4

Steam Boilers, Engines, Nozzles and Turbines

Conversion Factors for Pressure

	bar	*dyne/cm²*	*kgf/cm²*	*N/m²*	*mm Hg (21°C)*	*mm H_2O (21°C)*	*at m*
1 bar =	1	10^6	1.01972	10^5	750.06	10197.2	0.986923
1dyne/cm² =	10^{-6}	1	101.972×10^8	0.1	750.062×10^{-6}	10197×10^{-6}	986.923×10^{-5}
1kgf/cm² =	0.980665	980.665×10^3	1	98066.5	735.559	10000	967.861×10^{-3}
1 N/m² =	10^{-5}	10	0.03453	1	750.06×10^{-5}	10197.2×10^{-5}	0.986923×10^{-5}
1 mm Hg = (21°C)	1.333223×10^{-3}	1333.223	1.3595×10^{-3}	133.322	1	13.5951	1.315×10^{-3}
1 mm H_2O = (21°C)	98.0655×10^6	98.0669	10^4	9.80665	0.073556	1	96.7837×10^{-6}
1 atm=	1.0132	101.325×10^4	1.03323	133.322	760	10332.276	1

Properties of Liquids (Water and Steam)

For understanding the various properties of steam, let us start heating 1 kg of water at 0°C in closed vessel closed by a piston and continue till it becomes superheated steam. Let this 1 kg of water be subjected to continuous constant pressure P kg/cm². The volume occupied by water is nearly independent of the pressure. As heat is applied, the speed of movement of molecules in water increases and temperature starts rising. If we assume there are no other losses, then this heat energy is being stored in water and internal energy of water is increased. This is shown visibly by rise of temperature in thermometer. Change of volume will be very small, therefore work done will be negligible as long as it is in state of water.

So heat supplied to water

$$Q = U_2 - U_1 + \frac{W}{J}$$

U_2 = final internal energy

U_1 = initial internal energy

However H (Enthalpy), is defined as sum of internal energy and work.

$$\therefore \quad H = U + \frac{pV}{J}$$

But for water, $\frac{pV}{J}$ = negligible.

Hence $H = U$ (as long as water remains water)

As we keep on increasing the application of heat, a stage is reached when pressure exerted by the molecules of water equals the external pressure and boiling starts. The temperature corresponding to this stage is called *saturation temperature.* By still further addition of heat, the molecules of water start shooting off, thus push the piston away which is causing pressure. The temperature remains same. In other words for any particular value of pressure there is a definite temperature at which water will start boiling and change into steam. This temperature is called saturation temperature and the temperature remains same during complete formation of steam. The saturation temperature increases as the pressure of water is increased.

When water is boiling and steam is being formed at constant temperature, steam is said to be *saturated.* Its physical condition is such that slight disturbance in pressure or temperature will change it into water.

Wet steam is a mixture of steam and water which exist in equilibrium at saturation temperature.

When no water particles are present and temperature of steam corresponds to saturation temperature then steam is said to be dry saturated.

Sensible heat (h). It is the heat required to raise the temperature of 1 kg of liquid from 0°C to the boiling point.

Latent heat (L). It is the quantity of heat required to convert 1 kg of liquid at boiling point into dry saturated vapour at the same temperature.

Total heat (H). It is the quantity of heat required to convert 1 kg of liquid from the 0°C to dry saturated vapour at constant pressure.

During steam formation stage, the volume of the working substance goes on increasing. Therefore, work done is given by

$$\frac{p\,(V_s - V_w)}{J} \qquad \begin{bmatrix} V_s = \text{volume of steam} \\ V_w = \text{volume of water} \end{bmatrix}$$

V_w is very small in comparison with V_s, therefore, it is neglected. Term $\frac{pV_s}{J}$ is known as external work of evaporation.

Heat supplied or enthalpy of steam

$$= h + L + \frac{pV_s}{J}$$

As the term pV_s/J is spent in doing external work and not stored in the form of energy, therefore the expression $h - \frac{pV_s}{J}$ is called true or internal latent heat.

Total increase in internal energy of water changed into dry saturated steam = $h + L - \frac{pV_s}{J}$.

= heat of water from zero °C to saturation temperature, and $h + L$ is defined as enthalpy (H).

$$\therefore \qquad U_2 - U_1 = H - \frac{pV_s}{J}.$$

Dryness Fraction is the term used to define the wetness of steam. This is equal to ratio of weight of steam to total weight of mixture.

This is also defined as the ratio of actual heat supplied to total latent heat of evaporation.

i.e. $$x = \frac{H_w - h}{L}\ ;\ H_w = \text{total heat of steam at any stage}$$

$$\therefore \qquad H_w = h + xL.$$

Dryness fraction of steam can be determined by using separating calorimeter or throttling calorimeter or combined separating and throttling calorimeter, or barrel calorimeter.

Superheated Steam. If heat be still further added when whole of water is converted into steam, then molecules become more active and temperature will rise. The steam is then said to be superheated. The difference of superheated steam temperature and saturation temperature is called degree of superheat. Superheated steam follows gas law and heat added after the complete dry steam is formed is equal to

$$= C_p(t_s - t_d)$$

where t_s = superheated steam temperature

t_d = temperature of dry saturated steam.

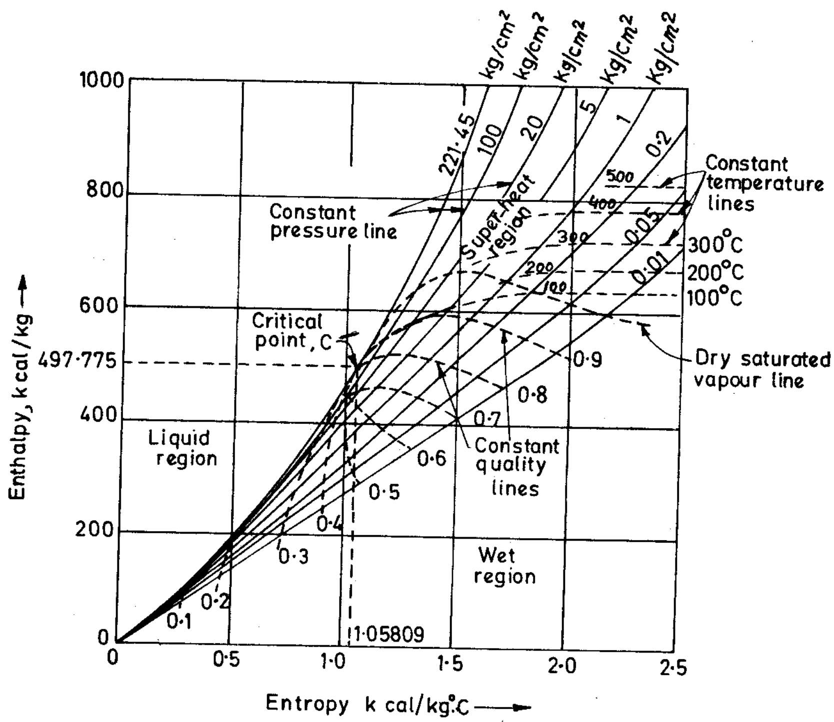

Fig. 4.1.

Based on the results of a number of experiments conducted over long periods of time, steam tables have been made which give all the properties of steam at various temperatures.

Mollier Diagram or Enthalpy-Entropy Chart

On mollier diagram, enthalpy is plotted on y-axis and entropy on x-axis. The critical point, corresponds to liquid enthalpy as well as enthalpy of dry saturated steam at 225.65 kg/cm^2.

The constant pressure lines, constant quality lines, constant temperature lines, constant specific volume lines and constant superheat lines are plotted on Mollier chart.

It any region, constant enthalpy process may be represented by a horizontal straight line from one specified point to the other, the reversible adiabatic or isentropic process by a vertical straight line from one specified point to the other. The constant temperature line coincides with constant pressure line in the wet regions.

Steam Boilers

According to A.S.M.E. boiler is defined "as a steam generating unit which is used for producing, furnishing and recovering heat, together with the apparatus for transferring heat, so made available to the fluid being heated and vaporised".

The boilers are mainly classified in two classes as follows :

(1) **Smoke Tube Boilers.** Boilers in which there is an external shell generally in whole or in part cylindrical, which contains a fire-box or one or more tubes large enough to hold grate, upon which the fuel is burned. From the fire-box or the furnace box, the products of combustion are led through the shell by one or more large tubes or by a considerable number of small tubes to a smoke-box or to some space outside the shell and discharged to atmosphere through chimney. In these types of boilers the water is circulated outside the tubes, and the gases inside the tube. Examples of this type are Cochren, Cornish, Lancashire, Locomotive and Marine boiler. These may be externally fired or internally fired. In externally fired boilers, fire is entirely external to the boilers and is suited to those boilers which are used for stationary installations. In internally fired boilers, fire takes place internal to boiler shell. Locomotive and marine boilers are of this type.

(2) **Water Tube Boilers.** Boilers which contain a large number of comparatively small tubes through which water circulates, the fire and hot gases being outside these tubes.

Comparison of Water Tube and Fire Tube Boilers

1. The water tube boiler is safer, because most of water at hottest part of the furnace is in small tubes which if ruptured, only a comparatively small volume of water is instantly released to flash in steam.
2. For efficient and economic working, fire tube boilers require less of skill than water tube boilers.
3. Fire tube boilers can be made of small sizes and cannot be made up in large sizes due to the difficulty in transportation of the shell whereas water tube boilers are uneconomical in small sizes and hence often made in large sizes.
4. The fire tube boilers are limited to a steam pressure of 16.0 kg/cm^2, the commonly used pressure being of 10.0 kg/cm^2, but the water tube boilers can be operated upto 160 kg/cm^2 or higher, the limitation in the former case being of large-shell thickness required.

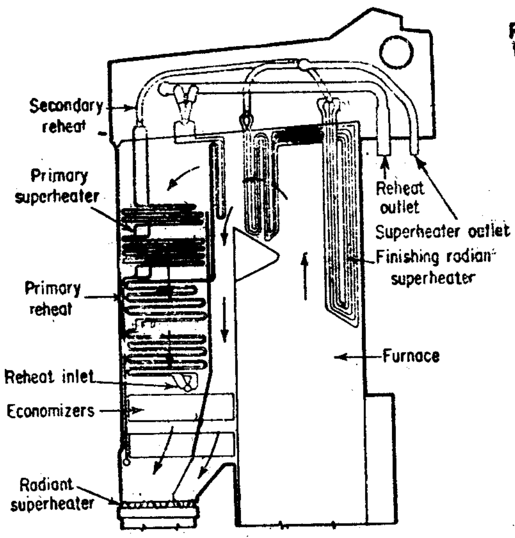

Fig. 4.2.

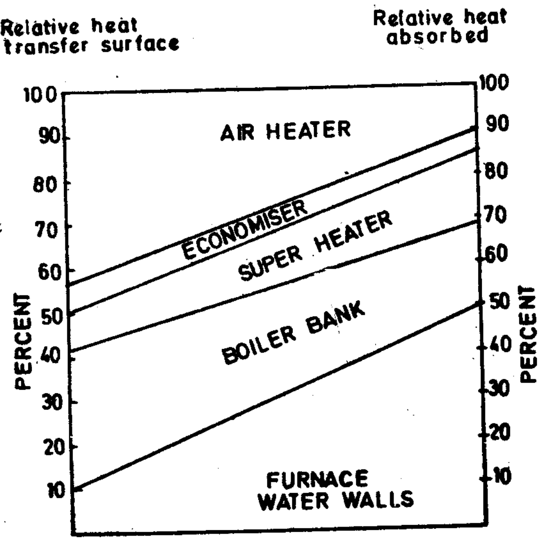

Fig. 4.3. Relative percentage of heat absorbed through heat transfer surfaces in case of large boilers.

5. Rate of steam production is higher in water tube boilers than gas tube boilers because of large heating surface, long gas travel, and rapid and positive water circulation.
6. The floor area required for smoke box boiler is much less than water tube boiler.
7. Chances of explosion in water tube boilers are more than in smoke box.
8. The flexibility of operation is more in fire tube boilers than water tube boilers due to comparatively large water quantity.
9. Need for feed water treatment is more in water tube boilers, because pitting and corrosion caused by impure water is more injurious to thin tubes than thick tubes.
10. Larger water tube boilers can carry much greater overloads and respond more rapidly to sudden changes and fluctuations in demand. The drum in water tube boilers is not exposed to the radiant heat of the fire.

Fig. 4.2 shows a typical water wall type pulverised coal fired boiler; in which the various heat exchange surfaces are clearly depicted. The relative percentage of heat absorbed and heat transfer surfaces in large boilers is shown in Fig. 4.3.

Other types of boilers are discussed below in brief.

Lancashire Boiler is a horizontal fire tube, natural circulation type boiler. It is approximately 7 to 9 m in length and 2 to 3 m in diameter. It has two parallel furnace flue tubes. It can generate steam pressure upto 16 kgf/cm^2 and evaporative capacity is of the order of 8000 kg/hr.

Cornish Boiler is a horizontal, fire tube boiler having only one furnace flue tube. Its length varies from 4 to 8 m and diameter from 1.25 to 1.75 m. The working pressure range and capacity of Cornish boiler is lower than Lancashire boiler.

Cochran Boiler is a vertical boiler with horizontal firetube and is portable type. It is approximately 2.75 m in diameter and 5.5 m in height. The working pressure range and evaporative capacity are of the order of 20 kgf/m^2 and 3500 kg/hr respectively.

Locomotive Boiler is a portable, horizontal, multi-tubular fire tube boiler. The fire box is separate and the shell is horizontal. The hot flue gases from fire box flow to smoke box through the tubes, which are surrounded with water in shell. The normal pressure range is upto 20 kgf/cm^2.

Benson Boiler is a light weight high pressure boiler, having no drum. It can generate steam within 20 minutes. The normal pressure range is from 30 kgf/cm^2 to 250 kgf/cm^2 and evaporative capacity is of the order of 135 tons of steam per hour.

La Mont Boiler is a high pressure, water tube type, forced circulation boiler. It produces around 100 tons of superheated steam per hour at a temperature of 500° C and at a pressure of about 135 kgf/cm^2.

Boiler Mountings :

Water level indicator idicates the level of water in the boiler.

Stop valve controls the steam supply as per requirements.

Safety valve. It prevents excessive steam pressure in the boiler. It may be spring loaded type, dead weight type, lever type or high steam and low water type.

Pressure gauge shows the pressure of the steam in the boiler.

Blow cock is used to remove mud, scale or sediments collected at the bottom of the boiler; and to empty the boiler.

Feed check valve is used to control the supply of water to boiler, to maintain constant level.

Fusible plug is used to put off the fire in the furnace of the boiler when the level of water falls to very low value.

Boiler Accessories :

Economiser is used for heating feed water thus utilising the heat in the exhaust flue gases.

Super heater is used to superheat wet steam at constant pressure by utilising the heat of combustion products.

Feed Pump pumps water at desired pressure to boiler.

Injector is used for pumping water to a vertical or a locomotive boiler and low pressure boilers.

Air Preheater heats the secondary air used for combustion and also primary air used to pulverise and carry coal air mixture by utilising the heat in the exhaust flue gases.

Steam Trap is used to drain off water resulting from the partial condensation of steam from steam pipes.

Steam Separator is used for separating suspended water particles from steam.

Pressure reducing valve is used to maintain desired steam pressure.

I.D. Fan (Induced Draft fan). It is used to exhaust flue gases from boiler and discharge them into chimney.

F.D. Fan (Forced Draft Fan). It is used to supply air for combustion.

P.A. Fan (Primary Air Fan). It is used to supply air for pulverising the fuel and carrying it to furnace.

Electrostatic Precipitators. These remove ash from flue gases so that clean gases are discharged into atmosphere.

Boiler Draught is defined as the small pressure difference (usually in mm of wcl) which removes products of combustion from boiler and allows fresh air to be sucked in. Draught may be natural draught produced by a chimney, or artificial type produced by fans, steam jet etc.

A chimney is made of masonry, steel or concrete. In big power plants, draft is created by I.D. fans and chimney is used mainly to discharge flue gases at high altitude so as to disperse the ash, etc over a wider area, thus causing minimum pollution.

Chimney Calculations (when it is used for producing draught only).

Pressure causing draught in the chimney.

$$= 353\,H\left[\frac{1}{T_1}-\left(\frac{m+1}{m}\right)\times\frac{1}{T}\right]\text{kgf/m}^2 \text{ or mm of } H_2O$$

where H = height of chimney in m

T_1 = absolute atmospheric temperature

T = absolute temperature of flue gases in the chimney.

Velocity of flue gases in the chimney can be determined by the relation

$$V=\sqrt{2g\,(H'-h_f)}$$

where H' = height of hot gas column in m

and h_f = loss due to friction.

For maximum discharge through the chimney, the height of chimney should be equal to the height of the hot gas column producing draught.

Boiler Performance

As already defined, a boiler is an apparatus used for converting water into steam at the desired pressure and temperature by the heat obtained from combustion of fuel. So the amount of water evaporated is the basic consideration for studying the performance of boilers. The evaporation is generally expressed in terms of kg of steam/kg of fuel or in kg of steam/sq metre of heating surface area/hour or simply by the total weight of water evaporated. The water evaporated in the boiler is under different conditions of temperature of feed water and steam generation pressure, temperature and dryness fraction and so takes different amount of heat for its evaporation.

For the purpose of comparison, it is therefore necessary to state a standard condition under which evaporation of water takes place. The standard which has been adopted is as follows : "The temperature of feed water and steam should be 100°C and the steam should be dry and saturated. So from the above fact, it is clear that under the standard conditions of evaporation the water takes its latent heat which has value of 539 kcal/kg.

Factor of evaporation. If under the standard conditions,

t = temperature of feed water which will have total heat h_f,

h = total heat of water at saturation temperature and pressure.

H = total heat of dry saturated steam at saturated pressure and temperature.

L = latent heat of steam at 100°C.

W = weight of steam produced,

W_1 = equivalent evaporation.

Then under these conditions, heat to produce one kg of steam is

$$(H - h) + (h - h_t)$$

and total heat required to produce W kg of steam will be

$$= W[(H - h) + (h - h_t)]$$

This amount of heat under the standard conditions will be able to evaporate water weighing W_1 kg which is given by

$$W_1 = \left\{\frac{[(H - h) + (h - h_t)]}{L}\right\}$$

W_1 is called the equivalent evaporation and term within brackets $\left[\frac{(H - h) + (h - h_t)}{L}\right]$ is called the *factor of evaporation.*

The *equivalent evaporation* is defined as 'The number of kg of water that would be evaporated from a feed water temperature of 100°C into dry and saturated steam at 100°C by the same amount of heat actually used under this condition. It is generally expressed in kg/kg of fuel or in kg/square metre of the heating area/hr. The water is evaporated under the standard conditions which have been adopted, are as follows, temperature of feed water and steam at 100°C and the steam is dry and saturated. Under the standard conditions, therefore the evaporation of kg of water represents the utilization of latent heat which is 539 kcal at 100°C.

Evaporation per square metre of Heating Surface. In certain cases the evaporation is also expressed in terms of kg/sq. metre of heating surface area/hr. The value of evaporation is more in fire tube boilers and less in water tube boilers. For evaporation the locomotive boiler has an average value of about 11 whereas water tube boiler has 4.

This can be obtained by finding the total weight of water evaporated in kg/hr, and finding the total surface area of heating. Then dividing the two, we get equivalent evaporation in kg/sq. m/hr.

Efficiency of Boiler. Efficiency of boiler is defined as heat utilized in raising the steam to the heat supplied by burning the fuel in a given period.

A boiler plant mainly consists of boiler itself but in addition it has several auxiliaries such as feed water heater, economiser and super heaters etc. All of them have their own efficiencies. When all these efficiencies are taken together, it is termed as the efficiency of the whole plant.

Heat Losses in a Boiler Plant. All the heat which is evolved due to combustion of fuel is not utilized in raising the steam. Certain amount of heat goes waste in one form or the other which is termed as 'lost heat'. The sources of losses are as follows :

1. Heat lost through hot gases in chimney.
2. Incomplete combustion of fuel *i.e.*, a certain amount of carbon burnt to CO instead of CO_2 due to lack of O_2.
3. Unburnt fuel which drops in ash-pit.
4. Heat wasted in the evaporation of moisture content in air and due to H_2 in fuel.
5. Radiation losses.

Fuels. Fuels may be solid, liquid or gaseous type. A good fuel should have low ignition temperature, high calorific value, easy to store, produce minimum smoke and be economical.

STEAM ENGINES

A *simple steam engine* is one in which the expansion of steam is completed in one cylinder, and the *compound steam engine* is one in which the steam progressively expands in two or more cylinders. Compound engines may have the cylinders arranged in various ways such as *tandem compound*—cylinders placed one behind the other in line; *angle compound* — cylinders axes placed at right angle to one another : *cross compound*—cylinders arranged side by side etc.

An engine may operate *non-condensing, i.e.* exhausting to atmosphere, or *condensing i.e.* exhausting to a condenser.

In a *single acting* engine, steam is admitted only to one side of the piston, and in *double-acting* engine, steam is admitted alternately to each side of the piston.

A *slide valve* is a valve which controls the inlet and exhaust of steam by sliding across the ports. A *poppet valve* is a disc, fitting a port which is raised or lowered for the control of inlet and exhaust, A *Corliss valve* has a cylindrical surface which oscillates through a small angle to open or close the port. In *uniflow engine,* piston itself acts as exhaust valve.

Steam *engine speed is governed* by (*i*) throttling the steam supply, or (*ii*) by varying the cut-off.

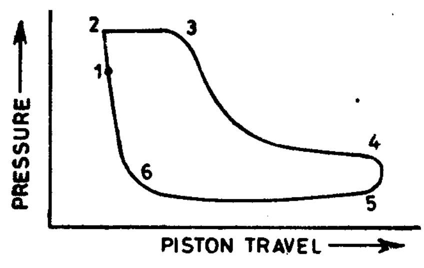

Fig. 4.4. Indicator diagram of a simple steam engine.

Indicator diagram shows the pressure in engine cylinder at each point of the stroke. It is used for determining the power developed in the cylinder, approximate steam consumption, and whether the valves are properly adjusted. Point 1 in Fig. 4.4 represents the *point of admission* at which the steam valve opens. 12 is the *admission line* where pressure rises due to admission of steam to the cylinder by opening the steam valve. 23 represents the *steam line* when the steam valve is open and steam is being admitted to the cylinder, 3 is the *point of cut-off* where admission of steam is stopped by closing of the valve. It is usually located where the outline of the diagram changes its curvature from convex to concave. 34 is the *expansion curve* as the steam expands doing the work, 4 is the *point of release* when the exhaust valve opens. 45 is the *exhaust line* representing the change in pressure that takes place when the exhaust valve opens. 56 shows the pressure against which the piston acts during its return stroke, known as *back pressure* line. 6 is the *point of exhaust* closure where the exhaust valve closes. 61 is the *compression curve* and shows the rise in pressure due to compression of the steam remaining in the cylinder after the exhaust valve has closed. *Indicated Mean Effective Pressure (i.m.e.p.)* is found by dividing the area by its length and multiplying the quotient by the pressure scale of the diagram.

$$\text{Diagram factor} = \frac{\text{Area of actual indicator diagram}}{\text{Area of hypothetical diagram}}$$

Indicated work for cycle = i.m.e.p. $\times A \times L$, where A = piston area, L = Length of stroke of piston. Theoretically work done per cycle = $p_1 v_1 (1 + \log_e r) - p_b v_2$.

$$\text{Ideal m.e.p.} = \frac{p_1}{r}[1 + \log_e r] - p_b$$

where p_1 = supply pressure

p_b = back pressure

r = hypothetical ratio of expansion

$$= \frac{\text{Total swept volume}}{\text{Swept volume at cut off}} = \frac{V_2}{V_1}$$

When clearance volume is also considered and if a = clearance volume/swept volume, then *m.e.p.*

$$= p_1\left[\frac{1}{r} + \left(\frac{1}{r} + a\right)\log_e\left(\frac{1+a}{1/r+a}\right)\right] - p_b$$

Ideal thermal efficiency of the steam engine is same as that of the Rankine cycle.

Losses of available energy in the steam engine are due to following causes (arranged in the approximate order of magnitude of loss).

(1) Initial or surface condensation

(2) Incomplete expansion and early release

(3) Throttling of steam flowing through partly open valves

(4) Leakage of steam past piston and closed valves

(5) Friction of moving parts

(6) Heat loss by radiation and conduction

(7) Compression and early admission.

Initial condensation can be reduced by adopting late cut off and small expansion ratio, superheat, separate valves for admission and exhaust, short valve passages, high compression compounding, and uniflow principle.

Saturation Curve and Missing Quantity

The actual volume of steam in cylinder is less due to condensation of steam.

The saturation curve in an indicator diagram is the curve showing the volume which steam would have occupied if the steam were totally dry and saturated. The difference in the theoretical and actual volume at any point is known as missing quantity. Missing quantity is due to initial cylinder condensation and a small amount of leakage past piston.

It can be reduced by steam jacketing of cylinder walls, use of superheated steam, compounding *i.e.* expanding the steam in two or more cylinders instead of a single cylinder.

In Fig. 4.5, curve AB represents saturation curve. *The missing quantity* is the difference in the theoretical volume and the actual volume at any point. Thus for pressure OC.

Dryness fraction at $D = \dfrac{CD}{CE}$

where CD = actual volume in the cylinder

CE = volume given by saturation curve

DE = missing quantity.

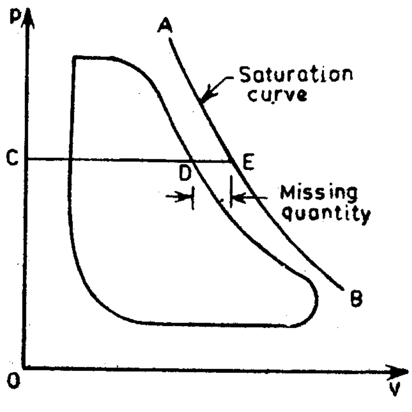

Fig. 4.5.

Governing of Simple Steam Engine

(*i*) **Throttle Governing.** In this method, steam inlet pressure is varied by a throttle valve, and the cut-off ratio is kept constant. In the throttle governing, the steam consumption is directly proportional to the indicated horse power. Hence graph between consumption of steam and indicated horse power is a straight line known as *Willan's line* which is shown in Fig. 4.6.

(*ii*) **Cut-off Governing.** In it the ratio of cut off is varied and the admission pressure is kept constant. It is more efficient and economical.

Compound Steam Engine. In a compound steam engine, the expansion of steam takes place in two or more than two cylinders in series. The expansion of steam first occurs in high pressure cylinder and expansion of steam takes place in the last in low pressure cylinders. The remaining cylinders are called intermediate pressure cylinders.

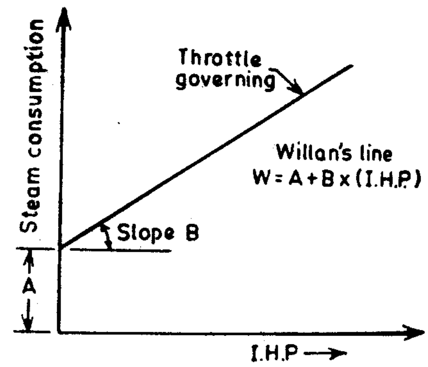

Fig. 4.6.

The *main advantages* of compound engines are : Small pressure range and small temperature range per cylinder and hence reduction in condensation and reduced steam leakage past piston and valves and hence less missing quantity; reduced stroke length, less weight and more uniform torque and thus lighter flywheel; reheating between high pressure cylinder and low pressure cylinder is possible which coupled with a saving in steam consumption results in increase in thermal efficiency.

Methods of Compounding Steam Engines are :

(*i*) **Tandem Compound Engine.** In this method, there is a common piston rod for the high pressure cylinder and low pressure cylinder, connected to the same crank. The exhaust steam from the high pressure cylinder is passed directly to the low pressure cylinder. Since the maximum turning moment from each cylinder acts simultaneously on the crank-shaft, a large flywheel is required.

(*ii*) **Woolfe Compound Engine.** In this method, two cranks are at 180° to each other. The two cylinders are arranged side by side and the exhaust steam from the H.P. cylinder passes directly to L.P. cylinder. This also requires large flywheel.

(*iii*) **Receiver Compound Engine.** In this method, two cranks are placed at 90° to each other. The exhaust from the H.P. cylinder is passed into a receiver and from there to the L.P. cylinder. As the two cycles are out of phase by 90°, turning moment on the crank shaft is more uniform and hence a smaller flywheel is required.

Rankine Cycle. (Refer Fig. 4.7)

1-2 → Pressurisation of water in pumps.

2-3-4-5→ Heat addition in boiler to boil water and raising the steam to super-heat condition.

5-6 → Expansion in turbine.

6-1 → Condensation of steam to get original water.

$$\eta_{th} = \frac{h_5 - h_6}{h_5 - h_1} \quad \text{[where } h = \text{enthalpy at respective point].}$$

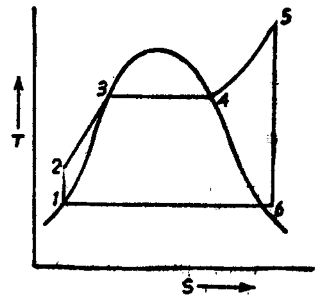

Fig. 4.7.

In addition to it, the modifications of Rankine cycle are non-expansion vapour cycle, incomplete-expansion vapour cycle, ideal vapour reheating cycle and regenerative cycle.

Rankine cycle efficiency can be improved by increasing the initial pressure and temperature. Further, in order to avoid excessive condensation in the lower pressure stages, the pressure increase must go hand-in-hand with increase in temperature.

Reheating Cycle. Process representing 6-7 in Fig. 4.8 is reheating. By this the thermal efficiency of plant is increased and steam at the end of expansion is not wet. Wet steam may be disastrous for turbines. Reheating is accomplished by passing the partially expanded steam from turbine through the steam superheater. The cycle efficiency is determined by totalling the available energy converted to work in each part of the expansion and dividing by the total heat supplied to the boiler. The main advantage of reheating is to accomplish reduction of excessive moisture formation in the last stages of turbine without the use of high initial steam temperature.

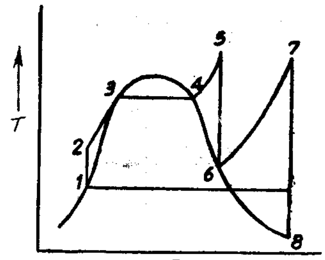

Fig. 4.8.

Regenerative Cycle (Feed heating cycle). In this cycle steam is withdrawn from the turbine at various points, *i.e.*, without full expansion taking place. The extracted steam supplies heat to the feedwater. This cycle results in a considerable gain in economy, because the extracted steam has already done work in the turbine and it does not lose its heat to the cooling water in a condenser but its latent heat is fully utilised within the cycle in heating the feed-water which is returned to the boiler.

The Mercury-Vapour Cycle. This cycle is based on the effects of the difference in thermodynamic properties of the two pure fluids, steam and mercury. Steam works under relatively high pressure with an attendant relative low temperature. Mercury, on other hand operates under low pressures, with attendant high temperature. The selection of pressures is done such that the mercury vapour condenses at a temperature higher

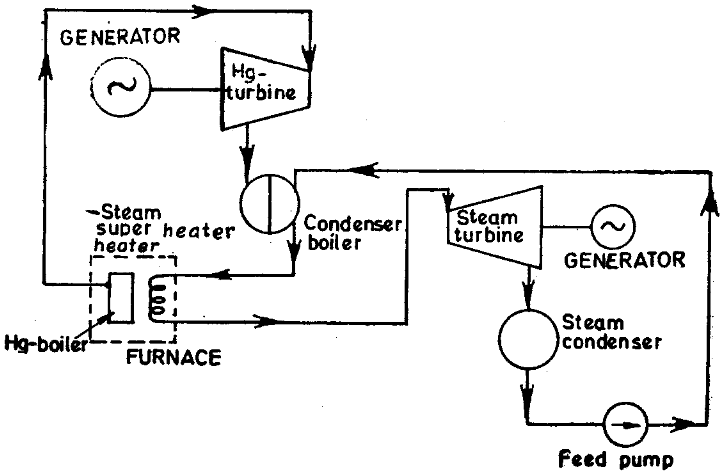

Fig. 4.9.

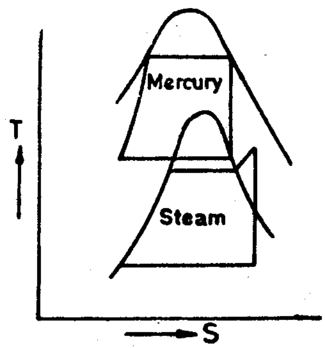

Fig. 4.10.

than that at which steam evaporates. The heart of the cycle is the condenser-boiler in which the processes of mercury vapour condensation and steam evaporation take place. In the steam portion of the cycle, condenser water carries away heat of steam condensation, and in the mercury portion of the cycle, steam picks up the heat of condensation of the mercury vapour. Obviously, this cycle results in a great saving in heat. Fig. 4.9 shows the flow diagram and Fig. 4.10 shows it on T-S diagram. It will be noted that the same furnace serves the mercury boiler and the steam superheater. Mercury vapour is only condensed and not superheated. Condenser boiler is located high enough above the mercury boiler, so that the head of mercury returns the liquid mercury to the boiler by gravity, thereby dispensing with the mercury feed pump.

Steam Nozzles

These are used to convert heat energy of steam into kinetic energy. The nozzles are classified into two types according to their shapes as (*i*) Convergent nozzle and (*ii*) Convergent and divergent nozzle.

Convergent nozzle. A convergent nozzle is one in which the cross-section decreases continuously (converges) right from the entrance upto the exit. This is essential to bring about the increase in kinetic energy of steam and hence the convergent portion is found in all types of nozzles. The exit end of such a nozzle is throat. This type of nozzle is rarely used in practice.

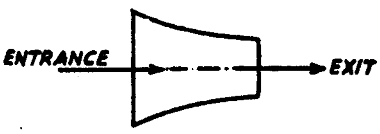

Fig. 4.11.

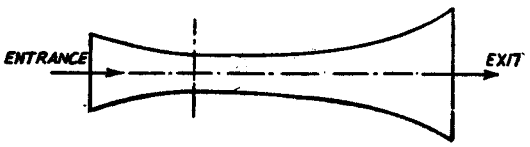

Fig. 4.12.

Convergent and divergent nozzle. The Fig. 4.12 shows the other type of nozzle which has its cross-section firstly decreasing up to throat (convergent) and then increasing in cross-section upto mouth (divergent). The cross-section at the throat is minimum and the corresponding pressure at throat is called the critical pressure. The divergent part of nozzle performs the function of change of quality of steam and also increases K.E. of leaving steam while the discharge remains constant. Divergent portion is essential because for expansion beyond critical pressure and at the same time with increase of velocity, divergent section is needed. This type of nozzle is mostly used.

The velocity of expanding steam

$$V = \sqrt{2g\,J\,Uk}$$
$$= \sqrt{2 \times 9.81 \times 427 \times U \times k} = 91.5\sqrt{k\,U}$$

Where k is called the coefficient of friction. It has been found that there is a loss of 10 to 15% of total heat when friction effects are taken into account, and $U = H_1 - H_2$.

If initial velocity is ignored, then velocity at outlet of nozzle

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

Condition for throat pressure for maximum discharge :

Ratio $\frac{p_2}{p_1} = \left[\frac{2}{(n+1)}\right]^{\frac{n}{n-1}}$ is known as critical pressure ratio and it gives maximum discharge for the nozzle. p_2 (throat pressure) for this ratio is called critical pressure.

For a saturated dry steam. the value of $n = 1.135$ and putting this value of n in above equation

$$\frac{p_2}{p_1} = 0.580$$

So the nozzle using dry and saturated steam in the beginning has the throat pressure = 0.58 of initial pressure for the maximum discharge. This throat pressure is called the *critical pressure.*

When the steam is initially super heated then the value of n is 1.3 and the ratio of $\frac{p_2}{p_1}$ will be 0.545

The maximum discharge is given

$$m = \frac{A_2}{v_1}\left[2g\left(\frac{n}{n+1}\right)p_1 v_1\left(\frac{2}{n+1}\right)^{2/(n-1)}\right]$$

where A = throat area

Effect of Friction. The flow of steam through a nozzle is not a streamline flow, but there is always present a certain amount of eddies or irregular motion which causes friction between the molecules of the steam. Also there is some friction due to the contact of moving steam and the surface of nozzle. All this friction has the effect of reducing the final kinetic energy of steam, but the energy spent in friction is converted into heat energy and so it changes the final condition of steam.

Heat drop due to friction is nearly 10 to 15% of total heat drop and equation of velocity changes to

$$V = 91.5\sqrt{k\,U}\ \text{m/sec.}$$

Where k is the coefficient which is responsible for frictional loss.

This effect of friction is show on Mollier-diagram in Fig. 4.13.

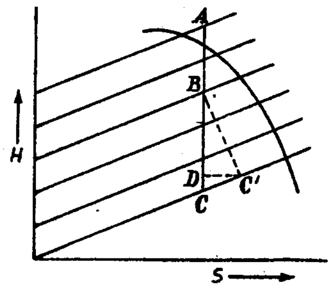

Fig. 4.13.

Let Point A represent the initial condition of steam, B at the throat and C at the exit when there is no loss of heat drop due to friction.

Suppose the loss of heat due to friction is CD then

$$\frac{AD}{AC} = k.$$

and hence the point D has been located, but the expansion should end on the same pressure line as C and hence draw a horizontal line DC' from D. The point C' gives the final condition of steam. It is seen from H–S. diagram that point C' has greater dryness fraction than C, hence the effect or friction is partly to dry the steam.

It has been found from experiments that loss of heat drop from enterance to throat is negligible in comparison to total drop in heat and hence for all purposes, we assume that effect of friction occurs only in divergent part of the nozzle. The flow of steam in that portion is shown by the dotted line BC' in Fig. 4.13.

The ratio of $\frac{AD}{AC}$ is called the efficiency of nozzle. The efficiency of nozzle is also defined as the ratio of actual kinetic energy of the jet to the kinetic energy with frictionless adiabatic flow.

Supersaturated flow. It occurs when the expansion through a nozzle, is so rapid that steam does not get enough time to condense, and thus remains dry.

Turbines

In turbines the internal energy of steam is converted into mechanical work by first converting steam energy into kinetic energy of a jet in stationary or moving nozzles, and then by change of direction of motion of jet while steam is passing over moving vanes or blades. In the *impulse turbines,* the steam pressure drop and consequent development of kinetic energy takes place solely in the stationary nozzles, and the work is obtained by the conversion of this kinetic energy into work on moving blades. In the *reaction turbine* only a part of the kinetic energy conversion occurs in the stationary nozzles, the remainder of the kinetic energy conversion being accomplished by a pressure drop in the steam as it passes through the moving blades. Steam turbines may be classified as

Turbines
- Impulse
  - De Leval
  - Simple curits
  - Rateau
  - Re-entry
    - Axial flow
    - Tangential type
  - Multi stage curtis
- Reaction
  - Axial flow, Parsons
  - Radial flow, Ljung-storm
- Combination types
  - Curtis Rateau
  - Curtis Parsons

The characteristic feature of *De Laval turbine* is the diverging nozzles which expand the steam to back pressure in a single stage, and a single steel disc, mounted on a slender flexible shaft, carrying the blades on its periphery. It has high steam and wheel velocities and large pressure drop in nozzle.

Simple Curtis impulse turbine has one nozzle or set of nozzles and single disc with two or more rows of blades, and intermediate reversing blades. One passage of steam takes place across each blade row. The wheel speed is moderate and pressure is same throughout the stage. Pressure drop in nozzle is large.

Rateau turbine consists of a number of De Laval elements in series with intermediate diaphragms carrying orifices. The steam expands in several pressure stages until completely expanded to the back pressure. Thus there is small pressure drop per stage, and most efficient ratio of wheel speed to steam speed can be secured.

In multistage Curtis turbine, series of simple Curtis wheels separated by diaphragms carrying orifices are arranged. Compared to Rateau, it usually has relatively few stages.

Parsons turbine has large number of rows of blades mounted on a rotor or revolving drum. Between each of these rows of blades is a row of stationary blades attached to the casing. The steam expands to a lower pressure in both sets of blades. A set of stationary blades and the following set of moving blades constitute a stage. There is small pressure drop per row and it is possible to achieve most efficient ratio of wheel speed to steam speed.

In radial flow reaction turbine, a series of radial rings of converging reaction blades are arranged with alternate rings revolving in opposite directions. It has elaborate steam packing devices. It also has high ratio of blade speed to steam speed.

Principle of action of steam turbine. In the reciprocating steam engine, the pressure energy of steam is utilised to overcome external resistance and the dynamic action of steam is negligibly small. Steam engines may be operated by using the full pressure without any expansion or drop of pressure in the cylinder. Such engines are said to work nonexpansively. The steam turbine could not be operated in this manner. A steam turbine depends wholly upon the dynamic action of the steam.

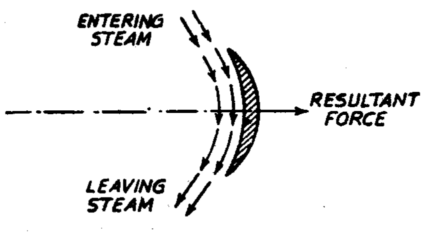

Fig. 4.14.

The steam is caused to fall in pressure during its passage in nozzle. Due to this fall in pressure, a certain amount of heat energy is converted into kinetic energy which causes the increase in velocity of steam. The rapidly moving particles of the steam enter the moving part of the turbine and suffer a change in direction of motion which gives rise to a change of momentum and therefore to a force. This constitutes the driving force of the machine. The process of expansion and direction changing may occur once, or a number of times in succession.

Impulse turbine. In its simplest form, it is shown diagrammatically in Fig. 4.15. The top portion of the figure shows a longitudinal section through the upper half of the turbine, the middle portion shows a development of nozzle and blading, while the lower part shows approximately how the absolute pressure and the absolute velocity of the steam vary from point to point during the passage of steam through the turbine.

This type of turbine is termed as simple impulse turbine because the expansion of steam takes place in one set of nozzles only. The pressure of steam falls from that in the steam chest to that existing in condenser while the steam flows through it and hence the pressure in the wheel chamber is practically equal to the condenser pressure. There is thus a relatively great ratio of expansion of the steam in the nozzles with the result, the steam issues from the nozzle at a very high velocity of the order of 1200 metres/sec. It will be shown later, that velocity of blades for maximum efficiency should be about one-half of the velocity of jet *i.e.* about 600 metres/sec in this case.

In practice this type of turbine is employed for small power ratings. The rotor diameter is kept fairly small and consequently the rotational speed becomes very high, of the order or 30,000 r.p.m. Since, only few driven machines require such a high speed and that is why, the chief object of the development of steam turbines is to reduce the speed of rotor to practical limits. There are several methods to reduce the speed, all of which employ several blade rings.

Another development effort in steam turbines is that if a single blade is used to utilise the kinetic energy of high velocity steam, the velocity of steam leaving the moving blades is in large proportion to the maximum absolute velocity of steam when leaving the nozzle. The exit velocity involves the loss of energy which is termed as 'Carry over losses' or 'Leaving losses'. A small final velocity is necessary in order that the steam may escape from the turbine into condenser with minimum losses. A moderate outlet velocity would involve only a small loss, since actual energy varies with the square of leaving velocity. But where a large velocity is inevitable, the corresponding leaving losses may reach a large value. In a De Laval turbine, the leaving velocity amounts to 33% of nozzle outlet velocity and hence the leaving losses may reach $(0.33)^2$ or 11% of initial kinetic energy of steam.

Another reason of development is that due to very high rotor speed, the stress in the shafting and blade material will be very high which requires the use of very good material and hardly any commonly used material can withstand these forces. If we use high stress material, the cost will be very high. Also high rotor speed causes a problem in design of turbine.

To overcome these difficulties, a development has been sought to bring the high rotor speed to practical limits and this has been achieved by using a series of blades which are keyed to a common-shaft in which jet velocity is absorbed is stages as it flows through the blades. This is called 'compounding'. The following are the chief methods of reducing rotor speed by compounding.

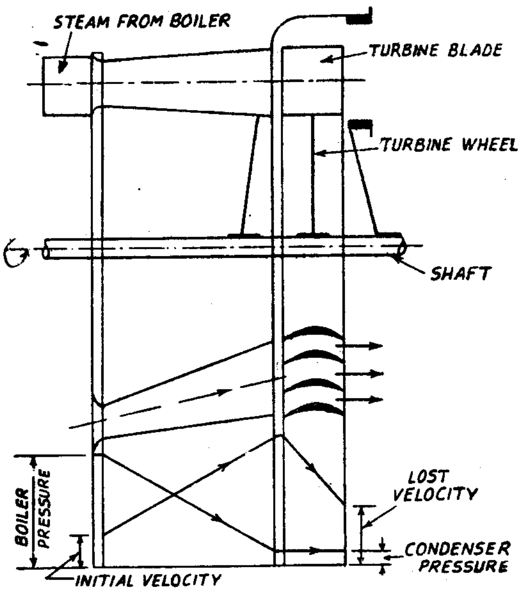

Fig. 4.15. Simple impulse turbine (De Laval).

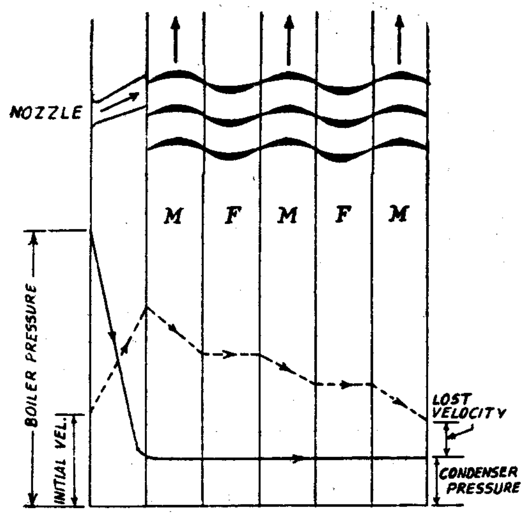

Fig. 4.16. Velocity compounding.

Velocity Compounding. This system consists of a nozzle or a set of nozzles and a wheel fitted with two or more rows of blades. This has been shown in Fig. 4.17 which has 3 rings of moving blades and such a wheel is referred to as three row-wheel. There are also a number of guide blades, suitably arranged between the moving blades and set in the reverse manner. With 3 row wheel, there are 2 rows of guide blades, placed between the first and second, and second and third rows of moving blades respectively. These fixed blades need not necessarily extend round the full circumference of casing. It is only necessary to arrange them roughly in front of nozzle or nozzles, but covering somewhat larger area of circumference than the nozzles themselves.

Steam entering the nozzle expands from the initial pressure to the exhaust pressure. Thus in general, the steam velocity is very high, as in single impulse turbine. The provision of two or more rows of moving blades, however, enables the blade velocity to be made appreciably less than would be the case with a turbine having a single row of moving blades. On passing through the first row of moving blades, the steam gives only a part of K.E. and issues from this row of blades with a fairly high velocity. It then enters the steam blade pair *i.e.* first in the guide blade and then on moving blade. There is a slight drop in velocity in the guide-blade due to friction. In passing through the second row of moving blades, the steam suffers a change in momentum and gives up a part of K.E. to rotor. Again the leaving steam from second row of moving blades is redirected to second row of guide blades. Thus doing work on third row of moving blades steam finally leaves the wheel in more or less axial direction with a certain residual velocity, about 2% of the initial velocity of steam at nozzle exit.

This arrangement of reducing the rotor speed to practical limit is called 'velocity compounding' *i.e.* whole of velocity is not utilised on one row of moving blades but in a number of stages. The example of this type is Curtis turbine.

Pressure Compounding. It is obvious that by arranging the expansion of the steam in a number of steps we can arrange a number of simple impulse machines in series on the same shaft, allowing the exhaust steam from one turbine to enter the nozzles of succeeding turbines. Each of the simple impulse turbine would be termed as 'stage' of the turbine, each stage comprising its sets of nozzles and blades.

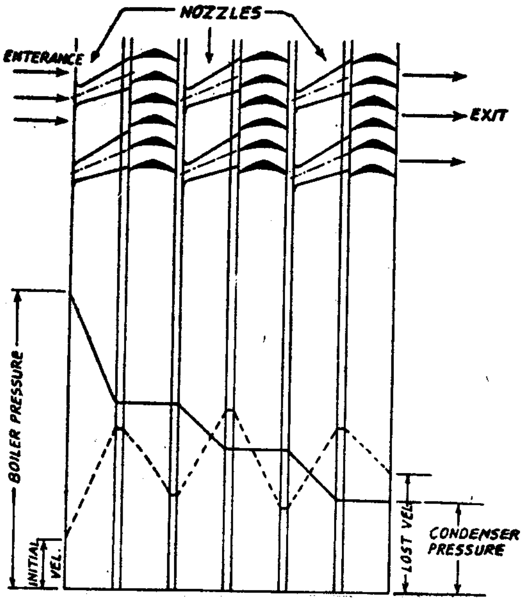

Fig. 4.17. Pressure Compounding.

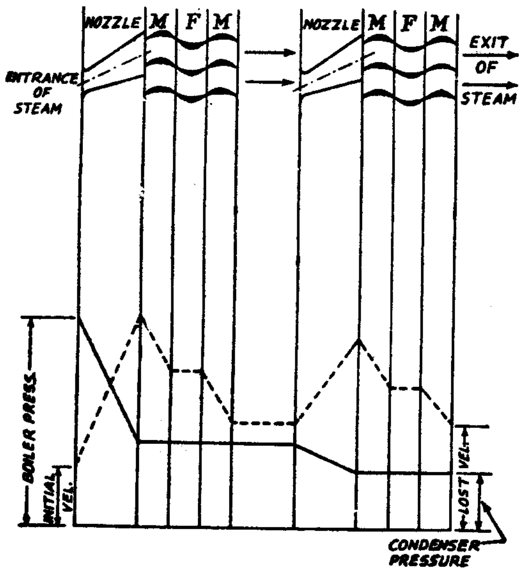

Fig. 4.18.

This is equivalent to splitting up the whole pressure drop, into a series of smaller-pressure drops, hence the term 'pressure-compounding'. The nozzles are usually fitted into partitions termed 'diaphragm' which separates one wheel chamber from the next. The wheels are mounted individually on the shaft or spindle and carry the blades on their periphery. Expansion of steam takes place wholly in nozzles, the space between any two diaphragms being filled with steam at constant pressure. The pressures on either side of diaphragm are therefore different, the greater difference of pressure occurring in first few stages. Hence the steam will tend to leak through the space between the bore of diaphragm and the surface of the shaft or the wheel hub. Special devices are fitted to minimise leakages.

The pressure compounding causes a smaller transformation of heat energy into K.E. to take place in each stage than in simple impulse turbine. Hence the steam velocity is much lower with the result that the blade velocity and rotational speed may be lowered.

Although the leaving velocity may bear the same ratio of the steam velocity at the nozzle outlet as it does in simple impulse-turbine, yet the K.E. at exit is a fraction of total energy, consequently the leaving loss in a pressure compounded turbine is also a fraction of heat associated with the simple impulse turbine and it is usually one or two per cent of the total available energy.

This type of turbine was developed by late Professor. A Rateau of Paris and on the other hand by Dr. Zoelly of Zurich and hence Rateau and Zoelly are the classifications of this type of turbine.

Pressure Velocity Compounding. It is a combination of pressure compounding as well as velocity compounding. The total pressure drop of the steam is divided in stages and then velocity of each stage is compounded. The advantage of combining these two types is to allow a bigger pressure drop in each stage and hence less stages are necessary which require a shorter turbine for a given pressure drop.

The pressure and velocity curves of this type of turbine are shown in Fig. 4.18. The diameter of such a type of turbine increases in each stage; this is done to accommodate for a larger volume of steam at the lower pressure. The pressure is constant in each stage and the turbine is therefore an impulse one.

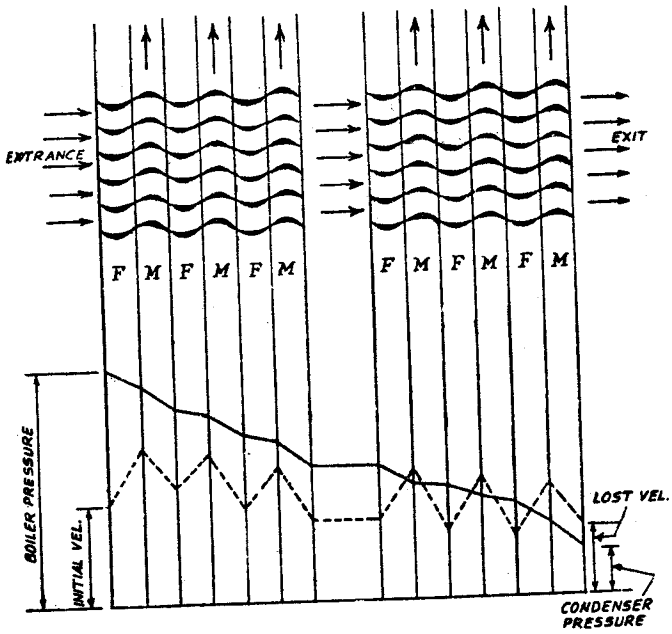

Fig. 4.19.

Reaction Turbines. (Refer Fig. 4.19). It aim is also to minimise loss of kinetic energy as in the case of impulse turbine. Instead of nozzles there are guide blades which convert the pressure energy into kinetic energy. The steam passing over the moving blades has the difference of pressure at the inlet tip and exit tip and hence there is a drop of pressure in steam while passing over the moving blades, but in the case of impulse turbine the pressure remains constant. The fixed blades serve the purpose of nozzles which change the direction of steam and at the same time allow it to expand to a higher velocity, the pressure of steam falls as it passes over the moving blades and hence these turbines are called reaction turbines. The diameter of each stage of this type of turbine must increase after each group of blade rings in order to accommodate the increased volume of steam at lower pressure.

Flow of steam through turbine blades. The following are the notations used in connection with velocity triangles of steam turbines.

v = Linear velocity of blades in metre/sec.

V_1 = Absolute velocity of steam entering the moving blades in metre/sec

V_2 = Absolute velocity of steam leaving the moving blades.

V_{w_1} = Velocity of whirl at entry of moving blades. (Tangential Component of V_1)

V_{w_2} = Velocity of whirl at exit. (Tangential component of V_2)

V_{f_1} = Velocity of flow at entrance (Axial component of V_1)

V_{f_2} = Velocity of flow at exit. (Axial component of V_2)

V_{r_1} = Velocity of steam relative to moving blade at entrance

V_{r_2} = Velocity of steam relative to moving blade at exit.

α = Angle which the entering steam makes with the tangent of wheel at entrance, or the exit angle of fixed blade but more commonly the nozzle angle.

β = Angle which the discharging steam makes with the tangent of wheel at exit, or the entrance angle of fixed blade.

θ = Inlet angle of moving blade.

ϕ = Exit angle of moving blade.

W = Weight of steam flowing over blades in kg/sec.

Q = Volume of steam flowing over blades in m^3/sec.

d = Diameter of blade drum in metre.

The suffix 1 represents the inlet condition of steam and suffix 2 represents the outlet condition of steam.

The jet of steam impinges on the moving blades at the angle α to the tangent of wheel with a velocity V_1 which is the absolute velocity of steam. This velocity V_1 has two components, one tangential component V_{w_1} and other axial component V_{f_1}. The former is called the whirl component and since it is in the same direction as the motion of blades, it is the actual component which does work on the blade. The latter is called flow component, since it is perpendicular to the direction of blade's motion and hence does no work, but this component is responsible for the flow of steam through the turbine. Also this component causes the axial thrust on the rotor.

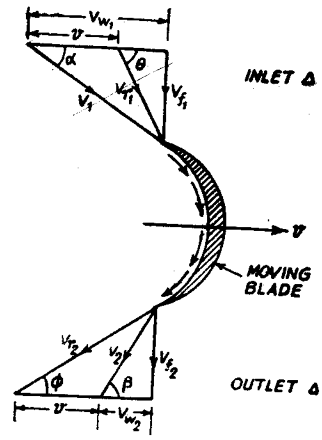

Fig. 4.20.

The moving blades are moving with a tangential velocity v in horizontal direction and the jet of steam with velocity V_1 at the angle α to the horizontal direction, both being in motion so actually steam enters the moving blades with a relative velocity V_r which can be obtained by subtracting the v and V_1 velocities and at some angle say θ_1. But in order to have a smooth flow of steam while passing over the moving blades, the jet of steam should enter the moving blade tip at the enterance angle of moving blade *i.e.* θ so that the angles, θ_1 and θ are same. When the angle θ_1 is equal to θ, the steam is said to glide over the moving blades and the flow of steam is smooth and steady. In actual practice both these angles are same and we denote them by θ.

Similarly the outlet velocity triangle can be explained.

Work done by steam on moving blades. The work done can be found out by change of momentum at inlet and outlet. The reaction of this change of momentum on the blade will be propulsive force. Considering that the W kg of steam is used.

Momentum at inlet $= \dfrac{W}{g} V_{w_1}$

Momentum at outlet $= \dfrac{W}{g} V_{w_2}$

Change of momentum $= \dfrac{W}{g} (V_{w_1} - V_{w_2})$

This change of momentum according to the Second law of the Newton is equal to impressed force or sinply force and hence tangential force on the blade is equal to change in momentum.

$$F = \frac{W}{g}(V_{w_1} - V_{w_2})$$

$$\left.\begin{array}{l}\text{Work done on blades}\\ \text{per sec.}\end{array}\right\} = \text{Force} \times \text{distance per sec.}$$

$$= \frac{W}{g}(V_{w_1} - V_{w_2}) \times v$$

$\therefore$ Horse power developed $= \dfrac{W}{g}\dfrac{v(V_{w_1} - V_{w_2})}{75}$

Work done per kg of steam $= \dfrac{v}{g}(V_{w_1} - V_{w_2})$

Energy supplied to blades per kg of steam $= \dfrac{V_1^2}{2g}$

Blade or diagram efficiency $= \dfrac{\text{Work done on blade}}{\text{Energy supplied}}$

$$= \frac{v(V_{w_1} - V_{w_2}) \times 2g}{g \times V_1^2} = \frac{2v(V_{w_1} - V_{w_2})}{V_1^2}$$

Stage efficiency $= \dfrac{\text{Work done on blade}}{\text{Total energy supplied per stage}}$

Stage efficiency, η $= \dfrac{(V_{w_1} - V_{w_2})\,v}{g \times \Delta h \times J}$

$$= \frac{\frac{1}{g}(V_{w_1} - V_{w_2})\,v}{\frac{V_1}{2g}} \times \frac{\frac{V_1^2}{2g}}{\Delta h \times J} = \text{blade } \eta \times \text{nozzle } \eta$$

Thus the stage efficiency is the same as blade efficiency except losses in nozzles.

Axial thrust is due to the difference of flow components of velocities as inlet and outlet.

Axial force on wheel $= \dfrac{W}{g}(V_{f_1} - V_{f_2})$

In all the turbines either the axial thrust is zero which is possible only when $V_{f_1} = V_{f_2}$ or thrust bearings must be provided so that axial thrust can be taken by it.

The efficiency of impulse turbine is maximum when blade speed

$$v = \frac{V_1 \cos\alpha}{2}$$

and the efficiency of reaction turbine is maximum when

$$v = V\cos\alpha$$

The maximum efficiencies of impulse and reaction turbines respectively are

$$\frac{\cos^2\alpha}{2}\left(\frac{V_{r_1}\cos\phi}{V_{r_2}\cos\theta} + 1\right) \quad \text{and} \quad \frac{2\cos^2\alpha}{1 + \cos^2\alpha}$$

Reheat Factor is defined as the ratio of cumulative enthalpy drop to isentropic enthalpy drop. Fig. 4.21 shows the expansion of steam in three stage turbine. For the first stage, A_1B_1 is the isentropic enthalpy drop, out of which A_1C_1 is the useful enthalpy drop. The ratio $\frac{A_1C_1}{A_1B_1}$ is called stage efficiency. At the exit from the first stage, the steam is at A_2 and not at B_1. Thus the quality of the steam at exit from first stage is improved.

2nd stage efficiency $= \frac{A_2C_2}{A_2B_2}$

and 3rd stage efficiency $= \frac{A_3C_3}{A_3B_3}$.

For the three stage turbine,

Reheat Factor (R.F.) $= \frac{\text{Cumulative heat drop}}{\text{Isentropic enthalpy drop}}$

$$= \frac{(A_1B_1 + A_2B_2 + A_3B_3)}{A_1D}$$

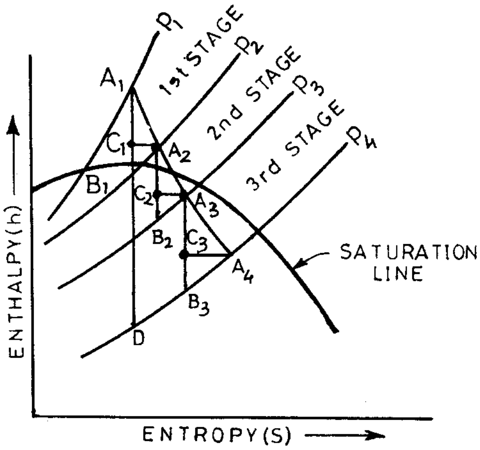

Fig. 4.21.

The value varies between 1.02 to 1.06.

The reheat factor depends on turbine stage efficiency, initial pressure and temperature, exit pressure, and number of stages.

Due to reheat factor the efficiency of a turbine as a whole is more than its individual stages and the efficiency of the complete turbine is known as **internal efficiency** of the turbine.

Internal efficiency $= \frac{\text{Total useful heat drop}}{\text{Isentropic heat drop}}$

$$= \frac{(A_1C_1 + A_2C_2 + A_3C_3)}{A_1D}$$

$$= \left(\frac{A_1C_1 + A_2C_2 + A_2C_3}{A_1B_1 + A_2B_2 + A_3B_3}\right) \times \text{R.F.}$$

$= \text{stage efficiency} \times \text{R.F.}$

assuming stage efficiency constant for all stages.

PROBLEMS

Provide single suitable word(s) for following statements :

PROPERTIES OF STEAM

1. At a pressure of 0.006112 bar, the melting and boiling temperature become equal and the change of phase ice-water-steam is shown on T-v diagram by ....
2. At a pressure of 0.006112 bar and temperature of 273.16°K, ice, water and steam coexist and such a condition is represented by ....
3. At pressures lower than 0.006112 bar, the ice .... to steam.
4. At a pressure of 221.2 bar and temperature of 374.15°C, the change of volume accompanying evaporation is ....
5. The condition of $p = 221.2$ bar and $t = 374.15$°C is called ....
6. At critical point, the latent heat of vaporisation is ....
7. There is no definite transition from liquid to vapour and two phases can't be distinguished visually.
8. Superheated steam behaves like .... gas.

9. At very high pressures, the volumes of saturated water .... by omitted from calculations.
10. During throttling process, ... remains unchanged.
11. Diagram showing properties of steam on enthalpy vs. entropy chart.
12. With increase in pressure, the enthalpy of evaporation ....
13. With increase in pressure, the enthalpy of dry saturated steam ....
14. For measuring dryness fraction of the order of 0.98, ... calorimeter is used.

STEAM BOILERS

15. Units used with pulverised coal fired boilers in order to reduce atmospheric pollution.
16. The coal and air pass directly from the pulvèriser to the burners and the desired firing rate is regulated by the ratio of pulverising.
17. Type of pulveriser adaptable to highly abrasive fuels having high silica content.
18. It mixes and directs the flow of fuel and air so as to ensure rapid ignition and complete combustion.
19. Air initially mixed with the fuel to obtain rapid ignition and to act as a conveyor for the fuel.
20. Air introduced to the burners outside of the primary-air ports, in order to ensure complete combustion.
21. Type of atomiser used for providing an operating range of 10 to 1 in oil burners.
22. The burner tips in corner fired boiler are tilted to control ....
23. A furnace designed to burn low-ash-fusion coals and to retain most of the coal ash in the slag.
24. The pressure limit upto which drum type natural or assisted circulation boiler are restricted because of circulation of steam-separation characteristics.
25. Component to add heat to steam after evaporation in furnace.
26. Component to remove heat from the moderately low temperature combustion gases after superheat/reheat sections of the boiler unit.
27. Type of air heater in which the stationary metal parts form a separating boundary between the heating and cooling fluids, and the heat passes by conduction through the metal wall.
28. Type of air heater in which heat transferring members are moved alternately through the gas and air streams undergoing successive heating and cooling cycles and transferring heat by the internal storage capacity of the members.
29. Circulation produced in boiler by the difference in the densities of the water in the unheated down-comers and the steam-water mixture in the heated tubes.
30. In assisted circulation type of boilers, the quantity of water pumped (usually) in comparison to the amount of steam evaporated is ....
31. Boiler requiring no steam drum and using relatively small-diameter tubes.
32. Effect caused by the difference in densities of flue gases in chimney and surrounding air.
33. The greatest factor in the corrosion of steel surfaces in contact with water in boilers is ....
34. Dissolved oxygen is removed from feedwater for boiler in ....
35. Corrosion in boilers is prevented or minimised by maintaining .... boiler water.
36. The pH of a water sample can be determined accurately by the measurement of its ....
37. The elimination of hardness in boiler water is necessary to prevent ....

STEAM ENGINE

38. Steam engines depend for their operation on the .... power of the steam.
39. The expansion ratio in steam engines is limited due to ....
40. The ratio of expansion in simple steam engines is of the order of ....
41. The efficiency of steam engines is dependend largely on the value of the ....
42. The ratio of area of actual indicator diagram to the area of theoretical card is called ....
43. The mechanical efficiency of steam increases as the load ....
44. Condensation losses in steam engine are related to the temperature difference existing in the cylinder which can be reduced by using ....
45. .... result in overall improvement in performance and water rate, and also reduces the cylinder condensation.
46. The efficiency of steam engines is expressed in terms of ....
47. In .... arrangement of steam engine, the temperature of the residual steam at the end of compression is high which results in economy of steam.
48. Steam engine economy may be improved by separation of inlet and outlet ports, applying steam jackets to cylinders and heads, and using ....
49. Losses in steam engine cylinders are due to incomplete expansion, initial condensation, radiation and ....
50. .... cycle is the accepted thermodynamic standard for comparing the performance of the steam engine.
51. There is no compression or clearance in the ....

STEAM TURBINES

52. According to details of .... design, the steam turbines may be classified as impulse or reaction.
53. The total pressure drop for the stage is taken across the nozzles.
54. The flow through the rotor blades is substantially at constant static pressure.
55. Multivelocity stages occur in .... turbine.
56. The total pressure drop assigned to the stage is divided equally between the stationary blades and the rotor blades.
57. The .... buckets do much more turning of the steam.
58. Blade length/steam passage width.
59. Leakage loss of steam between inner circumference of stationary element and rotor is minimised by maintaining minimum practical clearance and by use of ....
60. In stages with short blades, the best net efficiency obtains with near .... design.
61. The presence of moisture in the steam causes extra ......
62. The ratio of total internal used energy to the isentropic available energy.
63. In condensing turbines expanding to high vacuum, the ratio of volume of steam at exhaust to that at inlet is around ....
64. .... keep the rotor turning at slow speed to maintain uniform temperature when the turbine is shutdown and cooling.

65. .... on the turbine rotor is caused by pressure and velocity differences across the rotor blades.

66. The performance of central-station turbine-generators is generally expressed as ....

67. The unit of heat rate is ....

ANSWERS

1. a straight line	2. triple point line	3. sublimates	4. zero
5. critical point	6. zero	7. at critical point and above	
8. perfect	9. can not	10. enthalpy	11. Mollier diagram
12. decreases	13. decreases	14. throttling	15. electrostatic precipitators
16. direct fired system	17. slow-speed, rotating drum type		18. burner
19. Primary air	20. Secondary air	21. Steam and air type	22. steam temperature
23. cyclone	24. 183 kgf/cm^2	25. superheater	26. economiser
27. recuperative	28. regenerative	29. natural	30. 4–6 times
31. once through type	32. stack	33. dissolved oxygen	34. deaerator
35. alkaline	36. electrical potential	37. scale	38. expansive
39. practical losses	40. 4	41. ratio of expansion	42. diagram factor
43. increases	44. cylinders in series	45. superheating	46. kg of steam per h.p. hour
47. uniflow	48. multiple expansion	49. throttling	50. Rankine
51. Rankine cycle	52. stage	53. impulse stage	54. impulse
55. curtis	56. reaction stage	57. impulse	58. aspect ratio
59. labyrinth packings	60. impulse	61. losses	62. internal efficiency
63. 1000	64. turning gear	65. axial thrust	66. heat rate
67. kcal/kWhr			

5

Compressors, Gas Turbines and Jet Engines

Compressors

Compressors may be divided into two main classes : reciprocating and rotary.

Assuming clearance and leakage to be absent, total **work done** on reciprocating compressors is given by

$$W = \int_{p_1}^{p_2} v\,dp = \frac{\gamma}{\gamma - 1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] p_1 V_1 \quad \text{for adiabatic compression}$$

γ = ratio of specific heats.

This is also equal to $\frac{\gamma}{\gamma - 1} R\,(T_2 - T_1)$

Work done in case of **isothermal compression**

$$= p_1\, v_1 \log_e \frac{V_1}{V_2}$$

Work done is minimum when compression follows isothermal law, *i.e.* $pv = c$ or $n = 1$ and is maximum for adiabatic compression.

Compressor efficiency or Isothermal efficiency

$$= \frac{\text{Isothermal work done}}{\text{Indicated work done}} = \frac{p_1 v_1 \log_e \dfrac{V_1}{V_2}}{\dfrac{n}{n-1} p_1 v_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

Mechanical efficiency $= \dfrac{\text{Indicated h.p.}}{\text{B.H.P. of motor}}$

Adiabatic efficiency $= \dfrac{\text{Adiabatic h.p.}}{\text{B.H.P. of motor}} = \dfrac{\left\{\left(\dfrac{\gamma}{\gamma-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]\right.}{\left(\dfrac{n}{n-1}\right)\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$

Compression work is least for isothermal compression and attempts need to be made to approach same.

Advantages of **multi-stage compression** are :

(*i*) An opportunity is given to cool the air in external cooler on discharge from cylinder and before entering the next cylinder.

(*ii*) Better mechanical balance, and uniform torque of multi-crank machines and smaller flywheel.

(*iii*) Increased volumetric efficiency as a result of lower pressure in l.p. cylinder clearance.

(*iv*) Reduction of power to drive.

(*v*) Possibility of running at higher speeds.

(*vi*) Provision of better lubrication due to smaller working temperature.

(*vii*) Smaller leakage loss and lighter cylinders.

Ideal intercooler pressure p_2 for two stage polytropic compression operating between pressures p_1 and p_3 is given by

$$p_2 = \sqrt{p_1 p_3}$$

Work done for N stages in case of perfect intercooling will be

$$\frac{Nn}{n-1} p_1 v_1 \left[\left(\frac{p_{N+1}}{p_1} \right)^{\frac{n-1}{Nn}} - 1 \right]$$

Heat rejected with perfect intercooling

$$= \left[C_p + C_v \left(\frac{\gamma - n}{n-1} \right) \right] (T_2 - T_1)$$

Volumetric efficiency of compressor considering the clearance

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

where $\dfrac{v_c}{v_s}$ = Clearance ratio

$$= \frac{\text{Clearance volume}}{\text{Swept volume}}$$

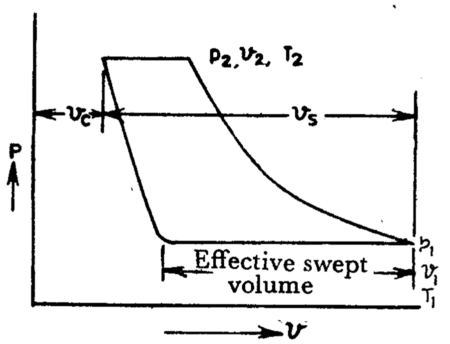

Fig. 5.1.

Fig. 5.1 shows the compressor cycle on p-v diagram

Net work done during a compression cycle

$$= \frac{n}{n-1} p_1 \times \text{Effective swept volume} \times \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

Fig. 5.1 shows theoretical cycle. In practice throttling of air at intake causes the suction pressure to fall; throttling of air at outlet causes the discharge pressure to rise; clearance causes expansion on the outstroke, thereby reducing the capacity of machine.

It is important of note that

(*i*) throttling the low-pressure suction increases the temperature range in successive stages;

(*ii*) increasing the clearance in one stage causes more work to be done in the lower stage.

Control of Compressors

The amount of air delivered by the compressor can be controlled by

(*a*) *Throttle control* — In this method, the suction valve of compressor is throttled to permit less suction air to be compressed.

(*b*) *Clearance control* — For high pressures in receiver, the clearance volume is increased by opening some valve, thereby reducing volumetric efficiency.

(*c*) *Blow-off control* — The compressed air from compressor is bypassed to atmosphere, when pressure in the receiver is high.

Centrifugal compressors. Single stage centrifugal compressors can produce pressure ratio of 4 : 1. These have smaller length and can perform efficiently over wide range of mass flows at any particular speed. The performance is not deteriorated by contaminated atmosphere. However, these have large frontal area as compared to axial flow compressor and have lower maximum efficiency. These find use in superchargers and turbopropeller units. These are preferred where simplicity, light weight and ruggedness are the criteria.

In centrifugal compressor, the air contained in its rotating passages is subjected to a centrifugal force which causes the air to flow radially outward. Pressure rise takes place in diffuser.

H.P. required to drive an ideal compressor

$$= \frac{U_2^2}{75g} \text{ per kg of flow/sec.}$$

(where U_2 = blade tip velocity)

$$= \frac{JC_pT_{01}\left[(r_p)^{(\gamma-1)/\gamma} - 1\right]}{75}$$

where r = static pressure ratio.

Roots Efficiency — It is the ratio of work required per minute for ideal isentropic compression to the work required per minute to drive the roots blower compressor. Mathematically

$$\eta_{roots} = \frac{\frac{\gamma}{\gamma-1}\left[(r_p)^{(\gamma-1)/\gamma} - 1\right]}{(r_p - 1)}$$

where r_p = pressure ratio

$$= \frac{\frac{C_p J}{R}\left[(r_p)^{(\gamma-1)/\gamma} - 1\right]}{r_p - 1}$$

Forward, radial and backward curved blades. Backward curved blades have slightly higher efficiency and are stable over a wider range of flow. Forward curved blades can produce the highest pressure ratio for a given blade tip speed, but it is inherently less stable and possesses a narrow operating range. Good performance can be had with radial impeller blades which is exclusively used in turbojet engine applications.

Characteristics of Centrifugal Compressors

The ideal radial vane centrifugal compressor will have constant pressure ratio with change in mass flow. But losses like friction losses (proportional to V^2 or mass^2) and incidence losses (proportional to drag coefficient × V^2) make the pressure ratio to first increase with increase in mass flow, reach a maximum value and then fall as shown in Fig. 5.2.

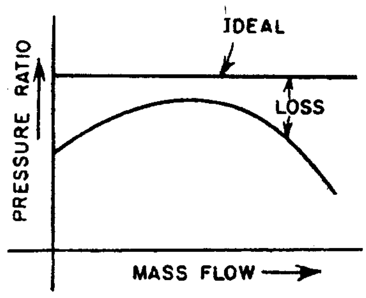

Fig. 5.2.

Surging and Choking

Phenomena of surging (on pumping) occurs in centrifugal compressors at low mass flow rates when compressor is operated at any point on left of the maximum pressure ratio point. This instability is very severe in compressors producing high pressure ratios and it may lead to physical change due to impact loads and high frequency vibration.

At higher mass flow rate points, a stage is soon reached when mass can't be increased any further, known as choking.

The surge line is marked on compressor characteristic curves and is found to be very close to peak efficiency line. When the compressor is used with the gas turbines then the characteristics must be matched properly, otherwise problems of surging or low efficiency will be experienced.

Axial Flow Compressors. The blading of axial flow compressors is designed on an aerodynamic basis. These find wide application in aircraft and industrial gas turbines because of higher efficiency and capability of producing higher pressure ratio on a single shaft. Stage pressure ratio of about 1.2 can be obtained with overall pressure ratio of upto 8 : 1 or even higher. A number of stages are clubbed together. These produce high thrust per unit frontal area. These have advantage of large air handling capacity with a small frontal area, a straight through flow system and high pressure ratios with relatively high efficiencies. However it is a complex unit with high initial cost. It consists of an alternating sequence of fixed and movable set of blades.

Stall and Surge. Blade stall is associated with reduced flow rate. At low speed, the absence of pressure in the last stages leads to choking. Stalling does not necessarily cause a complete breakdown of flow but this is only a mild instability. Sufficient extensive stall results in a sufficient reduction of total flow and the mechanism of surging takes place. Stalling is the phenomenon in which air stream is not able to follow the blade contour.

Characteristics. Axial flow compressor characteristics are quite similar to the centrifugal type but these cover a narrower range of mass flow than centrifugal type.

Curves at two ends are limited by surging and choking. Surge points in axial flow compressors are reached before the curves reach a maximum value. Therefore the design point which is always near to the peak of the characteristic is also very near the surge lines making operating range quite narrow, calling for great care in selection for a duty.

Gas Turbines

Gas turbines find application in power generation and are best suited for peaking loads due to quick starting. These are pre-eminent as an aircraft power plant and capacity-wise go beyond 35000 h.p. Compared to I.C. engines it has the advantage of weight, size and shape, but poor in fuel economy.

Dynamics of fluid flow. This can be analysed by considering

(*i*) Conservation of mass

(*ii*) Newton's second low

(*iii*) First law of thermodynamics (conservation of energy) and

(*iv*) Second law of thermodynamics.

Subsonic, supersonic and sonic flow. At subsonic flow (Mach no. (M, < 1), with increase in area, pressure and density increase but velocity decreases.

At supersonic flow ($M > 1$), with increase in area, pressure and density decrease but velocity increases.

At sonic flow ($M = 1$), area is minimum, the pressure and density changes near sonic condition become very large for even a small change in area.

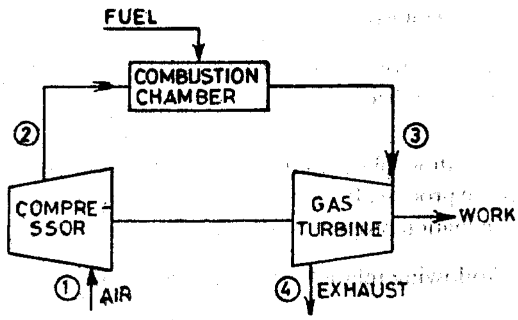

Fig. 5.3.

Also the change in velocity and change in density compensate near sonic velocities.

Fig. 5.3 shows a diagrammatic layout of a simple gas turbine plant based on Brayton cycle. Its cycle thermal efficiency

$$= 1 - \frac{1}{(P_r)^{(\gamma-1)/\gamma}}$$

P_r = pressure ratio for the expansion phase of cycle

The gas turbines may be classified as open cycle, closed cycle, and semi-closed cycle. The open cycle gas turbines are further classified as constant volume (working on Atkinson cycle) and constant pressure (working on Brayton cycle). Closed cycle gas turbines are similar to steam turbine plants. Semi-closed cycle employs two turbines, closed cycle gas turbine driving the compressor, and open cycle turbine driving the main generator.

Gas turbines are based on Brayton cycle for which following observations should be noted. With regeneration, efficiency reduces as the pressure ratio increases because with increase in pressure ratio, the compressor delivery temperature rises and ultimately exceeds that of the exhaust gas from the turbine. At this point heat in the regenerator is lost from the air to the exhaust.

In many cases, regeneration is not desirable. With high pressure ratios, efficiencies are higher without regeneration because of heat loss from compressed air to the exhaust gases.

Efficiency, with or without regeneration, rises very rapidly with increase in maximum temperature of the cycle.

Low pressure-ratios and high-temperatures are favourable for regenerative cycle due to possibility of large heat recoveries.

For a given temperature-ratio, there is a fixed pressure ratio at which efficiency is maximum.

Limitations in actual gas turbine cycles

(*i*) Compression and expansion processes are not frictionless. However these are nearly adiabatic because of negligible heat transfer.

(*ii*) Pressure losses occur in combustion chamber, heat exchanger, ducting etc.

(*iii*) Specific heats of the working fluid vary throughout the cycle due to temperature change and contamination of by-products of combustion.

(*iv*) Heat exchangers are not ideal.

(*v*) Negative work to drive compressor and other auxiliaries is more than theoretical.

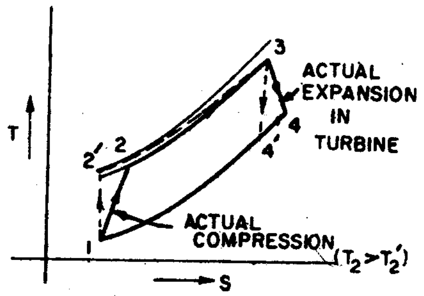

Fig. 5.4.

(*vi*) In ideal case it is assumed that velocity of gas stream and hence kinetic energy is same before and after the process. In actual cycle, the changes in kinetic energy are appreciable and hence the need to consider the stagnation properties (denoted by zero suffix at bottom) for calculating work done in various components.

Following relations may be noted. Efficiency of compressor

$$\eta_c = \frac{h_{02}' - h_{01}}{h_{02} - h_{01}}$$

If $\quad r_c$ = total-head pressure ratio, or stagnation pressure ratio for compression,

then
$$\eta_c = \frac{T_{01}\left(r_c^{\frac{\gamma-1}{\gamma}} - 1\right)}{T_{02} - T_{01}}$$

Efficiency of turbine
$$= \frac{h_{03} - h_{04}}{h_{03} - h_{04}'}$$

and if r_t = total-head pressure ratio for expansion

then
$$\eta_t = \frac{T_{03} - T_{04}}{T_{03}\left(1 - \frac{1}{r_t^{(\gamma-1)/\gamma}}\right)}$$

Actual work of compression
$$= \frac{C_p T_{01}}{\eta_c}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right)$$

Actual work of expansion
$$= \eta_t\, C_p \cdot T_{03}\left[\frac{r_p^{\frac{\gamma-1}{\gamma}} - 1}{r_p^{(\gamma-1)/\gamma}}\right]$$

If works of compression and expansion be equated, then minimum temperature ratio
$$\frac{T_{03}}{T_{01}} = \frac{r_p^{\frac{\gamma-1}{\gamma}}}{\eta_t \cdot \eta_c}$$

For gas turbine, the "negative" work of compression is a considerable fraction of the total turbine work and thus the compressor efficiency is very significant. The magnitude of effect of compressor and turbine efficiencies can be assessed by a parameter known as "work ratio", which is equal to ratio of net work to the total turbine work.

Work ratio
$$= 1 - \frac{r_p^{\frac{\gamma-1}{\gamma}}}{\eta_c \eta_t} \cdot \frac{T_{01}}{T_{03}}$$

It may be noted that work ratio is increased by high-temperature ratio $\frac{T_{03}}{T_{01}}$ and by a low pressure ratio r_p.

Overall cycle efficiency
$$\eta = \frac{C_{pg} \cdot \eta_t T_{03}\left[1 - \frac{1}{r^{\frac{\gamma_g-1}{\gamma_g}}}\right] - \frac{1}{\eta_{mech}} C_{pa} \cdot \frac{T_{01}}{\eta_c}\left[r^{\frac{\gamma_a-1}{\gamma_a}} - 1\right]}{C_{pg}\left[T_{03} - T_{01}\left\{\frac{1}{\eta_c} \cdot \left(r^{\frac{\gamma_a-1}{\gamma_a}} - 1\right) + 1\right\}\right.} right])$$

where C_{pa} and C_{pg} are mean specific heats of air and gas respectively ; γ_a and γ_g are ratio of specific heats for air and gas
$$r = \frac{p_{03}}{p_{01}}$$

The variations of overall cycle efficiency and specific output (HP/kg of air/sec) vs pressure ratio r_p for T_{03} of 900, 1000 and 1100°K are shown in Fig. 5.5 and 5.6. Its would be seen that effect of increasing

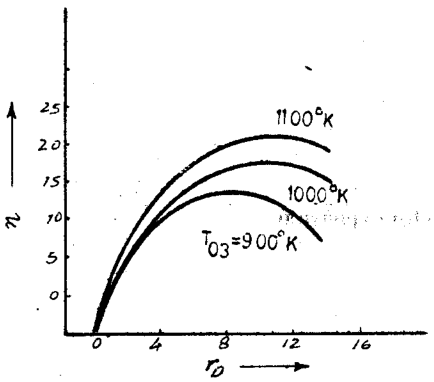

Fig. 5.5.

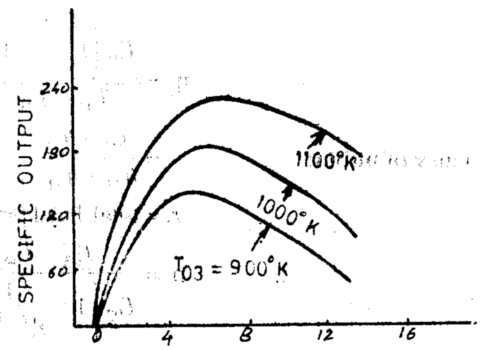

Fig. 5.6.

T_{max} is to increase both η and specific output. Both η and specific output have maximum values at certain value of r_p. Increase of T_{max} has more effect on specific work output than on efficiency.

Effect of Regeneration. Addition of heat exchanger in gas turbine cycle increases values of both overall cycle efficiency and specific output for given pressure ratio. The effect on specific output is small due to increased pressure losses and hence, reduced pressure ratio across the turbine. Regeneration lowers the pressure ratio at which the peak efficiency occurs, permitting use of simple and cheaper compressor of high efficiency.

Effect of intercooling. For ideal cycle, intercooling improves the work output but decreases the efficiency. In actual cycle, the specific output increases considerably and peaks at much higher pressure ratio. Efficiency is little changed at low pressure ratios but increases as the pressure ratio increases. This is because with irreversible compression the saving in negative work with intercooling overweighs the additional fuel efficiency, thus though not large, the intercooling achieves higher work output per kg of air, making more compact unit. Both intercooling and regeneration, in addition to increasing specific output also improve efficiency.

Work Ratio. It is the ratio of net work output from the plant to the work obtained from the turbine.

Fig. 5.7 shows a gas turbine cycle using intercooler, reheater, combustion chamber, heat exchanger, etc.

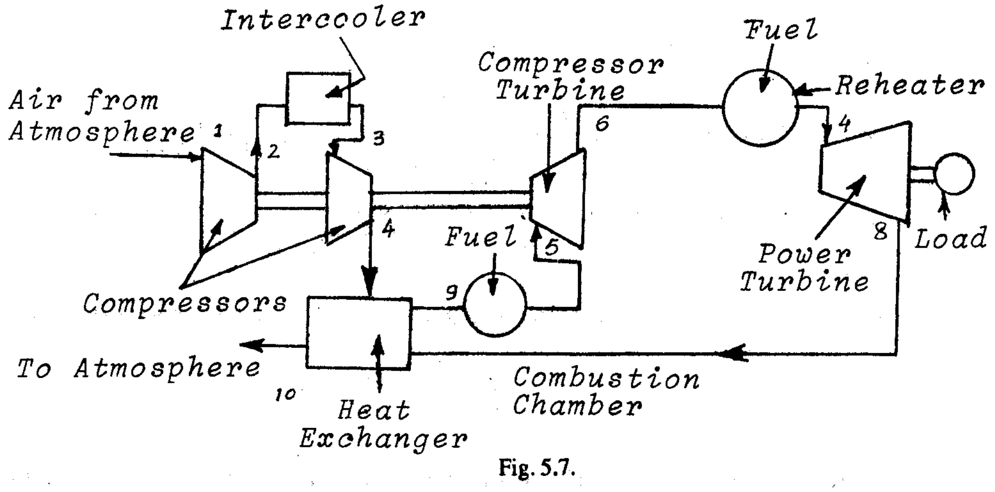

Fig. 5.7.

Fig. 5.8 shows the cycle shown in Fig. 5.7 on *T-S* diagram.

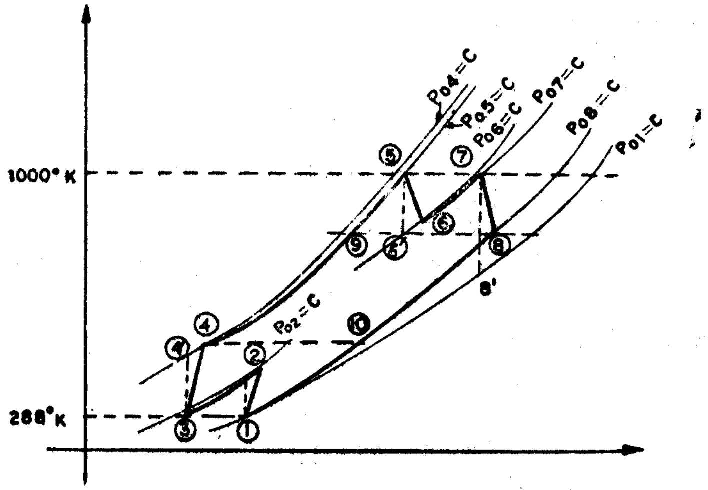

Fig. 5.8.

Jet Propulsion

Basic ideal cycle for a jet propulsion unit is Joule cycle. Actual cycle is shown in Fig. 5.9. The operation of this cycle is as under :

A—B Auto compression in the inlet duct of diverging shape; some kinetic energy being converted to heat which results in slight increase in volume.

B—C Compression due to vortex motion in impeller, there being further heating.

C—D. Diffuser converting some of kinetic energy of impeller discharge into pressure energy.

D—E. Heat absorption at constant pressure in combustion chamber.

E—F. Adiabatic expansion through turbine nozzles and blades.

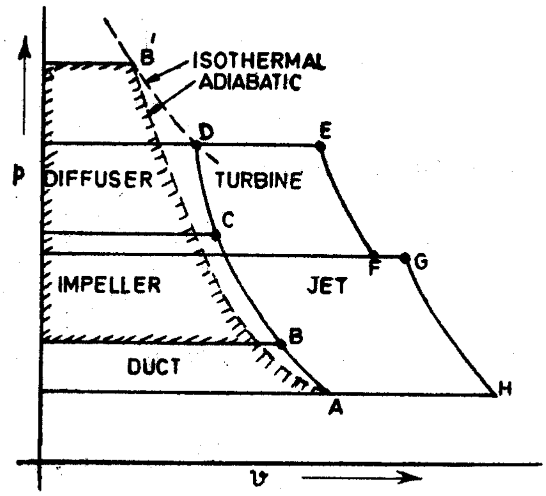

Fig. 5.9. *p-v* diagram for jet propulsion unit.

F—G. Reheating due to friction losses in nozzles and blades.

G—H. Adiabatic expansion in jet tube.

Propulsive power $= \dfrac{U^2 - V^2}{2g}$

and thrust power $= (U - V)\,V$

∴ Propulsive efficiency

$$= \frac{\text{Thrust power}}{\text{Propulsive power}} = \frac{2(U-V)\,V}{U^2 - V^2}$$

$$= \frac{2V}{U+V} = \frac{2}{1+\dfrac{U}{V}}$$

Exact value of propulsive efficiency

$$= \frac{2\left[\frac{V}{U}\left(1+\frac{1}{AFR}\right)-1\right]}{\left(\frac{V}{U}\right)^2 - 1 + \frac{1}{AFR}\left[\left(\frac{U}{V}\right)^2 + 1\right]}$$

where U = speed of jet

V = velocity of the products of combustion leaving the jet-tube.

AFR = air fuel ratio or mixture strength.

Thermal efficiency of jet propulsion unit

$$= \frac{\frac{V^2 - U^2}{2gJ}}{\frac{\text{Calorific value of fuel}}{AFR}}$$

More exact value of thermal efficiency

$$= \frac{\left[\left(\frac{V}{U}\right)^2 - 1 + \frac{1}{AFR}\left\{\left(\frac{V}{U}\right)^2 + 1\right\}\right]}{\frac{1}{AFR}\left[\frac{2gJ}{U^2} \times \text{Calorific value} + 1\right]}$$

Overall thermal efficiency

$$\eta_0 = \frac{2AFR\left[\frac{V}{U}\left(1+\frac{1}{AFR}\right)-1\right]}{\left[2gJ \times \frac{\text{Calorific value}}{V^2} + 1\right]}$$

In case of rocket $AFR = 0$

$$\eta_0 = \frac{2\frac{U}{V}}{\frac{2gJ \times C.V.}{U^2} + 1}$$

PROBLEMS

Jet engines are more efficient than screw propellers at speeds above sonic speed and also at high altitudes.

Provide single suitable word(s) for following statements :

1. As the value of index of compression 'n' increases, the work done in compressing air ....
2. If n is the index of compression, then work done in compressing air is proportional to ....
3. Work input to compressor is minimum when the law of compression followed is ....
4. The compression law $pv^{1.4}$ = constant on temperature – entropy diagram is represented by .... line.
5. The volumetric efficiency with fixed clearance volume and suction being from atmosphere and with increase in delivery pressure will ....
6. For the given pressure ratio, the leakage past the piston in multi stage compression in comparison to single stage compression is ....

7. If index of compression for first stage of compression is higher than second stage, then for perfect inter-cooling and minimum total work, the first stage shares .... work than second.
8. If m and n be the indices of compression for first and second stages of compression, then with perfect intercooling and for minimum work done, the ratio of first stage to second stage of work is ....
9. Low value of 'n' (index of compression) is obtained by ....
10. In case of compressor used with gas turbine, ... compression is preferable.
11. Amount of work done to compress a given mass of air under specified conditions is .... clearance ratio.

ANSWERS

1. increases	2. $\left(\frac{n}{n-1}\right)$	3. isothermal	4. vertical
5. decrease	6. less	7. more	8. $\left(\frac{m\,(n-1)}{n\,(m-1)}\right)$
9. cooling the compressor cylinder		10. adiabatic	11. independent of

6

Heat Transfer

Heat transfer may occur by conduction, convection and radiation. For heat transfer to take place, temperature difference must exist. (This is as per second law of thermodynamics). Conduction heat transfer takes place by molecular energy interchanges or by flow of valence electrons in conducting medium. Convection heat transfer takes place by motion of a fluid. If the convection is induced by density difference resulting from temperature differences within the fluid, it is known as natural convection. However, if fluid motion is aided by a pump or fan, then the process is called forced convection. Radiation heat transfer does not depend on any material medium, but takes place by means of electromagnetic waves which are propagated at a velocity comparable to that of light. In actual practice, heat transfer takes place by a combination of modes operating in series or parallel.

In heat transfer by *conduction,* heat flow dQ through an area dA in a plane normal to the direction of heat transfer in time dt is given by

$$dQ = -kdA\frac{dT}{dx}dt = -k\,dy\,dz\frac{dT}{dx}dt \qquad \text{(Fourier's law of conduction)}$$

where $\frac{dT}{dx}$ = temperature gradient in plane of heat transfer

and k = thermal conductivity, which depends on the state of the substance (solid, liquid or gas), and for a given state it varies somewhat with temperature (units of k are kcal/m hr °C) and rate of heat transfer

$$q = \frac{kA\,\Delta T}{\Delta x}.$$

where Δx = length of heat flow path or thickness of material.

Thermal diffusivity, property of the substance is defined as $\frac{k}{\rho C_p}$.

In case of laminated flat wall, shown in Fig. 6.1.

$$q = \frac{A(T_a - T_d)}{\frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3}}$$

In case of homogeneous tube wall, [Fig. 6.2].

$$q = \frac{2\pi Lk(T_i - T_0)}{\log\frac{r_0}{r_i}}$$

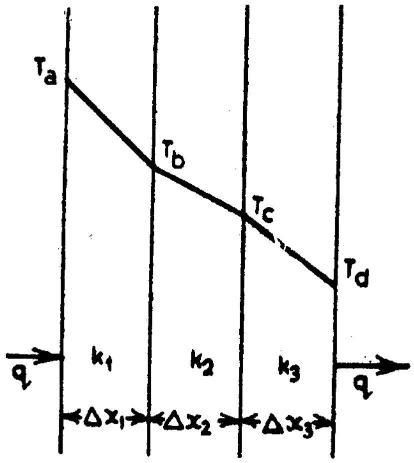

Fig. 6.1.

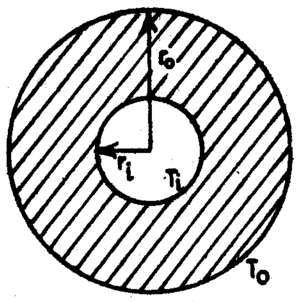

Fig. 6.2.

In case of thick spherical shell,

$$q = \frac{kA\,(T_i - T_0)}{r_2 - r_1}$$

where $\quad A$ = means area of heat transfer $= 4\pi r_1 r_2$

Heat exchangers. These could be of direct contact type, regenerative or storage type, recuperative or transfer type.

Convection heat transfer rate

$$q = \frac{k}{\Delta x} A \Delta T$$

where $\quad \Delta T$ = temperature difference between main stream inside tube and the wall surface.

$$\frac{k}{\Delta x} = h_e = \text{film coefficient.}$$

The value of h_e depends on the physical properties and the velocity of fluid, shape of wall, whether flow is vertical or horizontal, and several other factors.

$\frac{1}{h_c A}$ is equivalent of the thermal resistance of the film heat transfer process.

In case of forced convection heat transfer

$$h_e = 0.02\,\frac{h}{D}\left(\frac{\rho v D}{\mu}\right)^{0.8}\left(\frac{C_p \mu}{k}\right)^{0.4}$$

where $\quad v$ = fluid velocity, ρ = density

C_p = specific heat capacity.

k = thermal conductivity

μ = viscosity of fluid,

D = inside dia. of tube

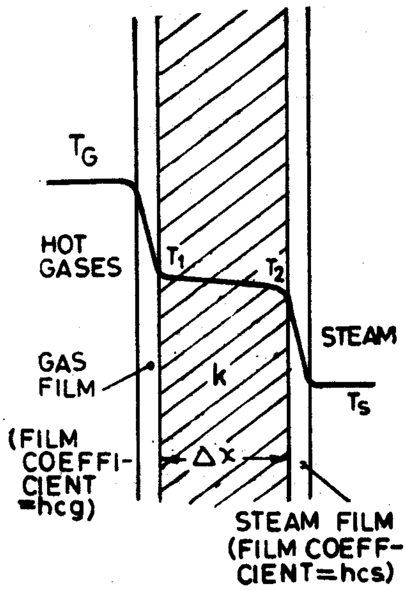

Fig. 6.3. Heat transfer by convection and conduction through a flat homogeneous wall.

$$\frac{\rho_v D}{\mu} = \text{Reynolds number}$$

$$\frac{h_c D}{k} = \text{Nusselt number}, \frac{C_p \mu}{k} = \text{Prandtl number.}$$

or $$\frac{h_c D}{k} = f\left[\frac{C_p \mu}{k}, \frac{\rho_v D}{\mu}\right]$$

or $$Nu = f(Pr, \text{Re})$$

The above equation for h_c is applicable for longitudinal flow outside of tube and cannot be used for flow normal to tubes.

For free convection transfer

$$\frac{hl}{k} = f\left[\left(\frac{C_p \mu}{k}\right), \left(\frac{\beta g \rho^2 l^3 \theta}{\mu^2}\right)\right]$$

where $\frac{\beta g \rho^2 l^3 \theta}{\mu^2}$ is known as Grashoff number.

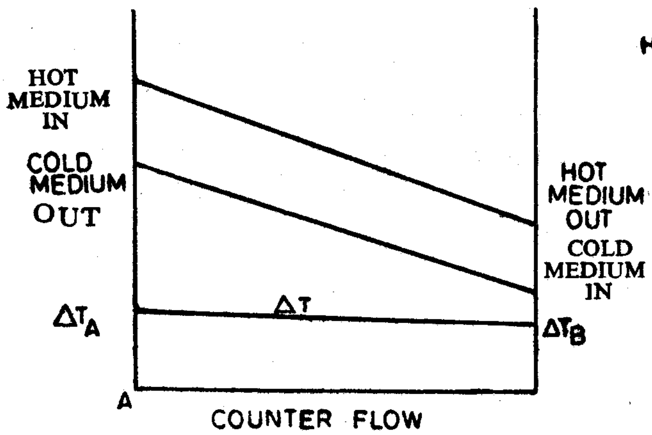

Fig. 6.4.

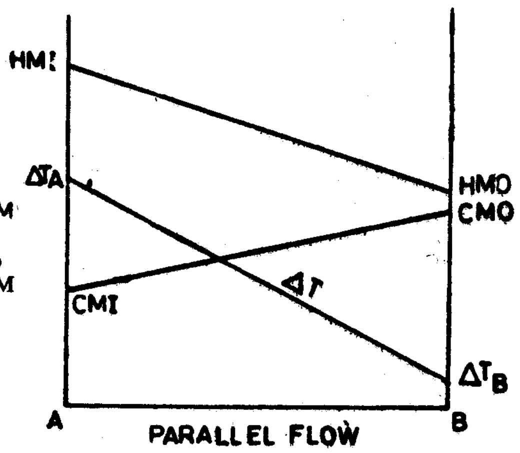

Fig. 6.5.
HMI—Hot Medium In, CMI—Cold Medium In,
HMO—Hot Medium Out, CMO—Cold Medium Out.

Thus $$Nu = f(Pr, Gr)$$

In case of heat transfer by convection and conduction, (Fig. 6.3) overall coefficient of heat transfer

$$U = \frac{1}{\frac{1}{h_{cg}} + \frac{\Delta x}{k} + \frac{1}{h_{cs}}}$$

In case of heat exchangers, heat flow $q = UA$ (LMTD)

where LMTD=Long mean temperature difference

$$= \frac{\Delta T_A - \Delta T_B}{\log \frac{\Delta T_B}{\Delta T_A}}$$ [Refer Fig. 6.4 and 6.5 for ΔT_A and ΔT_B]

Radiation is the process by which energy is continually emitted in the form of electromagnetic waves from the surfaces of all bodies. The rate of heat radiated from a given area is $q = \varepsilon\sigma A T^4$ (T = absolute temperature of the body, A = its area, ε= factor depending on kind of surface and on temperature and σ= constant independent of both surface and temperature). The waves incident upon the surface of a body may be partly absorbed, partly reflected and partly transmitted through the substance. If Q_i is total incident radiation, then $Q_i = Q_n + Q_r + Q_t$ where Q_n, Q_r and Q_t are radiation energies absorbed, reflected, and transmitted respectively.

or
$$\frac{Q_n}{Q_i} + \frac{Q_r}{Q_i} + \frac{Q_t}{Q_i} = 1$$

i.e.
$$\alpha + \rho + \tau = 1$$

Thus the sum of the absorptivity α, reflectivity ρ, and transmissibility τ, of a substance is unity.

A substance is black body, *i.e.* all radiation absorbed, if $\alpha = 1$, $\rho = 0$, $\tau = 0$.

A substance is opaque to radiation (transmits no radiation) if τ is zero.

and
$$\alpha + \rho = 1$$

A substance is transparent to radiation *i.e.* all radiation transmitted if τ is unity and $\rho = 0$, $\alpha = 0$.

A substance is ideal reflector (highly polished surface) or white body if $\alpha = 0$, $\rho = 1$, $\tau = 0$.

A substance is ideal absorber if $\alpha = 1$

A substance is ideal radiator if $\alpha = 1$, and ρ and $\tau = 0$.

Fig 6.7 shows the nature of monochromic emissive power E_λ (rate at which energy is radiated at a wavelength λ, per unit of area surface).

In Fig. 6.7, $T_1 > T_2$, from which it will be noted that at higher temperatures the distribution of energy is largely in the range of shorter waves, and maximum value of E_λ increases with increasing temperature.

Emissivity (ε) is defined as the ratio of the total emissive power of a body to the total emissive power of a black body.

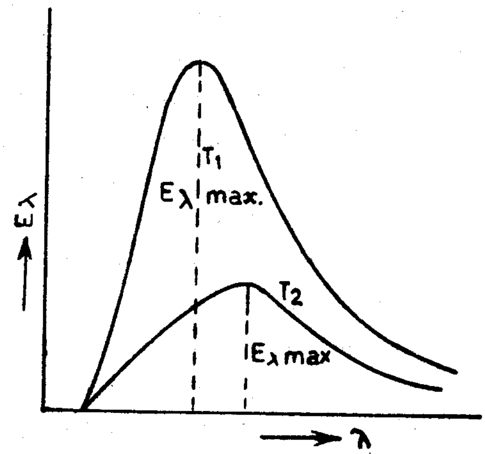

Fig. 6.7. Energy distribution of ideal reflector.

According to Kirchhoff's radiation law, a good radiator is also a good absorber. It states that the total emissive power for any body at a given temperature is equal to its absorptivity multiplied by the total emissive power of a perfect black body at the same temperature.

For maximum emissive power, the value of wavelength is given by Wien's law, according to which $\lambda_{max} \times T$ = constant.

A body is said to be *grey body* if the ratio of emission of a body to that of the black body at a given temperature is constant for all wave lengths.

Planck's Law

All substances emit radiation, the quantity and quality of which depends upon the absolute temperature and the properties of the material, composing the radiating body. The Planck's equation, which gives the distribution of energy versus wavelength for a black body, is

$$W = \frac{C_1 \lambda^{-5}}{e^{C^2/\lambda T} - 1}$$

where λ = wave length in metres

T = absolute temperature in °K

C_1 = constant having value 3.74×10^2

C_2 = constant having value 1.4385×10^7

W = spectral emittance in watts/metre^2 millimicron.

According to Stefan-Boltzmann law, ideal radiators emit radiant energy at a rate proportional to the fourth power of the absolute temperature : and accordingly the net rate of exchange of energy between two ideal radiators A and B is

$$q = \sigma\,(T_A^4 - T_B^4)\ \text{(Area)}$$

where σ = Stefan–Boltzmann constant = 5.672×10^{-12} watt/ cm^2 °K^4.

For non-ideal radiators, the geometry and position of radiating bodies, and the presence of absorbing media between the two bodies must also be considered and suitable factors used for these.

PROBLEMS

Provide single suitable word(s) for following statements :

1. Heat transfer rate per unit area by conduction is proportional to the normal temperature gradient.
2. A physical property of a substance that characterises the ability of the substance to transfer heat.
3. The ratio of (product of thermal conductivity and area) and the wall thickness.
4. Walls of several heterogeneous layers.
5. Mean radius for heat transfer by conduction through hollow cylinder of radii r_1 and r_2.
6. Mean radius for heat transfer by conduction through hollow sphere of radii r_1 and r_2.
7. If Δt_i and Δt_o be the inlet and outlet conditions of temperature difference in a heat exchanger, then log mean temperature difference is ....
8. The ratio of (product of coefficient of heat transfer and length) and thermal conductivity.
9. Heat transfer taking place by means of electromagnetic waves.
10. The sum of absorptivity (α), reflectivity (ρ) and transmissivity (τ).
11. The values of α, ρ, and τ for black body.
12. The values of α, ρ and τ for white body.
13. The values of α, ρ and τ for transparent body.
14. The values of α, ρ and τ for opaque body.
15. Relationship for black bodies between monochromatic emissive power and different wavelengths.
16. The total emission from a black body per unit time varies directly as the fourth power of the absolute temperature.
17. At thermal equilibrium, the absorptivity and emissivity are ....

ANSWERS

1. Fourier's law	2. thermal conductivity	3. thermal conductance	4. composite
5. $\left(\dfrac{r_2 - r_1}{\log_e (r_2/r_1)}\right)$	6. $\sqrt{r_1 r_2}$	7. $\left(\dfrac{\Delta t_o - \Delta t_i}{\log_e \dfrac{\Delta t_o}{\Delta t_i}}\right)$	8. Nusselt Number
9. radiation	10. unity	11. $\rho = \tau = 0, \alpha = 1$	12. $\rho = 1, \tau = \alpha = 0$
13. $\rho = 0, \tau = 1, \alpha = 0$	14. $\tau = 0, \rho + \alpha = 1$	15. Planck's law	16. Stefan Boltzman law
17. same			

7

Refrigeration and Air Conditioning

Refrigeration

A refrigeration system is an apparatus or combination of mechanical equipment in which a refrigerant is circulated for cooling, or extracting heat from spaces or bodies. Refrigeration system could be of two types, *viz.* compression and absorption. Compression system, also sometimes known as mechanical system is the more popular of the two.

A refrigerant having its boiling temperature less than atmospheric temperature will start boiling immediately if subjected to atmospheric conditions, absorbing heat from its surroundings, thereby producing cooling effect. To be economical it is essential that means must be available to capture and reuse the refrigerant vapour.

Fig. 7.1 shows the compression refrigeration system, the cycle of which is as follows :

(*i*) The refrigerant gas is compressed by the compressor and discharged into condenser, (*ii*) In condenser it is cooled and condensed to liquid; atmospheric air or water being used for abstracting heat from condenser, (*iii*) Condensed, and warm high pressure refrigerant then travels through strainer/drier which prevents plugging

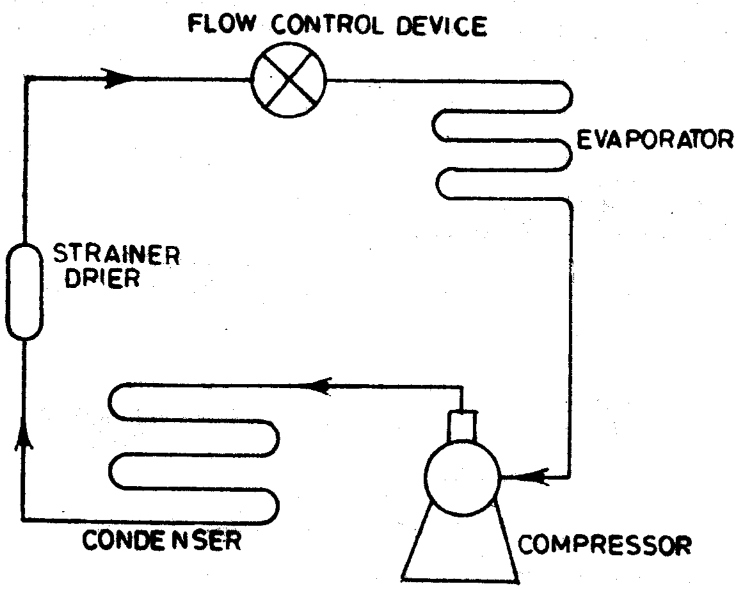

Fig. 7.1. Compression Refrigeration System.

of the flow control device by trapping scale, dirt and moisture, (*iv*) Flow control device controls the flow of refrigerant to .... and refrigerant is throttled, (*v*) in the evaporator, refrigerant is subjected to a much lower pressure due to the combined suction of the compressor and the pressure drop across the flow control device. As a result, refrigerant tends to expand and evaporate thereby cooling the air passing over evaporator. Refrigeration takes place from the absorption of heat during vaporisation in the cooling coils (*vi*) The gas is again drawn into compressor and the cycle is repeated.

Refrigeration systems may be classified as (*i*) flooded type and (*ii*) dry type depending on the condition of the refrigerant in the evaporator. In flooded system, there is a pool of refrigerant in the evaporator. Dry systems have only droplets of liquid in the evaporator, there being almost all refrigerant in vapour condition in evaporator.

Flooded systems are further classified as follows depending on the type of flow control device used (*i*) low side float system, (*ii*) high side float system, (*iii*) capillary tube or restrictor systems.

Dry type systems are of two types (*i*) automatic expansion systems and (*ii*) thermostatic expansion valve systems.

In flooded systems, the refrigerant level in the evaporator is maintained through the action of the refrigerant flow control device. Flooded system has following advantages over dry type.

(*i*) As system operates at higher average suction pressure, the operating cost is low, and efficiency is high.

(*ii*) It provides closer control of temperature.

(*iii*) More liquid on low pressure side of the system provides a greater area of wetted surface, and allows a higher rate of heat transfer through the evaporator walls and tubing.

(*iv*) Number of starts and stops of compressor are low.

(*v*) Operating cost is lower.

In dry system, the refrigerant passing through the valve is partially evaporated immediately after passing the orifice. The fine suspended droplets of liquid refrigerant are completely evaporated as they flow through the balance of the evaporator. Evaporator is generally made from one continuous length of tubing which ensures better control of liquid refrigerant when the flow control device is properly adjusted.

Low side float system is shown in Fig. 7.2, the working of which is self-explanatory. Important point to note is that refrigerant charge used in system is not critical (which is so in other systems) as any excess or over-charge will remain in the liquid receiver.

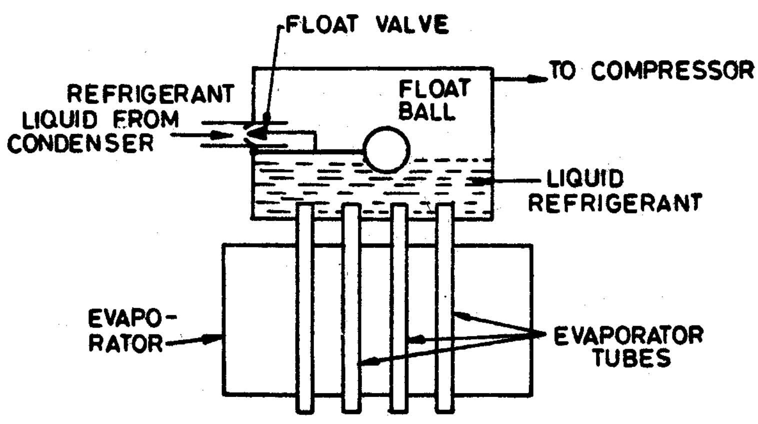

Fig. 7.2. Low Side Float System.

Fig. 7.3 shows high side float system. As the flow of liquid refrigerant in this system is controlled by both the high side float in the liquid receiver and the pressure reducing valve at the evaporator (to maintain

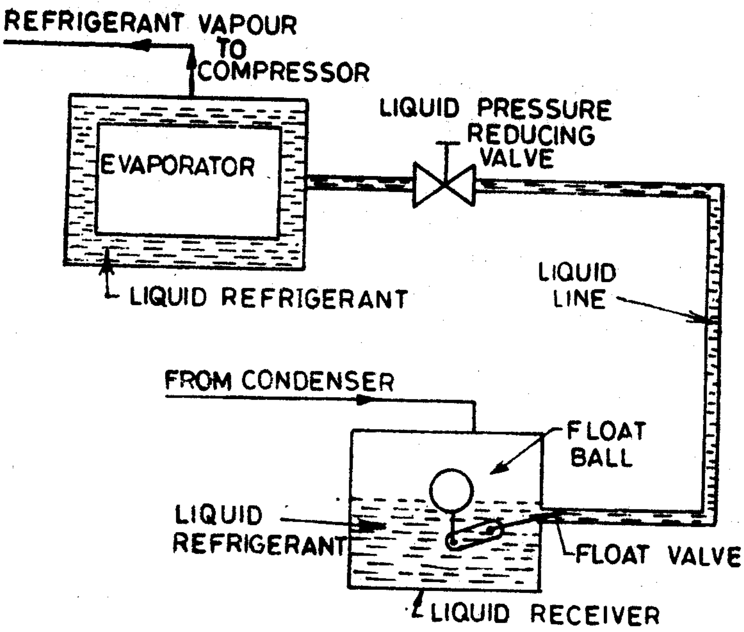

Fig. 7.3. High Side Float System.

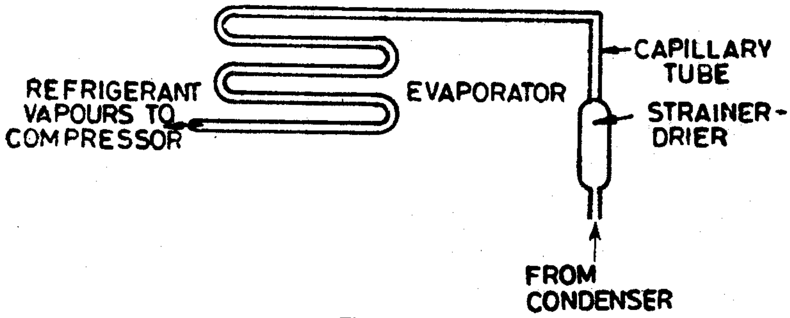

Fig. 7.4.

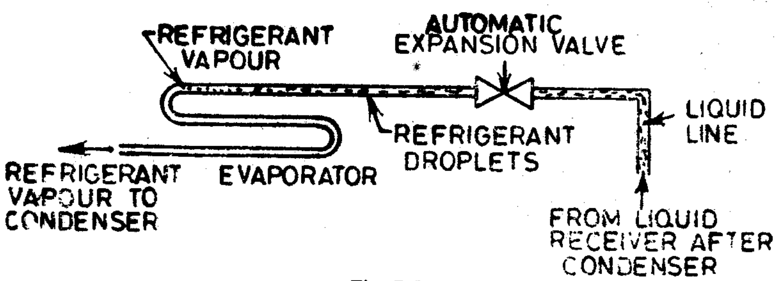

Fig. 7.5.

differential pressure between the refrigerant in the liquid line and the refrigerant in the evaporator, the refrigerant charge in the high side float system is very critical. Excess charge will result in flood back and liquid refrigerant in that case will flow into suction line of compressor causing frost to appear and may damage compressor. Insufficient charge may result in a starved evaporator and loss of refrigeration.

Capillary tube or restrictor system is widely used because of low cost, simplicity, trouble-free operation and achieving of hermetically sealed system. The amount of refrigerant allowed to flow is dependent upon the length and inside diameter of capillary tube provided between the filter and evaporator. This system also requires an accurate charge of refrigerant. Overcharge may result in floodback.

Automatic expansion valve system uses a pressure operated refrigerant flow control device which when properly adjusted, keeps the evaporator fully refrigerated.

In thermostatic expansion valve system, thermostatic control is used to maintain evaporator outlet temperature in combination flow of with evaporator pressure by automatic expansion valve (Refer Fig. 7.5). It provides more accurate control of refrigerant in the evaporator.

Coefficient of performance of a refrigerant is defined as the ratio of the refrigeration obtained to the net work done on the system in the cycle.

$$\text{C.O.P.} = \frac{h_C - h_A}{h_D - h_C}$$

Relative C.O.P. is the ratio of a actual and theoretical C.O.P.

It will be noted that the COP can be improved by subcooling the liquid refrigerant leaving the condenser, *i.e.*, point *A* being moved down on the saturated liquid line.

The capacity of a refrigerator is the rate at which refrigeration is produced and is expressed in tons of refrigeration. One ton of refrigeration is defined as the heat rate corresponding to melting of 1 ton of ice in 24 hours.

Assuming latent heat of fusion of ice as 80 kcal/kg; one ton of 1 ton of ice in 24 hours.

Assuming latent heat of fusion of ice as 80 kcal/kg : one ton of refrigeration = 50 kcal/mt = 210 kJ/min = 3.5 kW.

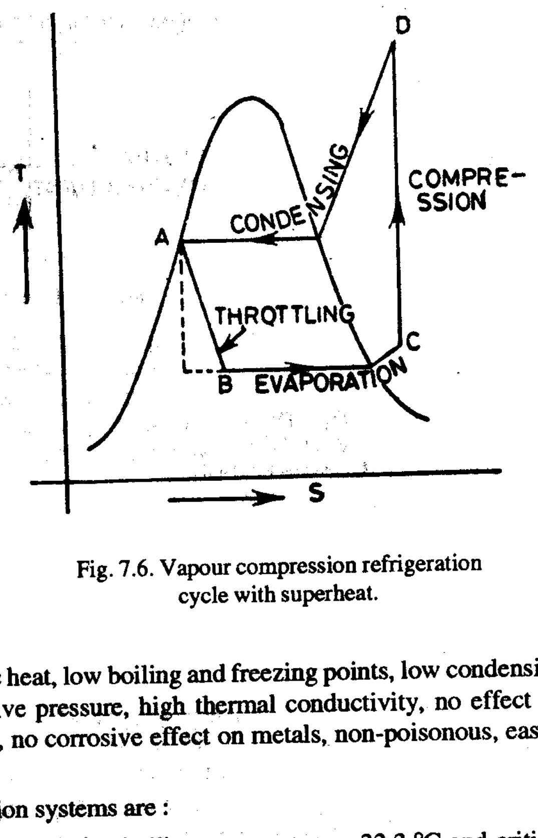

Fig. 7.6. Vapour compression refrigeration cycle with superheat.

A good refrigerant should have following properties :

High latent heat, low specific volume, low specific heat, low boiling and freezing points, low condensing pressure, high critical temperature, positive evaporative pressure, high thermal conductivity, no effect on moisture, safe and non-toxic, capacity to mix with oil, no corrosive effect on metals, non-poisonous, easily available and cheap.

The five refrigerants commonly used in refrigeration systems are :

(*i*) **Ammonia (NH_3).** Its freezing temperature is — 77.8 °C, boiling temperature – 33.3 °C and critical temperature 132.6 °C. It has high latent heat, moderate working pressure. It is highly toxic and has a strong smell. It attacks brass and bronze but is noncorrosive to iron and steel. It finds application in large commercial installations.

(*ii*) **Carbon dioxide (CO_2).** Its boiling point is — 78.5 °C and critical temperature 31°C. It has low specific volume (0.016 m^3/kg. at — 15 °C) and hence plant is very compact. It is non-toxic, noncorrosive and non-inflammable. It is cheap but it has high working pressure (about 70 kgf/cm^2). It finds application where space is precious such as in ships.

(*iii*) **Sulphur dioxide (SO_2).** It has boiling temperature — 25.6 °C and critical temperature 157.2 °C. It has low working pressure (4.5 kgf/cm^2). It is highly toxic, and is corrosive when in contact with moisture. It is used in small and domestic plants.

(*iv*) **Freon-12 (CCl_2F_2).** It is colourless, odourless and nontoxic. It is mostly used for domestic plants. It has operating pressure of about 8 kgf/cm^2. As compared to ammonia it has small specific volume but has a high cost.

(*v*) **Freon-22 ($CHClF_2$).** Its boiling temperature is — 40.8 °C and critical temperature 96.2 °C. It is used in small to medium commercial plants. As compared to Freon-12, it has low specific volume but has high cost.

AIR CONDITIONING

Absolute humidity. It is the weight of water vapour per unit volume.

Relative humidity is actual amount of moisture in air at any given temperature, divided by the greatest amount of moisture the same air could hold without condensation. Relative humidity could also be expressed in terms of pressures as the actual pressure exerted by moisture (vapour pressure), divided by highest vapour pressure (saturation condition) possible under the same conditions.

Fig. 7.7 shows highest pressure that water vapour can have without condensation, *i.e.* 100% relative humidity at various atmospheric temperatures.

Atmospheric pressure = Vapour pressure + pressure of dry air.

Dry bulb temperature is the temperature measured by an ordinary thermometer.

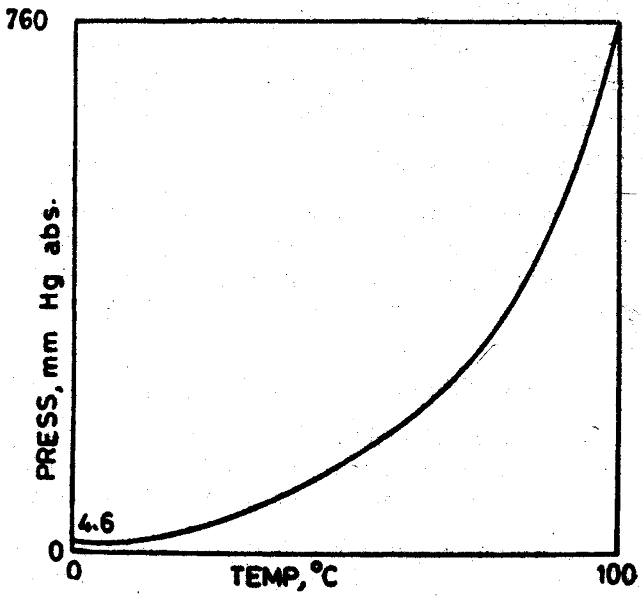

Fig. 7.7. Saturation vapour pressure vs. Atmospheric temperature.

Wet bulb temperature. If the bulb of an ordinary thermometer is surrounded by a wick wetted with water and air passed around it; water at surface will being to evaporate. The vaporisation takes heat-latent heat of vaporisation and the wick is cooled. The temperature finally reached when cooling stops is known as wet bulb temperature. At wet bulb temperature, the heat from the air to water surface equals the heat taken from the surface to vaporise water.

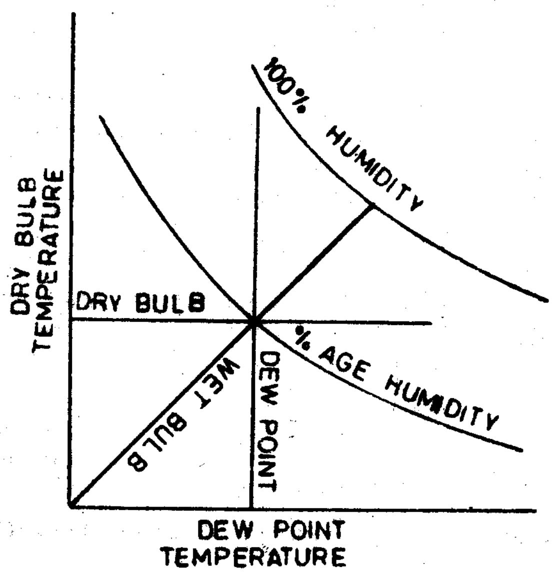

Fig. 7.8. Skelton form of Psychrometric chart.

Dew point temperature. It is the temperature at which moisture will condense out of the air for a given specific humidity and pressure, as air moisture temperature is reduced. It is important to note that dew point temperature remains constant as long as there is no change in moisture content of the air.

At 100% humidity, wet-bulb temperature is equal to dry bulb temperature. Drier is the air, more will be the difference between dry bulb and wet bulb temperatures.

Psychrometric chart. It is a graphic portrayal of various air-moisture mixtures. The vertical axis is taken as dry bulb temperature and horizontal axis as dew point temperature. Sometimes reverse of the above is adopted. With first arrangement, the lines of constant dry-bulb temperature are almost vertical, constant dew point temperature lines are horizontal, constant wet-bulb temperature lines slope downward to the right; constant percentage humidity lines are curved. If any two values of an air-moisture mixture are known then other conditions can be pinpointed.

Psychrometric Processes

1. **Mixing of streams.** The final condition or state of mixture after mixing of two air streams can be found on psychrometric chart by dividing the line joining the conditions of individual streams inversely in the ratio of the mixing masses.

2. **Sensible heating process.** Heating of air (raising dry bulb temperature at the same humidity ratio, *i.e.* without changing its moisture content per kg of dry air), *e.g.* passing air over heating coil, is the process of sensible heating. During sensible heating, the relative humidity decreases.

3. **Sensible cooling process.** It is reverse of sensible heating process.

4. **Humidification process.** It is the process in which moisture is added to air, maintaining its dry bulb temperature constant. During humidification process, relative humidity increases. True humidification process is possible only by keeping the spray water temperature (through which air is passed) at same temperature as the dry bulb temperature of entering air. In normal practice, humidification process is accompanied by cooling or heating.

5. **Dehumidification process.** This is reverse of humidification process.

6. **Cooling and dehumidification.** This process takes place when air passes over a cooling coil having effective surface temperature below the dew point temperature of entering air. As a result the condensation of moisture takes place which results in fall in the specific humidity ratio.

7. **Cooling and humidification.** This process takes place when air is washed through sprays of water maintained at temperatures higher than the dew point temperature of entering air, but lower than the dry bulb temperature of entering air. As a result the air gets cooled and humidified.

8. **Heating and humidification.** This process takes place when air is washed through the spray water at a temperature higher than the dry bulb temperature of the entering air. During this process, the humidity ratio, dry bulb and wet bulb temperature, dew point temperature, and the enthalpy of air increase. The relative humidity may increase or decrease.

9. **Heating and dehumidification.** This process takes place when air is passed through a solid absorbent surface or through a liquid absorbent spray. Dehumidification occurs due to lower water vapour pressure of the absorbent. Water vapour condenses out of air thereby heating the air sensibly due to liberation of latent heat of condensation. This process is reverse of adiabatic saturation and follows the constant wet bulb temperature line.

Some Commonly Used Terms in Psychrometry

Bypass factor. This represents the inefficiency of the heating coil or cooling coil in not being able to heat or cool the incoming air to the temperature of cooling coil.

Bypass factor for a heating coil

$$= \frac{\text{Temp. of heating coil} - \text{Temp. of air leaving}}{\text{Temp. of heating coil} - \text{Temp. of air entering}}$$

Sensible heat factor. It is the ratio of sensible heat of air to the total heat (sensible heat = latent heat).

Grand sensible heat factor. It is the ratio of total sensible heat to the grand total that the cooling coil should handle.

Effective surface temperature. It is the assumed uniform surface temperature of heating/cooling coil which would produce same conditions of air leaving the coil as the actual non-uniform surface temperature of the coil varying throughout the surface as air passes over it.

Humidifying efficiency. In actual practice the air from spray wash will come out at 100% relative humidity, the extent to which humidification is affected depending on the velocity of air and the depth of showers etc. Humidifying efficiency is expressed as

$$\frac{\text{Temp. of entering air} - \text{Dry bulb temp. of air at outlet}}{\text{Temp. of entering air} - \text{Wet bulb temp. of entering air}}$$

PROBLEMS

Provide single suitable word(s) for following statements :

1. The COP of reversed cycle refrigeration cycle is defined as the ratio of ....
2. The C.O.P. of a Reversed Carnot cycle refrigerator with lower temperature T_1 and higher temperature T_2 will be ....
3. If T_2 is kept fixed in a refrigerator working on reversed Carnot cycle, then COP with increase in T_1 will ....
4. In reversed Carnot cycle working on perfect gas, the isentropic work of compression is .... isentropic work of expansion.
5. In reversed Carnot cycle working on vapour, work done during isothermal process is ....
6. If r is the volume compression ratio for isentropic compression, then the COP of Carnot refrigerator is equal to ....
7. If the ratio of high temperature to low temperature for reversed Carnot refrigerator is 5/4, then COP will be ....
8. A reverse Carnot cycle has a COP of 4. The ratio of higher temperature to lower temperature will be ....
9. The COP of Carnot refrigerator used for comfort cooling is .... when compared to that used for making ice.
10. For the same range of temperature operation, the difference in COP of heat pump and refrigerator is ....
11. A refrigerator operating in same range will have .... COP when working substance is changed from R-12 to R-717.
12. The COP of heat pump operating on reversed Carnot cycle is defined as the ratio of ....
13. A heat pump operating between higher temperature T_2 and lower temperature T_1 has COP of ....
14. 1 ton of refrigeration in terms of kJ/min is equal to ....
15. 1 ton of refrigeration in terms of kW is equal to ....
16. 1 ton of refrigeration in terms of kcal/min is equal to ......
17. In case of wet vapour region, the work of isentropic compression is .... work of isentropic expansion.
18. Carnot refrigerator among all refrigerators operating between two fixed temperatures has .... coefficient of performance.
19. Air refrigerator cycle works on .... cycle.
20. Air refrigeration system is preferred in ....

21. The dense air system in comparison to open air system based on Bell-Coleman system and for given range of temperature has .... power/ton of refrigeration.

22. If r_p is the pressure ratio in Bell-Coleman refrigeration cycle, then COP is ....

23. Fluids .... heat while changing from a liquid phase to vapour phase.

24. A container in which the liquid is vaporised and heat is absorbed.

25. The temperature at which the liquid vaporises is tied up with the ....

ANSWERS

1. refrigeration effect and work done		**2.** $\left(\frac{T_1}{T_2 - T_1}\right)$	**3.** increase
4. equal to	**5.** zero	**6.** $\left(\frac{1}{r^{\gamma-1} - 1}\right)$	**7.** 4
8. 1.25	**9.** higher	**10.** unity	**11.** same
12. heat rejected and work done	**13.** $\left(\frac{T_2}{T_2 - T_1}\right)$	**14.** 211	**15.** 3.52
16. 50.4	**17.** more than	**18.** highest	**19.** Bell-Coleman
20. air crafts	**21.** lower	**22.** $\left[\frac{1}{(r_p^{\frac{\gamma-1}{\gamma}} - 1)}\right]$	**23.** absorb
24. evaporator	**25.** pressure		

8

Fluid Mechanics

A **fluid** is a substance that is capable of flowing, conforms to shape of containing vessel, deforms continuously when it is subjected to even a smallest shear stress. An ideal fluid is frictionless and incompressible. For *Newtonian fluids,* there is a linear relation between the magnitude of applied shear stress and the resulting rate of deformation. **Viscosity** is that property of a fluid by virtue of which it offers resistance to shear. Unit of viscosity is poise or 1 gm/cm/sec. $\left(1 \text{ poise} = \frac{0.1\,Ns}{m^2}\right)$. The absolute viscosity of water at 20.2°C is 1 centipoise and of air is 0.17 centipoise. $\left(1 \text{ centipoise} = \frac{1}{100} \text{ poise}\right)$. **Kinematic viscosity** is the ratio of absolute or dynamic viscosity and the density. Its unit is stoke which is equal to 1 cm^2/sec. Viscosity is practically independent of pressure and depends upon temperature only.

Shear stress = $\mu \frac{du}{dx}$ (μ = coefficient of viscosity).

$\frac{du}{dx}$ is known as velocity gradient or rate of shear strain and it is zero for fluids at rest. The law which states that shear stress is proportional to $\frac{du}{dx}$ is called Newton's law of viscosity. Fluids following this are known as **Newtonian fluids** and others Non-Newtonian fluids. An ideal fluid offers no resistance to flow but real fluid does. Thus viscosity is zero for ideal fluid but not for real fluid.

Liquid molecules are attracted to each other by equal forces in all directions. However on the surface, they are subjected to an inward attraction which is not balanced by the outward attraction. This causes the liquid surface to seek a minimum possible area by exerting surface tension tangent to the surface over the entire surface area. **Surface tension** is expressed as force per unit length. Small amounts of salt dissolved in water tend to increase the electrolytic content and hence the surface tension. Organic matter like soap decreases the surface tension in water. Surface tension of water is 74.16 dyn/cm at 0°C, 71.32 at 20°C, 60.71 at 90°C.

The phenomenon of rising water in a tube of smaller diameter dipped in water, is called **capillarity** of water.

The capillary rise of water in a tube, may be calculated from the formula

$$h = \frac{4\,\sigma \cos\alpha}{w \cdot d}$$

where h = capillary rise, d = inner diameter of the tube,
α = angle of contact of water surface,
σ = force of surface tension in kg/mm^2 per unit length of the periphery.

The force per unit area exerted by water on the sides and bottom of its container is called **water pressure.** The intensity of pressure at any point in water, is proportional to its depth from the surface, *i.e.* $p = wh$.

where w is the value of specific weight of water and h is the depth of the point below water surface.

According to **Pascal's law**, the intensity of pressure at any point in a fluid at rest, is the same in all directions.

The **compressibility** of a liquid is expressed by its bulk modulus of elasticity K which is equal to $\frac{-dp}{dv/v}$.

Capillary attraction is caused by *surface tension* and by the relative value of adhesion between liquid and solid to cohesion of the liquid. If a liquid has greater adhesion than cohesion, it would wet the surface.

Gauge pressure is measured with reference to atmospheric pressure and absolute pressure is measured with reference to complete vacuum.

Absolute Pressure = Atmospheric Pressure + Gauge Pressure

Local atmospheric pressure can be measured by a mercury barometer or by an aneroid barometer.

For differential manometer shown in Fig. 8.1.

$$h_A - h_B = h_1S_1 + h_2S_2 + h_3S_3$$

where S_1, S_2, S_3 are the applicable specific gravities of the liquids in the system.

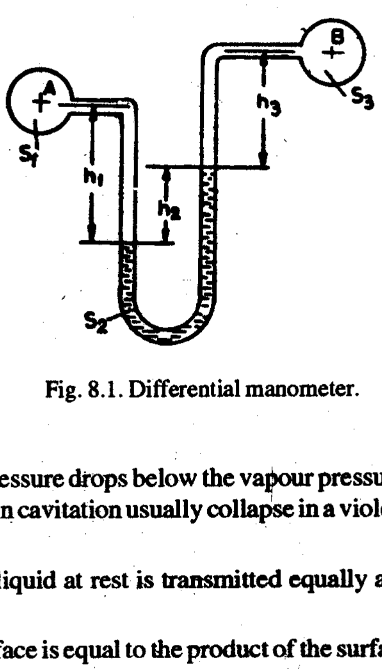

Fig. 8.1. Differential manometer.

The continuous impingement of vapour molecules on the liquid surface creates a pressure on the liquid surface known as the **vapour pressure.**

In a closed system, water vaporises rapidly in regions where the pressure drops below the vapour pressure. This phenomenon is known as **cavitation.** The vapour bubbles formed in cavitation usually collapse in a violent manner, which may cause considerable damage to the system.

According to **Pascal's law,** a pressure applied at any point in a liquid at rest is transmitted equally and undiminished is all directions to every other point in the liquid.

The total **hydrostatic pressure force** on any submerged plane surface is equal to the product of the surface area and the pressure acting at the centroid of the plane surface.

For a horizontal area subjected to static fluid pressure, the resultant force passes through the centroid of the area. The magnitude of force exerted on one side of a plane area submerged in a liquid is the product of the area and the pressure at its centroid. The point where the resultant force acts is known as **centre of pressure.**

The centre of pressure of any submerged plane surface is always below the centroid of the surface.

The location of centre of pressure y_P in case of vertically immersed plane surface is given by

$$y_P = \frac{I_G}{\bar{y} \cdot A} + \bar{y}$$

where I_G = M.I. of area through centroidal axis parallel to liquid surface.

and $\bar{y}$ = Depth of C.G. of the area below liquid surface.

A = Area.

Centre of pressure on an inclined immersed plane surface

$$y_p = \frac{I_G \sin^2 \theta}{A\bar{y}} + \bar{y}$$

and $x_P = x$, if the area is symmetrical about the centroidal axis parallel to the x-axis. If not,

$$x_P = \frac{\bar{I}_{xy}}{\bar{x}} + \bar{x}$$

where $\bar{I}_{xy} = I_{xy} - \bar{x}\,\bar{y}A$

I_{xy} = product of inertia

Horizontal component of pressure force on a curved surface is equal to the pressure force exerted on a projection of the curved surface. The vertical plane of projection is normal to the direction of the vertical component.

The vertical component of pressure force on a curved surface is equal to the weight of liquid vertically above the curved surface and extending up to the free surface. Line of action of the vertical force passes through the centroid of the volume, real or imaginary, that extends above the curved surface upto the real or imaginary free surface.

According to **Archimedes**, the weight of a submerged body is reduced by an amount equal to the weight of the liquid displaced by the body.

A floating body is partially submerged due to the balance of the body weight and the buoyancy force.

Buoyant force is the resultant force exerted on a body by a static fluid in which it is submerged or floating. Buoyant force acts through the centroid of the displaced volume of fluid, known as *centre of buoyancy*. This holds good for both submerged and floating bodies.

A body has linear stability when a small linear displacement in any direction sets up restoring forces tending to return the body to its original position. A body may float in stable, unstable or neutral equilibrium. A submerged body is rotationally stable only when its c.g. is below the centre of buoyancy.

The stability of a floating body is determined by the relative position of the centre of gravity of body and the centre of buoyance (centre of gravity of the liquid volume displaced by the body).

Metacentre '*M*' is the point where the buoyant force and the centre line intersect. When a body is given a small angular displacement, it starts oscillating about metacentre. Body is stable when M is above G, unstable when M is below G, and in neutral equilibrium when M is at G. The distance between the metacentre and c.g. is known as metacentric height.

$$\text{Metacentric height} = \frac{I}{V} - \text{Distance between c.g. and centre of buoyancy}$$

I = M.I. of the plan of floating body at water surface

and V = Volume of submerged body in water.

The line of vertical buoyancy force action meets the axis of symmetry at metacentre. Distance between metacentre and centre of gravity is called *metacentre height* and it is a measure of the floatation stability of the body. A floating body is stable if centre of gravity is below the metacentre, otherwise it is unstable. A submerged body is stable if the centre of gravity is below the centre of buoyancy.

Forced vortex motion is constituted when a fluid rotates about an axis, moving as a solid, and every particle of fluid has same angular velocity. In *free-vortex* motion, each particle moves in a circular path with a speed varying inversely as the distance from the centre. For free vortex $v \times r =$ constant.

In forced vortex motion, the vertical depth varies as the square of the radius from centre, and accordingly the surfaces of equal pressure are paraboloids of revolution. The shape of paraboloid depends only upon the angular velocity.

The volume of water which flows through a section of a channel or pipe per second, is known as **discharge.**

According to **fundamental equation of liquid flow,** or equation of continuity of liquid flow, if an incompressible liquid flows continuously through a pipe or channel, the discharge remains the same irrespective of the areas of cross-section.

The path followed by a fluid particle in motion, is called *path line.*

The imaginary line, tangent to which at any point indicates the direction of motion at that point, is called *stream line.*

The instantaneous positions of all fluid particles which have passed through a given point, is called *streak line*

The line joining the points of equal potential on adjacent flow lines, is called *potential line* or *equipotential line.*

The pattern obtained by the intersection of stream lines and potential lines, is called *flow net.*

The flow in which the velocities of liquid particles at all sections of the pipe or channel are equal, is called **uniform flow.** This generally refers to flows in channels.

The flow in which velocities of liquid particles at all sections of the pipe or channel are not equal, is called a **non-uniform flow.**

The flow in which the quantity of liquid flowing per second is constant is called **steady flow.**

The flow in which the quantity of liquid flowing per second is not constant is called **unsteady flow.**

The flow in which paths of individual particles of liquid do not cross each other, is called *stream line flow* or **laminar flow.** For laminar flow, Reynold's number is less than 2000.

The flow in which paths of individual particles cross each other and particles do not have definite paths, is called **turbulent flow.** For turbulent flow, Reynold number is more than 4000.

The flow whose stream lines may be represented by straight lines, is called *one-dimensional flow.*

The flow of liquid whose stream lines may be represented by a curve, is called *two-dimensional flow.*

The flow of liquid, whose stream lines may be represented in space along three mutually perpendicular axes, is called *three dimensional flow.*

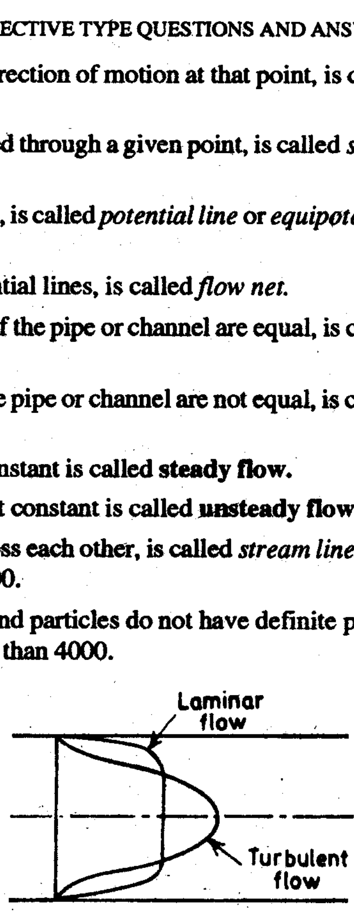

Fig. 8.2. Velocity profile.

According to Reynold, the transition from laminar to turbulent flow in a pipe depends on velocity (V), pipe diameter (D) and kinematic viscosity (γ) of the fluid.

Reynolds number = $\dfrac{DV}{\gamma}$

At critical Reynolds number of 2000 flow starts changing from laminar to turbulent.

Rate of energy loss in *pipe flow* varies as a function of the Reynolds number and the roughness of the pipe.

According to **continuity equation**

$$\text{Flow } Q = AV = A_1V_1 = A_2V_2$$

According to **Bernoulli's equation**

$$\frac{V_1^2}{2g} + \frac{P_1}{\rho g} + h_1 = \frac{V_2^2}{2g} + \frac{P_2}{\rho g} + h_2$$

According to **Darcy-Weisbach formula**

$$h_f = f\left(\frac{L}{D}\right)\frac{V^2}{2g}$$

where h_f = energy loss through friction in pipe line

f = friction factor

L & D = length and diameter

V = Velocity

The sum of the potential head, kinetic head and pressure head of a liquid particle, is called its total head *i.e.*

$$H = Z + \frac{v^2}{2g} + \frac{p}{w} \text{ metres of liquid}$$

According to Bernoulli's theorem, for a perfect incompressible liquid flowing in a continuous flow, the total energy of a particle remains the same; while the particle moves from one point to another. Mathematically,

$$Z + \frac{v^2}{2g} + \frac{p}{w} = \text{constant}$$

Discharge in pipes may be determined either by inserting a venturimeter or an orifice meter.

Discharge through venturimeter $Q = C_d a_2 v_2$

$$= C_d \cdot \sqrt{2gh} \cdot \frac{a_1 a_2}{\sqrt{a_1^2 - a_2^2}}$$

a_1 and a_2 are areas before and at throat

C_d is the coefficient of discharge of the venturimeter.

Pitot tube is used to measure velocity of fluid in a pipe or channel. $V = C_v \sqrt{2gh}$.

Orifice is a smaller opening on the side or at bottom of a tank. Mouthpiece is a short length of a pipe is 2—3 times its diameter in length, fitted in a tank of vessel containing the fluid.

The various hydraulic coefficients are

Coefficient of contraction. The ratio of the area of the jet at vena-contracta to the area of the orifice opening, is known as *coefficient of contraction*. Its average value is taken as 0.64.

Coefficient of velocity. The ratio of the velocity of the jet at venacontracta to the theoretical velocity, is known as coefficient of velocity. Its average value is taken 0.97. In terms of coordinates (x, y) for vena-contracta, $C_v = \frac{x}{\sqrt{4yH}}$, $H =$ height of fluid in tank.

Coefficient of discharge. The ratio of actual discharge through an orifice to the theoretical discharge, is known as coefficient of discharge. Its average value is taken as 0.62.

Coefficient of resistance. The ratio of loss of head in the orifice to the head of water available at the orifice exit, is known as coefficient of resistance. It is treated equal to zero for all practical purposes.

In laminar flow, fluid particles move along smooth paths in laminar, or layers, with one layer gliding smoothly over an adjacent layer. The losses in laminar flow vary proportional to velocity.

In turbulent flow (most prevalent in engineering practice) the fluid particles move in very irregular paths. It sets up greater shear stresses throughout the fluid and causes more irreversibilities or losses which are proportional to square of velocity.

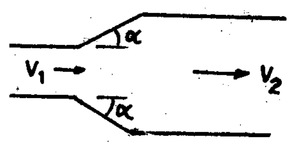

Fig. 8.3.

According to continuity equation

$\rho_1 v_1 A_1 = \rho_2 v_2 A_2$ $\qquad \rho =$ density, $v =$ velocity, $A =$ Area

According to Bernoulli's theorem

$$\frac{v_1^2}{2g} + \frac{p_1}{w_1} + Z_1 = \frac{v_2^2}{2g} + \frac{p_2}{w_2} + Z_2 + \text{Losses}$$

($p =$ pressure, $w =$ sp. wt., $Z =$ elevation)

Head loss due to pipe enlargement

$$= K_e \frac{V_1^2 - V_2^2}{2g}$$

where value of $K_e = 0.39$ for $\alpha = 10°$, 1.06 for $\alpha = 40°$ and 1.0 for 30°

In open channel flow, hydraulic radius $= \dfrac{\text{Water cross–sectional area}}{\text{wetted perimeter}}$

Best hydraulic section is one with the least wetted perimeter.

The best hydraulic trapezoidal section is a **half-hexagon.**

Hydraulic jump in an open channel is an abrupt reduction in flow velocity by means of a sudden increase of water depth in the downstream direction.

Hydraulic jump occurs when a rapidly flowing stream of liquid in an open channel suddenly changes to a slowly flowing stream with a larger cross-sectional area and a sudden rise in elevation of liquid surface occurs. This is the example of steady non-uniform flow. It is very effective device for creating irreversibilities and is commonly used at the end of chutes or the bottom of spillways to destroy much of kinetic energy in flow.

Any obstruction of a stream flow over which water flows can be called a **weir.**

If the downstream water level rises over the weir crest, the weir is said to be submerged.

A spillway is an essential part of a large dam and provides an efficient, safe means of releasing flow water that exceeds the design capacity of the reservoir.

Culverts are built at the point of lowest valley to pass water across the embarkments of highways or railroads.

Standpipe is used principally for alleviating the transient pressure in large pipe line systems. It acts as a pressure relief valve for the upstream pipe during the turbine shut off. Water hammer effects are also diminished noticeably.

Dimensional analysis. The analysis of the basic relationship of the various physical quantities, involved in the static and dynamic behaviours of water flow in a hydraulic structure is known as *dimensional analysis.*

Similarity between hydraulic models and prototype may be achieved in three basic forms :

(*i*) Geometric similarity implying similarity of form.

(*ii*) Kinematic similarity implying similarity in motion.

(*iii*) Dynamic similarity implying similarity in forces involved in motion.

Dimensionless parameters permit limited experimental results carried in laboratories to be applied to actual big size objects in fluids of different properties. The five most important parameters used in correlating experimental data are :

Pressure coefficient

$$= \frac{\Delta p}{\frac{\rho v^2}{2}} = \frac{\text{Pressure}}{\text{Dynamic pressure}} = \frac{\Delta p \times A}{\frac{\rho v^2 A}{2}} = \frac{\text{Pressure force}}{\text{Inertial force}}$$

$$= \frac{\frac{\Delta p}{\rho g}}{\frac{v^2}{2g}} = \frac{\frac{\Delta p}{\text{sp. wt.}}}{\frac{v^2}{2g}} = \frac{\Delta h}{\frac{v^2}{2g}}$$

For pipe flow the Darcy-Weisbach equation relates losses h_1 to length of pipe L, diameter D, and velocity v by a dimensionless friction factor f

$$h_1 = f\frac{L}{D}\frac{v^2}{2g} \quad \text{or} \quad \frac{h_1}{v^2/2g} = \frac{fL}{D}$$

Reynolds number

$$= \frac{\rho v D}{\mu} = \frac{\text{Inertial force}}{\text{Viscous force}}$$

A critical Reynolds number is the demarcation between laminar and turbulent flow in pipe.

Froude number

$$= \frac{v^2}{gl} = \frac{v^2 \rho A}{gl\rho A} = \frac{\text{Dynamic force}}{\text{Weight}}$$

Forude number decides whether the free liquid-surface flow is rapid or tranquil depending on whether Froude number is greater or less than unity. It is useful in calculations of hydraulic jump, in design of hydraulic structures, and in ship design.

Weber number

$$= \frac{v^2 l\rho}{\sigma} = \frac{\text{Inertial force}}{\text{Surface tension force}}$$

It is important at gas-liquid or liquid-liquid interfaces and also where these interfaces are in contact with a boundary.

Mach number $= v/\sqrt{K/\rho}$

It is a measure of the ratio of inertial forces to elastic forces and is very important when velocities are near or above local sonic velocities. (K = bulk modulus of elasticity).

Model studies are big aid to the designer. These permit visual observation of flow and make possible the obtaining of certain numerical data and picture of behaviour of actual big size models by conducting tests on models in laboratories. For obtaining accurate quantitative data from model study, there must be dynamic simlitude between model and prototype. For this purpose there must be exact (*i*) geometric simlitude and (*ii*) kinematic simlitude *i.e.* the ratio of dynamic pressure at corresponding points must be a constant. Geometric simlitude refers to the actual surface roughness of the model and prototype. For dynamic pressures to be in the same ratio at corresponding points in model and prototype, the ratios of various types of forces must be the same at corresponding points.

For laminar flow through circular tubes and circular annuli.

$$\Delta p = \frac{128 \mu L Q}{\pi D^4}$$

μ = coefficient of viscosity, L = length, Q = flow rate, D = pipe diameter.

Also head loss $h_1 = f \dfrac{4L}{D} \dfrac{v^2}{2g}$

Time for discharge from an orifice of area 'a' in a cylindrical vessel of cross-sectional area A to fall from height H_1 to H_2 is

$$t = \frac{2A}{C_d \cdot a\sqrt{2g}} (\sqrt{H_1} - \sqrt{H_2})$$

Similar time is case of two vessels of area A_1 and A_2 interconnected together

$$= \frac{2A_1 A_2 (\sqrt{H_1} - \sqrt{H_2})}{(A_1 + A_2)\, C_d \cdot a \cdot \sqrt{2g}}$$

Notch is a device used for measuring the rate of flow of a liquid through a small channel. A weir is a concrete or masonary structure placed in the open channel over which the flow occurs.

Flow over *rectangular/weir* or notch of length L and depth H is

$$Q = \frac{2}{3} C_d \sqrt{2g}\, L \cdot H^{3/2}.$$

Flow over *triangular weir* or notch of semi-angle θ and water depth H is

$$Q = \frac{8}{15} C_d \sqrt{2g} \tan \theta\, H^{5/2}$$

The error in discharge due to error in measurement of H for rectangular and triangular notch/weir is $\frac{dQ}{Q} = \frac{3}{2}\frac{dH}{H}$ and $\frac{5}{2}\frac{dH}{H}$ respectively.

Flow over broad crested weir of length L and water depth over weir before the weir H is

$$Q = 1.7\, C_d L \cdot H^{3/2}$$

Hydraulic gradient or hydraulic slope

$$= \frac{\text{Head lost due to friction}}{\text{Total length of pipe}}$$

Hydraulic mean depth or *hydraulic radius*

$$= \frac{\text{Area of cross–section } (A)}{\text{Wetted perimeter } (P) \text{ (surface in contact with water)}}$$

According to *Darcy equation*

$$h_f = \frac{fPl}{A} \cdot \frac{v^2}{2g}; \text{ and for pipe this is}$$

$$h_f = \frac{4fl}{D} \cdot \frac{v^2}{2g}$$

According to *Chezy's formula*

$$v = C\sqrt{mi}, \qquad C = \text{Chezy's constant.}$$

The *losses of head in pipes* are :

(*i*) Loss at the entrance = $\frac{0.5\, v^2}{2g}$, (*ii*) Loss at the outlet = $\frac{v^2}{2g}$

(*iii*) Loss due to friction = $\frac{4flv^2}{2g \cdot d}$.

(*iv*) Total loss of head in a pipe flow

$$= \frac{0.5v^2}{2g} + \frac{v^2}{2g} + \frac{4flv^2}{2g \cdot d}$$

where v is the velocity of flow, l is the length of pipe, d is diameter of pipe and f is fundamental resistance per unit area.

For maximum power transmitted by a nozzle, the head loss due to friction should be $\frac{1}{3}$ rd of supply head.

For maximum power transmission by a nozzle, the diameter d may be obtained from the formula,

$$d = \left(\frac{D^5}{8fL}\right)^{1/4}$$

where D = Diameter of main pipe, L = length of the pipe

f = Darcy's coefficient of friction

Again, for maximum power transmission by a nozzle, the ratio of the area of the pipe to the area of the nozzle is related by

$$\frac{A}{a} = \sqrt{\frac{8fL}{D}}$$

The pipes of different diameters connected with one another to form a pipe line, is called a *compound pipe* or *pipes in series.*

Total head loss $= \dfrac{fQ^2}{3}\left(\dfrac{l_1}{d_1^5}+\dfrac{l_2}{d_2^5}+\dfrac{l_3}{d_3^5}+\ldots+\dfrac{l_n}{d_n^5}\right)$

where, $l_1, l_2, l_3, \ldots l_n$ are the lengths of individual portions and $d_1, d_2, d_3, \ldots, d_n$ are the diameters of the respective portions.

The pipe of uniform diameter which may replace a compound pipe, keeping the loss of head and discharge same, in both cases, is called *equivalent pipe,* and its diameter is called equivalent size of the pipe.

Let, $d_1, d_2, d_3, \ldots, d_n$ be the diameters

$l_1, l_2, l_3, \ldots, l_n$ be the length of various pipes

L and D, the length and diameter of the equivalent pipe

then
$$\frac{L}{D^5}=\frac{l_1}{d_1^5}+\frac{l_2}{d_2^5}+\frac{l_3}{d_3^5}+\ldots+\frac{l_n}{d_n^6}$$

Water Hammer. Water hammer is caused due to sudden stoppage of water flow in a pipe. Sudden stoppage results in a pressure wave which moves upstream with velocity of sound in the medium which is reflected back and forth until dissipated by friction and imperfect elasticity.

Rise of pressure (p) due to sudden stopping of flow in a pipe

$$p=\frac{v}{\sqrt{\dfrac{g}{w}\left(\dfrac{1}{k}+\dfrac{D}{tE}\right)}}$$

v = velocity of flow in pipe of diameter D, thickness t and coefficient of elasticity E

K = coefficient of bulk modulus of fluid.

and w = specific weight of fluid.

Surge tanks are used to relieve the pipe line of excessive pressure.

Flow in Pipes

The following conditions must be satisfied in a network of pipes :

(*i*) The algebraic sum of the pressure drops around each circuit must be zero.

(*ii*) Flow into each junction must be equal flow out of the junction.

(*iii*) The Darcy equation must be satisfied for each pipe *i.e.* proper relation between head loss and discharge must be maintained for each pipe.

According to momentum equation, the net force acting on a fluid mass is equal to the change in momentum per second in that direction.

PROBLEMS

Provide single suitable word(s) for following statements :

1. The branch of applied mechanics dealing with the behaviour of fluids at rest and in motion.
2. When in equilibrium, fluids can't sustain .... forces.
3. Ratio of the mass of a body to the mass of an equal volume of a substance taken as a standard.
4. The property which determines the amount of its resistance to a shearing force.
5. Kinematic coefficient of viscosity is the ratio of absolute viscosity and ....
6. Viscosities of liquids .... affected by pressure changes.
7. Kinematic viscosity of gases varies .... as the pressure.

8. Surface molecules have .... energy than interior molecules in a liquid.
9. The work to be done to bring enough molecules from inside the liquid to the surface to form one new unit area of that surface.
10. The liquids rise in tubes when adhesion of liquid to walls is .... cohesion of the liquid.
11. The ratio of the change in unit pressure to the corresponding volume change per unit of volume.
12. A fluid in which the shear stress is proportional to the velocity gradient or shearing strain.
13. A fluid for which the resistance to shearing deformation is zero.
14. Longitudinal stress in thin-walled cylinders closed at the ends is equal to .... the hoop tension.
15. The line of action of force exerted by a liquid on a plane area passes through the ....
16. The position of the centre of pressure is always .... the centre of gravity of the area.
17. The force acting on any weight floating or immersed in a liquid, then force being equal to the weight of th liquid displaced.
18. The point through which the buoyant force acts is called the centre of buoyancy and it is located at the .... of the displaced liquid.
19. For stability of a submerged body, the centre of gravity of body must lie directly .... the centre of buoyance of the displaced liquid.
20. If the ratio of all corresponding dimensions of model and prototype are equal.
21. If the paths of homologous moving particles are geometrically similar and if the ratios of the velocities of homologous particles are equal.
22. If the ratios of all homologous forces in geometrically and kinematically similar models and prototypes are the same.
23. Inertial pressure force ratio.
24. Inertia-viscous force ratio.
25. Inertia-gravity force ratio.
26. Inertia-elasticity force ratio.
27. Square root of Cauchy number.
28. Inertia-surface tension ratio.
29. An incompressible fluid flow in which the direction and magnitude of the velocity at all points are identical.
30. A fluid flow in which the fluid particles move in planes or parallel planes and the streamline patterns are identical in each plane.
31. An ideal flow which can be represented by a flow net (*i.e.* no shear stresses and no torques exist).
32. A flow, in which, at any point, the velocity of successive fluid particles is the same at successive periods of time.
33. A flow in which the magnitude and direction of the velocity do not change from point to point in the fluid.
34. Imaginary curves drawn through a fluid to indicate the direction of motion in various sections of the flow of the fluid system.
35. Equation of .... results from the principle of conservation of mass.

36. Energy at any section, plus energy added, minus the energy lost and extracted, is equal to energy at the end.
37. The hydraulic grade line lies below the energy line by an amount equal to the .... at that section.
38. A flow in which the fluid particles move along straight, parallel paths in layers.
39. The .... velocity of practical interest is the velocity below which all turbulence is damped out by the viscosity of the fluid.
40. The upper limit of laminar flow of practical interest is represented by a Reynold number of about ....
41. The ratio of the cross sectional area to the wetted perimeter for non-circular cross sections.
42. A flow in which the particles of the fluid move in a haphazard fashion in all directions.
43. Velocity distribution at a cross section of variation for laminar flow.
44. In laminar flow the maximum velocity at the centre of pipe is .... the average velocity.
45. Used to measure the velocity head of flowing fluid.
46. The ratio of area of jet (pitot tube) at vena contracta and the area of orifice.
47. The product of coefficient of velocity and coefficient of contraction.
48. The ratio of loss of kinetic energy in the orifice and the actual kinetic energy.
49. If V_1 and V_2 be velocity at inlet and outlet, then loss of head due to sudden enlargement is proportional to ....
50. Loss of head due to sudden contraction is proportional to ....
51. Coefficient of discharge is unity in case of .... mouth piece.
52. Coefficient of contraction for an internal mouthpiece is ....
53. A sharp edged obstruction over which flow of a fluid takes place.
54. The sheet of water which flows over the notch or weir.
55. Flow over rectangular notch is proportional to ....
56. Flow over triangular notch is proportional to ....
57. A trapezoidal notch having side slops of one horizontal to four vertical.
58. The ratio of head lost due to friction and total length of pipe.
59. A pipe of uniform diameter which replaces the compound pipe consisting of several pipes of different diameters and lengths.
60. Transmission of power through pipe is maximum when loss of head due to friction in pipe is .... of the total head supplied at the entrance to the pipe.
61. According to Froude, the frictional resistance in pipe is proportional to ....
62. Flow in a pipe will be turbulent when Reynold's number is greater than ....
63. A flow measuring device in which indication is essentially linear with flow rate.
64. A flow in which the velocity, *i.e.* depth of flow varies from one section to another.
65. In case of rectangular open channel, the width of channel should be .... the depth for maximum discharge.
66. For maximum efficiency, the trapezoidal section of open channel should form a ....

67. For maximum discharge in circular shape open channel, depth should be equal to .... × diameter of pipe.

68. The depth of flow for the given discharge in a channel, corresponding to minimum specific energy.

ANSWERS

1. fluid mechanics	**2.** shear	**3.** relative density	**4.** viscosity
5. mass density	**6.** are not	**7.** inversely	**8.** more
9. surface tension	**10.** greater than	**11.** bulk modulus of elasticity	
12. Newtonian fluid	**13.** ideal fluid	**14.** half	**15.** centre of pressure
16. below	**17.** buoyant force	**18.** centre of gravity	**19.** below
20. geometric similitude	**21.** Kinematic similitude	**22.** dynamic similitude	**23.** Euler number
24. Reynolds number	**25.** Froude number	**26.** Cauchy number	**27.** Mach number
28. Weber number	**29.** true one dimensional flow		**30.** two-dimensional flow
31. irrotational flow	**32.** steady flow	**33.** uniform flow	**34.** streamlines
35. continuity	**36.** Bernoulli's theorem	**37.** velocity head	**38.** laminar flow
39. critical	**40.** 2000	**41.** hydraulic radius	**42.** turbulent flow
43. parabolic law	**44.** twice	**45.** pitot tube	**46.** coefficient of contraction
47. coefficient of discharge	**48.** coefficient of resistance	**49.** $[(V_1 - V_2)^2]$	**50.** $(V_2)^2$
51. convergent divergent	**52.** 0.5	**53.** notch	**54.** nappe or vein
55. $(H^{3/2})$	**56.** $(H^{5/2})$	**57.** cippoletti notch	**58.** hydraulic gradient
59. equivalent pipe	**60.** one-third	**61.** (V^2)	**62.** 4000
63. rotameter	**64.** non-uniform flow	**65.** twice	**66.** half hexagon
67. 0.95	**68.** critical depth		