B.E.

Fourth Semester Examination, May-2006

ENERGY CONVERSION

Note : Attempt any five questions.

Q. 1. (a) Sample of dry anthracite has the following composition by mass :

C : 90%; H : 30%; O : 2.5%; N : 1%; S : 0.5%; ash 3% calculate the stoichiometric air-fuel ratio.

Ans.

Element, wt (hg)	O_2 required (hg)
C =.90	$.90 \times \frac{8}{3} = 2.4$
H =.03	$.03 \times 8 = .24$
N = .01	–
S = .005	$.005 \times 1 = .005$
	Total $O_2 = 2.645$ kg

Weight of oxygen to be supplied per hg of

fuel $= 2.645 - .025 = 2.62$ kg

Weight of minimum air required for complete combustion

$$= \frac{2.62 \times 100}{23} = 11.39$$

Fuel air ratio $= \frac{1}{11.39} : 1$.

Q. 1. (b) Define the term 'Availability'. Derive an expression for Availability for a steady-flow system.

Ans. Availability : The theoretical amount of work which can be obtained from a system at any state P_1 and T_1 when operating with a reservoir at the constant pressure P_0 and temperature. T_0 is called availability.

Availability in steady flow systems : Consider a fluid flowing steadily with a velocity C_1 from a reservoir in which the pressure and temperature remain constant at p_1 and T_1 through an apparatus to atmospheric pressure of p_0. Let the reservoir be at a height Z_1 from the datum, which can be taken at exit from the apparatus, i.e. $Z_0 = 0$. For maximum work to be obtained from the apparatus the exit velocity, C_0, must be zero. It can be shown as for article 5.4 that a reversible best engine working between the limits would reject

$T_0(s_1 - s_0)$ units of heat, where T_0 is the atmospheric temperature. Thus, we have

$$W_{max} = \left(h_1 + \frac{C_1^2}{2} + Z_1 g\right) - h_0 - T_0(s_1 - s_0)$$

In several thermodynamic systems the kinetic and potential energy terms are negligible i.e.,

$$W_{max} = (h_1 - T_0 s_1) - (h_0 - T_0 s_0)$$

$$= b - b_0$$

The property, $b = h - T_0 s$ (per unit mass) is called the steady-flow availability function.

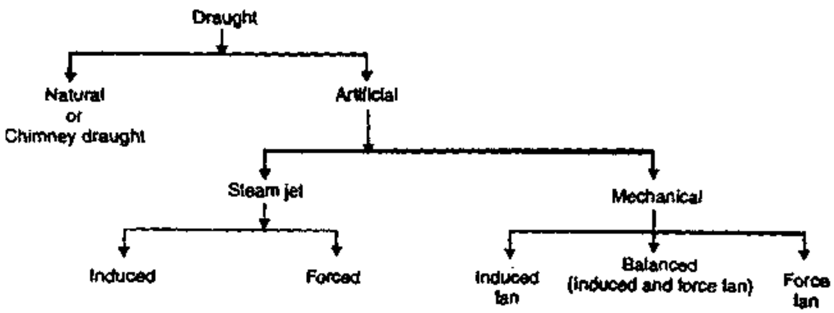
In the equation $b = h - T_0 s$; the function 'b' (like the function 'a') is a composite property of a system and s environment; this is also known as Keenan function.

Note :

1. The alternative names for availability and unavailable quantity $T_0 \Delta s$ are energy and a energy respectively.
2. The only different between $a = u + p_0 u - T_0 s$ function and $b = (h - T_0 s) = (u + pu - T_0 s)$ function is in pressure only.

Q. 2. (a) What is draught? Explain different types of draught.

Ans. Definitions and classification of draught : The small pressure difference which causes a flow of as to take place is termed as a draught. The function of the draught, in case of a boiler, is to force air to the fire nd to carry away the gaseous products of combustion. In a boiler furnace proper combustion takes place only hen sufficient quantity of air is supplied to the burning fuel.



Natural Draught-Chimney : Natural draught is obtained by the use of a chimney. The chimney in a oiler installation performs one or more of the following functions : (i) It produces the draught whereby the air id gas are forced through the fuel bed, furnace, boiler passed and settings; (ii) It carries the products of ombustion to such a height before discharging them that they will not be objectionable or injurious to surroundings. A chimney is vertical tubular structure built either of masonry, concrete or steel. The draught produced by the chimney is due to the density difference between the column of hot gases inside the chimney and the cold air outside.

Fig. 1. shows a diagrammatic arrangement of a chimney of height 'H' metres above the grate.

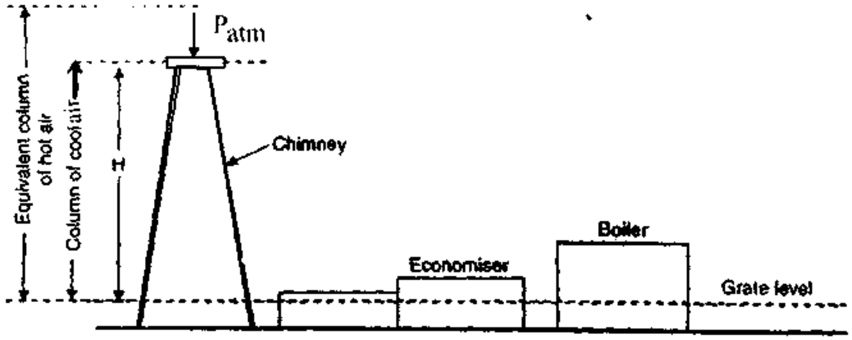


Fig. 1

We have $$p_1 = p_a + \rho_g . gH$$

Where, p_1 = Pressure at the grate level (Chimney side),

p_a = Atmospheric pressure at chimney top,

$\rho_g . gH$ = Pressure due to column of hot gas of height H metres, and

ρ_g = Average mass density of hot gas.

Similarly, $$p_2 = p_a + \rho_a . gH$$

Where, p_2 = Pressure acting on the gate on the open side,

$\rho_a . gH$ = Pressure exerted by the column of cold air outside the chimney of height H metres, and

ρ_a = Mass density of air outside the chimney.

$\therefore$ Net pressure difference causing the flow through the combustion chamber.

$$\Delta p = p_2 - p_1 = (\rho_a - \rho_g) gH$$

This difference of pressure causing the flow of gases is known as 'static draught'. Its value is small and is generally measured by a water manometer.

It may be noted that this pressure difference in chimney is generally less than 12 mm of water.

Artificial draugth : In the boiler installation of today the total static draught required may vary from 30 to 350 mm of water column.

It may not be possible to build a chimney high enough to produce draught of such a large magnitude. To meet this requirement artificial draught system should be used. It may be a mechanical draught or a steam jet draught or a steam jet draught. The former is used for central power stations and many other boiler installations while the latter is employed for small installations and in locomotives.

Forced draught : In a mechanical draught system, the draught is produced by a fan. In a forced draught system, a blower or a fan is installed near or at the base of the boiler to force the air through the cool bed and other passages through the furnace, flues, air preheater, economiser etc. It is a positive pressure draught. The enclosure for the furnace etc. has to be very highly sealed so that gases from the furnace do not leak out in the boiler house.

Induced draught : In this system a fan or blower is located at or near the base of the chimney. The pressure over the fuel bed is reduced blow that of the atmosphere. By creating a partial vacuum in the furnace and flues, the products of combustion are drawn from the main flue and they pass up the chimney. This draught is used usually when economisers and air preheaters are incorporated in the system. The draugth is similar in action to the natural draught.

Balanced draugth : It is a combination of the forced and induced Draugth systems. In this system the forced draught fan overcomes the resistance in the air preheater and chain grate stoke while the induced draught fan overcomes draught losses through boiler, economiser, air preheater and connecting flues.

The forced draught entails following advantages over induced draught :

1. Forced draught fan does not require water cooled bearings.
2. Tendency to air leak into the boiler furnace is reduced.
3. No loss due to in rush of cold air through the furnace doors when they are opened for fire and cleaning fires.
4. Fan size and power required for the same draught are 1/5 to 1/2 of that required from an induced draught fan installation because forced draught fan handles cold air.

Advantages or mechanical draught : The mechanical draught possesses the following advantages :

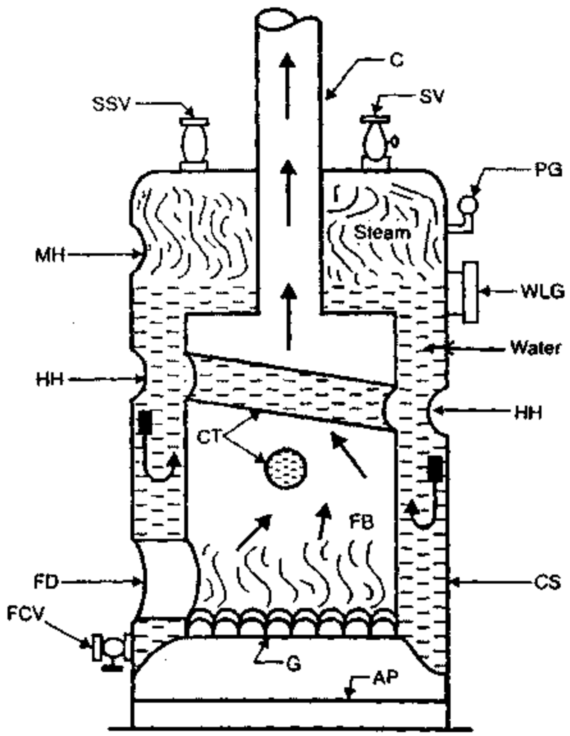
1. Easy control of combustion and evaporation.
2. Increase in evaporative power of a boiler.
3. Improvement in the efficiency of the plant.
4. Reduced chimney height.
5. Prevention of smoke.
6. Capability of consuming low grade fuel.
7. Low grade fuel can be used as the intensity of artificial draught is high.
8. The fuel consumption per H.P., due to artificial draught is 15% less than that for natural draught.
9. The fuel burning capacity of grate is 200 to 300 kg/m^2h with mechanical draught whereas it is hardly 50 to 100 kg/m^2-h with natural draught.

Q. 2. (b) Describe with the help of a neat sketch the working of a Nestler boiler.

Ans. Simple vertical boiler Nestler boiler : Fig. 1. It consists of a cylindrical shell, the greater portion of which is full of water (which surrounds the fire box also) and remaining the steam space. At the bottom o the fire box is grate on which fuel is burnt and the ash from it fall sin the ash pit.

The fire box is provided with two cross tubes. This increases the heating surface and the circulation o water. The cross tubes are fitted inclined. This ensures efficient circulation of water. At the ends of each cros: tube are provided hand holes to give access for cleaning these tubes. The combustion gases after heating th water and thus converting it into steam escape to the atmosphere through the chimney. Man hole, is providec to clear the interior of the boiler and exterior of the combustion chamber and chimney. The various mountin; shown in fig.2 are (i) Pressure gauge, (ii) Water level gauge or indicator, (iii) Safety valve, (iv) Steam sto| valve, (v) Feed check valve, and (vi) Man hole.

Flow of combustion gases and circulation of water in water jackets are indicated by arrows in fig.2.



CS = Cylindrical shell	C = Chimney
MH = Man hole	HH = Hand hole
CT = Cross tubes	FD = Fire door
G = Grate	FB = Fire box
PG = Pressure gauge	AP = Ash pit
SV = Safety valve	SSV = Steam Stop Valve
WLG = Water Level Gauge	FCV = Feed Check Valve

Fig. 2. Simple vertical boiler. Nester boiler.

The rate of production in such a boiler normally does not exceed 2500 kg/hr and pressure is normally limited to 7.5 to 10 bar.

A simple vertical boiler is self-contained and can be transported easily.

Q. 3. (a) What are the assumptions made in the analysis of Air-standard Otto cycle.

Ans. Assumptions :

1. The gas in the engine cylinder is a perfect gas i.e., it obeys the gas laws and has constant specific heats.
2. The physical constant 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.

$$c_p = 1.005\,\text{kJ/kg}-\text{K},\ c_p = 0.718\,\text{kJ/kg}-\text{K}.$$

3. The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.
4. No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot body or a cold body in constant with cylinder at appropriate points during the process.
5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

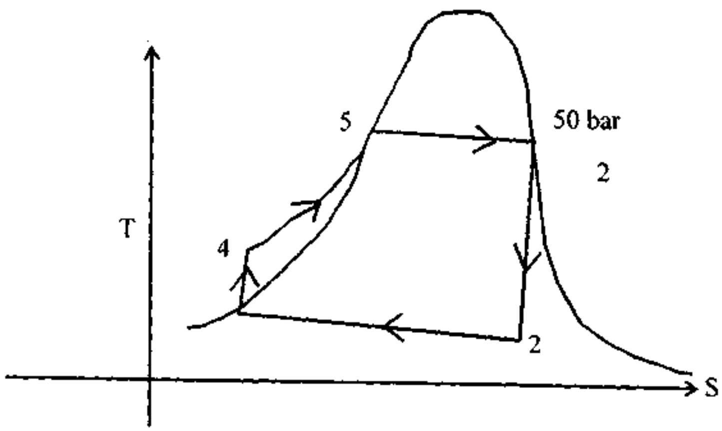
Q. 3. (b) A simple Rankine cycle operates between the pressures of 50 bar and 0.1 bar. If the steam is dry saturated at entry to turbine. Calculate

(i) Thermal efficiency

(ii) Work Ratio and

(iii) Specific Steam Consumption.

Ans.



From the steam table at 50 bar :

$$h_1 = 2794.2\,\text{kJ/kg} \qquad S_1 = 5.9735\,\text{kJ/kg k}$$

At .1 bar

$$h_{f_2} = h_{f_3} = 191.8\,\text{kJ/kg} \qquad h_{fg_2} = 2392.8\,\text{kJ/kg}$$

$$s_{f_2} = .649\,\text{kJ/kgk} \qquad s_{fg_2} = 7.501\,\text{kJ/kgk}$$

$$v_f = .001010\,\text{m}^3/\text{kg}$$

Considering turbine process 1–2

$$S_1 = S_2$$

$$5.9735 = .649 + x_2 \times 7.501$$

$$x_2 = .71$$

$$h_2 = h_{f_2} + x_2 h_{fg_2}$$

$$= 191.8 + .17 \times 2392.8$$

$$= 598.58\,\text{kJ/kg}.$$

Specific steam consumption $= \dfrac{3600}{W_{net}}$

$$= \frac{3600}{2190.63} = 1.64\,\text{kg/KWh}.$$

Q. 4. (a) Derive an expression for the mass flow rate of an ideal gas expanding isentropically through a nozzle in terms of the pressure ratio and specific heat ratio. Hence obtain the value of critical pressure ratio for maximum mass flow rate.

Ans. Discharge through the nozzle and conditions for its maximum value : Let

p_1 = Initial pressure of steam.

v_1 = Initial volume of 1 kg of steam at pressure

p_2 = Steam pressure at the throat,

v_2 = Volume of 1 kg of steam at pressure $p_2(m^3)$,

A = Cross-sectional area of nozzle at throat (m^3), and

C = Velocity of steam (m/s).

The steam flowing through the nozzle follows approximately the equation given below :

$$pv^n = \text{Constant}$$

Where, $n = 1.135$ for saturated steam.

$= 1.3$ for superheated steam.

For wet steam, the value of n can be calculated by Dr. Zenner's equation,

$n = 1.035 + 0.1x$, where x is the initial dryness fraction of steam.

Work done per kg of steam during the cycle (ankine area).

$$= \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

And, gain in kinetic energy = Adiabatic heat drop

= Work done during Raking cycle

Or $$\frac{C^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

$$= \frac{n}{n-1}p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right) \quad ...(1)$$

Also $$p_1v_1^n = p_2v_2^n$$

Or, $$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n} \quad ...(2)$$

Or, $$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/n} \quad ...(3)$$

Putting the value of v_2 / v_1 from equation (2) in equation (3), we get,

$$\frac{C^2}{2} = \frac{n}{n-1}p_1v_1\left[1 - \frac{p_2}{ip_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right] = \frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}\right]$$

$$= \frac{n}{n-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C^2 = 2\left(\frac{n}{n-1}\right)p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C = \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}} \qquad ...(4)$$

If m is the mass of steam discharged in kg/sec.,

Then $$m = \frac{AC}{v_2} \qquad ...(5)$$

Substituting the value of v_2 from equation (3) and (5)

$$m = \frac{AC}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}}$$

Or $$m = \frac{A}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}}\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \frac{A}{v_1}\sqrt{\left[2\left(\frac{n}{n-1}\right)p_1v_1\left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\left(\frac{p_2}{p_1}\right)^{2/n}\right\}\right]}$$

$$= \frac{A}{v_1}\sqrt{\left[2\left(\frac{n}{n-1}\right)p_1v_1\left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}\right]} \qquad ...(6)$$

It is obvious from above equation that there is only one value of the ratio (called critical pressure ratio) p_2 / p_1 which will produce the maximum discharge. This can be obtained by differentiating 'm' with respect to

(p_2/p_1) and equating it to zero.

As other quantities except the ratio p_2/p_1 are constant,

$$\therefore \quad \frac{d}{d\left(\frac{p_2}{p_1}\right)}\left[\left(\frac{p_2}{p_1}\right)^{2/n}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]=0$$

Or $$\frac{2}{n}\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1}-\left(\frac{n+1}{n}\right)\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}-1}=0$$

Or, $$\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1}=\frac{n+1}{n}\left(\frac{p_2}{p_1}\right)^{1/n}$$

Or, $$\left(\frac{p_2}{p_1}\right)^{2-n}=\left(\frac{n+1}{n}\right)^{n}\left(\frac{p_2}{p_1}\right)$$

Or, $$\left(\frac{p_2}{p_1}\right)^{2-n-1}=\left(\frac{n+1}{2}\right)^{n}$$

Or, $$\frac{p_2}{p_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \qquad ...(7)$$

Hence the discharge through the nozzle will be the maximum when critical pressure ratio, i.e.,

$$\frac{\text{Throat pressure}}{\text{Inlet pressure}}=\frac{p_2}{p_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

For saturated steam : $n = 1.135$

$$\frac{p_2}{p_1}=\left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}=\left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}}=0.58$$

For superheated steam : $n = 1.3$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = \left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}} \approx 0.546$$

Substituting the value of $\frac{p_2}{p_1}$ from equation (6) into equation (5), we get the maximum discharge,

$$m_{max} = \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left[\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\} - \left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{n+1}{n}}\right]}$$

$$= \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}-\frac{n+1}{n-1}} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{-1} - 1\right]}$$

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

i.e.,
$$m_{max} = A\sqrt{n\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

From the above equation it is evident that the maximum mass flow depends only on the initial condition of the steam (p_1, v_1) and the throat area and is independent of the final pressure of steam, i.e., at the axis of the nozzle. The addition of the divergent part of the nozzle after the throat does not affect the discharge of steam passing through the nozzle but it only accelerates the steam leaving the nozzle.

It may be noted that the discharge through nozzle increases as the pressure at the throat of the nozzle (p_2) decreases, when the supply pressure p_1 is constant. But once the nozzle pressure p_2 reaches the critical value (given by equation), the discharge reaches a maximum and after that the throat pressure and mass flow remains constant irrespective of the pressure at the exit.

The velocity of steam at the throat of the nozzle when the discharge is maximum is obtained by substituting the value of $\frac{p_2}{p_1}$ from equation (7) into (8)

$$C_{max} = \sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left[1-\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n+1}}\right\}^{\frac{n-1}{n}}\right]}$$

$$= \sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left(1-\frac{2}{n+1}\right)}$$

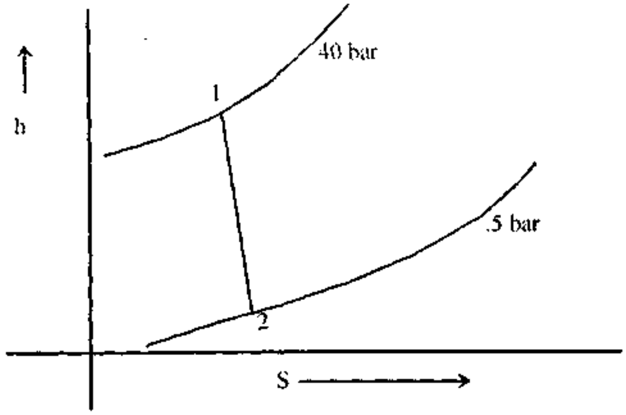
$$= \sqrt{2\left(\frac{n}{n-1}\right)p_1 v_1\left(\frac{n-1}{n+1}\right)}$$

i.e.
$$C_{max} = \sqrt{2\left(\frac{n}{n+1}\right)p_1 v_1}$$

The above equation indicates that the velocity is also dependent on the initial conditions of the steam.

Q. 4. (b) Calculate the exit velocity of steam expanding isentropically through a nozzle if the inlet conditions are p = 40 bar, (dry sat) and the exit pressure is 0.5 bar.

Ans.



From the miller diagram,

At 40 bar $\quad h_1 = 2800.3\,kJ/kg$

At .5 bar $\quad h_2 = 2646.0\,kJ/kg$

$$h_1 = h_1 - h_2 = 154.3\,kJ/kg$$

$$C_2 = 44.72\sqrt{h_d}$$

$$= 44.72 \times 12.42$$

$$= 555.5\,m/sec.$$

Q. 5. (a) Explain the different types of compounding schemes used in steam turbines.

Ans. Method of reducing wheel or rotor speed : As already discussed under the heading 'simple impulse turbine' that if the steam is expanded from the boiler pressure to condenser pressure in one stage the speed of the rotor becomes tremendously high which crops up practical complicacies. There are several methods of reducing this speed to lower value; all these methods utilise a multiple system of rotor in series, keyed on a common shaft and the steam pressure or jet velocity is absorbed in stages as the steam flows over the blades. This is known as 'compounding'. The different methods of compounding are :

1. Velocity compounding.
2. Pressure compounding.
3. Pressure velocity compounding.
4. Reaction turbine.

1. Velocity compounding : Steam is expanded through a stationary nozzle from the boiler or inlet pressure to condenser pressure. So the pressure in the nozzle drops, the kinetic energy of the steam increases due to increase in velocity. A portion of this available energy is absorbed by a row of moving blades. The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to re-direct the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig. 3 shows a cut away section of such a stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades. The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to re-direct the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig. 3 shows a cut away section of such a stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades.

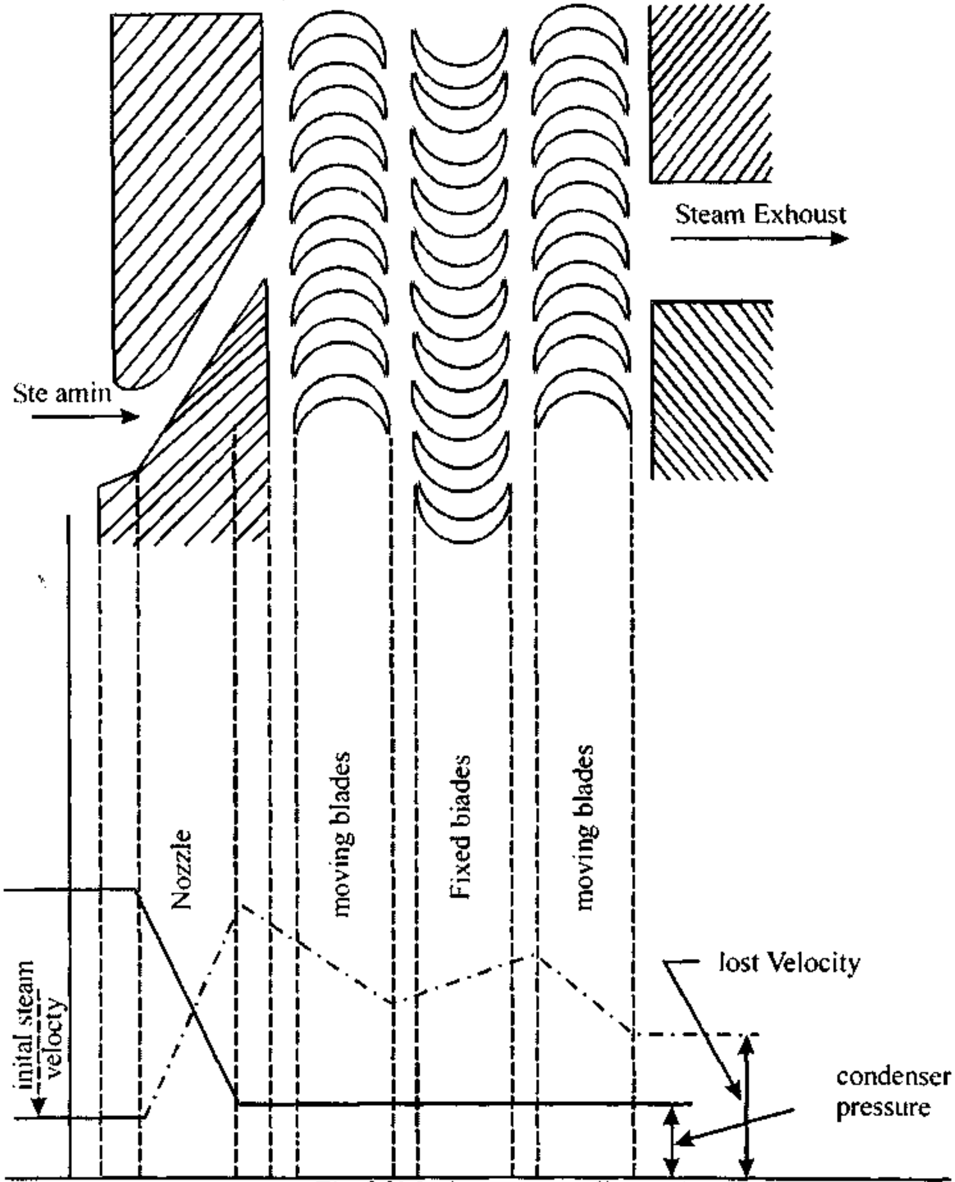


Fig. 3 Velocity compounding

Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.

2. Pressure Compounding : Fig. 4. shows rings of fixed nozzles incorporated between the rings of moving blades. The steam at boiler pressure enters the first set of nozzles and expands partially. The kinetic energy of the steam thus obtained is absorbed by the moving blades (stage 1). The steam then expands partially in the second set of nozzles where its pressure again falls and the velocity increases; the kinetic energy so obtained is absorbed by the second ring of moving blades (stage 2). This is repeated in stage 3 and steam finally leaves the turbine at low velocity and pressure. The number of stages (or pressure reductions) depends on the number of rows of nozzle through which the steam must pass.

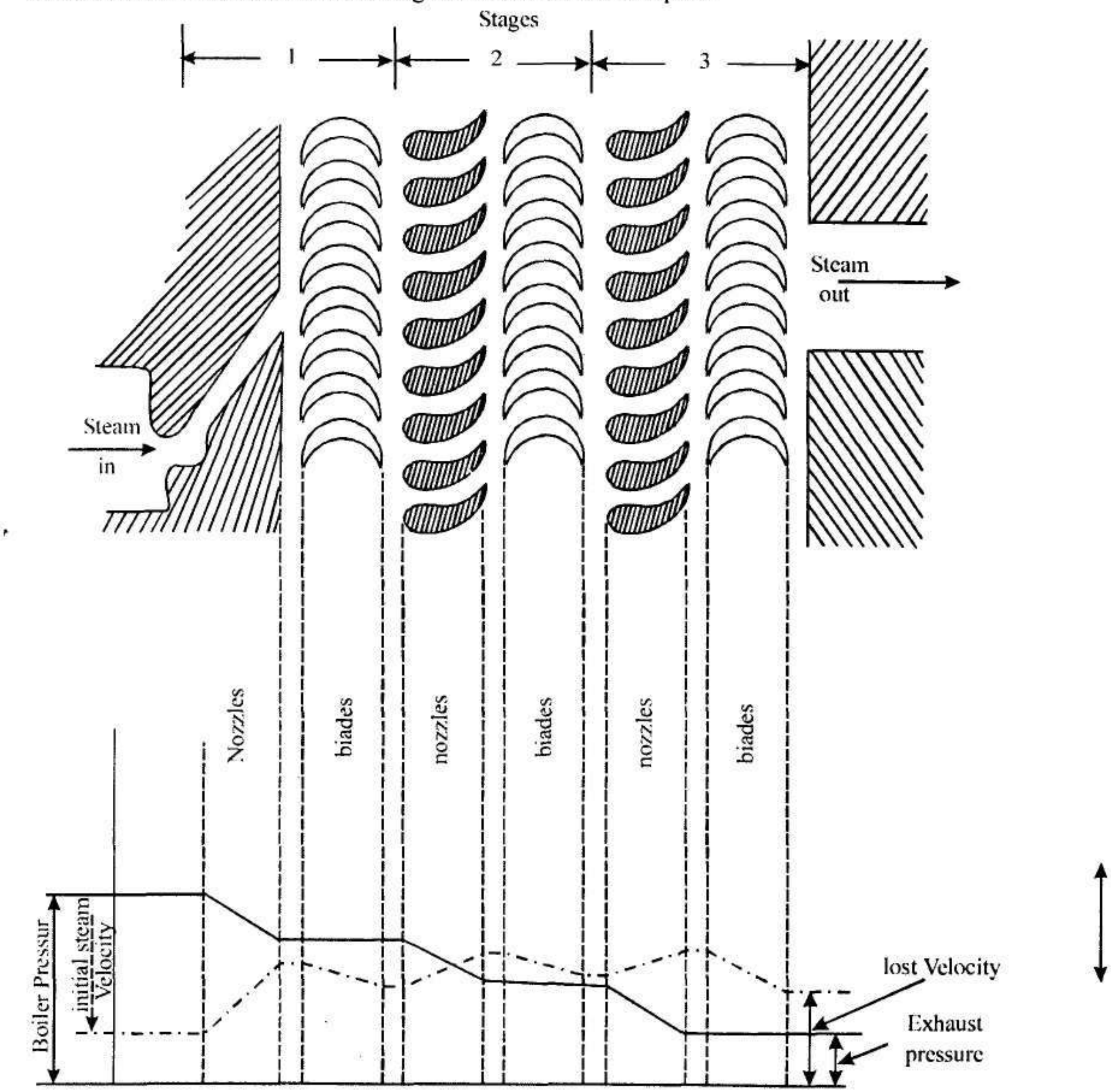


fig pressure compounding.

This method of compounding is used in Reteau and Zoelly turbine. This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages.

3. Pressure Velocity Compounding : This method of compounding is the combination of two previously discussed method. The total drop in steam, pressure is divided into stages and the velocity obtained in each stage is also compounded. The signs of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage. The changes in pressure and velocity are shown in fig. 5.

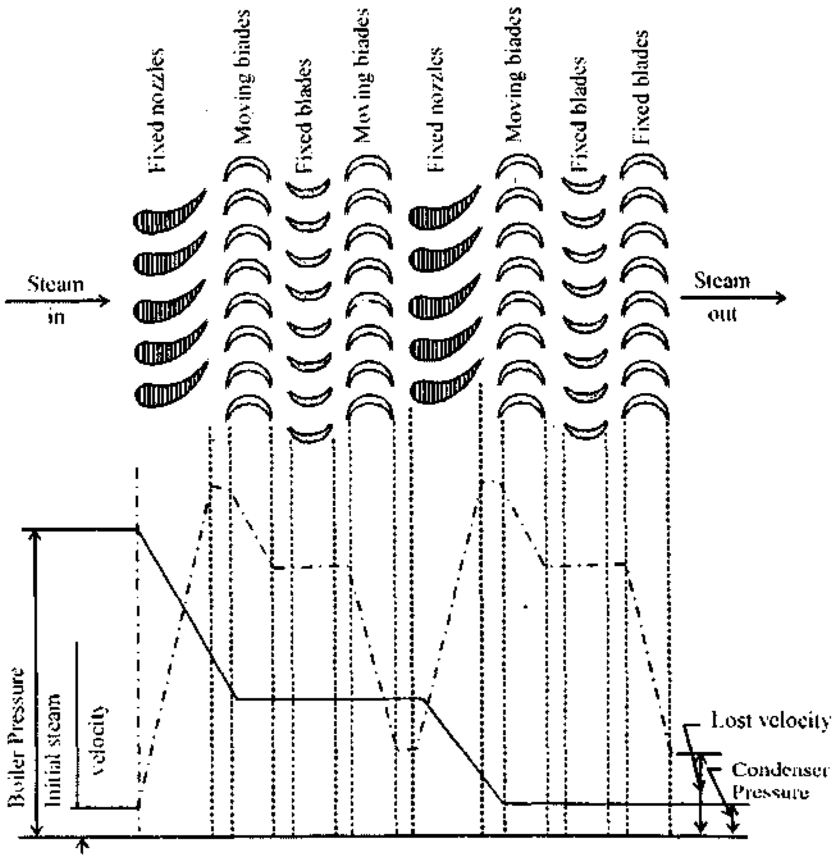


Fig. Pressure velocity compounding

This method of compounding is used in Curits and Moore turbine.

Q. 5. (b) The velocity of steam leaving the nozzles of an impulse turbine is 900 m/s and the nozzle angle is 20°. The blade velocity is 300 m/s and the blade velocity coefficient is 0.7. Calculate for a mass flow of 1 Kg/s, and symmetrical balding (i) the blade inlet angle, (ii) the driving force on the wheel and the axial thrust.

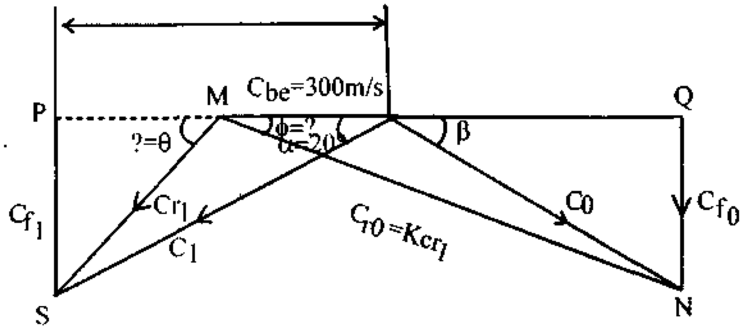
Ans $G = 900\,m/s$

Nozzle angle $\alpha = 20°C$

$C_{be} = 300 \, m/s$

$K = .7$

$\dot{m} = 1 \, kg/sec$



By measurement,

$$\theta = 30°, \; C_{r1} = 830 \, m/sec$$

$$\phi = \theta = 30° \text{ (given)}$$

$$c_{r_2} = kc_{r_1} = .7 \times 830 = 64 \, m/sec.$$

(i) Blade Inlet angle $\theta = \phi = 30°$

(ii) Tangential force on blades

$$\dot{m}_s(\omega_1 + c\omega_0)$$

$$= \frac{1}{60 \times 60}(1310) = .36 \, N$$

(iii) Axial thrust : $\dot{m}_s\left(c_{f_0} + c_{f_1}\right)$

$$= \frac{1}{60 \times 60}(1225) = .34 \, N.$$

Q. 6. (a) Define the term 'Vacuum Efficiency' of a Condenser. On what factors does its value depend?

Ans. Vacuum Efficiency : It is defined as the ratio of the actual vacuum to the maximum obtainable vacuum. The latter vacuum is obtained when there is only steam and no air is present in the condenser.

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Maximum obtained vacuum}}$$

$$= \frac{\text{Actual vacuum}}{\text{Barometer pressure} - \text{Absolute pressure of steam}}$$

Note : In case of the absolute pressure of steam corresponding to the temperature of condensate being equal to the absolute pressure in the condenser, the efficiency would be 100%. Actually some quantity of air is also present in the condenser which may leak in and be accompanied by the entering steam. The vacuum efficiency, therefore, depends on the amount of air removed by the air pump from the condenser.

Condenser efficiency : It is defined as the ratio of the different between the outlet and inlet temperatures of cooling water to the difference between the temperature corresponding to the vacuum in the condenser and inlet temperature of cooling water, i.e.,

$$\text{Condenser efficiency} = \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temp. corresponding to vacuum} \\ \text{in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temperature} \\ \text{of cooling water}\end{bmatrix}}$$

Or

$$= \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temperature corresponding to} \\ \text{the absolute pressure in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temperature of} \\ \text{cooling water}\end{bmatrix}}.$$

Q. 6. (b) Why are steam condenser equipped with Air extraction pump? Explain, with the help of a neat sketch, the working of an air extraction pump.

Ans. Air pumps : The main function which an air pump performs is that it maintains vacuum in the condenser as nearly as possible equal to that corresponding to the exhaust steam temperature, by removing air from the condenser. It may also remove consdensate together with air from the condenser.

An air pump which removes the moist air alone is called a dry air pump whereas that which removes both air an condensate is called a wet air pump.

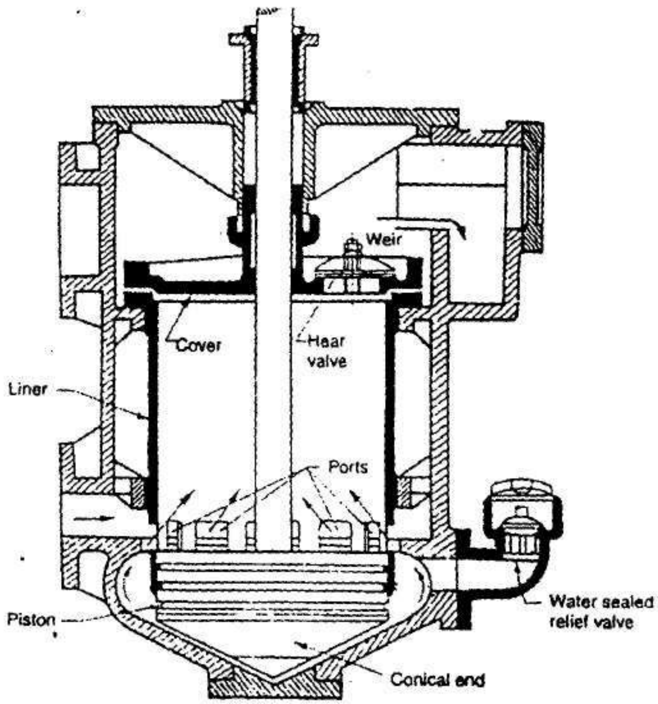
Types of air pumps :

Air pumps may be classified as follows :

1. Reciprocating piston or bucket pumps.
2. Rotary pumps.
3. Steam jet air pumps (ejectors)
4. Wet jet pumps.

1. Edward's pump : Fig. 4 shows Edward air pump which is a design of reciprocation piston or bucket wet air pump. The feature of this pump is the absence of inaccessible foot the bucket valves. This is effected by conical end to the piston and piercing the base of the liner with ports which communicate with air pump suction pipe down which the condensate gravities.

On the down stroke of the piston, a partial vacuum is produced above it, since the head valves are closed and sealed by water. Immediately the piston uncovers ports, air and water vapour rush into the space above the piston; further motion of the piston causing its conical end to displace the condensate rapidly through the ports. The rising piston traps the water, air and steam above the piston, and raises the pressure to slightly over that of the atmosphere until head valves open and allow the water vapour and air to pass to the waste, and the condensate to gravitate to the hot well over the weir which retains sufficient water above cover to seal the valves against air leakage. A water-sealed relief valve is placed in the base of the cylinder to release the pressure should it, for any reason, exceed atmospheric pressure.



Endward's air pump

2. Rotary dry air pump : The reciprocating air pumps, because of limited speed of operation, become very bulky for higher vacuum or large powers and it is due to this reason that rotary dry air pumps and steam jet ejectors have been developed.

Prof. Mauri Leblance invented a rotary dry air pump which is faily widely used and resembles one stage of radial flow steam turbine. The revolving vanes project thin film of water, at a velocity of about 39 m/sec. down a collecting cone in which these films act as pistons, the air being entrained between successive sets of water. Although the pump is charged with the water, it is intended to handle only air, the water and air being discharged through a diverging cone which raises its pressure to slightly greater than atmospheric. The water and air pass on to a slightly relevated tank in which the water is cooled prior to its return to the pump.

3. Steam-operated air ejector : Steam-operated air ejectors find a very wide of field of applications for the production of high vacuum because of the following reasons :

(i) Simple in construction, (ii) Cheap to construct,

(iii) No moving parts, and (iv) Occupy very little space.

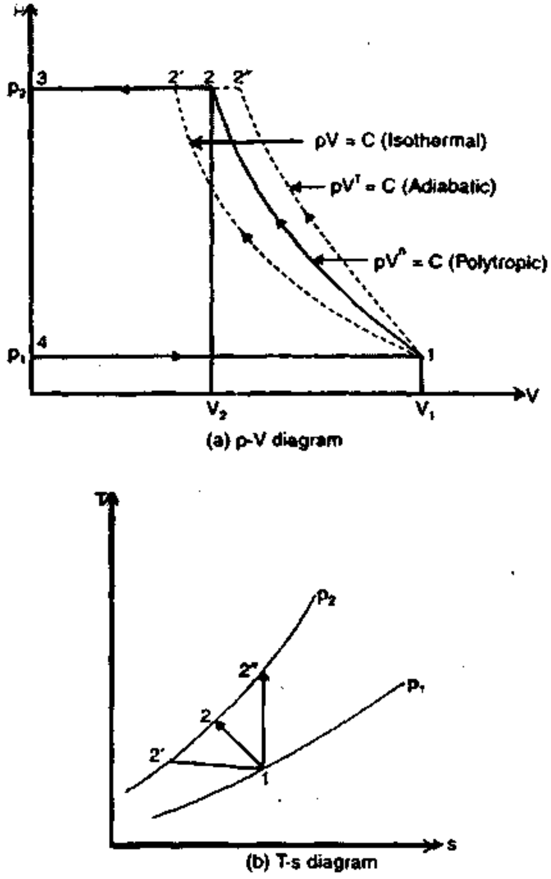
The operation of the ejector consists in utilizing the viscous drag of high velocity steam jet for the ejection of air and other incontestable gases from a chamber; it is chiefly used for exhausting the air from steam condensers. The steam jet flows through an air chamber where it entrains the air and any other gases which are adjacent to tis surface; the kinetic energy of the resulting mixture is then converted to pressure energy by being passed through a diverging cone, a diffuser. The increase of pressure thus obtained enables the mixture to be discharged against a pressure which is higher than that of the entraining chamber. The entraining operation is due to the viscous drag between the air and steam jet.

For steam plants where a high vacuum pressure is maintained it is imperative to use two or three sectors in series to obtain sufficient increase of pressure in the mixture for its discharge into the atmosphere.

Fig. 5 shows two-stage air injector. It consists of two injectors in series having a surface cooler between the stages and after the last stage. The function of the surface coolers is to condense the steam used in the ejector. The latent heat thus absorbed by the cooling water is recovered by being transferred to the condensate. The surface coolers also cool the steam and reduce its volume before it is passed on to the next stage. The operating steam for the two stages is controlled by stop valves 'A_1' and 'A_2'. The steam for first stage enters the steam box B_1 and that for the second stage the steam box B_2. The first stage operating steam is expanded through the nozzle 'C' into the mixing chamber 'D', where the jet of steam, entrains the air and vapour coming from the boiler.

Q. 7. (a) Define the term 'Isothermal Efficiency' of a reciprocating compressor. What does it signify?

Ans.



PVr T-S diagram for reciprocating compressor

Fig. 3. Theoretical p-V and T-s diagrams for a single-stage reciprocating air compressor

The sequence of operations as represented on the diagram, are as follows :

(i) Operation 4-1 : Volume of air V_1 aspirated into the compressor at pressure p_1 and temperature T_1 ;

(ii) Operation 1-2 : Air compressed according to the law $pV^n = C$ from p_1 to pressure p_2. Volume decreases from V_1 to V_2. Temperature increases from T_1 to T_2.

(iii) Operation2-3 : Compressed air of volume V_2 and at pressure p_2 with temperature T_2 delivered from the compressor.

During compression, due to its excess temperature above the compressor surroundings, the air will lose some heat. Thus, neglecting the internal effect of friction which is small in the case of the reciprocating compressor, the index n is less than γ, the adiabatic index. Since work must be put into an air compressor to run it, every effort is made to reduce this amount of work input. Inspection of p-V diagram shows the frictionless diabetics as $1-2^n$ and that if compression were along the isothermal 1-2' instead of polytropic 1-2 then the work done, given by the area of the diagram, would be reduced and, infect, would then be 'minimum'. Isothermal compression cannot be achieved in practice but an attempt is made to approach the isothermal case by cooling the compressor either by addition of cooling fins or a water jacket to the compressor cylinder. For a reciprocating compressor, a comparison between the actual work done during compression and the ideal isothermal work done is made by means of the isothermal efficiency.

This is defined as,

$$\text{Isothermal efficiency} = \frac{\text{Isothermal work done}}{\text{Actual work done}}$$

Thus, the higher the isochronal efficiency, the more nearly has the actual compression approached the ideal isothermal compression.

Total shaft work done/cycle, W = Area 41234

Or W = Area under 4–1 – Area under 1-2 – ARea under 2-3

$$= p_1V_1 - \frac{p_2V_2 - p_1V_1}{n-1} - p_2V_2$$

$$= (p_1V_1 - p_2V_2) - \left(\frac{p_2V_2 - p_1V_1}{n-1}\right) = (p_1V_1 - p_2V_2) + \left(\frac{p_1V_1 - p_2V_2}{n-1}\right)$$

$$= \left(1 + \frac{1}{n+1}\right)(p_1V_1 - p_2V_2)$$

$$\therefore \quad W = \left(\frac{n}{n-1}\right)(p_1V_1 - p_2V_2) \quad ...(1)$$

This equation can be modified as follows :

$$W = \frac{n}{n-1}(p_1V_1 - p_2V_2) = \frac{n}{n-1}.p_1v_1\left(1 - \frac{p_2V_2}{p_1V_1}\right) \quad ...(2)$$

Now $\quad p_1V_1^n = p_2V_2^n$

$\therefore \quad \dfrac{V_2}{V_1} = \left(\dfrac{p_1}{p_2}\right)^{1/n}$

And substituting this into equation we have,

$$W = \frac{n}{n-1}p_1V_1\left\{1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right\} = \frac{n}{n-1}p_1V_1\left\{1 - \frac{p_2}{p_1}\left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}\right\}$$

$$= \frac{n}{n-1}p_1V_1\left\{1 - \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}\right\} = \frac{n}{n-1}p_1V_1\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}$$

The solution to this equation will always come out negative showing that work must be done on the :ompressor. Since only the magnitude of the work done is required from the expression.

Q. 7. (b) A single-stage reciprocating compressor takes 1m3 of air per minute at 1.013 bar and 15°C and delivers it at 7 bar. If the index of compression is 1.35, and the clearance is negligible, calculate he indicated power.

Ans. Volume of air taken in, $V_1 = 1m^3/min$

Intake pressure, $p_1 = 1.013\,bar$

Initial temperature, $T_1 = 15 + 273 = 288\,K$

Delivery pressure, $p_2 = 7\,bar$

Law of compression : $pV^{1.35} = constant$

Indicated Power I.P. :

Mass of air delivered per min.,

$$m = \frac{p_1V_1}{RT_1} = \frac{1.013 \times 10^5 \times 1}{287 \times 288} = 1.226\,kg/min$$

Delivery temperature, $T_2 = T_1\left(\frac{p_2}{p_1}\right)^{(n-1)/m}$

$$= 288\left(\frac{7}{1.013}\right)^{(1.35-1)/1.35}$$

$$= 475.2 \text{ K}$$

Indicated work $= \frac{n}{n-1} mR(T_2 - T_1) \text{kJ/min}$

$$= \frac{1.35}{1.35-1} \times 1.226 \times 0.287(475.2 - 288) = 254 \text{ kJ/min}$$

i.e., Indicated power I.P. $= \frac{254}{60} = 4.23 \text{ kW}$.

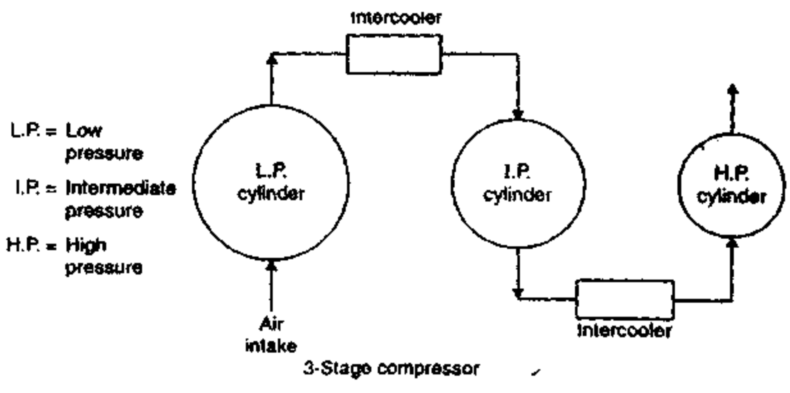
Q. 8. Write short notes on any two :

(a) Multi-stage Reciprocating Compressor.

(b) Mean Effective pressure

(c) Regenerative Rankine Cycle.

Ans. (a) Multi-stage Reciprocating Compressors : Multistage compression is a series arrangement of cylinders in which compressed air from the cylinder before, becomes the intake air for the cylinder which follows. This is illustrated in fig. 7. The low pressure ratio in the low-pressure cylinder means that the clearance air expansion is reduced and the effective swept volume of this cylinder is increased. Since in this cylinder which controls the mass flow through the machine, because it is this cylinder which introduces the air into the machine, then there is greater mass flow through the multi-stage arrangement than the single-stage machine.



L.P. = Low Pressure
I.P. = Intermediate pressure
H.P. = High pressure

3-Stage compressor

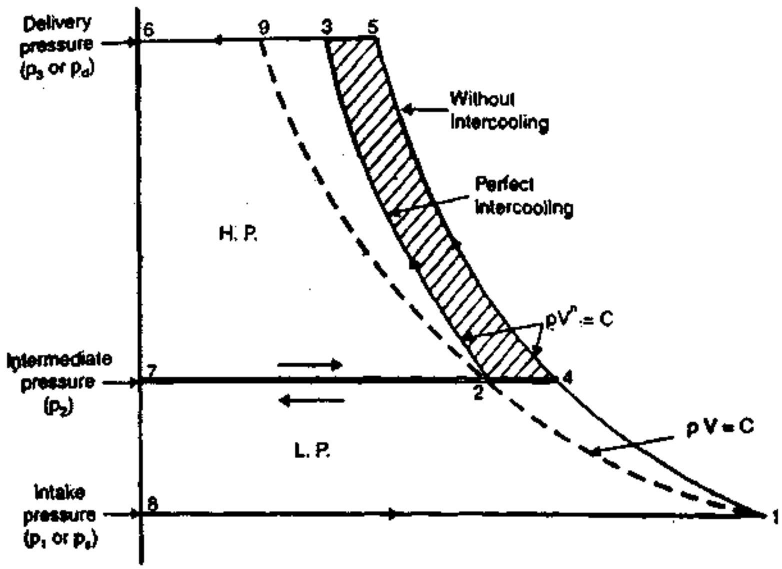
If an intercooler is installed between cylinders, in which the compressed air is cooled between cylinders, then the final delivery temperature is reduced. This reduction in temperature means a reduction in internal energy of the delivered air, and since this energy must have come from the input energy required to drive the machine, this results in a decrease in input work requirement for a given mass of delivered air.

It is common to find machines with either two or three stages or compression. The complexity of the machinery limits the number of stages.

Refer fig. 7. The cylinders are shown with diameters which decrease as the pressure increases. This is because, as the pressure increases, so the volume of a given mass of gas decreases. There is continuity of mass flow through a compressor and hence each following cylinder will require a smaller volume due to its increased pressure range. This reduction in volume is usually accomplished by reducing the cylinder diameter.

Fig. 8 shows cycle arrangements in the development of the ideal conditions required for multi-stage compression. For simplicity, clearance is neglected.

Referring to Fig. 8, the overall pressure range is p_1 to p_3. Cycle 8156 is that of single-stage compressor. Cycles 8147 and 7456 are that of a two-stage compressor without intercooling between cylinders. Cycles 8147 and 7236 are that of a two-stage compressor with perfect intercooling between cylinders.



'Perfect intercooling' means that after the initial compression in the L.P. cylinder, with tis consequent temperature rise, the air is cooled in an intercooler back to its original temperature. This means, referring to fig. 21.9, $T_2 = T_1$, in which case point 2 lies on isothermal through point 1. This shows that multi-stage compression, with perfect intercooling, approaches more closely the ideal isothermal compression than in the case with single-stage compression.

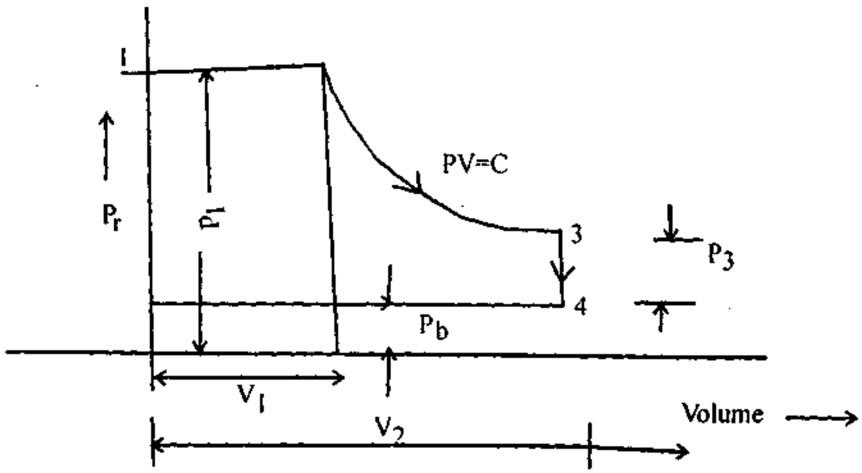
Ideal conditions for multi-stage compressors :

Case : 1. Single-stage compressor :

As earlier stated cycle 8156 is that of a single-stage compressor, neglecting clearance. For this cycle,

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_5}{p_1} \right)^{\frac{n-1}{n}} - 1 \right].$$

(b) Mean effective pressure :



Let A = ARea of the indicator's diagram

L = Base width of indicator diagram.

K = Ordinate or Pr. scale in $N/m^2/m$

Then the mean effective pressure

$$P_m = \frac{Am^2}{lm} \times k\left(N/m^2\right)/m$$

$$= \frac{Ak}{l} N/m^2$$

If a expressed in m^2,. l in m and K in bar/m.

Then

$$P_m = \frac{Am^2}{lm} \times k \text{ bar / m} = \frac{Ak}{l} \text{bar} .$$

(c) Regenerative Rankine Cycle : In the Rankine cycle it is observed that the condensate which is fairly t low temperature has an irreversible mixing with hot boiler water and this results in decrease of cycle effi-iency. Methods are, therefore, adopted to hat the feed water from the hot well of condenser irreversibly by nterchange of heat within the system and thus improving the cycle efficiency. This heating method is called egenerative feed heat and the cycle is called regenerative cycle.

The principle of regeneration can be practically utilised by extracting steam from the turbine at several ocations and supplying it to the regenerative heaters. The resulting cycle is known as regenerating or bleeding ycle. The heating arrangement comprises of : (i) For medium capacity turbines–not more than 3 heaters; (ii) 'or high pressure high capacity turbines–not more than 5 to 7 heaters; and (iii) For turbines of super critical arameters 8 to 9 heaters. The most advantageous condensate heating temperature is selected depending on he turbine throttle conditions and this determines the number of heaters to be used. The final condensate eating temperature is kept 50 to 60°C below the boiler saturated steam temperature so as to prevent evapora-ion of water in the feed mains following a drop in the boiler drum pressure. The conditions of steam bled for ach heater are so selected that the temperature of saturated steam will be 4 to 10°C higher than the final ondensate temperature.

Fig. 5 (a) shows a diagrammatic layout of a condensing steam power plant in which a surface condenser s used to condense all the steam that is not extracted for feed water heating. The turbine is double extracting nd the boiler is equipped with a superheater. The cycle diagram (T-s) would appear as shown in fig. 15.15 (b). 'his arrangement constitutes a regenerative cycle.

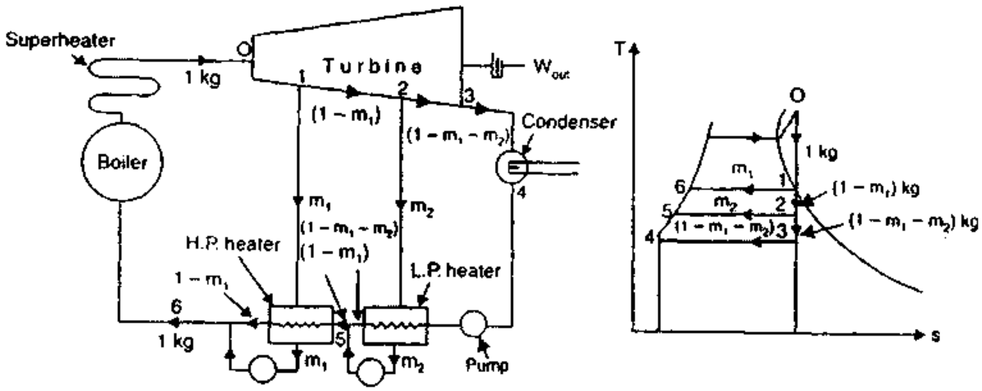


Fig. Regenerative cycle

Let $\quad m_1$ = kg of high pressure (H.P.) steam per kg of steam flow,

m_2 = kg of low pressure (L.P.) steam extracted per kg of steam flow, and

$(1 - m_2 - m_2)$ = kg of steam entering condenser per kg of steam flow.

Energy Heat balance equation for H.P. heater :

$$m_1 = \left(h_1 - h_{f_6}\right) = (1 - m_1)\left(h_{f_6} - h_{f_6}\right)$$

Or $m_1\left[\left(h_1 - h_{f_6}\right) + \left(h_{f_6} - h_{f_5}\right)\right] = \left(h_{f_6} - h_{f_5}\right)$

Or,
$$m_1 = \frac{h_{f_6} - h_{f_5}}{h_1 - h_{f_5}}$$

Energy/Heat balance equation for L.P. heater :

$$m_2\left(h_2 - h_{f_6}\right) = (1 - m_1 - m_2)\left(h_{f_6} - h_{f_3}\right)$$

Or
$$m_2\left[\left(h_2 - h_{f_5}\right) + \left(h_{f_6} - h_{f_3}\right)\right] - (1 - m_1 - m_2)\left(h_{f_5} - h_{f_3}\right)$$

Or
$$m_2 = \frac{(1 - m_1)\left(h_{f_5} - h_{f_3}\right)}{\left(h_2 - h_{f_3}\right)}$$

All enthalpies may be determined; therefore m_1 and m_2 may be found. The maximum temperature tô which the water can be heated is dictated by that of bled steam. The condensate from the bled steam is added to feed water.

Neglecting pump work :

The heat supplied externally in the cycle

$$= \left(h_0 - h_{f_6}\right)$$

Isentropic work done $= m_1(h_0 - h_1) + m_2(h_0 - h_2) + (1 - m_1 - m_2)(h_0 - h_3)$

The thermal efficiency of regenerative cycle is

$$\eta_{thermal} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{m_1(h_0 - h_1) + m_2(h_0 - h_2) + (1 - m_1 - m_2)(h_0 - h_3)}{(h_0 - h_{f_6})}$$

The work done by the turbine may also be calculated by summing up the products of the steam flow and the corresponding heat drop in the turbine stages.

i.e., Work done $= (h_0 - h_1) + (1 - m_1)(h_1 - h_2) + (1 - m_1 - m_2)(h_2 - m_3)$

Advantages of regenerative cycle over simple rankine cycle :

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