :

Rate of heat transfer per unit area,

$$q = \frac{Q}{A} = \frac{k(t_1 - t_2)}{L}$$

or

$$q = \frac{0.51 \times (60 - 35)}{0.22} = 57.95 \text{ W/m}^2$$



15. **Factors affecting thermal conductivity :**

Some of the important factors which affect thermal conductivity ( $k$ ) of the insulators (the value of  $k$  should be always low to reduce the rate of heat flow) are as follows :

- **Temperature :** For most of the insulating materials, the value of  $k$  increases with increase in temperature.
- **Density :** There is no mathematical relationship between  $k$  and  $\rho$  (density). The common understanding that high density insulating materials will have higher values of  $k$  is not always true.
- **Direction of heat flow :** For most of the insulating materials (except few like wood) the effect of direction of heat flow on the values of  $k$  is negligible.

- **Moisture** : It is always considered necessary to prevent ingress of moisture in the insulating materials during service, it is however difficult to find the effect of moisture on the values of  $k$  of different insulating materials.
  - **Air pressure** : It has been found that the value of  $k$  decreases with decrease in pressure.
  - **Convection in insulators** : The value of  $k$  increases due to the phenomenon of convection in insulators.
16. If the radiative properties,  $\alpha$ ,  $\rho$ ,  $\tau$  of a body are assumed to be uniform over the entire wavelength spectrum, then such a body is called gray body. A gray body is also defined as one whose absorptivity of a surface does not vary with temperature and wavelength of the incident radiation  
 $[\alpha = (\alpha)\lambda = \text{constant}]$   
 A coloured body is one whose absorptivity of a surface varies with the wavelength of radiation  
 $[\alpha \neq (\alpha)\lambda]$ .
17. "Entropy is a function of a quantity of heat which shows the possibility of conversion of that heat into work. The increase in entropy is small when heat is added at a high temperature and is greater when heat addition is made at a lower temperature. Thus for maximum entropy, there is minimum availability for conversion into work and for minimum entropy there is maximum availability for conversion into work."
18. (i) A finite volume of gas contains large number of molecules.  
 (ii) The collision of molecules with one another and with the walls of the container are perfectly elastic.  
 (iii) The molecules are separated by large distances compared to their own dimensions.  
 (iv) The molecules do not exert forces on one another except when they collide.  
 As long as the above assumptions are valid the behaviour of a real gas approaches closely that of an ideal gas.
19. Diameter of the wire,  $d = 1.5 \text{ mm} = 0.0015 \text{ m}$   
 Length of the wire,  $l = 150 \text{ mm} = 0.15 \text{ m}$   
 $\therefore$  Surface area of the wire (exposed to heat transfer),  
 $A = \pi d l = \pi \times 0.0015 \times 0.15 = 7.068 \times 10^{-4} \text{ m}^2$   
 Wire surface temperature,  $t_s = 120^\circ\text{C}$   
 Water temperature,  $t_f = 100^\circ\text{C}$   
 Convective heat transfer coefficient,  $h = 4500 \text{ W/m}^2\text{C}$   
 Electric power to be supplied :  
 Electric power which must be supplied = Total convection loss (Q)  
 $\therefore$   

$$Q = hA(t_s - t_f)$$

$$= 4500 \times 7.068 \times 10^{-4} (120 - 100) = 63.6 \text{ W.}$$
20. **Assumptions** :
- The gas in the engine cylinder is a perfect gas i.e., it obeys the gas laws and has constant specific heats.
  - The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures i.e., the molecular weight of cylinder gas is 29.  
 $c_p = 1.005 \text{ kJ/kg-K,}$   
 $c_v = 0.718 \text{ kJ/kg-K.}$
  - The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.
  - No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in contact with cylinder at appropriate points during the process.
  - The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.



**[PART : B]**

## 21. Irreversibility

$$I = W_{\max} - W_{\text{act}}$$

$$W_{\max} = \text{Change in internal energy} - T_0 \times \text{Change in entropy}$$

or

$$W_{\max} = (u_2 - u_1) - T_0(s_2 - s_1) = W_{\text{rev}}$$

or

$$W_{\max} = c_v(T_2 - T_1) - T_0[c_p \ln(T_2/T_1) - R \ln(p_2/p_1)]$$

$$= 0.716(370 - 300) - 293 \times [1.005 \ln(370/300) - 0.287 \ln(6.8/1)]$$

or

$$W_{\max} = -149.53 \text{ kJ/kg} = W_{\text{rev}}$$

(negative sign indicates that work is done on air)

The index of compression 'n' is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{[(n-1)/n]}$$

or

$$\frac{n-1}{n} = \frac{\ln(T_2 - T_1)}{\ln(p_2 - p_1)} = \frac{\ln(370/300)}{\ln(6.8/1)}$$

or

$$n = 1.123$$

$$W_{\text{actual}} = \frac{mR(T_1 - T_2)}{n-1} = \frac{1 \times 0.287(300 - 370)}{1.123 - 1} = -163.33 \text{ kJ/kg}$$

$$I = W_{\text{rev}} - W_{\text{act}}$$

$$= -149.53 - (-163.33) = 13.8 \text{ kJ/kg}$$

22. From p-v-T measurements, we find that an equation of state is not the only useful information which can be obtained. When the experimental results are plotted as a series of constant pressure lines on a v-T diagrams, as in figure(a), the slope of a constant pressure line at any given state is  $\left(\frac{\partial v}{\partial T}\right)_p$ . If the gradient is divided by the volume at that state, we have a value of a property of the substance called its co-efficient of cubical expansion  $\beta$ . That is,

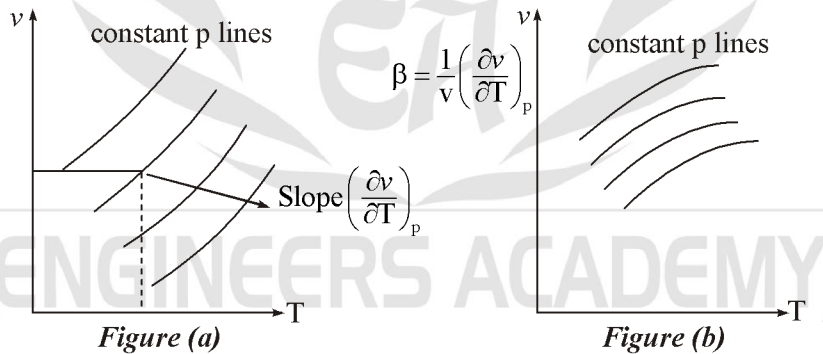


Figure : Determination of co-efficient of expansion from p-v-T data

$$\beta = \frac{1}{v} \left(\frac{\partial v}{\partial T}\right)_p$$

Value of  $\beta$  can be tabulated for a range of pressures and temperatures, or plotted graphically as in figure (b). For solids and liquids over the normal working range of pressure and temperature, the variation of p is small and can often be neglected. In tables of physical properties  $\beta$  is usually quoted as an average value over a small range of temperature, the pressure being atmospheric. This average coefficient may be symbolised by  $\beta$  and it is defined by

$$\beta = \frac{v_2 - v_1}{v_1(T_2 - T_1)}$$

23. Change in entropy during constant volume process

$$= mc_v \ln \left( \frac{T_2}{T_1} \right) \quad \dots(1)$$

Change in entropy during polytropic process ( $pv^n = \text{constant}$ )

$$= mc_v \left( \frac{\gamma - n}{n - 1} \right) \ln \left( \frac{T_2}{T_1} \right) \quad \dots(2)$$

For the same entropy, equation (1) and (2), we have

$$\frac{\gamma - n}{n - 1} = 1 \text{ or } (\gamma - n) = (n - 1)$$

or  $2n = \gamma + 1$

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

24. When air passes over a dry surface which is at a temperature greater than its (air) dry bulb temperature, it undergoes sensible heating. Thus the heating can be achieved by passing the air over heating coil like electric resistance heating coils or steam coils. During such a process, the specific humidity remains constant but the dry bulb temperature rises and approaches that of the surface. The extent to which it approaches the mean effective surface temperature of the coil is conveniently expressed in terms of the equivalent by-pass factor. 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. Thus on figure., air at temperature  $t_{db1}$ , passes over a heating coil with an average surface temperature  $t_{db3}$  and leaves at temperature  $t_{db2}$ . The by-pass factor is expressed as follows :

$$BF = \frac{t_{db3} - t_{db2}}{t_{db3} - t_{db1}}$$

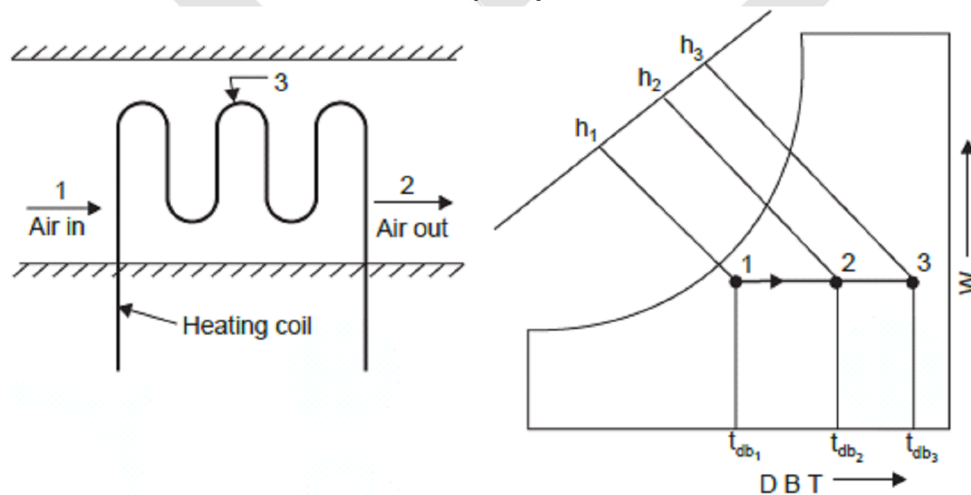


Figure : Sensible Heating

Or in terms of lengths on the chart, it is length 2 - 3 / length 1 - 3. The value of the by-pass factor is a function of coil design and velocity. The heat added to the air can be obtained directly from the entering and leaving enthalpies ( $h_2 - h_1$ ) or it can be obtained from the humid specific heat multiplied by the temperature difference ( $t_{db2} - t_{db1}$ ). In a complete air conditioning system the preheating and reheating of air are among the familiar examples of sensible heating.

*Note: 'By-pass factor' can be considered to represent the fraction of air which does not come into contact with coil surface.*

25. Dry bulb temperature,  $t_{db} = 17^\circ\text{C}$   
 Relative humidity,  $\phi = 60\%$   
 Barometric or total pressure,  $p_t = 1.01325 \text{ bar}$

**Specific humidity,  $W$  :**

Corresponding to  $17^\circ\text{C}$ , from steam tables,

$$p_{vs} = 0.0194 \text{ bar}$$

Also, 
$$\phi = \frac{p_v}{p_{vs}}$$

i.e., 
$$0.6 = \frac{p_v}{0.0194}$$

$\therefore$  
$$p_v = 0.6 \times 0.0194 = 0.01164 \text{ bar.}$$

specinc humidity, 
$$W = \frac{0.622p_v}{p_t - p_v} = \frac{0.622 \times 0.01164}{1.01325 - 0.01164}$$
  

$$= 0.00723 \text{ kg/kg of dry air}$$

**Dew point temperature,  $t_{dp}$  :**

If the air is cooled at constant pressure the vapour will begin to condense at the saturation temperature corresponding to 0.01164 bar By interpolation from steam tables, the dew point temperature,  $t_{dp}$  is then

$$t_{dp} = 9 + (10 - 9) \times \frac{0.01164 - 0.01150}{0.01230 - 0.01150} = 9.18^\circ\text{C}$$

26. • **Coal** : Its main constituents are carbon, hydrogen, oxygen, nitrogen, sulphur, moisture and ash. Coal passes through different stages during its formation from vegetation. These stages are enumerated and discussed below :
- **Plant debris** : Peat—Lignite—Brown coal—sub-bituminous coal—Bituminous coal—Semibituminous coal—Semi-anthracite coal—Anthracite coal—Graphite.
  - **Peat** : It is the first stage in the formation of coal from wood. It contains huge amount of moisture and therefore it is dried for about 1 to 2 months before it is put to use. It is used as a domestic fuel in Europe and for power generation in Russia. In India it does not come in the categories of good fuels.
  - **Lignites and brown coals** : These are intermediate stages between peat and coal. They have a woody or often a clay like appearance associated with high moisture, high ash and low heat contents. Lignites are usually amorphous in character and impose transport difficulties as they break easily. They burn with a smoky flame. Some of this type are suitable for local use only.
  - **Bituminous coal** : It burns with long yellow and smoky flames and has high percentages of volatile matter. The average calorific value of bituminous coal is about 31350 kJ/kg. It may be of two types, namely caking or noncaking.
  - **Semi-bituminous coal** : It is softer than the anthracite. It burns with a very small amount of smoke. It contains 15 to 20 per cent volatile matter and has a tendency to break into small sizes during storage or transportation.
  - **Semi-anthracite** : It has less fixed carbon and less lustre as compared to true anthracite and gives out longer and more luminous flames when burnt.
  - **Anthracite** : It is very hard coal and has a shining black lustre. It ignites slowly unless the furnace temperature is high. It is non-caking and has high percentage of fixed carbon. It burns either with very short blue flames or without flames. The calorific value of this fuel is high to the tune of 35500 kJ/kg and as such is very suitable for steam generation.

- **Wood charcoal** : It is obtained by destructive distillation of wood. During the process the volatile matter and water are expelled. The physical properties of the residue (charcoal), however depends upon the rate of heating and temperature.
- **Coke** : It consists of carbon, mineral matter with about 2% sulphur and small quantities of hydrogen, nitrogen and phosphorus. It is solid residue left after the destructive distillation of certain kinds of coals. It is smokeless and clear fuel and can be produced by several processes. It is mainly used in blast furnace to produce heat and at the same time to reduce the iron ore.
- **Briquettes** : These are prepared from fine coal or coke by compressing the material under high pressure.

27. 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.

28. Though Carnot cycle is simple (thermodynamically) and has the highest thermal efficiency for given values of  $T_1$  and  $T_2$ , yet it is extremely difficult to operate in practice because of the following reasons :

- It is difficult to compress a wet vapour isentropically to the saturated state as required by the process 3 - 4.
- It is difficult to control the quality of the condensate coming out of the condenser so that the state '3' is exactly obtained.
- The efficiency of the Carnot cycle is greatly affected by the temperature  $T_1$  at which heat is transferred to the working fluid. Since the critical temperature for steam is only  $374^\circ\text{C}$ , therefore, if the cycle is to be operated in the wet region, the maximum possible temperature is severely limited.
- The cycle is still more difficult to operate in practice with superheated steam due to the necessity of supplying the superheat at constant temperature instead of constant pressure (as it is customary).

In a practical cycle, limits of pressure and volume are far more easily realised than limits of temperature so that at present no practical engine operates on the Carnot cycle, although all modern cycles aspire to achieve it.

29. *The Rankine cycle efficiency can be improved by :*

- (i) Increasing the average temperature at which heat is supplied.
- (ii) Decreasing/reducing the temperature at which heat is rejected.

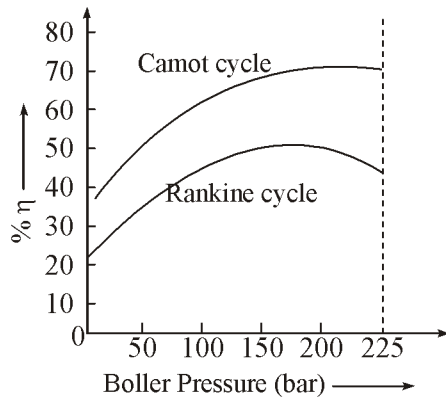


Figure (a)

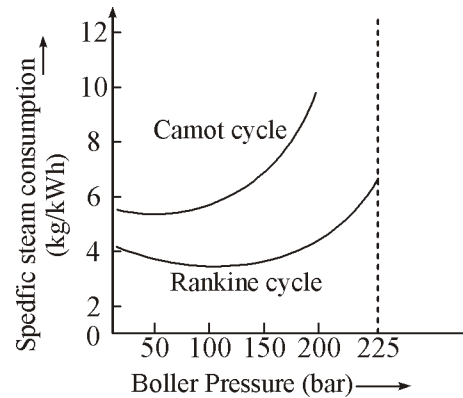


Figure (b)

**Condensation, as discussed below :**

- **Increasing boiler pressure :**

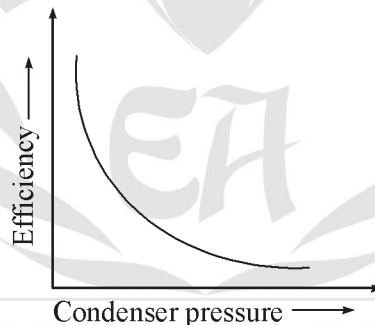
It has been observed that by increasing the boiler pressure (other factors remaining the same) the cycle tends to rise and reaches a maximum value at a boiler pressure of about 166 bar (a).

- **Superheating :**

All other factors remaining the same, if the steam is superheated before allowing it to expand the Rankine cycle efficiency may be increased [figure (b)]. The use of superheated steam also ensures longer turbine blade life because of the absence of erosion from high velocity water particles that are suspended in wet vapour.

- **Reducing condenser pressure :**

The thermal efficiency of the cycle can be amply improved by reducing the condenser pressure [figure (c)] (hence by reducing the temperature at which heat is rejected), especially in high vacuums. But the increase in efficiency is obtained at the increased cost of condensation apparatus.



30. Mean effective pressure,  $p_m = 7.5$  bar  
 Compression ratio,  $r = 12.5$   
 Initial pressure,  $p_t = 1$  bar  
 The mean effective pressure is given by

$$p_m = \frac{p_t r^\gamma [\gamma(\rho - 1) - r^{1-\gamma}(\rho^\gamma - 1)]}{(\gamma - 1)(r - 1)}$$

$$7.5 = \frac{1 \times (12.5)^{1.4} [1.4(\rho - 1) - (12.5)^{1-1.4}(\rho^{1.4} - 1)]}{(1.4 - 1)(12.5 - 1)}$$

$$7.5 = \frac{34.33[1.4\rho - 1.4 - 0.364\rho^{1.4} + 0.364]}{4.6}$$

$$7.5 = 7.46(1.4\rho - 1.036 - 0.364\rho^{1.4})$$

$$1.005 = 1.4\rho - 1.036 - 0.364\rho^{1.4}$$

$$2.04 = 1.4\rho - 0.364\rho^{1.4}$$

or  $0.364\rho^{1.4} - 1.4\rho + 2.04 = 0$



Solving by trial and error method, we get

$$\rho = 2.24$$

$$\therefore \% \text{cut-off} = \frac{\rho - 1}{r - 1} \times 100 = \frac{2.24 - 1}{12.5 - 1} \times 100 = 10.78\%$$

31. **Efficiency Versus Compression Ratio :**

Figure shows the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles at various compression ratios and with given cut-off ratio for the Diesel and Dual combustion cycles. It is evident from the Fig. that the air standard efficiencies increase with the increase in the compression ratio. For a given compression ratio Otto cycle is the most efficient while the Diesel cycle is the least efficient. ( $\eta_{\text{otto}} > \eta_{\text{dual}} > \eta_{\text{diesel}}$ ).

*Note :* The maximum compression ratio for the petrol engine is limited by detonation. In their respective ratio ranges, the Diesel cycle is more efficient than the Otto cycle.

**For the Same Compression Ratio and the Same Heat Input :**

A comparison of the cycles (Otto, Diesel and Dual) on the p-v and T-s diagrams for the same compression ratio and heat supplied is shown in the figure.

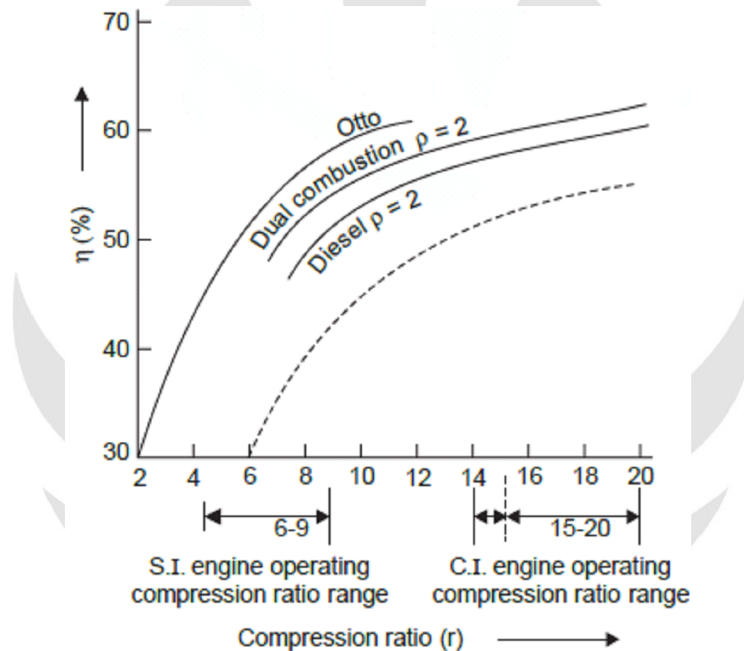


Figure : Comparison of efficiency at various compression ratios

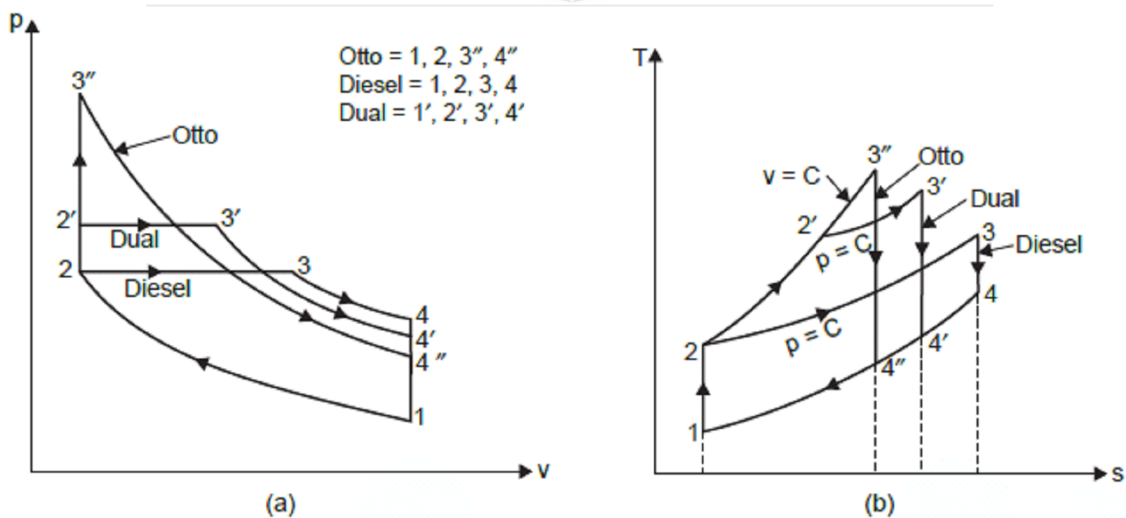


Figure (a) p-v diagram and (b) T-s diagram

We know that, 
$$\eta = 1 - \frac{\text{Heat rejected}}{\text{Heat supplied}}$$

Since all the cycles reject their heat at the same specific volume, process line from, state 4 to 1, the quantity of heat rejected from each cycle is represented by the appropriate area under the line 4 to 1 on the T-s diagram. As is evident from the equation the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and Diesel cycle is the least efficient of the three cycles.

i.e., 
$$\eta_{\text{otto}} > \eta_{\text{dual}} > \eta_{\text{diesel}}$$

32. *The difference between vapour compression and vapour absorption systems :*

S. No.	Particulars	Vapour compression system	Vapour absorption system
1.	Type of energy supplied	Mechanical-a high grade energy	Mainly heat -a low grade energy
2.	Energy supply	Low	High
3.	Wear and tear	More	Less
4.	Performance at part loads	Poor	System not affected by variations of loads.
5.	Suitability	Used where high grade mechanical energy is available	Can also be used at remote places as it can work even with a simple kerosene lamp (of course in small capacities)
6.	Charging of refrigerant	Simple	Difficult
7.	Leakage of refrigerant	More chances	No chance as there is no compressor or any reciprocating component to cause leakage.
8.	Damage	Liquid traces in suction line may damage the compressor	Liquid traces of refrigerant present in piping at the exit of evaporator constitute no danger.

[PART : C]

33. Clausius-Claperyon equation is a relationship between the saturation pressure, temperature, the enthalpy of evaporation, and the specific volume of the two phases involved. This equation provides a basis for calculations of properties in a two-phase region. It gives the slope of a curve separating the two phases in the p-T diagram.

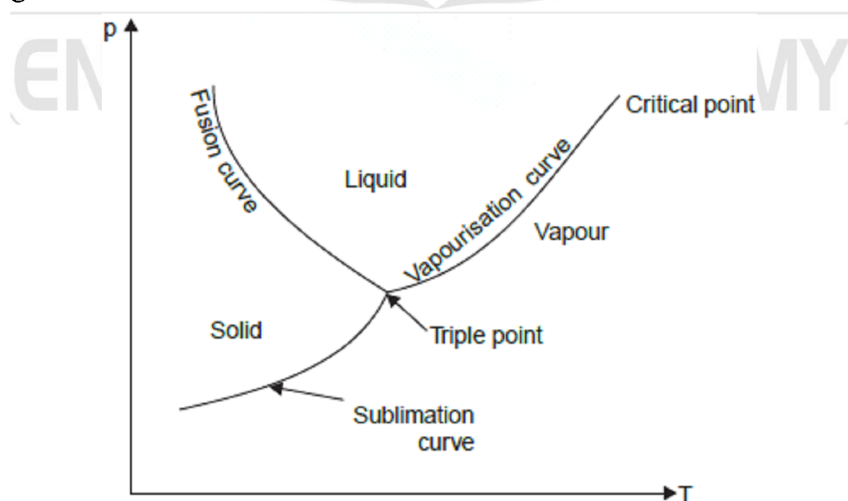


Figure : p-T diagram

The Clausius-Claperyon equation can be derived in different ways. The method given below involves the use of the Maxwell relation :

$$\left(\frac{\partial p}{\partial T}\right)_v = \left(\frac{\partial s}{\partial v}\right)_T$$

Let us consider the change of state from saturated liquid to saturated vapour of a pure substance which takes place at constant temperature. During the evaporation, the pressure and temperature are independent of volume :

$$\left(\frac{\partial p}{\partial T}\right) = \frac{s_g - s_f}{v_g - v_f}$$

Where,

$s_g$  = Specific entropy of saturated vapour

$s_f$  = Specific entropy of saturated liquid

$v_g$  = Specific volume of saturated vapour

and

$v_f$  = Specific volume of saturated liquid

Also,

$$s_g - s_f = s_{fg} = \frac{h_{fg}}{T}$$

and

$$v_g - v_f = v_{fg}$$

Where,

$s_{fg}$  = Increase in specific entropy

$v_{fg}$  = Increase in specific volume

and

$h_{fg}$  = Latent heat added during evaporation at saturation temperature T

$$\frac{dp}{dT} = \frac{s_g - s_f}{v_g - v_f} = \frac{s_{fg}}{v_{fg}} = \frac{h_{fg}}{T \cdot v_{fg}}$$

This is known as **Clausius-Claperyon or Claperyon equation** for evaporation of liquids.

The derivative  $\frac{dp}{dT}$  is the slope of vapour pressure versus temperature curve. Knowing this slope and the specific volume  $v_g$  and  $v_f$  from experimental data, we can determine the enthalpy of evaporation,  $(h_g - h_f)$  which is relatively difficult to measure accurately.

Equation is also valid for the change from a solid to liquid, and from solid to a vapour. At very low pressures, if we assume  $v_g \approx v_{fg}$  and the equation of the vapour is taken as  $p v = RT$ , then equation becomes :

$$\frac{dp}{dT} = \frac{h_{fg}}{T v_g} = \frac{h_{fg} p}{RT^2}$$

or

$$h_{fg} = \frac{RT^2}{p} \frac{dp}{dT}$$

Equation may be used to obtain the enthalpy of vapourisation. This equation can be rearranged as follows:

$$\frac{dp}{p} = \frac{h_{fg}}{R} \cdot \frac{dT}{T^2}$$

Integrating the above equation, we get

$$\int \frac{dp}{p} = \frac{h_{fg}}{R} \int \frac{dT}{T^2}$$

$$\ln \frac{p_2}{p_1} = \frac{h_{fg}}{R} \left[ \frac{1}{T_1} - \frac{1}{T_2} \right]$$

Knowing the vapour pressure  $p_1$  at temperature  $T_1$  we can find the vapour pressure  $p_2$  corresponding to temperature  $T_2$  from equation.

From equation, we see that the slope of the vapour pressure curve is always +ve, since  $v_g > v_f$  and  $h_{fg}$  is always +ve. Consequently, the vapour pressure of any simple compressible substance increases with temperature.

- It can be shown that the slope of the sublimation curve is also +ve for any pure substance.
- However, the slope of the melting curve could be +ve or -ve.
- For a substance that contracts on freezing, such as water, the slope of the melting curve will be negative.

34. Francis turbine is a radial inward flow turbine and is the most popularly used one in the medium head range of 60 to 300 m. Francis turbine was first developed as a purely radial flow turbine by James B. Francis, an American engineer in 1849. But the design has gradually changed into a mixed flow turbine of today.

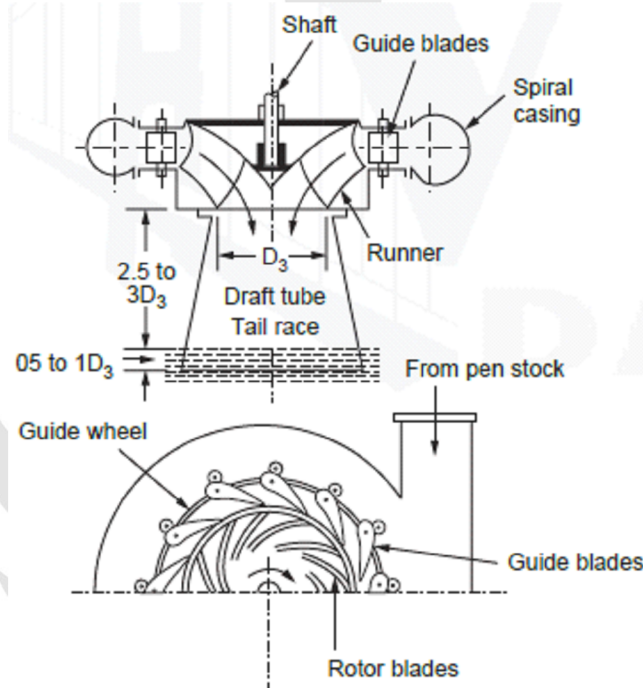


Figure : Typical sectional and front view of a modern Francis turbine.

A sectional view of a typical Francis turbine of today is shown in figure.

The main components are :

- The spiral casing
- Guide vanes
- Runner
- Draft tube
- Governor mechanism.

Most of the machines are of vertical shaft arrangement while some smaller units are of horizontal shaft type.

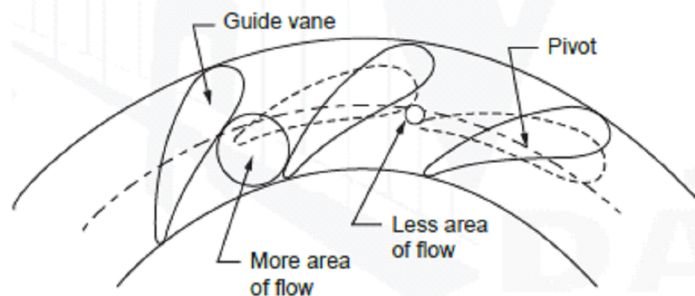
**The spiral casing :**

The spiral casing surrounds the runner completely. Its area of cross section decreases gradually around the circumference. This leads to uniform distribution of water all along the circumference of the runner. Water from the penstock pipes enters the spiral casing and is distributed uniformly to the guide blades placed on the periphery of a circle. The casing should be strong enough to withstand the high pressure.

**Guide Blades :**

Water enters the runner through the guide blades along the circumference. The number of guide blades are generally fewer than the number of blades in the runner. These should also be not simple multiples of the runner blades. The guide blades in addition to guiding the water at the proper direction serves two important functions. The water entering the guide blades are imparted a tangential velocity by the drop in pressure in the passage of the water through the blades. The blade passages act as a nozzle in this aspect. The guide blades rest on pivoted on a ring and can be rotated by the rotation of the ring, whose movement is controlled by the governor. In this way the area of blade passage is changed to vary the flow rate of water according to the load so that the speed can be maintained constant.

The variation of area between guide blades is illustrated in Figure. The control mechanism will be discussed in a later section.



**Figure : Guide Vanee and Giude wheel**

**The Runner :**

The runner is circular disc and has the blades fixed on one side. In high speed runners in which the blades are longer a circular band may be used around the blades to keep them in position.

The shape of the runner depends on the specific speed of the unit. These are classified as (a) slow runner (b) medium speed runner (c) high speed runner and (d) very high speed runner.

The shape of the runner and the corresponding velocity triangles are shown in figure . The development of mixed flow runners was necessitated by the limited power capacity of the purely radial flow runner. A larger exit flow area is made possible by the change of shape from radial to axial flow shape. This reduces the outlet velocity and thus increases efficiency. As seen in the figure the velocity triangles are of different shape for different runners. It is seen from the velocity triangles that the blade inlet angle  $\alpha_1$  changes from acute to obtuse as the speed increases. The guide vane outlet angle  $\alpha_1$  also increases from about  $15^\circ$  to higher values as speed increases. The runner blades are of doubly curved and are complex in shape. These may be made separately using suitable dies and then welded to the rotor. The height of the runner along the axial direction (may be called width also) depends upon the flow rate which depends on the head and power which are related to specific speed. As specific speed increases the width also increase accordingly, the fluid.

**Draft Tube :**

The turbines have to be installed a few meters above the flood water level to avoid inundation. In the case of impulse turbines this does not lead to significant loss of head. In the case of reaction turbines, the loss due to the installation at a higher level from the tailrace will be significant. This loss is reduced by connecting a fully flowing diverging tube from the turbine outlet to be immersed in the tailrace at the tube outlet. This reduces the pressure loss as the pressure at the turbine outlet will be below atmospheric due to the arrangement. The loss in effective head is reduced by this arrangement. Also because of the diverging section of the tube the kinetic energy is converted to pressure energy which adds to the effective head. The draft tube thus helps (a) to regain the lost static head due to higher level installation of the turbine and (b) helps to recover part of the kinetic energy that otherwise may be lost at the turbine outlet.



35. If at any point in the flow the pressure in the liquid is reduced to its vapour pressure, the liquid will then will boil at that point and bubbles of vapour will form. As the fluid flows into a region of higher pressure the bubbles of vapour will suddenly condense or collapse. This action produces very high dynamic pressure upon the adjacent solid walls and since the action is continuous and has a high frequency the material in that zone will be damaged. Turbine runners and pump impellers are often severely damaged by such action. The process is called cavitation and the damage is called cavitation damage. In order to avoid cavitation, the absolute pressure at all points should be above the vapour pressure.

Cavitation can occur in the case of reaction turbines at the turbine exit or draft tube inlet where the pressure may be below atmospheric level. In the case of pumps such damage may occur at the suction side of the pump, where the absolute pressure is generally below atmospheric level.

In addition to the damage to the runner cavitation results in undesirable vibration noise and loss of efficiency. The flow will be disturbed from the design conditions. In reaction turbines the most likely place for cavitation damage is the back sides of the runner blades near their trailing edge. The critical factor in the installation of reaction turbines is the vertical distance from the runner to the tailrace level. For high specific speed propeller units it may be desirable to place the runner at a level lower than the tailrace level.

To compare cavitation characteristics a cavitation parameter known as Thoma cavitation coefficient,  $\sigma$ , is used. It is defined as

$$\sigma = \frac{h_a - h_r - z}{h}$$

where  $h_a$  is the atmospheric head  $h_r$  is the vapour pressure head,  $z$  is the height of the runner outlet above tail race and  $h$  is the total operating head. The minimum value of  $\sigma$  at which cavitation occurs is defined as critical cavitation factor  $\sigma_c$ . Knowing  $\sigma_c$  the maximum value of  $z$  can be obtained as

$$z = h_a - h_r - \sigma_c h$$

$\sigma_c$  is found to be a function of specific speed. In the range of specific speeds for Francis turbine  $\sigma_c$  varies from 0.1 to 0.64 and in the range of specific speeds for Kaplan turbine  $\sigma_c$  varies from 0.4 to 1.5. The minimum pressure at the turbine outlet,  $h_0$  can be obtained as

$$h_0 = h_a - z - \sigma_c H$$

There are a number of correlations available for the value of  $\sigma_c$  in terms of specific speed, obtained from experiments by Moody and Zowski. The constants in the equations depends on the system used to calculate specific speed.

For Francis runners 
$$\sigma_c = 0.006 + 0.55 \left( \frac{N_s}{444.6} \right)^{1.8}$$

For Kaplan runners 
$$\sigma_c = 0.1 + 0.3 \left( \frac{N_s}{444.6} \right)^{2.5}$$

36. Given:

$$\dot{m}_s = \dot{m}_c = \frac{800}{60} = 13.33 \text{ kg/s}$$

$$\dot{m}_g = \dot{m}_h = \frac{1350}{60} = 22.5 \text{ kg/s}$$

$$t_{h_1} = 650^\circ\text{C}; t_{c_1} (= t_{\text{sat}}) = 180^\circ\text{C}; t_{c_2} = 350^\circ\text{C}$$

$$d = 30 \text{ mm} = 0.03 \text{ m}; L = 3 \text{ m}$$

Number of tubes required,  $N$  :

Heat lost by gases = Heat gained by steam

$$\dot{m}_h \times c_{ph} \times (t_{h_1} - t_{h_2}) = \dot{m}_c \times c_{pc} \times (t_{c_2} - t_{c_1})$$

$$22.5 \times 1 \times (650 - t_{h_1}) = 13.33 \times 2.71 \times (350 - 180)$$

$$t_{h_2} = 377^\circ\text{C}$$

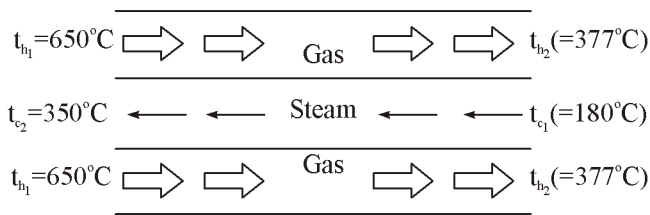


Figure : (a) Flow Arrangement

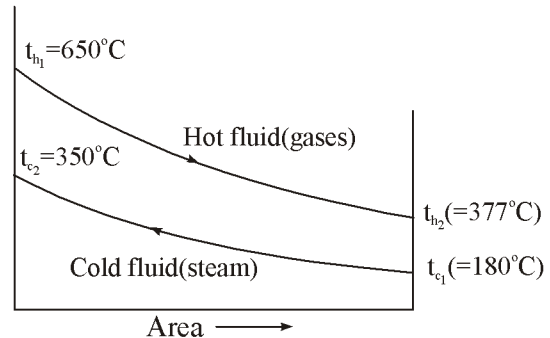


Figure : (b) Temperature distribution

Figure : Counter-flow heat exchanger

Overall heat transfer coefficient is given by

$$\frac{1}{U} = \frac{1}{h_g} + \frac{d_o}{d_i} \frac{1}{h_s} = \frac{1}{h_g} + \frac{1}{h_s} \text{ as } d_i \approx d_o$$

or

$$U = \frac{h_g \times h_s}{h_g + h_s} = \frac{250 \times 600}{250 + 600} = 176.5 \text{ W/m}^2\text{C}$$

Total heat transfer rate is given by

$$Q = U A \theta_m$$

Where

$$A = N \times (\pi d L) = N \times \pi \times 0.03 \times 3 = 0.2827 \text{ Nm}^2$$

$$Q = 22.5 \times (1 \times 10^3) \times (650 - 377) = 6142.5 \times 10^3 \text{ W}$$

$$\theta_m = \frac{\theta_1 - \theta_2}{\ln(\theta_1 / \theta_2)} = \frac{(t_{h_1} - t_{c_2}) - (t_{h_2} - t_{c_1})}{\ln[(t_{h_1} - t_{c_2}) / (t_{h_2} - t_{c_1})]}$$

$$= \frac{(650 - 350) - (377 - 180)}{\ln[(650 - 300) / (377 - 180)]} = \frac{300 - 197}{\ln(300/197)} = 244.9^\circ\text{C}$$

Substituting the values in equation, we get

$$6142.5 \times 10^3 = 176.5 \times 0.2827 \text{ N} \times 244.9$$

or

$$N = \frac{6142.5 \times 10^3}{176.5 \times 0.2827 \times 244.9} = 503 \text{ tubes}$$

37. Given,

$$r_1 = 5 \text{ cm}; r_2 = 10 \text{ cm}; r_3 = 15 \text{ cm}$$

$$T_1 = 1000 \text{ K}; T_3 = 300 \text{ K}$$

$$\epsilon_1 = \epsilon_2 = \epsilon_3 = 0.05$$

For steady state heat flow,  $Q_{12} = Q_{23}$

$$\text{or } \frac{A_1 \sigma (T_1^4 - T_2^4)}{\left(\frac{1 - \epsilon_1}{\epsilon_1}\right) + \frac{1}{F_{1-2}} + \left(\frac{1 - \epsilon_2}{\epsilon_2}\right) \frac{A_1}{A_2}} = \frac{A_2 \sigma (T_2^4 - T_3^4)}{\left(\frac{1 - \epsilon_2}{\epsilon_2}\right) + \frac{1}{F_{2-3}} + \left(\frac{1 - \epsilon_3}{\epsilon_3}\right) \frac{A_2}{A_3}}$$

In this case

$$F_{1-2} = F_{2-3} = 1$$

and

$$\frac{A_1}{A_2} = \frac{r_1}{r_2} = \frac{5}{10} = 0.5$$

$$\frac{A_2}{A_3} = \frac{r_2}{r_3} = \frac{10}{15} = 0.67$$

$$\therefore \frac{2\pi r_1 L \left[ \left( \frac{1000}{100} \right)^4 - \left( \frac{T_2}{100} \right)^4 \right]}{\left( \frac{1-0.05}{0.05} \right) + 1 + \left( \frac{1-0.05}{0.05} \right) \times 0.5} = \frac{2\pi r_2 L \left[ \left( \frac{T_2}{100} \right)^4 - \left( \frac{300}{100} \right)^4 \right]}{\left( \frac{1-0.05}{0.05} \right) + 1 + \left( \frac{1-0.05}{0.05} \right) \times 0.67}$$

$$\text{or } \frac{0.05(10000 - x^4)}{29.4} = \frac{0.1(x^4 - 81)}{32.73}$$

$$\text{or } (1000 - x^4) = \frac{29.5 \times 0.1}{32.73 \times 0.05} (x^4 - 81) = 1.8(x^4 - 81)$$

$$\text{or } 2.8x^4 = 10000 - 145.8 = 9854.2$$

$$\text{or } x = \left( \frac{9854.2}{2.8} \right)^{1/4} = 7.7$$

$$\text{or } \frac{T_2}{100} = 7.7 \text{ or } T_2 = 770 \text{ K}$$

38. Consider top jet : x distance travelled in time t is 10 m

$$\therefore V_{x01} t = 10 \quad \dots(1)$$

$$t = \frac{10}{V_{x01}}$$

The height drop is as  $V_{z0}$  as start is zero,

$$\therefore V_{z01} = H = \frac{1}{2} g t^2 \quad \dots(2)$$

Substituting for t

$$H = \frac{1}{2} g \frac{100}{V_{x01}^2}$$

$$\therefore V_{x01}^2 = \frac{50g}{H}$$

As jet issues from the nozzle it has any x directional velocity  $V_{x01}$  is present

$$V_{x01}^2 = 2g4 = 8g \quad \dots(3)$$

(as head available in 4 m)

$$\text{Substituting, } 8g = \frac{50g}{H} \text{ or } H = 6.25 \text{ m}$$

$$\text{Considering the second jet } V_{x02} t = 10; t = \frac{10}{V_{x02}}$$

The head drop in (H - h) m. As in the previous case  $V_{z0c} = 0$  at start

$$H - h = \frac{1}{2} g t^2$$

$$\text{Substituting } H - h = \frac{1}{2} g \frac{100}{V_{x02}^2} = \frac{50g}{V_{x02}^2} \quad \dots(4)$$

$$\text{As at start only } V_{x02} \text{ present, } V_{x02}^2 = (4 + h)g \times 2$$

Substituting in (4) 
$$H - h = \frac{50g}{(4+h)g \times 2} = \frac{25}{4+h}$$

As 
$$H = 6.25 \text{ m}$$

$$6.25 - h = \frac{25}{4+h}$$

This leads to 
$$h^2 - 2.25 h = 0 \text{ or } h = 2.25 \text{ m}$$

It may be also noted from problem

$$H \times 4 = (H - h)(4 + h)$$

$$6.25 \times 4 = 4 \times 6.25$$

Hence this condition is also satisfied.

39. Brayton cycle, popularly used for gas turbine power plants comprises of adiabatic compression process, constant pressure heat addition, adiabatic expansion process and constant pressure heat release process. A schematic diagram for air-standard Brayton cycle is shown in figure. Simple gas turbine power plant working on Brayton cycle is also shown here.

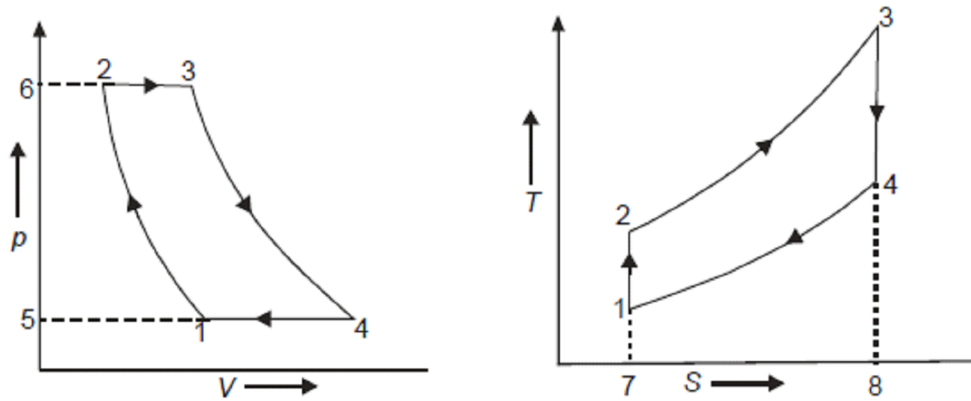
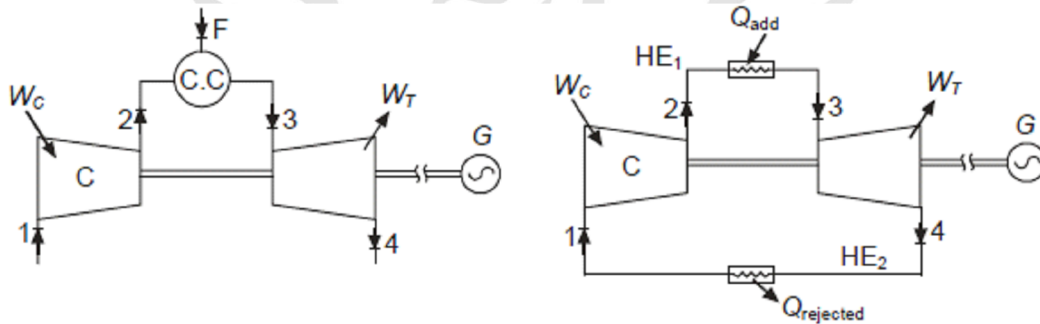


Figure : Brayton cycle on p-v and T-s diagram



C : Compressor  
CC : Combustion chamber

F : Fuel  
G : Generator  
HC : Heat exchanger

(a) Open type

(b) Closed type

Figure : Simple gas turbine plant

Thermodynamic cycle shows following processes :

- 1 - 2 : Adiabatic compression, involving (-ve) work,  $W_c$  in compressor.
- 2 - 3 : Constant pressure heat addition, involving heat  $Q_{add}$  in combustion chamber or heat exchanger.
- 3 - 4 : Adiabatic expansion, involving (+ve) work,  $W_t$  in turbine.
- 4 - 1 : Constant pressure heat rejection, involving heat,  $Q_{rejected}$  in atmosphere or heat exchanger.

In the gas turbine plant layout shown process 1–2 (adiabatic compression) is seen to occur in compressor, heat addition process 2–3 occurs in combustion chamber having open type arrangement and in heat exchanger in closed type arrangement. Process 3–4 of adiabatic expansion occurs in turbine.

In open type arrangement exhaust from turbine is discharged to atmosphere while in closed type, heat rejection occurs in heat exchanger. In gas turbine plant of open type, air entering compressor gets compressed and subsequently brought up to elevated temperature in combustion chamber where fuel is added to high pressure air and combustion occurs. High pressure and high temperature combustion products are sent for expansion in turbine where its' expansion yields positive work. Expanded combustion products are subsequently discharged to atmosphere. Negative work required for compression is drawn from the positive work available from turbine and residual positive work is available as shaft work for driving generator.

In gas turbine plant of closed type the working fluid is recycled and performs different processes without getting contaminated. Working fluid is compressed in compressor and subsequently heated up in heat exchanger through indirect heating. High pressure and high temperature working fluid is sent for getting positive work from turbine and the expanded working fluid leaving turbine is passed through heat exchanger where heat is picked up from working fluid. Thus, the arrangement shows that even costly working fluids can also be used in closed type as it remains uncontaminated and is being recycled.

Air standard analysis of Brayton cycle gives work for compression and expansion as;

$$W_C = m_1 \cdot (h_2 - h_1)$$

$$W_T = m_3 \cdot (h_3 - h_4)$$

for air standard analysis.  $m_1 = m_3$  where as in actual cycle

$$m_3 = m_1 + m_f; \text{ in open type gas turbine}$$

$$m_3 = m_1; \text{ in closed type gas turbine}$$

For the fuel having calorific value CV the heat added in air standard cycle;

$$Q_{\text{add}} = m_1(h_3 - h_2)$$

where as

$$Q_{\text{add}} = m_f \times \text{CV for actual cycle.}$$

$$\text{Net work} = W_T - W_C$$

$$W_{\text{net}} = \{m_3(h_3 - h_4) - m_1(h_2 - h_1)\}$$

$$\text{Air standard cycle efficiency} = \frac{W_{\text{net}}}{Q_{\text{add}}} = \frac{m_1\{(h_3 - h_4) - m_1(h_2 - h_1)\}}{m_1(h_3 - h_2)}$$

$$\eta_{\text{Brayton}} = \left\{1 - \frac{(h_4 - h_1)}{(h_3 - h_2)}\right\} = \left\{1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}\right\}$$

$$\text{as,} \quad (h_4 - h_1) = c_p(T_4 - T_1)$$

$$\text{and} \quad (h_3 - h_2) = c_p(T_3 - T_2)$$

For processes 1 – 2 and 3 – 4, which are of isentropic type,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma-1)}{\gamma}} \quad \text{and} \quad \frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{(\gamma-1)}{\gamma}}$$

$$\text{Let the pressure ratio be,} \quad \frac{p_2}{p_1} = r,$$

$$\frac{T_2}{T_1} = (r)^{\frac{(\gamma-1)}{\gamma}} \quad \text{and} \quad \frac{T_3}{T_4} = (r)^{\frac{(\gamma-1)}{\gamma}}$$



So,

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

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

$$\frac{T_4 - T_1}{T_1} = \frac{T_3 - T_2}{T_2}$$

$$\frac{(T_4 - T_1)}{(T_3 - T_2)} = \frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}$$

Making substitution for  $\left\{\frac{(T_4 - T_1)}{(T_3 - T_2)}\right\}$  in cycle efficiency :

$$\eta_{\text{Brayton}} = \left(1 - \frac{T_1}{T_2}\right) = 1 - \left(\frac{p_1}{p_2}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} = 1 - \frac{1}{(r)^{\frac{\gamma-1}{\gamma}}}$$

Air standard Brayton cycle efficiency :

$$\eta_{\text{Brayton}} = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}$$

Thus, it is obvious from the expression of efficiency that it depends only on pressure ratio ( $r$ ) and nature of gas ( $\gamma$ ). For pressure ratio of unity, efficiency shall be zero. For a particular gas the cycle efficiency increases with increasing pressure ratio. Here the variation of efficiency with pressure ratio is shown for air ( $\gamma = 1.4$ ) and monoatomic gas as argon ( $\gamma = 1.66$ )

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