

ELEMENTS OF HEAT ENGINES

VOLUME II

(IN SI UNITS)

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Preface to the Sixteenth Edition

It is a matter of great satisfaction to the author of this book that he is required to bring out the Sixteenth Edition of this volume in such a short span of time. This by itself speaks in favour of this book in regard to its usefulness and popularity among engineering students.

The author will very thankfully appreciate suggestions from readers for the improvement of the book.

The author thanks Shri J. C. Shah of Acharya Publications, Vadodara and Shri Surendra J. Shah of Parijat Printery, Ahmadabad, for getting the book printed so nicely and in time.

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Preface to the Second Edition

The second edition has been rewritten in M.K.S. Units to comply with the decision of the All India Council of Technical Education for change over in the System of Units. As most of the Universities and State Technical Education Boards have switched over to M.K.S. Units, the authors feel that this edition would be quite useful both to the teachers teaching the subject and to the students.

The major part of the book is overhauled to enhance the utility of the book, by making it more lucid in expression, more simplified in the subject matter and by suitable modification in the general arrangement and addition of subject matter at some places without unduly increasing the size of the book.

The authors take this opportunity of thanking all those who have been good enough to draw our attention to some of the slips that had inadvertently crept in the first edition and those who offered valuable suggestions and comments with a view to make the book really one of the best of its kind. The authors will gratefully appreciate all constructive comments and suggestions from the readers.

Before concluding, the authors feel it their duty to thank Sarvashri M.C. Karamchandani, B.E. (Mech.), Hon., M.M. Patel, B.E.(Mech.), K. H. Patel, B.E. (Mech.), M.C. Shah, B.E. (Mech.), B.C. Patel, D.M.E., B.C. Karamchandani, D.M.E., for checking the calculations and reading through the proofs. Our thanks are also due to Shri Karkhanis for redrawing the sketches. Our sincere thanks are also due to the publishers for getting the book published in time.

Baroda
1st July, 1966

R. C. Patel
C. J. Karamchandani

Preface to the First Edition

This volume is intended for the use of students preparing for the final diploma examinations in Mechanical and Electrical Engineering of the Universities and State Technical Boards.

Despite the fact that several books on this subject of Heat Engines have been written and published, the long felt need of many Indian Diploma students for a Book covering completely the Heat Engines Syllabus written in a simple style has led the authors to bring out this volume in the form most suitable for Indian students preparing for the above examinations.

This book which is an outcome of a very long experience of the authors in the teaching of the subject, has a special feature. Neat and simple diagrams to be found herein have gone a great way in simplifying the subject matter and have made its presentation instructive and interesting.

Another useful feature of this book is a large number of examples at the end of each chapter, which are fully worked out to inspire faith and confidence in the students, who otherwise cram theory without understanding and fail to apply theory rightly and correctly in solving examples. These worked out examples may help the students not only to understand clearly the basic principles underlying them, but may also lead them to attempt without frustration the solution of examples of varied types appearing in examination papers. The practice problems added at the end of each chapter are for the benefit of students and the teachers who may use the book.

The authors found it more convenient to cover the entire Heat Engines Syllabus in three volumes (Volume I, II and III). This has prevented the volume from being too bulky. *Volume I is intended for the use of students preparing for the second year diploma examinations in Mechanical and Electrical Engineering, Volume II for the final year diploma Examination in Electrical Engineering, and Volumes II and III for the final year diploma Examination in Mechanical Engineering.*

Utmost care has been taken in numerical calculations. They are made with the aid of slide rule, and no pains have been spared to avoid errors. And yet it is too much to be sure that all slips and errors have been detected and rectified. Authors will, therefore, very thankfully appreciate comments and suggestions from readers for the improvement of the book.

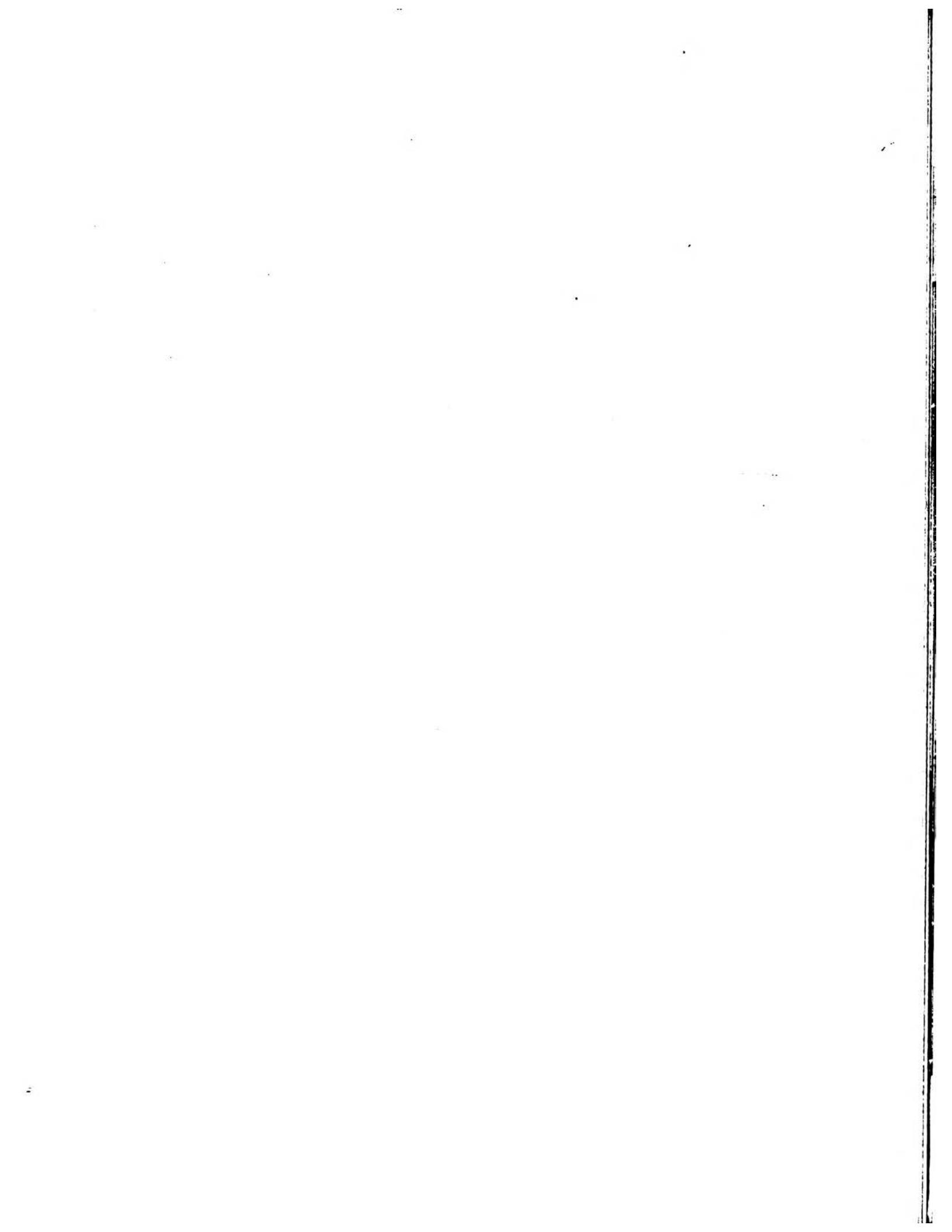
The authors will feel delighted and more than compensated if the book satisfies the end in view and meets with the need of students.

Before concluding, the authors feel it their duty to thank Shri M.C. Karamchandani, B.E. (Mech.) of the Polytechnic, Baroda for carefully going through the manuscript and checking it as also for solving examples.

The authors also take this opportunity of expressing their thankfulness to Shri Jayantilal C. Shah of M/s. Acharya Book Depot, Baroda for getting the volume published so nicely and in time.

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R. C. Patel
C. J. Karamchandani



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STEAM CONDENSERS

1.1 Introduction

Steam condensers are devices in which the exhaust steam from the steam engines or steam turbines is condensed by means of cooling water. The condensate (condensed steam) thus formed together with the air and other non-condensable gases is continuously removed by pumps or similar other devices.

The *primary object* of a condenser is to maintain a very low back pressure on the exhaust side of the piston of the steam engine or the rotor of the steam turbine. This enables the steam to expand to a greater extent which results in an increase in available energy for converting into mechanical work. The shaded area in fig. 1-1 shows the increase in work obtained by fitting a condenser to a non-condensing steam engine. The *secondary object* of a condenser is to supply to the boiler pure and hot feed water, as the condensed steam which is discharged from the condenser and

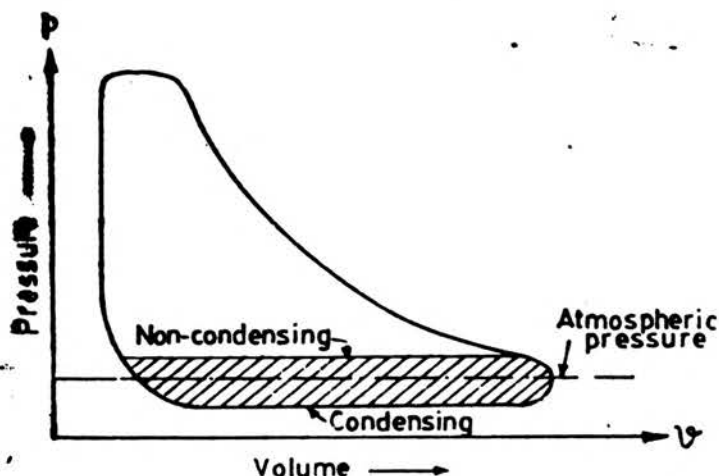


Fig. 1-1. $p-v$ diagram for non-condensing and condensing steam engines. The condensed steam which is collected in a hot-well, can be used over again as feed water for the boiler. Thus by fitting a condenser, the thermal efficiency of the steam power plant can be greatly increased and also the capacity, without increasing size.

The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low pressure steam or water vapour. If a kilogram of dry steam at 1.01325 bar and with a volume of 1.6729 m^3 , contained in a steam tight vessel, is condensed into water at a temperature of 100°C , the liquid would occupy only 0.001044 m^3 . The volume of steam would be thus $\frac{1}{1,644}$ part of the space inside the vessel, and the pressure would fall to 0.2 bar. This means the back pressure in the steam engine or steam turbine can be lowered from 1.01325 to 0.2 bar or even less.

Steam engines, however, cannot take the advantage of a very low vacuum in the condenser because they are intermittent (non-continuous) flow machines and have to force the expanded steam out of the cylinder through restricted exhaust ports and passages. The lowest exhaust pressure, therefore, for most steam engines is 150 to 200 mm of mercury, i.e. about 0.2 to 0.27 bar.

Steam turbines can, however, take advantage of very low vacuum because they are continuous-flow machines and have large exhaust outlets through which the steam may be discharged after expansion. Steam turbines may expand steam to a pressure of 25 mm of mercury absolute or even less, i.e. about 0.034 bar.

1.2 Elements of a Steam Condensing Plant

The main elements of a steam condensing plant are :

- a *condenser* in which the exhaust steam is condensed,
- *supply of cooling or injection water*, for condensing exhaust steam,
- a *pump* to circulate the cooling water in case of a surface condenser,
- a pump, called the *wet air pump*, to remove the condensed steam (condensate), the air, and uncondensed water vapour and gases from the condenser (separate pumps may be used to remove air and condensed steam),
- a *hot-well*, where the condensed steam can be discharged and from which the boiler feed water is taken, and
- an arrangement (*cooling pond or cooling tower*) for cooling the circulation water when a surface condenser is used and the supply of water is limited.

1.3 Types

Condensers may be classified broadly into two major groups according to the manner in which the cooling water cools and condenses the exhaust steam; these are :

- ... **Jet condensers**, in which cooling water comes in direct contact with the exhaust steam and the steam as a result is condensed. The condensing or cooling water is usually sprayed into the exhaust steam so that rapid condensation of the steam occurs.
- ... **Surface condensers**, in which the cooling water and exhaust steam do not actually mix; the cooling water passes through a number of tubes while the exhaust steam passes over the outer surfaces of the tubes.

The most common type is a surface condenser which has the great advantage that the condensate (condensed steam) is not thrown to waste but is returned to the boiler through feed water system. A jet condenser is a much simpler and less costly apparatus than a surface condenser. The jet condenser should be installed where a cheap source of boiler feed water is available.

1.4 Jet Condensers

There are three main types of jet condensers :

- ... **Low-level condenser**, in which the condensing chamber is at low level and overall height of the unit is low enough so that condenser may be directly placed beneath the steam engine or steam turbine. Combined pump or separate pumps are required to extract (remove) the cooling water, condensate, and air from the condenser.
- ... **High-level or Barometric condenser**, in which the condensing chamber is placed at sufficiently high level (above 12 metres) above the point of discharge to enable the water to drain away by gravity through the tail pipe. No water pump is required to remove the condensate and cooling water, but an air pump is required to remove air from the condenser.
- ... **Ejector condenser**, in which exhaust steam and water mix in a series of combining cones, and the kinetic energy of water is utilised to assist in removing the condensate and air from the condenser. The mixture is then discharged into the hot-well against the pressure of atmosphere. No separate pump is required to remove condensate and air.

Low-level Jet Condensers may be sub-divided according to the direction of flow of water and exhaust steam as under :

- *Counter flow jet condenser*, in which exhaust steam and water flow in opposite directions; exhaust steam enters at the bottom and flows upwards, while the water enters at the top and flows downwards, the air pump is at the top.
- *Parallel flow jet condenser*, in which exhaust steam and water flow in the same direction; the steam and water enter at the top and fall together to the bottom where the mixture is removed by an extraction pump. The parallel flow arrangement is generally best suited for turbine work where the exhaust steam comes from the underside of the turbine.

The counter flow jet condenser is normally arranged with two or three water trays perforated with holes to break the water in small jets. Sufficient water head is arranged over the trays to ensure that they are always full of water.

In the condenser (fig. 1-2), the exhaust steam and any air mixed with it, enter the lower part of the condenser and rise to the top through the falling water which enters at the side near the top of the condenser. Rapid condensation of exhaust steam takes place. The condensate and cooling water descend through a vertical pipe to the extraction pump (generally of the centrifugal type), which delivers water to the hot-well. The boiler feed pump returns some water to the boiler, while the surplus water flows by gravity back to the cooling pond. The air and uncondensed water vapour is withdrawn at the top of the condenser by a separate dry air pump or an ejector. The air pump required to remove air and uncondensed water vapour will be of small capacity for two reasons, firstly, it has to handle air and water vapour

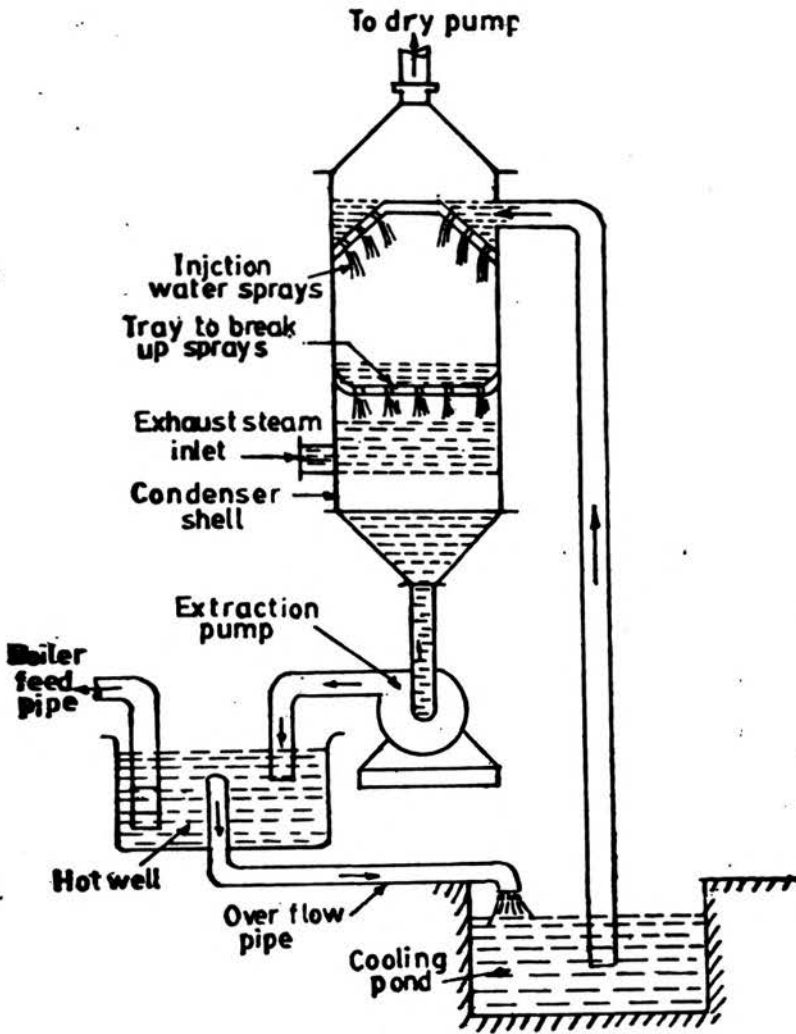


Fig. 1-2. Low-level counter flow jet condenser.

alone, and secondly, it has to deal with smaller volume of air and water vapour since the volume of air and water vapour is reduced due to their cooling while rising through the stream of cooling water. The cooling water is usually lifted from the source of supply (water tank or cooling pond) to the condenser head (top) by means of the vacuum created within the condenser. The greatest height to which water can be lifted by this means is about 5.5 metres with a vacuum of about 710 mm of mercury in the condenser. A cooling water pump is used when necessary, to aid in lifting the water to the condenser head.

In **parallel flow jet condensers**, the cooling water and exhaust steam flow in the same direction, downwards to the bottom of the condenser from where they are all

withdrawn by a single pump known as wet air pump and delivered to the hot-well. Since the pump has to deal with the condensate, air and water vapour, the capacity of the condenser to maintain vacuum is limited and the vacuum produced in such a condenser will, therefore, be low. The exhaust steam enters either at the top or the side of the condensing vessel and the cooling water enters immediately below. The cooling water is usually lifted from the source of supply (water tank or cooling pond) to the injection head by means of the vacuum created within the condenser as in the case of counter flow jet condenser. A cooling water pump is used when necessary, to help in lifting the water to the condenser head (top).

High-level or Barometric Jet Condensers is so named because it is placed at a

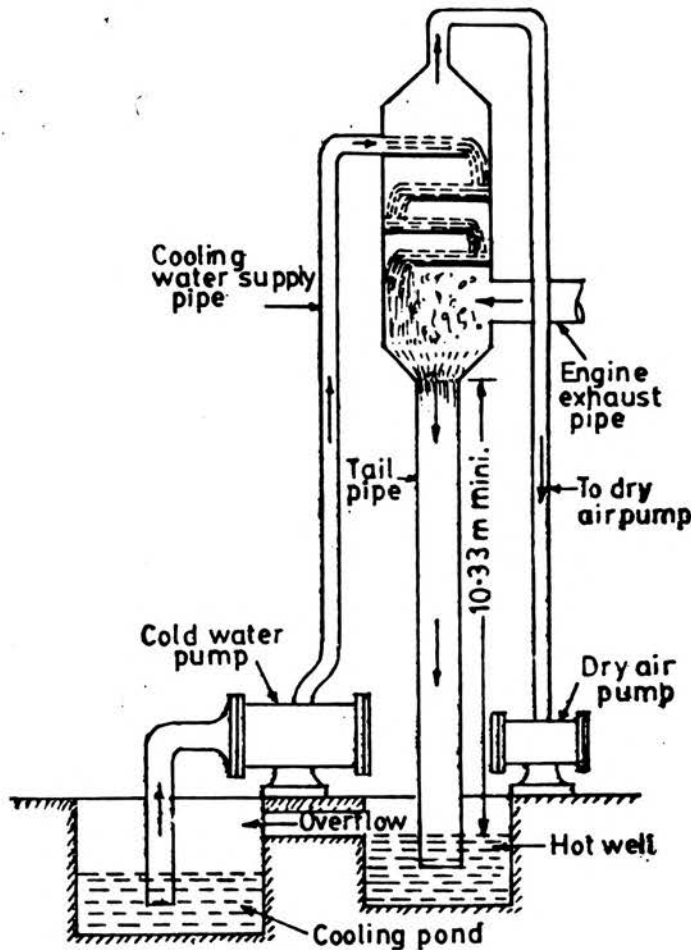


Fig. 1-3. High-level or barometric jet condenser.

The cooling water enters at the side near the top of the condenser. Rapid condensation of exhaust steam takes place. The condensate and cooling water flow out of the condenser by gravity into the hot-well. There is, thus, no need of a water extraction pump. The air released from the condensed steam passes upwards through the descending (falling) cold water and gets cooled. It is then removed by a separate dry air pump or an ejector at the top of the condenser. The size of the dry air pump required is comparatively small at it has not to handle the condensate.

Unless a supply of cooling water under pressure is available, an injection water pump will have to be provided in this type of condenser. The injection water pump will have to first lift water through a head of about 12 metres, but after the vacuum is formed in the condenser, the pump will have to lift water through a head of $12 - 5.5 = 6.5$ metres

height greater than that of water barometer. If a long pipe, over 10 metres in length, closed at one end, and filled with water, inverted without spilling (throwing out) any water, and the open end submerged in an open tank of water, the atmospheric pressure would hold the water up in the pipe to a height of 10 metres at sea level. This fact is made use of in a barometric condenser by making the tail pipe more than 10 metres in height and thus making it impossible for any vacuum in the condenser, to cause the water to rise high enough in the tail pipe and flood the engine.

Figure 1-3 illustrates a high-level counter flow jet condenser. The condenser is elevated at about 10.33 metres above the hot-well, and in general design it is similar to low-level counter flow jet condenser. The water outlet of tail pipe extends from the bottom of the condenser to the ground level and has its lower end immersed in the hot-well. The exhaust steam enters the lower part of the condenser through the engine exhaust pipe and rises to the top through the falling cooling water.

only (5.5 metres being the lift of water due to vacuum formed in the condenser).

The principle of operation of **Ejector condenser** is that the momentum of flowing

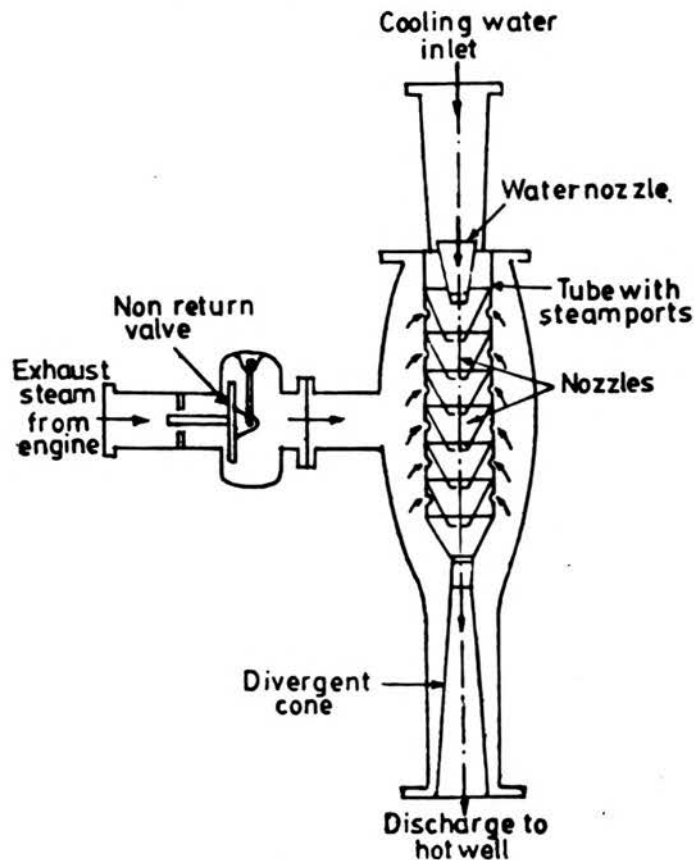


Fig. 1-4. Ejector condenser.

water ejects (throws out) the condensate and air without the aid of a pump. A common form of an ejector condenser is shown in fig. 1-4. The condenser consists of a central vertical tube in which are fixed a number of cones or converging nozzles.

The exhaust steam enters at the left and surrounds this central tube. In the central tube there are a number of steam ports. The cooling water enters the converging nozzle at the top under a head from 5 to 6 metres and attains high velocity while passing through it. In flowing past the steam ports the water produces a vacuum. The vacuum causes the exhaust steam and air to flow through the ports in the tube and mix with the cooling water. The exhaust steam gets condensed, as a result of which the vacuum is further increased. The condensate and air then pass on to the diverging cone where the kinetic energy is partly transformed into pressure energy so that the condensate and air will be discharged into the hot-well

against the pressure of the atmosphere. The condenser acts as an air pump as well as a condenser. This type of condenser is usually fitted with a non-return valve as shown, to prevent a sudden backward rush of water into the engine exhaust pipe in case of sudden failure of water supply to the condenser. To ensure satisfactory working under all conditions of load, the ejector condenser must be supplied with cooling water at inlet branch on the condenser of not less than 6 metres head of water. This head is usually derived from a centrifugal pump or from overhead tank.

An ejector condenser requires more cooling water than any other type of jet condenser. The vacuum obtained is about 620 mm of mercury. It is a small jet condenser as compared with other condensers and there being no pump, the first cost is low. It is also simple and reliable but can be used only for small power units.

1.5 Surface Condensers

Surface condensers may be sub-divided into :

- ... Surface condenser in which exhaust steam passes *over* a series of tubes through which the cooling water is flowing.
- ... The evaporative surface condenser in which exhaust steam passes *through* a series of tubes and water is allowed to flow in the form of thin film outside the tubes while air passes upwards outside the tubes.

A surface condenser, as illustrated in fig. 1-5, consists of a cast iron shell, cylindrical in shape and closed at each end to form a water box. A tube plate is located between each cover head and the shell. A number of water tubes are fixed to the tube plates.

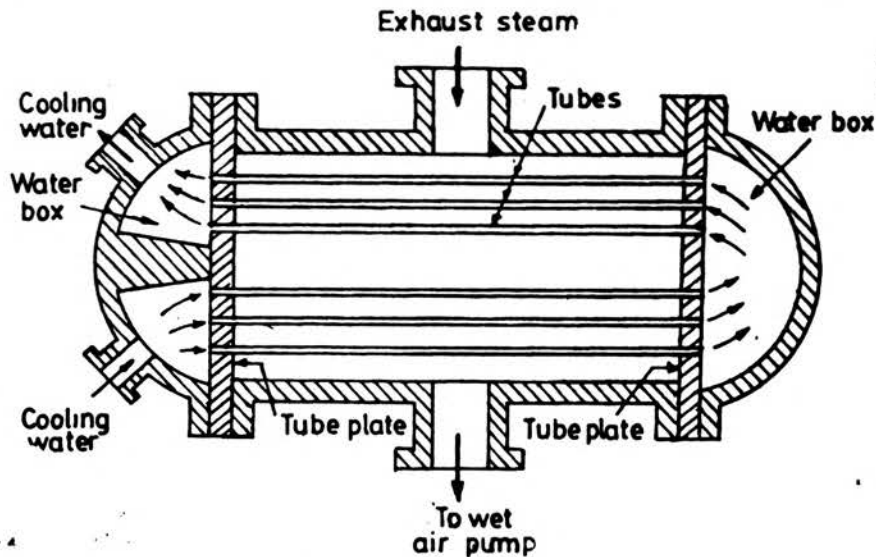


Fig. 1-5. Two-flow surface condenser.

The exhaust steam from the engine enters at the top of the condenser and is condensed by coming in contact with the cold surface of the tubes through which cooling water is being circulated. The cooling water enters at one end of the tubes situated in the lower half of the condenser and after flowing to the other end returns in the opposite direction through the tubes situated in the upper half of the condenser. The resulting water from the condensation of the exhaust steam and the air associated with the uncondensed water vapour, are extracted from the bottom of the condenser where the temperature is the lowest, so that the work of the wet air pump is reduced.

The surface condenser of this type requires two pumps, namely, wet air pump to remove air and condensate, and a water circulating pump to circulate the cooling water under pressure through the tubes of the condenser. Steam driven reciprocating pumps are used, but electric driven centrifugal pumps are used very extensively (commonly) for circulating water and condensate removal. Steam ejectors are also sometimes used for air removal.

Surface condensers may be classified as, *two-flow* or *multi-flow* condensers. Surface condenser, as illustrated in fig. 1-5, is a two-flow condenser because the circulating water traverses (travels) the whole length of the condenser twice. By introducing more partitions in the water boxes, the condenser may be converted into a three-flow condenser or even four-flow condenser. The velocity of cooling should be increased with the increase of number of flows. The rate of transmission of heat through the tubes to the circulating water, increases with the increase of number of flows, but the power required to circulate the water is increased.

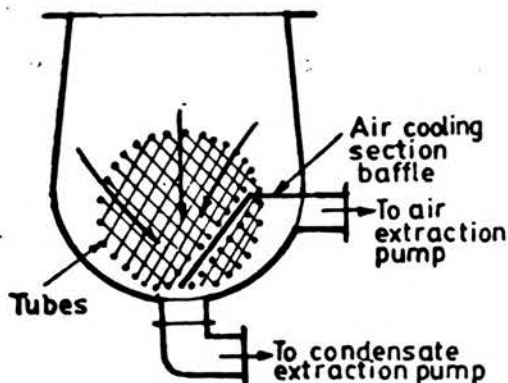


Fig. 1-6. Schematic view of a down flow surface condenser.

Another way of classifying surface condensers is according to the direction of flow of exhaust steam viz. *down-flow*, *central flow*, *inverted flow*, etc.

Figure 1-6 shows a sectional view of a **down flow condenser**. The exhaust steam enters at the top and flows downwards over the tubes through which the cooling water is flowing. The exhaust steam as a result is condensed and the condensate is extracted from the bottom by the condensate extraction pump. The cooling water enters at one end of the tubes situated in the lower half of the

condenser and after flowing to the other end returns in the opposite direction through the tubes situated in the upper half of the condenser. The temperature of condensate, therefore, decreases as the exhaust steam passes downwards, and hence partial pressure of steam decreases from top to bottom of the condenser. The air exit is shielded from the down stream of the condensate by means of a baffle plate, and thus air is extracted with only a comparatively small amount of water vapour. As the air passes downwards, it is progressively cooled and becomes denser (partial pressure of air increases) and hence it is extracted from the lowest convenient point. In a condenser of this type, therefore, the partial pressure of steam decreases, the partial pressure of air correspondingly increases, as the mixture passes from top to the bottom of the condenser. The result of all these effects is that the condensate temperature falls below the the exhaust steam temperature which enters at the top. Thus, by cooling the air, the capacity of the air pump is considerably reduced. (See illustrative problem-15).

In **Central Flow Surface Condenser** (fig. 1-7), the suction pipe of the air extraction pump is placed in the centre of the tubes nest; this causes the condensate to flow radially towards the centre as shown by the arrows in the figure. The condensate leaves at the bottom where the condensate extraction pump is situated. The air is withdrawn from the centre of the nest of tubes. This method is an improvement on the down flow type as the exhaust steam is directed radially inward by a volute casing around the tubes nest; it has thus access to the whole periphery of the tubes.

Where the supply of cold water is extremely limited, the **evaporative condenser** is

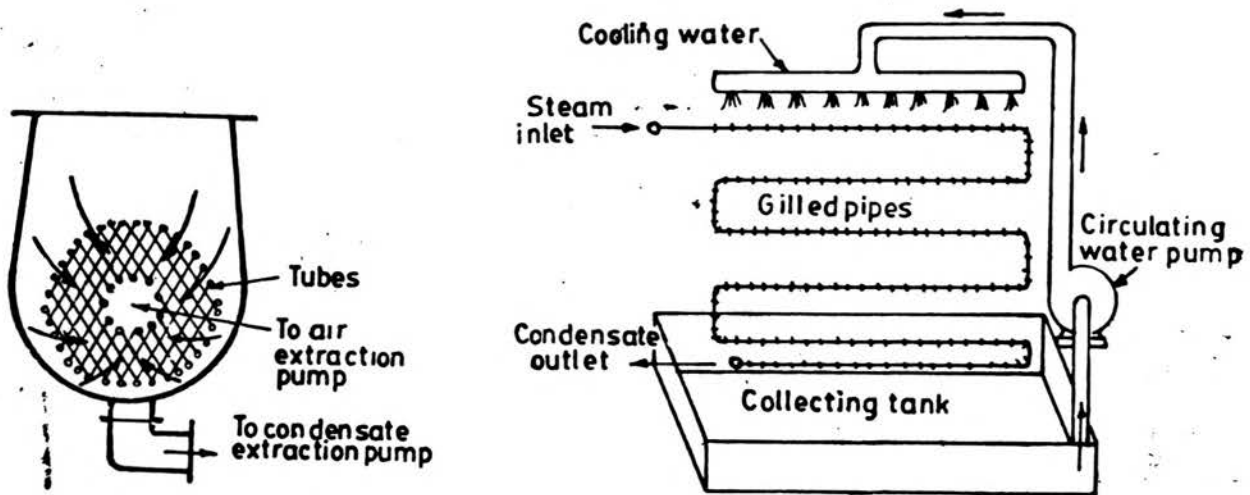


Fig. 1-7. Sectional view of a central flow surface condenser.

Fig. 1-8. Evaporative condenser.

the only suitable type which can be run on a minimum supply of cooling water, and even without cooling water in cold weather and on light loads. Exhaust steam from the engine is exhausted into a coil of gilled pipes or series of tubes, the outlet of which is connected to the wet air pump (fig. 1-8). Cooling water is allowed to flow in a thin film over the outside of the tubes. A natural or forced air current causes rapid evaporation of this film of water. The effect of this is that not only the steam inside the tubes is condensed but some of the cooling water is also evaporated on the outside of the tubes. The process of evaporation cools the water. The film of water on the outside of the tubes is maintained by allowing water to trickle (fall) over them continuously.

The water which is not evaporated falls into an open tank or collecting tank under the condenser, from which it can be drawn by circulating water pump and used over

again. The evaporative condenser is placed outside in the open air. On account of nuisance which would result from the production of clouds of steam, this type of condenser is restricted to small power plants.

1.6 Comparison of Jet and Surface Condensers

Jet condensers have low first cost, occupy small space, and attain high vacuum. However, they have more air to remove which requires large air pumps. In large plants, jet condensers are not used because, apart from the loss of the condensate, the power consumption of jet condenser pumps and the first cost of the water and air pumps, out-weigh the advantage of the high vacuum produced by them.

Surface condensers provide both higher vacuum and the recovery of the condensate. The construction of surface condenser is more complicated, and its first cost is greater. It occupies large space, and attention required is greater than in the case of a jet condenser. It provides pure feed water for the boilers which out-weighs its disadvantages. The necessity of having pure feed water for the boilers makes the use of a surface condensers universal for marine services.

1.7 Sources of Air in Condensers

Following are the *chief sources of air* found in condensers :

- ... Air leaks in condenser from atmosphere at the joints of the parts which are internally under a pressure less than that of atmosphere. The amount of air leaking in, mainly depends upon the accurate workmanship and can, with care in the design and making of the vacuum joints, be reduced to a very small quantity.
- ... Air also comes in with the steam from the boiler into which it enters dissolved in feed water. The amount of air coming in depends upon the treatment the feed water receives before it enters the boiler. The air entering through this source is relatively small.
- ... In case of jet condensers, some air comes in with the injection water (cooling water) in which it is dissolved.

In the surface condensers of well designed and properly maintained steam turbine plants, the amount of air entering condensers is about 5 kg per 10,000 kg of steam. With reciprocating steam engines, the air entering is about 15 kg per 10,000 kg of steam.

In case of jet condensers the amount of air dissolved in injection water is about 0.5 kg per 10,000 kg of water.

The important *effects of presence of air* in the condenser are as follows :

- ... With the increased amount of air in the condenser the condenser pressure or back pressure is increased. This reduces the useful work done in the primemover.
- ... Presence of air also lowers the partial pressure of steam and therefore lowers the saturation temperature of steam. With the lowering of the saturation temperature, the evaporation enthalpy (latent heat) of steam increases and therefore more cooling water will be required in the condenser.

1.8 Measurement of vacuum

The vacuum in a condenser is usually expressed in millimetres of mercury and it is the difference between the barometric pressure (or barometric height) and absolute pressure in condenser (fig. 1-9). In order to know the absolute pressure in the condenser, both the vacuum gauge and barometer must be read. The difference between the barometer and vacuum gauge readings will give the absolute pressure in the condenser.

Barometric pressure is a variable quantity and varies from place to place. Hence, it is more convenient for the purpose of comparison to refer vacuum gauge readings to a

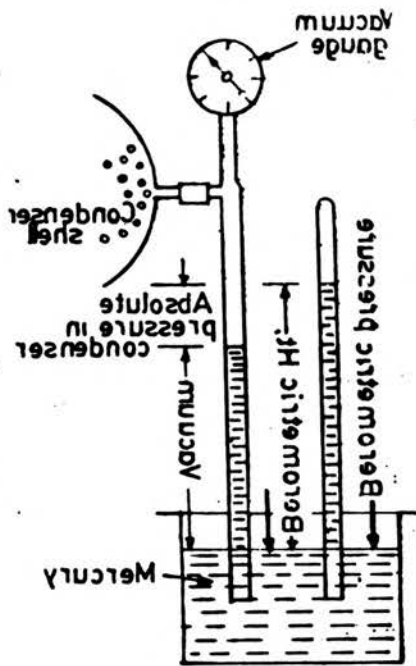


Fig. 1-9. Measurement of vacuum.

Problem-1 : The vacuum gauge reading in a condenser is 714 mm of Hg when the barometer reads 759 mm. In another case the vacuum is 709 mm of Hg while the barometer is 756 mm. Correct these vacuum gauge readings to the standard barometer of 760 mm.

If the barometric pressure is raised to 760 mm without any change in absolute pressure in condenser, then mercury in the vacuum gauge will rise to $(760 - 759) + 714 = 715$ mm, i.e. the standard or corrected vacuum gauge reading is **715 mm of Hg.**

Similarly, corrected vacuum gauge reading in the second case will be

$$(760 - 756) + 709 = 713 \text{ mm of Hg.}$$

1.9 Estimation of Quantity of Cooling Water

To find the quantity of cooling water necessary to condense a given quantity of steam is a problem of a simple heat exchange between two fluids.

Let m_w = mass of cooling water required in kg per hr.,

m_s = mass of exhaust steam condensed in kg per hr.,

t_s = saturation temperature of the exhaust steam (corresponding to the condenser vacuum),

t_c = temperature of the condensate leaving the condenser,

t_1 = inlet temperature of cooling water,

t_2 = outlet temperature of the cooling water,

x = dryness fraction of exhaust steam entering the condenser,

L = enthalpy of evaporation of exhaust steam entering the condenser, in kJ/kg. and

K = specific heat of cooling water = 4.187 kJ/kg K.

The exhaust steam entering the condenser is usually slightly wet even when steam supplied to the steam engine or steam turbine is usually superheated. The exhaust steam (wet steam) on condensing gives up first all its evaporation enthalpy xL and then a portion of its sensible enthalpy $(t_s - t_c)$ due to the cooling of the condensate below the saturation temperature corresponding to the vacuum; this latter effect is due to air which leaks into the condenser. The heat given up or lost by the exhaust steam per

standard barometer of 760 mm of mercury (or 1.01325 bar).

Standard or corrected vacuum in mm of Hg.

$$= 760 \text{ mm of mercury} - \text{absolute pressure in condenser in mm of Hg.}$$

$$= 760 \text{ mm of Hg} -$$

$$\left[\begin{array}{l} \text{Barometer reading} \\ \text{in mm of Hg.} \end{array} - \begin{array}{l} \text{Vacuum gauge reading} \\ \text{in mm of Hg.} \end{array} \right] \dots(1.1)$$

Since one standard atmosphere = 760 mm of Hg = 1.01325 bar (101.325 kPa),

$$1 \text{ bar} = \frac{760}{1.01325} = 750 \text{ mm of Hg.}$$

$$\therefore \text{Pressure equivalent of 1 mm of Hg} = \frac{1.01325}{760}$$

$$= 0.001333 \text{ bar or } = 0.1333 \text{ kPa.}$$

hour in condensing is, therefore, equal to $m_s [xL + (t_s - t_c)]$ kJ/kg. The heat removed or gained by cooling water per hour is equal to $m_w (t_2 - t_1)$ kJ/kg.

Assuming that all the heat given up (lost) by wet exhaust steam in condensing is removed (gained) by the cooling water,

i.e. heat lost by exhaust steam in kJ per hour = heat gained by cooling water in kJ per hour.

$$\text{i.e. } m_s [xL + (t_s - t_c)K] = m_w (t_2 - t_1) K$$

\therefore The mass of cooling water required per hour,

$$m_w = \frac{m_s [xL + K(t_s - t_c)]}{K(t_2 - t_1)} \text{ kg/hr,} \quad \dots(1.2a)$$

and dryness fraction of exhaust steam entering the condenser,

$$x = \frac{\frac{m_w}{m_s} (t_2 - t_1) K - (t_s - t_c)K}{L} \quad \dots(1.2b)$$

The above equations apply to *surface condenser* only. Since in *jet condenser* steam and cooling water mix together, the temperature of the condensate t_c will be the same as that of the outlet temperature of cooling water t_2 , i.e. $t_c = t_2$. The mass of cooling water m_w kg per hour that is necessary to condense m_s kg of exhaust steam per hour in case of a jet condenser will be given by the expression

$$m_w = \frac{m_s [xL + K(t_s - t_2)]}{K(t_2 - t_1)} \text{ kg per hour,} \quad \dots(1.3a)$$

and the dryness fraction of exhaust steam entering the jet condenser,

$$x = \frac{\frac{m_w}{m_s} (t_2 - t_1) - K(t_s - t_2)}{L} \quad \dots(1.3b)$$

Problem-2 : In a surface condensing plant the following data were obtained :

Temperature of exhaust steam entering the condenser	... 42.67°C
Temperature of condensate leaving the condenser	... 35°C
Inlet temperature of cooling water	... 16.5°C
Outlet temperature of cooling water	... 30°C
Quantity of cooling water per hour	... 46,250 kg
Quantity of condensate per hour	... 1,190 kg

Calculate the dryness fraction of exhaust steam entering the condenser.

From steam (pressure) tables, pressure of steam corresponding to 42.67°C is 0.085 bar.

From steam (pressure) tables, at 0.085 bar, $L = 2,400.3$ kJ/kg.

Let the unknown dryness fraction of exhaust steam entering the condenser = x .

Then, heat lost by *one kg of exhaust steam* of dryness fraction x at 42.67°C (0.085 bar) in condensing to water at 42.67°C and in being cooled from 42.67°C to 35°C

$$= xL + K(t_s - t_c) = 2,400.3x + (42.67 - 35) \times 4.187 \text{ kJ per kg of steam.}$$

Heat removed or gained by cooling water *per kg of exhaust steam*

$$= \text{Mass of cooling water required to condense one kg exhaust steam} \times \text{specific heat of water} \times \text{rise in temperature of cooling water.}$$

$$= \frac{m_w}{m_s} (t_2 - t_1) K = \frac{46,250}{1,190} (30 - 16.5) 4.187 = 2,196.9 \text{ kJ per kg of exhaust steam.}$$

Neglecting losses,

heat lost by 1 kg of exhaust steam = heat gained by cooling water per kg of exhaust steam

$$\text{i.e. } 2,400.3x + (42.67 - 35) 4.187 = 2,196.9$$

$$\therefore x = 0.902 \text{ (dryness fraction of exhaust steam entering the condenser)}$$

Alternatively, using eqn. (1.2b), dryness fraction of exhaust steam entering the condenser,

$$x = \frac{\frac{m_w}{m_s} (t_2 - t_1) K - (t_s - t_c) K}{L}$$

$$\therefore x = \frac{\frac{46,250}{1,190} (30 - 16.5) 4.187 - (42.67 - 35) 4.187}{2,400.3} = 0.902 \text{ (same as before)}$$

Problem-3 : The following particulars relate to a test of a surface condenser of a steam turbine :

Absolute pressure of exhaust steam, 0.06 bar; temperature of condensate, 32°C; temperature of cooling water in the condenser at inlet and outlet, 15°C and 30°C respectively; mass of condenser cooling water per kg of steam, 32 kg. Assuming that all heat lost by exhaust steam is taken up by cooling water, determine the dryness fraction of the steam as it enters the condenser. Take specific heat of water as 4.187 kJ/kg K.

From steam (pressure) tables, at 0.06 bar, $h = 151.53$ kJ/kg,

$$L = 2,415.9 \text{ kJ/kg, and } t_s = 36.16^\circ\text{C.}$$

Heat lost by one kg of exhaust steam

$$= xL + K (t_s - t_c)$$

$$= x \times 2,415.9 + 4.187 (36.16 - 32) = 2,415.9x + 17.418 \text{ kJ/kg of exhaust steam.}$$

Heat removed or gained by cooling water per kg of exhaust steam

$$= \frac{m_w}{m_s} \times K (t_2 - t_1) = 32 \times 4.187 \times (30 - 15) = 2,009.76 \text{ kJ per kg of exhaust steam}$$

Neglecting losses,

Heat lost by 1 kg exhaust steam = Heat gained by cooling water per kg of exhaust steam

$$\text{i. e. } 2,415.9x + 17.418 = 2,009.76$$

$$\therefore x = 0.8246 \text{ (dryness fraction of steam entering the condenser)}$$

1.10 Dalton's Law of Partial Pressures

Dalton's law states that in a mixture of perfect gases which do not react chemically with one another, total pressure exerted by the mixture is the sum of partial pressures which each gas would exert if it separately occupied the whole volume, and was at the same temperature as the mixture. In other words, in such a mixture each constituent gas obeys its own characteristic equation as if the other constituent gases were absent.

Let a mixture of gaseous substances be made up of constituents a, b, c, etc Then, according to Dalton's law the total pressure p_m exerted by a mixture of gases is the sum

of the partial pressures. A partial pressure is the pressure which one constituent, such as a , would exert if it alone occupied the whole volume (volume of the mixture), and was at the same temperature as the mixture. We therefore have,

$$p_m = p_a + p_b + p_c + \dots$$

If a mixture of gases a , b , c , etc. is contained in a volume v , each gas occupies the whole volume v , exerts a partial pressure p_a , p_b , p_c , etc. and all the constituents are at an absolute temperature T . Then, for any constituent such as a ,

$$p_a \times v_a = m_a \times R_a \times T_a$$

$$\text{or } m_a = \frac{p_a \times v_a}{R_a \times T_a} \quad \text{or } v_a = \frac{m_a \times R_a \times T_a}{p_a}$$

Dalton's law holds good approximately for mixture of gases and vapours which do not combine chemically.

In steam condensers, we have mixture of steam (water vapour) and air, and the total absolute pressure which exists in the condensers is the sum of the pressures exerted by the steam and non-condensable gases. These non-condensable gases consist chiefly of air and carbon dioxide. The carbon dioxide is in extremely small quantity in comparison with the air and may be neglected. We, therefore have $p_m = p_a + p_s$ where, p_m is the total pressure in the condenser, p_a is the partial pressure of air, and p_s is the partial pressure of steam.

If the effective volume of the condenser is $v \text{ m}^3$, then according to Dalton's law, each constituent part occupies the whole volume at its partial pressure. We, therefore, have $v = v_s = v_a$ where, v_s is the volume of steam in m^3 , and v_a is the volume of air in m^3 .

If the temperature of the mixture in the condenser is $t^\circ\text{C}$, then according to Dalton's law, temperature of each constituent part will be the same as the temperature of the mixture.

We, therefore, have $t = t_a = t_s$

where, t_a is the temperature of air, and t_s is the temperature of steam.

The application of Dalton's law of partial pressures to condensers and air pumps is illustrated by the following problems :

Problem-4 : *The vacuum in a steam condenser is 685 mm of mercury (barometer 760 mm) and the temperature is 28.96°C . What is partial pressure of air present in the condenser? If the volume of condenser steam space is 8.5 m^3 , what is the mass of air present in the condenser ? Take $R = 0.287 \text{ kJ/kg K}$ for air.*

The combined pressure of steam and air in the condenser,

$$p_m = 760 - 685 = 75 \text{ mm of Hg,}$$

Or total absolute pressure in the condenser, $p_m = \frac{75}{750} = 0.1 \text{ bar}$

$$(\because 1 \text{ bar} = 750 \text{ mm of Hg})$$

At 28.96°C , partial pressure of steam, $p_s = 0.04 \text{ bar}$ [from steam (Pressure) tables].

Hence, by Dalton's law, the partial pressure of air,

$$p_a = p_m - p_s = 0.1 - 0.04 = 0.06 \text{ bar}$$

By Dalton's law, air and steam occupy the same volume at their partial pressure and have the same temperature, and hence air present in the condenser will occupy a volume of 8.5 m^3 at 0.06 bar and will be at a temperature of 28.96°C .

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

$$\text{Mass of air, } m_a = \frac{p_a v_a}{R_a T_a} = \frac{(0.06 \times 10^5) 8.5}{(0.287 \times 10^3) (28.96 + 273)} = 0.5885 \text{ kg.}$$

Mass of air present in the condenser = **0.5885 kg.**

Problem-5 : The temperature in a surface condenser is 42°C and the vacuum is 685 mm of mercury (barometer 749 mm). Correct the vacuum to a standard barometer of 760 mm and determine the pressure of steam and air, and mass of air associated with one kg of steam. Take $R = 0.287 \text{ kJ/kg K}$ for air.

If the barometer is raised from 749 to 760 mm of Hg without any change in absolute pressure in condenser, then the mercury in the vacuum gauge will rise by $760 - 749 = 11 \text{ mm}$.

\therefore Corrected vacuum to standard barometer = $11 + 685 = 696 \text{ mm of Hg}$.

Total pressure in the condenser, $p_m = 749 - 685 = 64 \text{ mm of Hg}$.

Now, 760 mm of Hg = 1 standard atmosphere = 1,01,325 $p_a = 1.01325 \text{ bar}$.

$$\therefore 1 \text{ mm of Hg} = \frac{1.01325}{760} = 0.001333 \text{ bar.}$$

\therefore Total pressure in the condenser, $p_m = 64 \times 0.001333 = 0.08512 \text{ bar}$

From steam (temperature) tables, at 42°C , partial pressure of water vapour (steam),

$$p_s = 0.08208 \text{ bar}$$

Hence by Dalton's law, the partial pressure of air

$$p_a = p_m - p_s = 0.08512 - 0.08208 = 0.00304 \text{ bar}$$

Specific volume of one kg of steam at 0.08208 bar (saturation temperature 42°C)

$$= 17.671 \text{ m}^3/\text{kg.} \text{ [from steam (temperature) tables]}$$

By Dalton's law, air and steam occupy the same volume at their partial pressure and have the same temperature and hence the air present per kg of steam will occupy a volume of 17.671 m^3 at 0.00304 bar and will be at a temperature of 42°C .

From characteristic equation of air,

$$m_a = \frac{p_a v_a}{R_a T_a} = \frac{(0.00304 \times 10^5) \times 17.671}{(0.287 \times 10^3) \times (42 + 273)} = 0.05941 \text{ kg.}$$

Problem-6 : A closed vessel of 0.7 m^3 capacity contains saturated water vapour and air at a temperature of 42.67°C and pressure of 0.127 bar. Due to further air leakage into the vessel, the pressure rises to 0.28 bar and temperature falls to 37.63°C . Calculate the mass of air which has leaked in. Take $R = 0.287 \text{ kJ/kg K}$ for air.

Initially : From steam (pressure) tables, at 42.67°C , partial pressure of water vapour (steam), $p_s = 0.085 \text{ bar}$

Hence by Dalton's law, the partial pressure of air,

$$p_a = p_m - p_s = 0.127 - 0.085 = 0.042 \text{ bar}$$

\therefore Mass of air, present initially in the vessel of 0.7 m^3 capacity,

$$m_{ai} = \frac{p_a v_a}{R_a T_a} = \frac{(0.042 \times 10^5) \times 0.7}{(0.287 \times 10^3) (42.67 + 273)} = 0.0325 \text{ kg.}$$

Finally, from steam (pressure) tables, at 37.63°C , partial pressure of steam

$$p_s = 0.065 \text{ bar.}$$

Hence, by Dalton's law, the partial pressure of air,

$$p_a = p_m - p_s = 0.28 - 0.065 = 0.215 \text{ bar}$$

∴ Mass of air present *finally* in the vessel of 0.7 m³ capacity,

$$m_{af} = \frac{p_a V_a}{R_a T_a} = \frac{(0.215 \times 10^5) \times 0.7}{(0.287 \times 10^3) (37.63 + 273)} = 0.1688 \text{ kg}$$

Hence, air leakage into the vessel = $m_{af} - m_{ai} = 0.1688 - 0.0325 = 0.1363 \text{ kg}$.

1.11 Vacuum Efficiency

In a steam condenser we have a mixture of steam and air, and the total pressure which exists in the condenser is the sum of the partial pressures exerted by the steam and air. With no air present in the condenser, the total absolute pressure in the condenser would be equal to partial pressure of steam corresponding to the temperature of condenser, and maximum vacuum would be obtained in the condenser. The ratio of the actual vacuum obtained at the steam inlet to the condenser, to this maximum vacuum (or ideal vacuum) which could be obtained in a perfect condensing plant (with no air present) is called the *vacuum efficiency*, i. e.

$$\begin{aligned} \text{Vacuum efficiency} &= \frac{\text{Actual vacuum at the steam inlet to the condenser}}{[\text{Barometric pressure}] - \left[\begin{array}{c} \text{Absolute pressure corresponding} \\ \text{to the temperature of} \\ \text{condensation} \end{array} \right]} \\ \text{or} &= \frac{[\text{Barometric pressure}] - \left[\begin{array}{c} \text{Absolute pressure in the} \\ \text{condenser} \end{array} \right]}{[\text{Barometric pressure}] - \left[\begin{array}{c} \text{Absolute pressure corresponding} \\ \text{to temperature of condensation} \end{array} \right]} \quad \dots(1.4) \end{aligned}$$

If the absolute pressure of steam corresponding to the temperature of condensation were equal to the absolute pressure in the condenser, the vacuum efficiency would be 100%. In fact, there will always be some air present in the condenser due to leakage and dissolved air present in the steam entering the condenser. The value of vacuum efficiency, therefore, depends upon the quantity of air removed from the condenser by the air pump.

Problem-7 : Steam enters a condenser at 32.88°C and with barometer standing at 760 mm of Hg, a vacuum of 685 mm of Hg was produced. Determine the vacuum efficiency.

From steam (Pressure) tables, at 32.88°C, partial pressure of steam
= 0.05 bar = 0.05 × 750 = 37.5 mm of Hg.

Using eqn. (1.4),

$$\begin{aligned} \text{Vacuum efficiency} &= \frac{\text{Actual vacuum}}{[\text{Barometric pressure}] - \left[\begin{array}{c} \text{Absolute pressure corresponding} \\ \text{to temperature of} \\ \text{condensation} \end{array} \right]} \\ &= \frac{685}{760 - (0.05 \times 750)} = \frac{685}{722.5} = 0.9481 \text{ or } 94.81 \% \end{aligned}$$

$$\begin{aligned} \text{or Vacuum efficiency} &= \frac{\text{Actual vacuum}}{\left[\begin{array}{c} \text{vacuum corresponding to saturation} \\ \text{temperature of condensate} \end{array} \right]} \\ &= \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{685}{760 - 37.5} = 0.9481 \text{ or } 94.81 \% \text{ (same as before).} \end{aligned}$$

Problem-8 : In a surface condenser test the following observations were made :

Vacuum 700 mm of Hg; barometer 765 mm of Hg; mean temperature of condensation 36.16°C; hotwell temperature 30°C; mass of cooling water 47,500 kg/hour; inlet temperature of cooling water 17°C; outlet temperature of cooling water 32°C; mass of condensate 1,500 kg/hour. Find : (a) the mass of air present per m³ of condenser volume, (b) the state of exhaust steam entering the condenser, and (c) the vacuum efficiency.

Take $R = 0.287 \text{ kJ/kg K}$ for air.

(a) Total or combined pressure of steam and air in the condenser,

$$p_m = \frac{765 - 700}{750} = 0.0867 \text{ bar}$$

At 36.16°C, partial pressure of steam, $p_s = 0.06 \text{ bar}$ [from steam (pressure) tables].

Hence, by Dalton's law, partial pressure of air,

$$p_a = p_m - p_s = 0.0867 - 0.06 = 0.0267 \text{ bar}$$

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

Mass of air present per m³ of condenser volume,

$$m_a = \frac{p_a v_a}{R_a T_a} = \frac{(0.0267 \times 10^5) \times 1}{(0.287 \times 10^3) \times (36.16 + 273)} = 0.0301 \text{ kg}$$

(b) Heat removed or gained by cooling water per kg of exhaust steam

$$= \frac{m_w}{m_s} \times (t_2 - t_1) K = \frac{47,500}{1,500} (32 - 17) \times 4.187 = 1,988.83 \text{ kJ}$$

From steam (pressure) tables, pressure of steam corresponding to 36.16°C is 0.06 bar.

From steam (pressure) tables, at 0.06 bar, $L = 2,415.9 \text{ kJ/kg}$ of steam.

Let the dryness fraction of steam entering the condenser be x . Then, heat given up (lost) by one kg of wet steam of dryness fraction x at 36.16°C (0.06 bar) in condensing to water at 36.16°C and in being cooled from 36.16°C to 30°C

$$= xL + (t_s - t_c) K = 2,415.9x + (36.16 - 30) \times 4.187 \text{ kJ/kg of exhaust steam.}$$

Neglecting losses, heat lost by one kg of exhaust steam = heat gained by cooling water per kg of exhaust steam.

$$\text{i.e. } 2,415.9 + (36.16 - 30) \times 4.187 = 1,988.83$$

From which, $x = 0.8126$, i.e. exhaust steam entering the condenser is **81.26% dry**.

Alternatively, using eqn. (1.2b), dryness fraction of exhaust steam entering the condenser,

$$x = \frac{\frac{m_w}{m_s} (t_2 - t_1) K - (t_s - t_c) K}{L}$$

$$x = \frac{\frac{47,500}{1,500} (32 - 17) 4.187 - (36.16 - 30) \times 4.187}{2,415.9} = 0.8126 \text{ (same as before)}$$

(c) Using eqn. (1.4), Vacuum efficiency

$$= \frac{\text{Actual vacuum in mm of Hg}}{\text{Vacuum corresponding to temperature of condensation in mm of Hg}}$$

$$= \frac{\text{Actual vacuum}}{\text{Ideal vacuum}} = \frac{700}{765 - (750 \times 0.06)} = \frac{700}{765 - 45} = 0.9722 \text{ or } 97.22\%$$

1.12 Condenser Efficiency

There is no standard method of determining the efficiency of a condenser, but a method adopted by the well known makers of steam turbines, Messers Parson & Co., has been widely used in engineering practice. By this method the thermal efficiency of a condenser is stated as the ratio of the difference between the outlet and inlet temperatures of cooling water, to the difference between the saturation temperature corresponding to the absolute pressure in the condenser and inlet temperature of the cooling water, i.e.

$$\text{Condenser efficiency} = \frac{\text{Rise in temperature of cooling water}}{\left[\text{saturation temperature corresponding to the absolute pressure in the condenser} \right] - \left[\text{Inlet temperature of cooling water} \right]} \quad \dots(1.5)$$

Problem-9 : The vacuum in a surface condenser is found to be 707.5 mm of Hg with barometer reading 760 mm of Hg. The cooling water enters the condenser at 15°C and leaves at 36°C. Find the condenser efficiency.

$$\begin{aligned} \text{Absolute pressure in the condenser} &= 760 - 707.5 = 52.5 \text{ mm of Hg} \\ &= \frac{52.5}{750} = 0.07 \text{ bar.} \end{aligned}$$

Saturation temperature corresponding to 0.07 bar is 39°C [from steam (Pressure) tables].

Using eqn. (1.5), Condenser efficiency,

$$\begin{aligned} &= \frac{\text{Rise in temperature of cooling water}}{\left[\text{saturation temperature corresponding to the absolute pressure in the condenser} \right] - \left[\text{Inlet temperature of cooling water} \right]} \\ &= \frac{36 - 15}{39 - 15} = 0.875 \text{ or } 87.5\% \end{aligned}$$

1.13 Air Pumps

The primary function of an air pump is to maintain vacuum in the condenser, as nearly as possible, equal to that corresponding to the exhaust steam temperature, by removing air from the condenser. Another common but not essential function of the air pump is to remove condensate together with the air from the condenser. An air pump which removes both air and condensate is called a *wet air pump*, while one which removes the moist air only is called a *dry air pump*.

Air pumps may be divided into :

- (i) Reciprocating piston or bucket pumps, (ii) Rotary pumps, which are generally dry pumps, (iii) Steam jet air pumps (ejectors), which are generally dry pumps, and (iv) Water jet pumps, which are always wet pumps.

A design of reciprocating piston or bucket wet air pump which has been largely used and is highly efficient, is the Edward's air pump shown in fig. 1-10. In this pump, foot and bucket valves are eliminated and the condensate is arranged to flow by gravity from condenser into the pump. The bucket or the piston has a conical bottom which fits into the conical bottom of the pump cylinder. On the down stroke of the piston, a partial vacuum is produced above it, since the head discharge valves or delivery valves are closed and sealed with water. Immediately the piston uncovers the ports, air and water

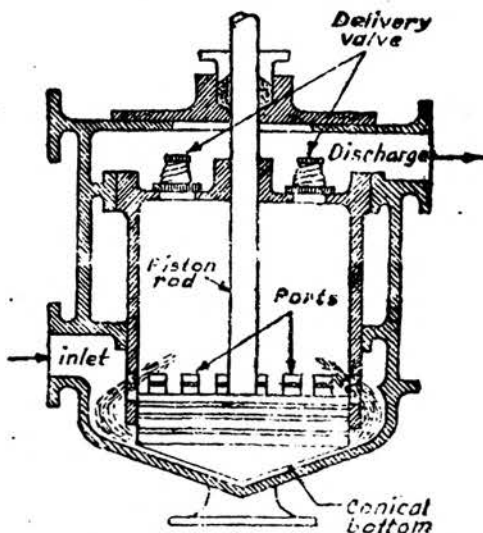


Fig. 1-10. Edward's air pump.

vacuum is produced above it, since the head discharge valves or delivery valves are closed and sealed with water. Immediately the piston uncovers the ports, air and water

vapour rush into the space above the piston. With the further downward motion of the piston, the conical part of the piston enters the condensate which has flowed into the conical bottom of the pump from the condenser and drives it through the ports into the barrel (pump cylinder) above the piston.

The rising piston traps the condensate, air and water vapour above the piston and raises the pressure slightly over that of the atmosphere until the head discharge valves open. The water vapour and air then pass to waste and the condensate gravitates to the hot-well.

Since the speed of reciprocating air pumps is very limited, they become very bulky for higher vacuum or large powers. For this reason rotary dry air pumps and steam jet air ejectors are widely used.

Rotary pump is a dry air pump and handles only air, although it is charged with water for the purpose of its operation. The water and air is discharged through a diverging cone which raises its pressure slightly greater than atmospheric. The water and air then pass to a slightly elevated tank in which water is cooled and returned to the pump.

Steam jet operated air ejectors are almost universally used for the production of high vacuum demanded in modern land condensing plants, owing to their simplicity the small space occupied, and the absence of moving parts with the consequent reliability. There are many types of ejectors now in the market and they all follow the same general principle. In the simplest type of air ejector described below, the air is compressed in two stages—the first stage compresses the air from condenser vacuum to about 650 mm Hg vacuum, and the second stage compressing from this vacuum 650 mm Hg to

atmospheric pressure. Both stages follow the same general principle, i.e. enthalpy of the steam is transformed in a convergent-divergent steam nozzle into kinetic energy. The rapidly moving jet of steam entrains (drags) the air and non-condensable gases, and the combined mass of entrained air and steam is discharged into a diffuser nozzle or diverging cone where portion of kinetic energy is re-transformed into pressure energy so that the air is discharged against a pressure higher than the ejector suction pressure.

Figure 1-11 illustrates the simple type of air ejector and is known as Lablanc's steam air ejector. The moist air from the condenser is drawn in at 1 and then compressed and discharged at not less than atmospheric pressure. Compression of air takes place in two stages. In the first stage there is one steam nozzle 2 and in the second stage there is a group of steam nozzles 3. All steam nozzles are of the De-Laval Type.

Steam generally at a pressure not less than 8 bar enters at 4, through a stop valve (not shown) and supplies steam directly to the second stage nozzles 3. The steam supply to the first stage nozzle 2 is through the steam pipe 5 in which there is a controlling valve 6. The steam expands in the nozzles and comes out from them with a velocity of above 1,000 m/sec. and at a small absolute pressure depending upon the vacuum in the condenser.

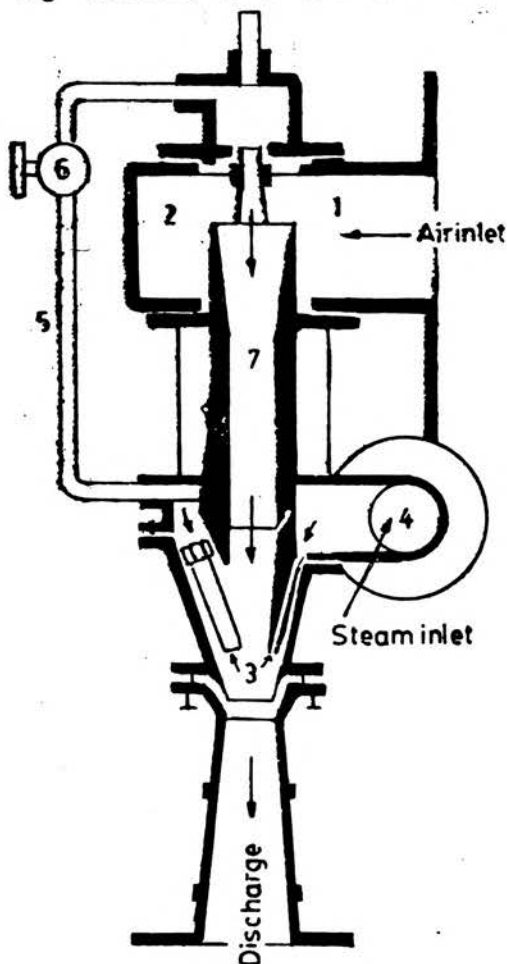


Fig. 1-11. Steam jet air ejector.

The steam issuing from the nozzle 2. entrains (drags) the air and water vapour entering at the air inlet 1. The whole mass of air, water vapour and steam is discharged into the compression pipe with considerable velocity and a small pressure rise is obtained in the diffuser or diverging cone 7. Before reaching the second stage, both air and steam is compressed to about seven times the pressure in the space. 1. Further compression required takes place in the second stage, aided by the steam jets from the nozzles 3. The steam issuing from nozzles 3 entrains the total mass of air, vapour and steam discharged from the first stage and compresses this mass along with its own steam in the second stage diverging cone or diffuser to atmospheric pressure. The steam then enters the boiler feed tank where steam is condensed.

Problem-10 : The vacuum at the air extraction pipe in a condenser is 710 mm of Hg (barometer 760 mm of Hg) and the temperature is 36.16°C. The air leakage into the condenser is 4 kg per 10,000 kg of steam. Determine : (a) the volume of air to be dealt with by the air pump per kg of steam entering the condenser, and (b) the mass of water vapour associated with this air. Take $R = 0.287 \text{ kJ/kg K}$ for air.

(a) Absolute pressure in the condenser,

$$p_m = 760 - 710 = 50 \text{ mm of Hg} = 50/750 = 0.0667 \text{ bar.}$$

From steam (pressure) tables, at 36.16°C, partial pressure of steam, $p_s = 0.06 \text{ bar}$.

Hence by Dalton's law, partial pressure of air,

$$p_a = p_m - p_s = 0.0667 - 0.06 = 0.0067 \text{ bar.}$$

Mass of air leaking per kg of steam, $m_a = 4/10,000 = 0.0004 \text{ kg}$.

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

$$\begin{aligned} \text{Volume of air, } v_a &= \frac{m_a R_a T_a}{p_a} = \frac{0.0004 \times (0.287 \times 10^3) \times (36.16 + 273)}{(0.0067 \times 10^5)} \\ &= 0.053 \text{ m}^3/\text{kg of steam} \end{aligned}$$

(b) By Dalton's law, 0.053 m³ is also the volume of water vapour.

From steam (Pressure) tables, specific volume of steam at 0.06 bar = 23.739 m³/kg.

Hence, mass of water vapour (steam) associated with this air = $\frac{0.053}{23.739} = 0.00223 \text{ kg}$.

Problem-11 : The temperature of steam entering a surface condenser is 53.97°C and the temperature of air pump suction is 45.81°C. The barometer reading is 757 mm of Hg.

Find : (a) the condenser vacuum,

(b) the water vapour pressure and the air pressure near the air pump suction and,

(c) If the effective capacity of the dry air pump on the suction stroke is 8.5 m³ per minute, find the mass of air entering the condenser per minute and the mass of steam carried over per minute in the air discharged from the air pump.

Take $R = 0.287 \text{ kJ/kg K}$ for air.

(a) From steam (pressure) tables, at 53.97°C, partial pressure of steam $p_s = 0.15 \text{ bar}$.

Partial pressure of air at steam inlet is very small and can be neglected. Hence, the total pressure in the condenser (p_m) may be taken as 0.15 bar.

Hence, the condenser pressure = $0.15 \times 750 = 112.5 \text{ mm of Hg}$.

∴ Condenser vacuum = $757 - 112.5 = 644.5 \text{ mm of Hg}$.

(b) From steam (Pressure) tables, partial pressure of steam, p_s at 45.81°C = 0.1 bar.

Hence, by Dalton's law, the partial pressure of air at suction,

$$p_a = p_m - p_s = 0.15 - 0.1 = 0.05 \text{ bar.}$$

(c) Assuming that the air pump deals with moist air only and not with condensate (dry air pump), the mass of air m_a is given by $p_a v_a = m_a R_a T_a$

$$\text{i.e. } (10^5 \times 0.05) \times 8.5 = m_a \times (0.287 \times 10^3) \times (45.81 + 273)$$

$$\therefore \text{Mass of air entering condenser, } m_a = 0.4654 \text{ kg per min.}$$

By Dalton's law, volume of steam entering the pump per minute is also 8.5 m^3 .

From steam (pressure) tables, specific volume of steam at 0.1 bar = 14.674 m^3 .

$$\therefore \text{Mass of } 8.5 \text{ m}^3 \text{ of steam per minute} = \frac{8.5}{14.674} = 0.5793 \text{ kg/min.}$$

Mass of the steam carried over in the air discharged = 0.5793 kg/min.

Problem-12 : A surface condenser deals with 12,500 kg of steam per hour. The leakage air in the system amounts to 1 kg per 2,500 kg of steam. The vacuum in the air pump suction is 705 mm of Hg (barometer 760 mm of Hg) and the temperature is 34°C . Compute the suction capacity of the wet air pump which removes both air and condensed steam in m^3 per minute, taking volumetric efficiency of the air pump as 80 percent.

If the air pump is single-acting and runs at 1 r.p.s. and piston stroke is 1.25 times the diameter of the pump, find the dimensions of the air pump.

Take $R = 0.287 \text{ kJ/kg K}$ for air.

From steam (temperature) tables, at 34°C , partial pressure of steam, $p_s = 0.05324 \text{ bar}$.

Combined absolute pressure of steam and air in the condenser,

$$p_m = 760 - 705 = 55 \text{ mm Hg} = 55 \times 0.001333 = 0.07315 \text{ bar.}$$

Hence, by Dalton's law, the partial pressure of air,

$$p_a = p_m - p_s = 0.07315 - 0.05324 = 0.01991 \text{ bar.}$$

$$\text{Mass of air leakage in the condenser per min, } m_a = \frac{12,500}{2,500 \times 60} = 0.0834 \text{ kg.}$$

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

Volume of air leakage in the condenser per minute,

$$v_a = \frac{m_a R_a T_a}{p_a} = \frac{0.0834 \times (0.287 \times 10^3) \times (34 + 273)}{(0.01991 \times 10^5)} = 3.691 \text{ m}^3$$

$$\text{Mass of steam condensed per min.} = \frac{12,500}{60} \text{ kg}$$

As density of water is $1,000 \text{ kg/m}^3$, volume of condensate/min.

$$= \frac{12,500}{60 \times 1,000} = 0.2083 \text{ m}^3$$

Volume of mixture actually discharged by wet air pump/min.

$$= 3.691 + 0.2083 = 3.8993 \text{ m}^3$$

$$\text{Suction (or swept) capacity of wet air pump per min.} = \frac{3.8993}{0.8} = 4.874 \text{ m}^3$$

$$\text{Suction capacity of wet air pump per stroke} = \frac{4.874 \times 10^6}{1 \times 60} = 81,223 \text{ cm}^3$$

$$\therefore \frac{\pi}{4} d^2 \times 1.25 d = 81,233$$

$$d = \sqrt[3]{\frac{4 \times 81,233}{1.25 \times 3.14}} = \sqrt[3]{82,785} = 43.583 \text{ cm (diameter of air pump),}$$

and air pump piston stroke, $l = 43.583 \times 1.25 = 54.48 \text{ cm}$

Problem-13 : A jet condenser has to condense 3,800 kg of steam per hour. The volume of the injection water used is 290 m³/hour and its initial temperature is 25°C. The volume of air at atmospheric pressure dissolved in the injection water is 5% of the volume of water. The air which comes in with the steam and that which leaks into the condenser amounts to 5 kg per 10,000 kg of steam. The vacuum in the air pump suction is 675 mm of Hg (barometer 760 mm of Hg) and the temperature of condensate is 36.15°C.

Determine the suction capacity of the wet air pump in m³ minute to remove the air and water from the condenser. The volumetric efficiency of the air pump is 80% and the mass of one m³ of air at 0°C and atmospheric pressure of 1.01325 bar (N.T.P) is 1.293 kg.

$$\text{Total pressure (of steam and air) in the condenser, } p_m = \frac{760 - 675}{750} = 0.1133 \text{ bar}$$

From steam (pressure) tables, partial pressure of steam, p_s at 36.16°C is 0.06 bar.

Hence, from Dalton's law of partial pressures, the partial pressure of air,

$$p_a = p_m - p_s = 0.1133 - 0.06 = 0.0533 \text{ bar}$$

$$\left[\begin{array}{l} \text{volume of air dissolved in the injection} \\ \text{water at atmospheric pressure and } 25^\circ\text{C} \end{array} \right] = 290 \times \frac{5}{100} = 14.5 \text{ m}^3/\text{hr.}$$

Volume of this air at N.T.P. (0°C and atmospheric pressure 1.01325 bar) dissolved

$$\text{in the injection water is given by } \frac{1.01325 \times 14.5}{(25 + 273)} = \frac{1.01325 \times v_{a1}}{(0 + 273)}$$

$$v_{a1} = 13.28 \text{ m}^3 \text{ per hr.}$$

Volume of air dissolved in injection water at N.T.P., $v_{a1} = 13.28 \text{ m}^3/\text{hr.}$

$$\text{Mass of leakage air} = 3,800 \times \frac{5}{10,000} = 1.9 \text{ kg/hr.}$$

$$\therefore \text{Volume of this leakage air at N.T.P., } V_{a2} = \frac{1.9}{1.293} = 1.47 \text{ m}^3/\text{hr.}$$

$$\therefore \text{Total volume of air per minute at N.T.P.} = \frac{v_{a1} + v_{a2}}{60} = \frac{13.28 + 1.47}{60} = 0.2458 \text{ m}^3$$

This volume (0.2458 m³) of air at 1.01325 bar and 0°C (N.T.P.).

\therefore Volume of this air (v_a) at 0.0533 bar and 36.16°C is given by

$$\frac{1.01325 \times 0.2458}{0 + 273} = \frac{0.0533 \times v_a}{36.16 + 273}$$

$$v_a = 5.292 \text{ m}^3/\text{min.}$$

$$\text{Volume of condensate per min.} = \frac{3,800}{60 \times 1,000} = 0.0633 \text{ m}^3.$$

$$\text{Volume of injection water per min.} = \frac{290}{60} = 4.833 \text{ m}^3$$

$$\therefore \text{Volume of mixture (air, condensate and injection water) actually discharge per min.} \\ = 5.292 + 0.0633 + 4.833 = 10.1883 \text{ m}^3.$$

$$\therefore \text{Suction capacity of the wet air pump} = \frac{10.1883}{0.8} = 12.735 \text{ m}^3/\text{min}$$

It may be noted that the volume of condensate is negligible in comparison with the volume of injection (cooling) water and air.

Problem-14 : A condenser is to deal with steam of dryness fraction 0.95 and temperature 37.63°C, at the rate of 5,000 kg/hr. If the estimated air leakage is 5 kg/hr., determine : (i) the water flow lost from the feed circuit in kg per hr., (ii) the air pump capacity in m³ per hr., and (iii) the additional heat required to be supplied in the boiler in kg/hr. as a result of undercooling for condensate temperatures of 36.16°C, 32.88°C, 30.62°C and 28.08°C. Take $R = 0.287 \text{ kJ/kg K}$ for air.

The total absolute pressure at entry to the condenser is equal to the sum of partial pressure of air and saturation pressure of steam entering the condenser. This total pressure is assumed as constant throughout the condenser, since the velocity of steam flow is small.

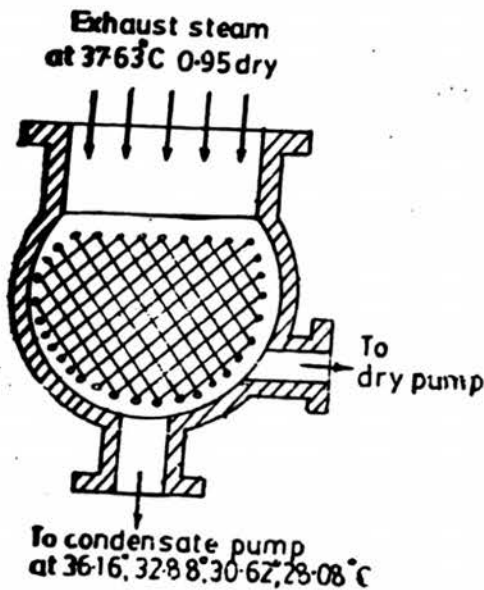


Fig. 1-12.

Referring to fig. 1-12,

At 37.63°C, partial pressure of steam,

$p_s = 0.065 \text{ bar}$, and specific volume of dry saturated steam at 37.63°C, $v_s = 22.014 \text{ m}^3/\text{kg}$ (from steam tables). The total volume of steam entering the condenser/hr.,

$$v = x \times v_s \times 5,000$$

$$= 0.95 \times 22.014 \times 5,000 = 1,04,600 \text{ m}^3/\text{hr.}$$

Air associated with this steam at 37.63°C is 5 kg. By Dalton's law, the associated air volume is equal to volume of steam.

Hence, air volume, $v_a = 1,04,600 \text{ m}^3/\text{hr.}$

Now, partial pressure of air, $p_a = \frac{m_a R_a T_a}{v_a}$

$$= \frac{5 \times (0.287 \times 10^3) \times (37.63 + 273)}{1,04,600 \times 10^5}$$

$$= 0.0000422 \text{ bar}$$

Total pressure in the condenser, $p_m = p_s + p_a = 0.065 + 0.0000422 = 0.06504 \text{ bar}$

This pressure (p_m) is constant throughout the condenser.

When condensate temperature is 36.16°C :

At 36.16°C, partial pressure of steam and specific volume of steam (from steam pressure tables) are : $p_s = 0.06 \text{ bar}$ and $v_s = 23.739 \text{ m}^3/\text{kg}$ respectively.

Hence, partial pressure of air, $p_a = p_m - p_s = 0.06504 - 0.06 = 0.00504 \text{ bar}$

The amount of air, m_a to be extracted is 5 kg/hr.

Using ideal gas equation for air,

$$\text{Volume of air per hr., } v_a = \frac{m_a R_a T_a}{p_a} = \frac{5 \times (0.287 \times 10^3) \times (36.16 + 273)}{0.00504 \times 10^5} = 880.25 \text{ m}^3/\text{hr.}$$

Hence, air pump capacity per hour = 880.25 m³.

By Dalton's's law, volume of steam per hour is also 880.25 m³.

Specific volume of dry saturated steam, v_s at 36.16°C = 23.739m³/kg.

The amount of steam flow lost with this air = $\frac{880 \cdot 25}{V_s}$ kg/hr.

∴ Amount of steam flow lost from feed circuit, $m_s = \frac{880 \cdot 25}{23 \cdot 739} = 37 \cdot 1$ kg per hr.

∴ The ratio, $\frac{m_s}{m_a} = \frac{37 \cdot 1}{5} = 7 \cdot 42$

The steam is entering the condenser at 37·63°C. The condensate temperature is 36·16°. Therefore condensate is cooled below saturation temperature, i.e. 37·63°C. The cooling of condensate below saturation temperature is known as undercooling. As a result of this, additional heat to be supplied in the boiler will increase.

At 36·16°C, the additional amount of heat supplied.

$$= 5,000 \times (37 \cdot 63 - 36 \cdot 16) \times 4 \cdot 187 = 30,800 \text{ kJ/hr.}$$

Similarly, calculations can be made for the condensate temperatures of 32·88°, 30·62°C and 28·08°C and results be tabulated as under :

Condensate temp., °C	36·16	32·88	30·62	28·08
Air pump capacity, m ³ /hr	880·25	291·85	207·08	159·78
Water flow lost from the circuit, kg/hr.	37·1	10·35	6·51	4·37
m_s/m_a	7·42	2·07	1·302	0·874
Additional heat required to be supplied in boiler, kJ/hr.	30,800	99,400	1,46,800	2,00,000

The following *effects of the reduction in condensate temperature* can be derived from the above table :

- Water flow lost from the feed circuit decreases with decrease in condensate temperature,
- Air pump capacity decreases with decrease in condensate temperature. This reduces the power required for driving air pump,
- Additional heat is supplied in the boiler, as a result of reduced temperature of feed water. This decreases the efficiency of the cycle.

Evidently a compromise must be made between the conflicting requirements of lesser loss of feed water, low capacity of air pump, and high condensate temperature, i. e. low capacity of air pump without undercooling. This is made possible by using down flow surface condenser as shown in fig. 1-6. Further improvement in performance can be obtained by using central flow surface condenser as shown in fig. 1-7. In this type of condenser, condensate temperature approaches that of the steam at inlet.

Problem-15 : A steam condenser fitted with separate air and condensate pumps, has portion of tubes near the air pump suction screened off and the condenser tubes at this point contain the coldest water. Explain the object of this arrangement.

In such a condenser, the temperature of entering steam is 37·63°C, of the condensate is 36·16°C, and of the air entering the pump is 35·58°C. If the quantity of air entering the condenser is 2·25 kg/hour, calculate the volume of air dealt with by the air pump in m³/hour. Compare this with the amount of air to be handled using a combined air and condensate pump. Assume constant vacuum throughout the condenser, and condensate temperature to be unaltered. Take $R = 0 \cdot 287$ kJ/kg K for air.

Referring to fig. 1-13, from steam (Pressure) tables, at 37·63°C, partial pressure of steam, $p_s = 0 \cdot 065$ bar.

The partial pressure of air at entry is very small and can be neglected. Hence, total pressure in the condenser (p_m) may be taken as 0·065 bar.

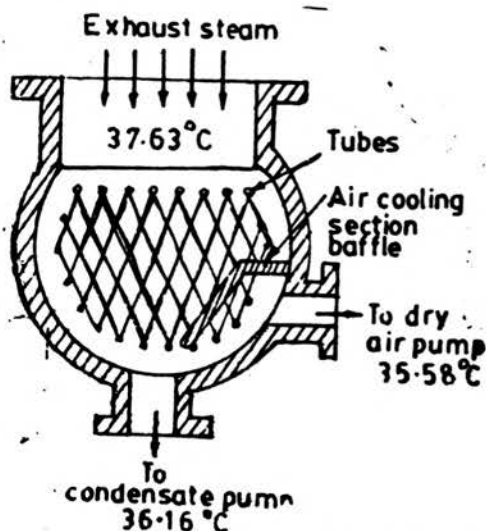


Fig. 1-13. Surface condenser fitted with air cooling section baffle.

At air pump suction temperature is 35.58°C.

At 35.58°C, partial pressure of steam, $p_s = 0.055$ bar [from steam (pressure) tables].

∴ From Dalton's law of partial pressures, partial pressure of air,

$$p_a = p_m - p_s = 0.065 - 0.055 = 0.01 \text{ bar.}$$

Applying the characteristic equation for air,

$$p_a v_a = m_a R_a T_a,$$

Volume of air at air pump suction to be dealt with by the air pump,

$$v_a = \frac{m_a R_a T_a}{p_a} = \frac{2.25 \times (0.287 \times 10^3) (35.58 + 273)}{0.01 \times 10^5} = 199.266 \text{ m}^3/\text{hour.}$$

When the condenser is not screened

At condensate extraction pump, temperature is 36.16°C.

At 36.16°C, partial pressure of steam, $p_s = 0.06$ bar [from steam (pressure), tables].

∴ Partial pressure of air (if cooling had not been provided),

$$p_a = p_m - p_s = 0.065 - 0.06 = 0.005 \text{ bar.}$$

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a,$

Volume of air to be dealt with by the air pump,

$$v_a = \frac{m_a R_a T_a}{p_a} = \frac{2.25 \times (0.287 \times 10^3) (36.16 + 273)}{0.005 \times 10^5} = 399.28 \text{ m}^3/\text{hour}$$

Thus the capacity of air pump is reduced to about half, by screening a section of tubes (that is, the air is cooled) near the air pump suction.

Problem-16 : Explain the benefit of fitting an air cooling section to the steam condenser.

In a surface condenser, a section of the tubes near by pump is screened off from steam so that air is cooled to a temperature below that of the condensate, separate extraction pumps being provided to deal with air and condensate respectively. The steam condensed per hour is 4,500 kg and the air leakage is 4 kg per hour. The temperature of the steam entering the condenser is 32.88°C, temperature of condensate at entrance to the air cooler is 30.62°C, and the temperature at the air pump suction is 26.19°C.

Assuming a constant vacuum throughout the condenser, find : (a) the volume of air in m^3 to be dealt with by the air pump per hour, (b) the mass of the steam condensed in the air cooler per minute, and (c) the percentage reduction in the air pump capacity following the cooling of the air. Take $R = 0.287 \text{ kJ/kg K}$ for air.

(a) From steam (pressure) tables, at 32.88°C, partial pressure of steam, $p_s = 0.05$ bar and specific volume of steam = 28.192 m^3/kg .

∴ Total volume of dry steam per hour = 28.192 × 4,500 = 1,26,864 m^3 , and from Dalton's law, this volume is also the volume of 4 kg of air per hour.

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a,$ partial pressure of air,

$$p_a = \frac{m_a R_a T_a}{v_a} = \frac{4 \times (0.287 \times 10^3) (32.88 + 273)}{10^5 \times 1,26,864} = 0.0000276 \text{ bar, which is negligible.}$$

Hence the total pressure, p_m in the condenser may be taken as 0.05 bar.

At air pump suction, temperature is 26.19°C.

Partial pressure of steam at 26.19°C, $p_s = 0.034$ bar (from steam pressure tables).

∴ From Dalton's law of partial pressure, partial pressure of air,

$$p_a = p_m - p_s = 0.05 - 0.034 = 0.016 \text{ bar.}$$

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

Volume of air to be dealt with by the air pump,

$$v_a = \frac{m_a R_a T_a}{p_a} = \frac{4 \times (0.287 \times 10^3) (26.19 + 273)}{0.016 \times 10^5} = 214.67 \text{ m}^3/\text{hour}$$

(b) According to Dalton's law, 214.67 m³ is also the volume of steam. Specific volume of steam at 26.19°C (0.034 bar) = 40.572 m³/kg (from steam tables).

Hence, mass of steam mixed or removed with the air per minute = $\frac{214.67}{60 \times 40.572} = 0.0882 \text{ kg}$

When condenser is not screened.

At entrance to air cooler, condensate temperature is 30.62°C.

Partial pressure of steam at 30.62°C, $p_s = 0.044$ bar [from steam (pressure) tables].

∴ Partial pressure of air (if cooling had not been provided)

$$p_a = p_m - p_s = 0.05 - 0.044 = 0.006 \text{ bar}$$

Applying the characteristic equation for air, $p_a v_a = m_a R_a T_a$,

Volume of air to be dealt with by the air pump,

$$v_a = \frac{m_a R_a T_a}{p_a} = \frac{4 \times (0.287 \times 10^3) \times (30.62 + 273)}{(0.006 \times 10^5)} = 580.92 \text{ m}^3/\text{hr.}$$

And from Dalton's law, this volume (580.92 m³ per hr.) is also the volume of steam per hour.

Specific volume of steam at 30.62°C (0.044 bar) = 31.806 m³/kg (from steam tables)

∴ Mass of steam associated with air per minute = $\frac{580.92}{60 \times 31.806} = 0.3044 \text{ kg.}$

∴ Mass of steam condensed in the air cooler per minute = 0.3044 - 0.0882 = 0.2162 kg.

(c) Percentage reduction in the air pump capacity due to cooling of air = $\frac{580.92 - 214.67}{580.92} = 0.6305$ or **63.05%**

The air cooling section reduces the required capacity of the air pump and the mass of water vapour removed by the air pump.

The benefit of fitting an air cooling section to the condenser is explained on page 7.

1.14 Cooling Water Supply

In marine practice and land practice adjacent to sea, the sea provides an ideal source of cooling water. In land practice, a river or canal may provide an ample supply of cooling water, for which reason, large power stations are built on the banks of rivers.

Where the supply of condensing water is limited, as when supplied by wells or brought from a water supply undertaking, it may be cooled and used again. This may be done in various ways. Where the adjacent land is cheap, a large open *water cooling spray pond* can be installed, into which warm condensing water from the surface condenser is

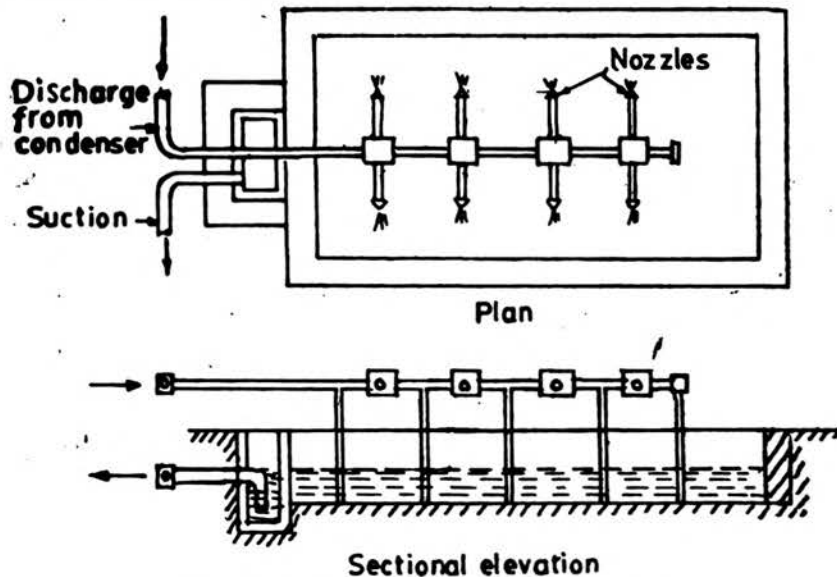


Fig. 1-14. Cooling pond with spray nozzles.

led at one end and withdrawn from the other end after being cooled. The cooling effect is mainly due to evaporation from the surface of water, and may be increased by a system of piping and spray nozzles over the cooling pond, from which the warm water is allowed to issue out as a fine spray (fig. 1-14). The disadvantage of a spray pond, however, is that a large amount of water may be lost in windy weather.

The most compact arrangement is *water cooling tower* which can be of the open or closed type. In the closed or chimney type of cooler, which is commonly used, the troughs and hurdles are fixed within a casing made of timber, steel or ferro-concrete. The casing is provided with large air openings near the base. Air enters at these openings and rises upward by natural draught. Most of the cooling towers are provided with chimneys, the purpose of which is to create an upward current of air, although in restricted places, forced fan draught is some times employed. Fan draught is also used in tropics, where the air and water temperature difference is so small, that a reversal of air current might occur if natural draught water were depended upon.

Wooden and steel towers are usually of rectangular section, some 25 to 30 metres high, while largest reinforced concrete towers are built 50 metres high. These are of circular cross-section and can be made of double hyperbolic (longitudinal section) in order to obtain a venturi effect. This produces air velocity increasing to maximum at the smallest cross-section of the tower where the warm water is introduced.

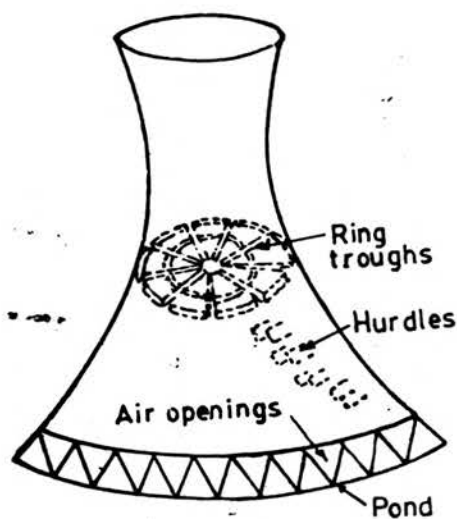


Fig. 1-15. Ferro-concrete hyperbolic cooling tower.

In the ferro-concrete hyperbolic cooling tower, illustrated in fig. 1-15, the hot condensing water is pumped to troughs which are placed about 10 metres above the ground level. Nozzles situated in the bottom of the troughs, project the water on to spray cups, which thin out jets of water into sheets. These sheets of water break up under the action of gravity and hurdles, and fall into a pond or tank situated below the tower. The falling water gives up its heat to the rising column of air.

The cooling tower is built over a tank into which cooling water collects. The cooling water is pumped from the tank and returned to the condenser. Make up water must be supplied with all these cooling arrangements and in the case of cooling towers, some 3 to 5 percent of water may have to be replaced.

Of the methods, the cooling pond is much cheaper, but the cooling results obtained from the cooling towers, are better.

Tutorial - 1

1. Delete the phrase which is not applicable from the following statements :

- (i) The primary object of a condenser is to maintain a very low/high back pressure on the exhaust side of the steam engine or steam turbine.
- (ii) The lowest possible exhaust pressure in case of a condensing steam turbine plant is less/higher than that of a condensing steam engine.
- (iii) A jet condenser is a much simpler and less costly/complicated and more costly piece of apparatus as compared to a surface condenser.
- (iv) In case of a marine steam power plant a jet condenser / a surface condenser is used.
- (v) Locomotive steam engines are generally condensing / non-condensing engines.
- (vi) In a surface condenser, the exhaust steam and the cooling water do not/do come in direct contact.
- (vii) The work output of steam engine will increase if its back pressure is increased/decreased.

[(i) high, (ii) higher, (iii) complicated and more costly, (iv) a jet condenser, (v) condensing, (vi) do, (vii) increased]

2. Fill in the blanks to complete the following statements :

- (i) A vessel having vacuum of 60 cm of Hg will have absolute pressure equal to _____ mm of Hg when barometer reads 750 mm of Hg.
- (ii) A pump which extracts both air and condensate from the condenser is known as _____ pump.
- (iii) A pump which extracts moist air only from the condenser is known as _____ pump.
- (iv) The vacuum gauge reading in a condenser is 713 mm of Hg when the barometer reads 758 mm of Hg. The corrected vacuum gauge reading to standard barometer of 760 mm in this case is _____ mm of Hg.
- (v) By Dalton's law, air and steam occupy the same _____ at their partial pressures and have the same temperature.
- (vi) The secondary object of a condenser is to supply to the _____ pure and hot feed water.

[(i) 150, (ii) wet air, (iii) dry air, (iv) 715, (v) volume, (vi) boiler]

3. Indicate the correct answer out of the suggested groups of phrases :

- (i) Air from a condenser is extracted from
 - (a) the coldest zone in the condenser
 - (b) the hottest zone in the condenser
 - (c) any where in the condenser
 - (d) the centre of the condenser.
- (ii) Air from a condenser is extracted from the coldest zone because
 - (a) the amount of air to be handled by the air pump will be low
 - (b) the air removed from the coldest zone will contain least water vapour
 - (c) less work is required to operate the air pump
 - (d) air pump of lower quality can be used.
- (iii) Vacuum efficiency of condenser would be 100% if
 - (a) there were no air present in the condenser
 - (b) there were maximum air present in the condenser
 - (c) the temperature of condensate falls below saturation temperature
 - (d) the condenser is of surface type.
- (iv) In surface condensers provided on steam turbines, the amount of air leakage should not exceed
 - (a) 10 kg/10,000 kg of steam condensed
 - (b) 5 kg/10,000 kg of steam condensed
 - (c) 10 kg/1,000 kg of steam condensed
 - (d) 5 kg/1,000 kg of steam condensed.
- (v) In surface condensers provided on steam engines, the amount of air leakage is about
 - (a) 5 kg/10,000 kg of steam condensed
 - (b) 10 kg/10,000 kg of steam condensed
 - (c) 15 kg/10,000 kg of steam condensed
 - (d) 20 kg/10,000 kg of steam condensed.

[(i) a, (ii) b, (iii) a, (iv) b, (v) c]

4. What is the function of a condenser in a modern steam condensing power plant?

The vacuum in the condenser is 716 mm of Hg when the barometer reads 748 mm. In another case the vacuum in the condenser is 705 mm of Hg when the barometer reads 754 mm. Correct these vacuum gauge readings to a standard barometer of 760 mm.

[728 mm; 711 mm]

5. State the different types of steam condensers. Sketch and described the working of any one of them.

The following particulars relate to a test of the surface condenser of a steam turbine :

Absolute pressure of the exhaust steam entering the condenser, 0.06 bar; temperature of condensate, 32°C; temperature of cooling water at inlet and outlet, 15°C and 30°C; respectively; mass of cooling water per kg of steam, 32 kg. Assuming that all the heat lost by the exhaust steam is taken up by the cooling water, determine the dryness fraction of the steam as it enters the condenser.

[0.825]

6. Taking the data of the preceding problem except the final temperature of the cooling water is to be taken to apply to a jet condenser, calculate the mass of injection (cooling) water required per kg of exhaust steam.

[32.15 kg]

7. Compare the merits and demerits of surface condensers over jet condensers.

Exhaust steam having a dryness fraction of 0.85, enters a surface condenser at a pressure of 0.1 bar and is condensed to water at 38°C. The cooling water enters at 15°C and leaves at 30°C. Calculate the mass of cooling water required per kg of exhaust steam.

[32.9 kg]

8. (a) Enumerate the sources of air leakage in a steam condenser. Briefly state the effects of air leakage into a condenser.

(b) Explain Dalton's law of partial pressures as applied to the condenser of a steam plant.

9. The temperature in a condenser is 37°C (corresponding saturation pressure is 0.06281 bar) and the vacuum is 700 mm of Hg (barometer 755.2 mm Hg). Correct the vacuum gauge reading to a standard barometer of 760 mm of Hg and hence determine : (i) the partial pressures of steam and air, and (ii) the mass of air associated with one kg of steam. Take $R = 0.287$ kJ/kg K for air.

[704.8 mm Hg; (i) 0.06281 bar, 0.010606 bar, (ii) 0.27104 kg]

10. (a) What do you understand by 'partial pressure' as applied to the condenser of a steam plant and what is the law connecting them ?

(b) A closed vessel of 0.35 m³ capacity contains saturated water vapour and air at a temperature of 42.67°C and pressure of 0.1336 bar. Due to further air leakage into the vessel, the pressure rises to 0.253 bar and temperature falls to 36.16°C. Calculate the mass of air which has leaked in. Take $R = 0.287$ kJ/kg K for air.

[0.0573 kg]

11. State the law of partial pressures and show how it applies to the condenser of a steam plant.

The following observations were made on a condensing plant in which the temperature of condensation was measured directly by thermometers : the recorded vacuum was 710 mm of Hg (barometer 765 mm), mean temperature of condensation 35.58°C, temperature of hot-well 28°C, mass of condensate per hour 2,000 kg, mass of cooling water per hour 64,000 kg, inlet temperature of cooling water 14.5°C, outlet temperature of cooling water 30°C.

Find : (a) the state of steam entering the condenser, and (b) the mass of air present per m³ of condenser volume.

[84.52% dry; (b) 0.0207 kg]

12. In a surface condenser, the following data were obtained :

Temperature of exhaust steam entering the condenser, 42.67°C; temperature of condensate leaving the condenser; 35°C; inlet temperature of condenser cooling water 17.5°C; outlet temperature of condenser cooling water, 31°C; quantity of condenser cooling water per hour, 46,250 kg; quantity of condensate per hour, 1,190 kg. Calculate the dryness fraction of exhaust steam entering the condenser.

[0.903]

13. What do you understand by the term 'Vacuum Efficiency' of a condensing plant ?

The vacuum at the steam inlet to a condenser is found to be 710 mm of Hg (barometer 760 mm) and the temperature of steam in the condenser is 36.16°C. Find the vacuum efficiency. [99.3 %]

14. Define the term 'Vacuum efficiency' of condenser. On what factors does this efficiency depend ?

Steam enters a condenser at a temperature of 35.58°C and the barometer standing at 749 mm, a

vacuum of 703 mm of Hg was produced. Determine the vacuum efficiency.

[99-34%]

15. (a) Explain fully the importance of a low vacuum in steam turbine practice.
 (b) In a particular steam power plant, air is believed to leak into the condenser. To check whether this is so, the plant is run until the conditions are steady and then the steam supply from the engine is shut off; simultaneously the air and condensate extraction pumps are closed down, so that the condenser is isolated. At shut down, the temperature and vacuum are observed to be 39°C and 702 mm of mercury respectively. After five minutes these values were 26.19°C and 483 mm of mercury. The barometer reads 757 mm of mercury. The effective volume of the condenser is 0.57 m³.

Determine from the data, the mass of air leakage into the condenser during the observed period. Assume $R = 0.287$ kJ/kg K for air.

[0-2178 kg]

16. Define the term "condenser efficiency" of a steam condensing plant.
 The following data were obtained from a test of a surface condenser :
- | | |
|---|-----------------|
| Inlet temperature of circulating water | ... 21°C |
| Outlet temperature of circulating water | ... 35°C |
| Vacuum in the condenser | ... 707.5 mm Hg |
| Barometer | ... 760 mm Hg |
- Determine the efficiency of the condenser.

[77-78%]

17. Enumerate the sources of air leakage in a condenser and describe briefly with suitable sketches any one method you know for extracting air from a condenser.

The temperature of the steam entering a surface condenser is 45.81°C and the temperature at the air pump suction is 42.67°C. The barometer reading is 754 mm of mercury. Find : (a) the condenser vacuum, and (b) the vapour pressure and the air pressure near to the air pump suction.

If the effective capacity of the air pump on the suction stroke is 11 m³ per minute find : (i) the mass of air entering the condenser per minute, and (ii) the mass of steam carried over per minute in the air discharged from the air pump.

Assume that the air pump deals with moist air only and not with the condensate.

[(a) 679 mm Hg; (b) 0.085 bar; 0.015 bar; (i) 0.18033 kg/min; (ii) 0.6433 kg/min.]

18. (a) Describe in detail the various methods used in steam power condensing plants to obtain the highest possible vacuum.
 (b) Discuss the factors which may influence the efficiency of a condensing plant.
 (c) The vacuum at the air extraction pump in a condenser is 706 mm of mercury (barometer 760 mm) and the temperature is 37.63°C. The air leakage into the condenser is 5 kg per 10,000 kg of steam. Determine : (i) the volume of air to be dealt with by the dry air pump per kg of steam entering the condenser, and (ii) the mass of water vapour associated with this air.

[(i) 0.0637 m³, (ii) 0.0029 kg]

19. (a) What is the function of an air pump in a steam power condensing plant ?
 (b) A surface condenser deals with 5,000 kg of steam per hour. The air leakage into the condenser is 0.5 kg per 1,000 kg of steam. The vacuum in the air pump suction is 670 mm of mercury (barometer 755 mm of mercury) and the temperature is 34°C (corresponding saturation pressure from steam tables is 0.05324 bar). Find the volumetric efficiency of a single-acting air pump required to remove the condensate and air having cylinder diameter of 24 cm and stroke of 40 cm. The speed of the air pump is 1 r.p.s.

[64.21%]

20. What factor contribute to loss of efficiency in a surface condenser ?

The air pump, for the removal of the air and condensed steam, for a surface condenser is single-acting and has a diameter of 40 cm and a stroke of 60 mm. The air pump speed is 60 r.p.m. The mass of the condensed steam per minute is 75 kg. The pressure in the air pump suction is 0.06 bar and the temperature is 32.88°C (corresponding saturation pressure is 0.05 bar). Taking the volumetric efficiency of the air pump as 80 per cent, calculate the mass of air passing through the air pump per 10,000 kg of steam condensed.

[5.406 kg]

21. Make a neat diagrammatic sketch of a barometric jet condenser and explain its working.

A condenser of this type deals with 400 kg of steam per hour, maintaining a vacuum of 625.75 mm of mercury; the barometer standing at 754 mm. The entering steam has a dryness fraction of 0.95 and the air leakage amounts to 0.3 kg per 100 kg of steam. The cooling water has an initial temperature of

15°C and the mixture of water and condensate leaves at 37°C. The temperature at the air pump suction is 42.67°C.

Determine : (i) the mass of water vapour removed along with the air per hour, and (ii) the mass of cooling water required per hour.

[(i) 0.7069 kg; (ii) 10,005 kg]

22. Describe with a neat sketch the working of a two-flow surface steam condenser.

Exhaust steam having a dryness fraction of 0.8 enters a surface steam condenser where the vacuum is 695.25 mm of mercury (barometer 759 mm) and is condensed to water at 37.63°C. The temperature of hot-well is 32.9°C. The circulating water enters at 15.5°C and leaves at 30°C. Determine : (a) the mass of the air extracted per kg of steam, (b) the mass of circulating water required per kg of steam, and (c) the vacuum efficiency.

[(a) 0.494 kg; (b) 32.27 kg; (c) 97.9%]

23. Describe briefly, with sketches, some form of a surface condenser.

Steam consumption of a turbine installation is 40,000 kg per hour, the quantity of air leaking in is 24 kg per hour, and the swept volume of air pump is 17.4 m³ per min.

Find the volumetric efficiency of the dry air pump when the vacuum in the air pump suction is 725 mm of Hg (barometer 770 mm) and temperature is 26.19°C.

Take $R = 0.287$ kJ/kg for air.

[75.92%]

24. Make a diagrammatic sketch of a counter-flow low level jet condenser and explain its working.

25. Describe briefly, with the aid of a sketch, any one type of condenser air pump.

26. Make a neat diagrammatic sketch of a two-flow surface condenser with an air pump and explain its working.

27. Describe, with neat sketches, a modern surface steam condenser showing how the air is cooled before it enters the air extraction pump.

28. Describe an arrangement suitable for reducing the water vapour loss at the air extraction of a condenser.

29. What is undercooling in a surface condenser ? State its merits and demerits.

30. What do you understand by "undercooling" in a surface condenser ? Discuss its effect on the following :

- (i) power required for air extraction pump,
- (ii) make-up water required, and
- (iii) efficiency of the cycle.

31. Explain the benefit of fitting an air cooling section to the steam condenser.

In a surface condenser a section of tubes near the pump suction is screened off so that the air is cooled to a temperature below that of the condensate, separate extraction pumps being provided to deal with air and condensate respectively. The steam condensed per hour is 15,000 kg and air leakage is 12 kg per hour. The temperature of the steam entering the condenser is 32.88°C, temperature of condensate at entrance of the air cooler is 30.62°C, and the temperature at the air pump suction is 25.16°C. Assuming a constant vacuum throughout the condenser, find :

- (a) the mass of steam condensed in the cooler per hour,
- (b) the volume of air in m³ per hour to be dealt with by the air pump, when the condenser is not screened and,
- (c) The percentage reduction in the air pump capacity following the cooling of the air.

[(a) 41.522 kg, (b) 1,743 m³, (c) 67.27]

32. Describe, with a neat sketch, the operation of :

- (i) an evaporative condenser, (ii) an ejector condenser, (iii) a steam jet air ejector, (iv) Edward's air pump, and (v) a high-level jet condenser.

33. What is the function of cooling tower in a modern condensing plant ? Describe with sketches the construction and working of any one type of cooling tower.

2

COMPOUND STEAM ENGINES

2.1 Introduction

A simple steam engine may be defined as one in which each of the engine cylinder receives steam direct from the boiler, and exhausts into the atmosphere or into a condenser. In modern steam engine practice high pressure steam is used, as the use of such a steam gives greater efficiency, and the plant requires less floor space per unit power developed. But if high pressure steam is used with a large range of expansion in a single-cylinder engine, serious difficulties and disadvantages follow. To overcome the difficulties and obtain certain advantages, compound or multiple-expansion steam engines are built.

Compound engine is one in which the steam from the boiler expands to a certain extent in one cylinder and then exhausts into a larger cylinder, where the expansion may be completed. The first cylinder is called the *high-pressure* or *H.P. cylinder*, while the second is called the *low-pressure* or *L.P. cylinder*. A compound steam engine with two cylinders is called *duplex* steam engine.

The expansion of the steam may be carried out in three or even four cylinders in succession as in the case of *triple* expansion or *quadruple* expansion engines. The H.P. cylinder, in such a case, is that in which the first expansion stage is performed and the L.P. cylinder is that in which the last expansion stage is performed. The cylinders between H.P. and L.P. cylinders are known as *intermediate pressure cylinders* or *I.P. cylinders*. Compound steam engines are generally condensing engines.

The main objections to working the high pressure steam through large range of expansion in a single-cylinder are :

- ... The cylinder must have a large volume for the required amount of expansion and it must be sufficiently strong to withstand safely the maximum pressure. The working parts of the engine have to be made large enough to transmit the maximum load. Therefore, a single-cylinder engine is excessively heavy and costly in proportion to the power developed.
- ... If the high pressure steam is expanded down to the condenser pressure in one cylinder, the stroke of the piston will be very large.
- ... The large range of steam pressure (pressure difference) between the initial pressure and exhaust pressure, causes a correspondingly large range of temperature in the cylinder. This causes condensation of steam since the high pressure hot steam will come in contact with relatively cold cylinder during admission period. Condensation of steam is a source of loss of power and also causes mechanical trouble in the cylinder. The accumulation of water in the clearance space might cause excessive pressure to break the cylinder head.

2.2 Advantages

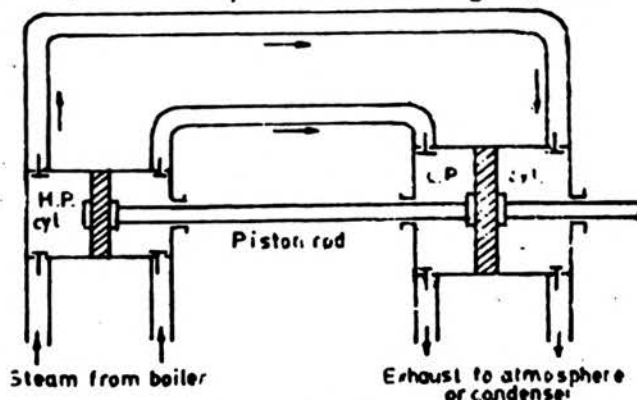
The following are the advantages of multiple-expansion (compound) steam engines :

- ... Temperature range in each cylinder is reduced, with a corresponding reduction in initial cylinder condensation and temperature stresses. The temperature range is the difference between the highest and lowest temperatures of steam within the cylinder.
- ... Re-evaporation of moisture at the end of expansion stroke in H.P. cylinder adds to the live steam supply to L.P. cylinder.
- ... If simple engine is to utilise the same expansion ratio as a compound engine, its cylinder will have to be made strong enough to withstand the high pressure steam and large enough to accommodate the large volume of low pressure steam at the end of expansion. The thickness required for cylinder walls depends on the maximum pressure to which they are subjected and the diameter of the cylinder. In compound engine the high-pressure cylinder only is subjected to maximum pressure, but its diameter is small. This results in reduction in total weight of cylinders.
- ... The uniformity of turning moment is improved due to the possible out of phase crank arrangement (different pistons are coupled to separate cranks) and also due to the fact that the pressure difference at the beginning and end of the strokes is reduced and in turn reduces the maximum load on the piston.
- ... Better mechanical balance is achieved, which allows adoption of higher speeds. The higher speed gives improved thermal economy.
- ... The forces on the working parts are reduced as the forces are distributed over more components of the engine.
- ... The leakage of steam past the pistons and valves is reduced due to the reduced pressure difference across them.
- ... The steam can be re-heated after expansion in one cylinder before entering the next cylinder which will reduce cylinder condensation.
- ... The compound steam engine may start in any position which is an advantage for locomotives and marine engines.
- ... The engine may be modified to run at reduced load, at the time of breakdown. This is an advantage for marine propulsion.

2.3 Types

Compound steam engines may be divided into two main classes, namely,

— *Tandem compound steam engine*, and



(a) Schematic diagram

Fig. 2-1. (a) Schematic diagram

— *Cross compound steam engine*.

Cross compound steam engines may be further classified as

Woolfe type and Receiver type compound steam engines.

In **Tandem Type** of compound steam engine, the cylinders are so arranged that they have a common axis, have a common piston rod and working on the same crank (fig. 2-1a). These cylinders may be

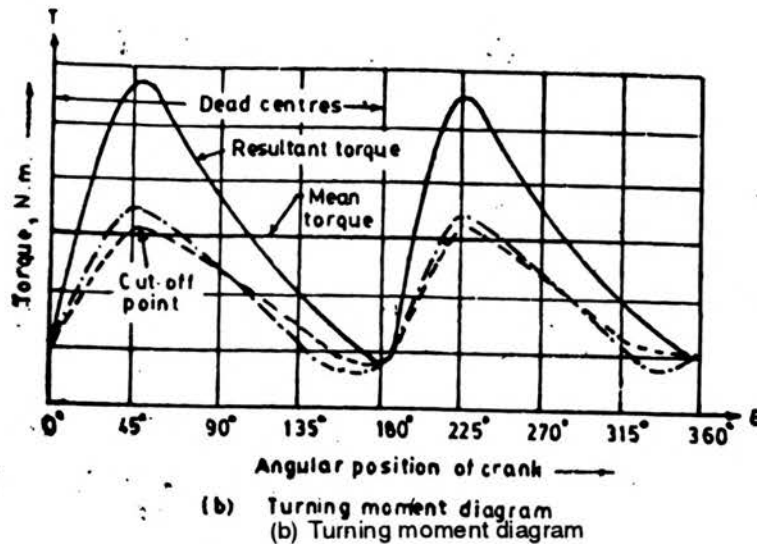


Fig. 2-1. Tandem type compound steam engine.

economy in view of the fewer components (parts) of the engine.

In Woolfe Type of compound steam engine, the cylinders are arranged side by side with their axis parallel to each other as shown in fig. 2-2. In an engine of this type with two cylinders, the cranks of the two cylinders are at 180° to each other. The piston of the H.P. and L.P. cylinders, begin and end their strokes together. The time during which the exhaust takes place in the H.P. cylinder coincides more or less with the time of admission of steam to the L.P. cylinder. The exhaust steam from the H.P. cylinder passes directly into the L.P. cylinder. The expansion is therefore, continuous during the stroke. Since the two cranks are at 180° to each other, the two cycles are in phase and this causes large variation in the turning moment on the crankshaft. This is the same disadvantage as in the case of tandem type engine.

In Receiver Type of compound steam engine, cranks of the two cylinder are placed at 90° to each other. The arrangement of cylinders is same as in the Woolfe type, As the cranks are at 90° to each other, steam from the H.P. cylinder cannot, exhaust directly into the L.P. cylinder. In order to overcome this difficulty, a container known as *receiver* is used (fig. 2-3). The H.P. cylinder exhausts steam into this receiver while the L.P.

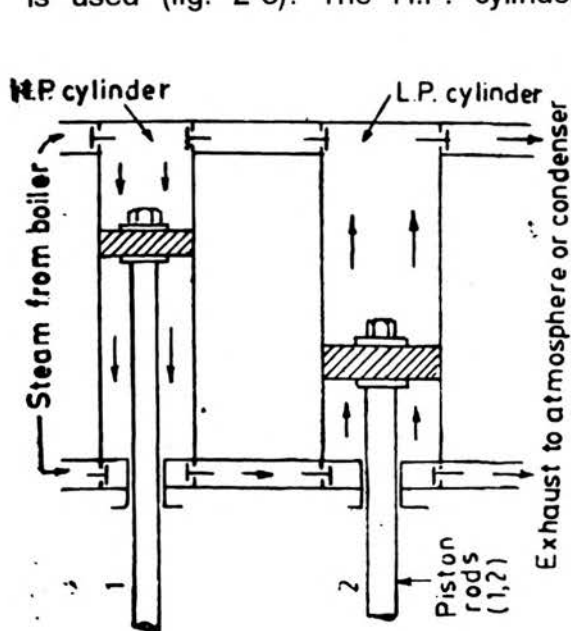
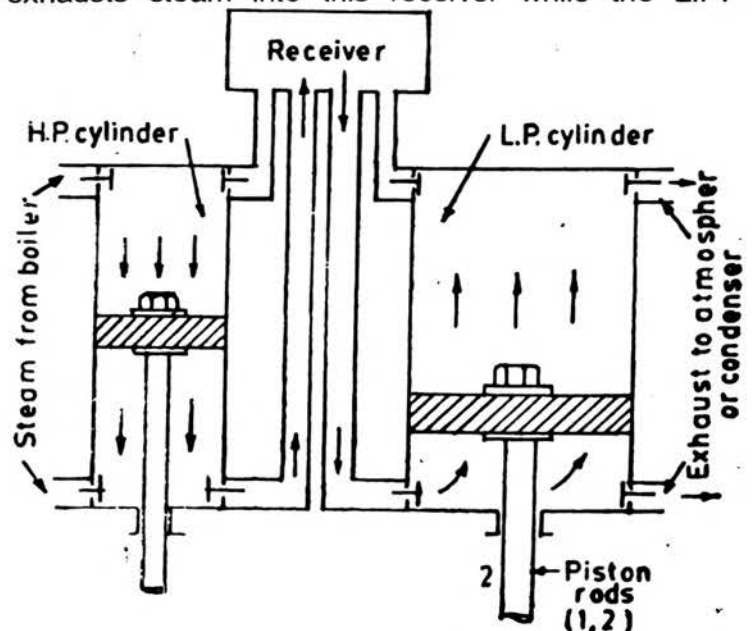
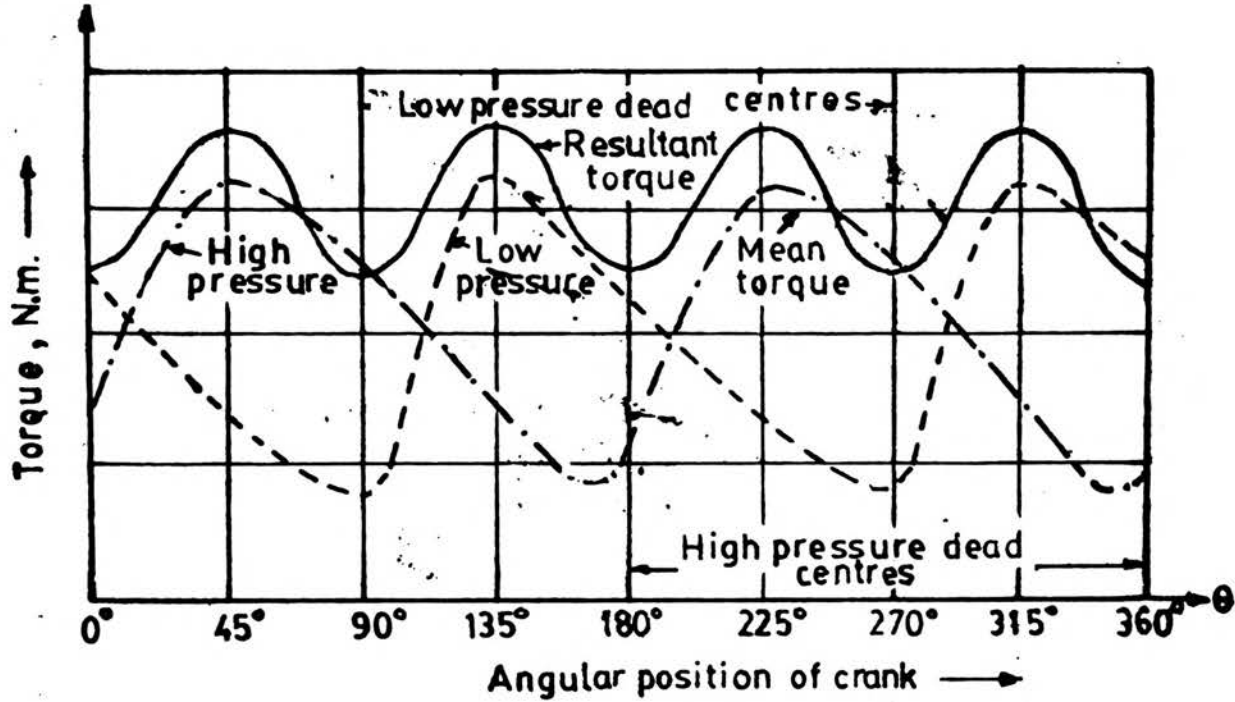


Fig. 2-2. Schematic diagram of a Woolfe type compound steam engine.

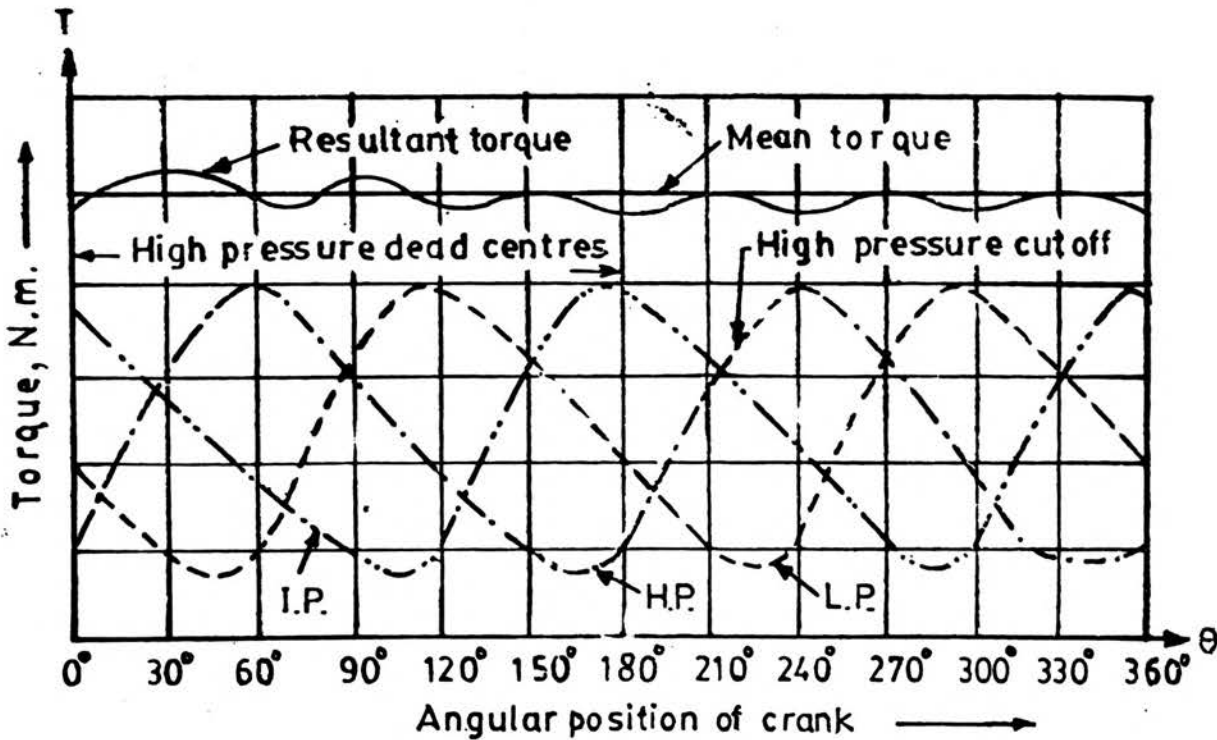
Fig. 2-3. Schematic diagram of a receiver type compound steam engine having cranks at 90° to each other.

regarded as having cranks at zero degree to each other. The exhaust from the H.P. cylinder passes directly into the L.P. cylinder, as both the pistons are at the end of their strokes at the same time. As the cycles of the two cylinders are in phase, the maximum turning moment, due to each cylinder, will act at the same time as shown in fig. 2-1b. The turning moment in this type of engine is, therefore, not uniform which is a disadvantage. Since the torque is not uniform, a larger flywheel is required. However, the tandem arrangement gives constructional

cylinder draws steam from the receiver. The receiver acts as a reservoir. The resultant turning moment (fig. 2-4a) in the receiver type compound engine is more uniform as the two cycles are out of phase by 90°, which is an advantage. With three-cylinder engine, the cranks are arranged at 120° to each other, which will result in still more uniform turning moment on the engine crankshaft as shown in fig. 2-4b.



(a) Two cylinders having cranks at 90° to each other.



(b) Three cylinders having cranks at 120° to each other.

Fig. 2-4. Turning moment diagrams of Receiver type compound steam engine.

The receiver should be large enough to keep the pressure in it fairly constant. The volume of receiver should be about 1.5 times the H.P. cylinder volume. The receiver is often jacketed to reduce condensation and the jacket is supplied with steam direct from the boiler.

The number of cylinders provided in a **Multi-cylinder compound steam engine**, will depend upon the total range of pressure through which the steam is to be expanded. Simple engines are usually non-condensing with initial pressure ranging from 6 to 7 bar. The two cylinder compound condensing engines may have initial pressures ranging from 7 to 10 bar and triple expansion engines from 10 to 15 bar. The initial pressure in quadruple expansion engine may be from 15 to 25 bar.

The L.P. release pressure for condensing engine may vary from 0.7 to 0.9 bar and for non-condensing engines may vary from 1.4 to 1.8 bar.

In very large engines, for example, when the diameter of L.P. cylinder exceeds 2.5 metres, it is customary to fit two L.P. cylinders thereby producing a four-cylinder triple expansion engine. One H.P. cylinder and two L.P. cylinders are also used on high speed engines. The famous Webb's compounded locomotive engine had the reverse of the arrangement, i.e. it had two H.P. cylinders and one L.P. cylinder.

2.4 Typical Terms

Before estimation of cylinder dimensions of a compound steam engine, following typical terms should be understood.

Cylinder volume ratio – It is the ratio of the displacement or swept volume of the low-pressure cylinder to that of the high-pressure cylinder. Where the strokes of the two cylinders are the same, the cylinder volume ratio may be taken as the square of the ratio of cylinder diameters. The cylinder volume ratio in compound steam engines varies from about 2 to 1 to about 8 to 1.

Total ratio of expansion or ratio of expansion for the whole engine – It is the ratio of the final volume of the steam in the L.P. cylinder to its volume at cut-off in the H.P. cylinder.

Free or unresisted expansion – It is the expansion of the steam in the receiver and passages between cylinders. It is measured by the mean difference between the pressures along the exhaust line of the H.P. cylinder and that along the admission line of L.P. cylinder.

Terminal drop – It is the difference between the pressure in the H.P. cylinder at release and the average receiver pressure.

2.5 Estimation of Cylinder Dimensions

The problem of estimation of cylinder dimensions presents considerable difficulty to beginners. Fig. 2-5 represents the hypothetical pressure-volume diagram for the complete expansion in H.P. cylinder in a two-cylinder, non-receiver type, compound steam engine, neglecting clearance and compression.

In the absence of condensation and other losses, L.P. cylinder can be regarded as capable of developing combined power of the H.P. and L.P. cylinders, when supplied with same mass of high pressure steam as originally supplied to

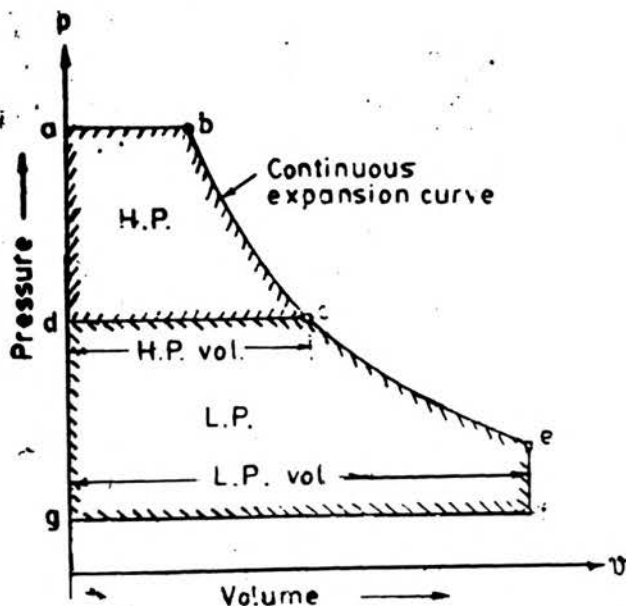


Fig. 2-5. Combined theoretical indicator diagram for non-receiver type, two cylinder compound steam engine with complete expansion in H.P. cylinder and incomplete expansion in L.P. cylinder.

the H.P. cylinder. Thus, combined diagram *a b c e f g* can be regarded as produced by continuous expansion *be* in L.P. cylinder, so that the swept volume of L.P. cylinder at cut-off (*d c*) is equal to the total swept volume of the H.P. cylinder. The average height of the combined indicator diagram is termed as the mean effective pressure referred to the low-pressure cylinder.

In practice, however, for reasons which will appear later, there is generally a pressure drop ($p_2 - p_3$) between the high-pressure cylinder release and the receiver, as shown in fig. 2-6.

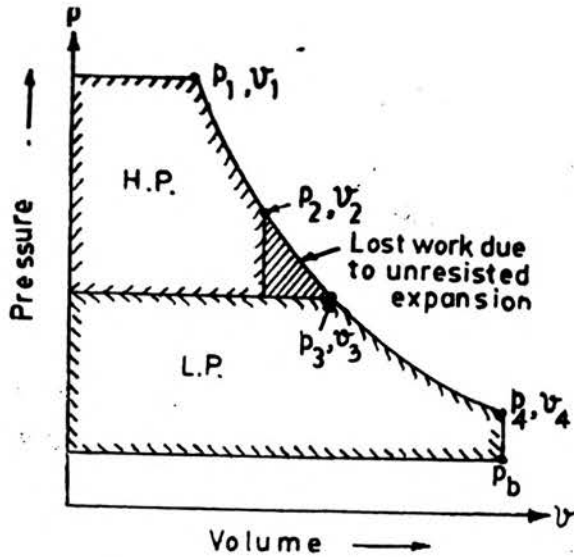


Fig. 2-6. Combined theoretical indicator diagram for two-cylinder compound steam engine fitted with receiver and with incomplete expansion in both cylinder.

The loss of work due to unresisted expansion in the receiver is shown shaded in fig 2-6. Although this drop of pressure after release of H.P. cylinder is wasteful, yet it is partly counter-balanced by the drying effect on the steam which it produces.

Let p_1 = initial steam pressure in bar,

p_2 = release pressure in H.P. cylinder in bar,

p_3 = receiver pressure in bar,

p_4 = release pressure in L.P. cylinder in bar,

p_b = condenser pressure in bar,

v_1 = volume at cut-off in H.P. cylinder in m^3 ,

v_2 = Volume of H.P. cylinder in m^3 ,

v_3 = Volume at cut-off in L.P. cylinder in m^3 ,

v_4 = Volume of L.P. cylinder in m^3 ,

l = Length of piston stroke in m,

N = Speed of the engine in r.p.s.,

d_1 = Diameter of H.P. cylinder in m, and

d_2 = Diameter of L.P. cylinder in m.

Then, neglecting clearance volume,

$$\text{H.P. cylinder volume, } v_2 = \frac{\pi}{4} (d_1)^2 \times l$$

$$\text{and L.P. cylinder volume, } v_4 = \frac{\pi}{4} (d_2)^2 \times l$$

$$\text{Cylinder volume ratio or ratio of cylinder volumes, } r = \frac{v_4}{v_2}$$

$$\text{Ratio of expansion in H.P. cylinder, } r_1 = \frac{v_2}{v_1}$$

$$\text{Ratio of expansion in L.P. cylinder, } r_2 = \frac{v_4}{v_3}$$

$$\text{Ratio of expansion for whole engine or total number of expansion, } R = \frac{v_4}{v_1}$$

As expansion is assumed hyperbolic, $p_1 v_1 = p_2 v_2 = p_3 v_3 = p_4 v_4 = \text{constant}$.

$$\text{Cylinder volume ratio, } r = \frac{v_4}{v_2} = \frac{v_4}{v_1} \times \frac{v_1}{v_2} = \left(\frac{d_2}{d_1} \right)^2$$

or = total expansion ratio (R) \times cut-off in H.P. cylinder $\left(\frac{1}{r_1}\right)$

i.e. Cylinder volume ratio, $r = \frac{v_4}{v_2} = \frac{\text{total ratio of expansion, } R}{\text{ratio of expansion in H.P. cylinder, } r_1}$

The M.E.P. referred to the L.P. cylinder may be calculated from the equation

$$\text{M.E.P.} = f \left[\frac{p_1}{R} (1 + \log_e R) - p_b \right] \text{ bar} \quad \text{if } p_1 \text{ and } p_b \text{ are in bar} \quad \dots (2.1)$$

where, f = overall diagram factor for the combined indicator diagram.

Using this M.E.P., indicated power of the double-acting engine

$$\begin{aligned} & (10^5 \times \text{M.E.P.}) \times \frac{\pi}{4} (d_2)^2 \times l \times N \times 2 \quad \dots (2.2) \\ & = \frac{\quad}{1,000} \text{ kW} \end{aligned}$$

In the design of an engine, the power, speed N , initial pressure p_1 and back pressure P_b are usually known. So in eqns. (2.1) and (2.2), with a tentative value of the diagram factor f , there are but two unknowns – the length of the piston stroke, l and diameter of L.P. cylinder, d_2 .

Now, the length of stroke is determined from the known rotational speed N , and permissible rubbing speed of the piston. As it stands today, it is not advisable to run the steam engines with piston speed in excess of 5 metres/sec.

$$\text{Average piston} = 2 l N \text{ m/sec.} \quad \dots (2.3)$$

Thus, from eqns. (2.1), (2.2) and (2.3), the L.P. cylinder diameter d_2 and stroke l can be evaluated.

In order to determine the cylinder bore (diameter), d_1 of H.P. cylinder, two conditions are desirable :

- Equal development of work by the H.P. and L.P. cylinders to give a uniform tuning moment, and
- The initial load on the piston exerted by the steam at the commencement of stroke should be the same for two cylinders.

The two conditions just mentioned cannot be simultaneously satisfied when the expansion curve is continuous, as shown in fig. 2-5. Hence, a compromise is made by having a pressure drop at release ($p_2 - p_3$) in the H.P. cylinder as shown fig. 2-6. This causes a loss of energy due to unresisted expansion in the receiver.

For equal work done in each cylinder (fig. 2-6),

$$p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 v_2 = p_3 v_3 \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b v_4 \quad \dots (2.4)$$

Dividing throughout by v_2 , we get,

$$\begin{aligned} p_1 \times \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 \times \frac{v_2}{v_2} &= p_3 \times \frac{v_3}{v_2} \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b \times \frac{v_4}{v_2} \\ &= p_3 \times \frac{v_3}{v_4} \times \frac{v_4}{v_2} \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b \times \frac{v_4}{v_2} \end{aligned} \quad \dots (2.5)$$

$$\therefore \left[\frac{p_1}{r_1} \{ 1 + \log_e (r_1) \} - p_3 \right] = r \left[\frac{p_3}{r_2} \{ 1 + \log_e (r_2) \} - p_b \right]$$

i.e. For equal work to be done in the two cylinders,

Mean effective pressure of H.P. cylinder

= Mean effective pressure of L.P. cylinder × cylinder volume ratio.

For equal initial loads on the two piston (fig. 2-6),

$$(p_1 - p_3) \times \frac{\pi}{4} (d_1)^2 = (p_3 - p_b) \times \frac{\pi}{4} (d_2)^2$$

$$\text{i.e. } p_1 - p_3 = (p_3 - p_b) \left(\frac{d_2}{d_1}\right)^2$$

$$\text{i.e. } p_1 - p_3 = (p_3 - p_b) r \quad \dots (2.6)$$

Thus, by using eqns. (2.4), (2.5) or (2.6), the value of cylinder volume ratio, r can be calculated and hence the diameter of H.P. cylinder, d_1 .

Problem-1 : A compound steam engine develops indicated power of 147 kW at 1.5 r.p.s. when taking in steam at 9.5 bar and exhausting it at 0.16 bar. Cut-off in the H.P. cylinder takes place at 0.45 stroke, and the ratio of low-pressure to high-pressure cylinder volumes is 3.2. Calculate the cylinder dimensions of the engine. Assume an overall diagram factor of 0.75 and an average piston speed of 2.25 metres per second. The strokes are equal in the two cylinders.

Estimate the fraction of stroke at which cut-off should take place in the L.P. cylinder for approximately equal initial loads on both pistons, and state the resulting receiver pressure. Neglect clearance and assume hyperbolic expansion.

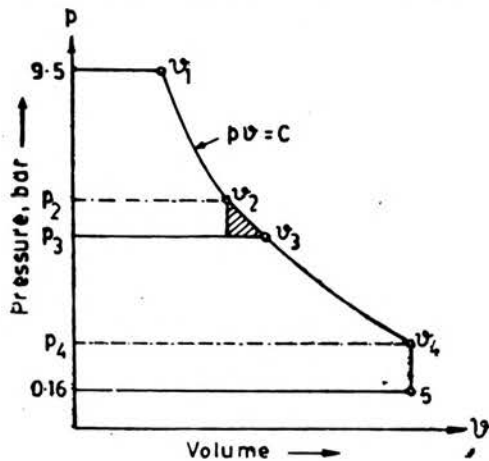


Fig. 2-7.

Referring to fig. 2-7,

$$\frac{v_1}{v_2} = 0.45, \quad \frac{v_4}{v_2} = r = 3.2$$

Total expansion ratio,

$$R = \frac{v_4}{v_1} = \frac{v_4}{v_2} \times \frac{v_2}{v_1} = r \times r_1$$

$$= 3.2 \times \frac{1}{0.45} = 7.1$$

Average piston speed = $2 \text{ l } N$

$$\text{i.e. } 2.25 = 2 \times l \times 1.5$$

∴ Piston stroke, $l = 0.75 \text{ m}$ i.e. 75 cm.

Total indicated power developed by the engine

$$= \frac{10^5 \times 2 \times f \times N}{1,000} \times \left[p_1 v_1 \left\{ 1 + \log_e \left(\frac{v_4}{v_1} \right) \right\} - p_b v_4 \right]$$

$$\text{i.e. } 147 = \frac{10^5 \times 2 \times 0.75 \times 1.5}{1,000} \left[9.5 v_1 \{ 1 + \log_e 7.1 \} - 0.16 \times 7.1 v_1 \right]$$

$$\therefore v_1 = 0.0242 \text{ m}^3.$$

$$\text{Now, } \frac{v_1}{v_2} = 0.45 \quad \therefore v_2 = \frac{v_1}{0.45} = 0.0242 \times \frac{1}{0.45} = 0.054 \text{ m}^3$$

$$\text{But, } v_2 = \frac{\pi}{4} (d_1)^2 \times l \quad \text{i.e. } 0.054 = \frac{\pi}{4} (d_1)^2 \times 0.75$$

Hence $(d_1)^2 = 0.0917 \text{ m}^2$ and $d_1 = 0.303 \text{ m}$, i.e. 30.3 cm (dia. of H.P. cylinder)

$$\text{Now, } \frac{v_4}{v_2} = 3.2 \quad \therefore v_4 = v_2 \times 3.2 = 0.054 \times 3.2 = 0.173 \text{ m}^3$$

$$\text{But, } V_4 = \frac{\pi}{4} (d_2)^2 \times l \quad \text{i.e. } 0.173 = \frac{\pi}{4} (d_2)^2 \times 0.75$$

Hence, $(d_2)^2 = 0.2937 \text{ m}^2$ and $d_2 = 0.542 \text{ m}$ i.e. **54.2 cm** (dia. of H.P. cylinder).

Considering equal initial loads on the two pistons.

$$(p_1 - p_3) \times \frac{\pi}{4} (d_1)^2 = (p_3 - p_b) \times \frac{\pi}{4} (d_2)^2$$

$$\text{i.e. } (p_1 - p_3) \frac{\pi}{4} (d_1)^2 \times l = (p_3 - p_b) \frac{\pi}{4} (d_2)^2 \times l \quad \text{i.e. } p_1 - p_3 = (p_3 - p_b) \frac{V_4}{V_2}$$

$$\text{i.e. } 9.5 - p_3 = (p_3 - 0.16) \times 3.2 \quad \therefore \text{Receiver pressure, } p_3 = 2.38 \text{ bar}$$

Considering points of cut-off in H.P. and L.P. cylinders on hyperbolic curve,

$$p_1 v_1 = p_3 v_3$$

$$\text{i.e. } 9.5 \times 0.0242 = 2.38 \times v_3 \quad \text{Hence, } v_3 = 0.0966 \text{ m}^3.$$

$$\text{Cut-off in L.P. cylinder, } \frac{v_3}{v_4} = \frac{0.0966}{0.173} = \mathbf{0.56 \text{ of stroke.}}$$

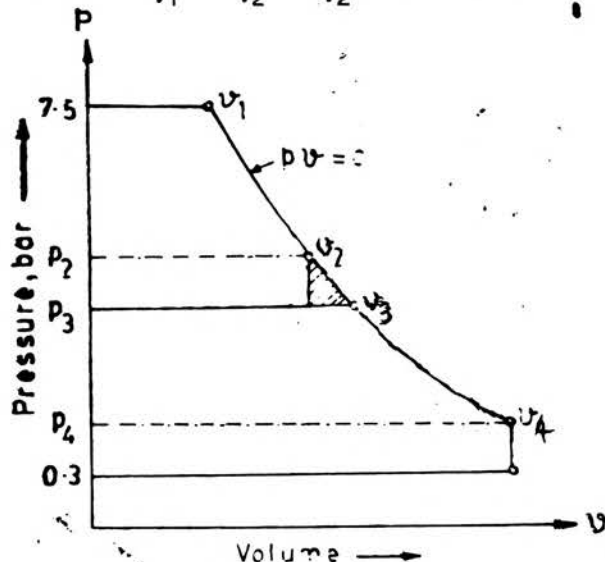
Problem-2 : A double-acting, two-cylinder, compound steam engine receives steam, at a pressure of 7.5 bar and exhausts into a condenser at a pressure of 0.3 bar. The cylinder volume ratio is 4 to 1 and cut-off in the H.P. cylinder takes place at half stroke. Draw the hypothetical indicator diagram for the engine, assuming the expansion to be hyperbolic and neglecting the clearance.

Calculate the fraction of the stroke at which cut-off must take place in the L.P. cylinder, if the initial loads on the two piston are to be equal. Also find the mean effective pressure of each cylinder referred to the L.P. cylinder and the percentage loss of work due to the incomplete expansion in the H.P. cylinder. The strokes are equal in the two cylinders.

Referring to fig. 2-8, $p_1 = 7.5 \text{ bar}$; $p_b = 0.3 \text{ bar}$; $\frac{v_1}{v_2} = 0.5$ or $r_1 = \frac{v_2}{v_1} = 2$, and

$$r = \frac{v_4}{v_2} = 4 \text{ (where } r = \text{ratio of cylinder volumes).}$$

$$\therefore R = \frac{v_4}{v_1} = \frac{v_4}{v_2} \times \frac{v_2}{v_1} = r \times r_1 = 4 \times 2 = 8 = \text{ratio of expansion for the whole engine.}$$



For equal initial loads on the two pistons,

$$p_1 - p_3 = (p_3 - p_b) \frac{v_4}{v_2}$$

$$\text{i.e. } 7.5 - p_3 = (p_3 - 0.3) \times 4$$

$$\therefore \text{Receiver pressure, } p_3 = 1.74 \text{ bar}$$

$$\text{Now, } p_4 v_4 = p_1 v_1$$

$$\text{i.e. } p_4 = p_1 \times \frac{v_1}{v_4} = 7.5 \times \frac{1}{8} = 0.9375 \text{ bar}$$

and $p_3 v_3 = p_4 v_4$, i.e. cut-off in L.P. cylinder,

$$\frac{v_3}{v_4} = \frac{p_4}{p_3} = \frac{0.9375}{1.74} = \mathbf{0.5388 \text{ of stroke.}}$$

Ratio of expansion in L.P. cylinder,

$$r_2 = \frac{v_4}{v_3} = \frac{1}{0.5388} = 1.85$$

$$p_m \text{ h.p.} = \frac{p_1}{r_1} [1 + \log_e r_1] - p_3$$

$$= \frac{7.5}{2} [1 + \log_e 2] - 1.74 = 4.61 \text{ bar (M.E.P. of H.P. cylinder)}$$

∴ M.E.P. of H.P. cylinder referred to L.P. cylinder

$$= 4.61 \times \frac{v_2}{v_4} = 4.61 \times \frac{1}{4} = 1.15 \text{ bar.}$$

$$p_m \text{ l.p.} = \frac{p_3}{r_2} [1 + \log_e r_2] - p_b = \frac{1.74}{1.85} [1 + \log_e 1.85] - 0.3 = 1.22 \text{ bar.}$$

∴ Total M.E.P. referred to L.P. cylinder = 1.15 + 1.22 = 2.37 bar.

Work lost due to incomplete expansion in H.P. cylinder is shown by the shaded area of fig. 2-8.

Overall M.E.P. (considering whole diagram including shaded area or considering overall diagram) referred to L.P. cylinder

$$= \frac{p_1}{R} [1 + \log_e R] - p_b = \frac{7.5}{8} [1 + \log_e 8] - 0.3 = 2.59 \text{ bar.}$$

∴ Loss of M.E.P. due to incomplete expansion in H.P. cylinder

$$= 2.59 - 2.37 = 0.22 \text{ bar}$$

∴ Percentage loss of work due to incomplete expansion in H.P. cylinder

$$= \frac{0.22}{2.59} \times 100 = 8.49\%$$

Problem-3 : The following data apply to a two-cylinder, double-acting, condensing compound steam engine :

Steam supply pressure	... 12.5 bar
Ratio of cylinder volumes	... 3.6 to 1
Back pressure in L.P. cylinder	... 0.2 bar
Cut-off in H.P. cylinder	... 0.4 of the stroke
Cut-off in L.P. cylinder	... 2/3rd of the stroke.

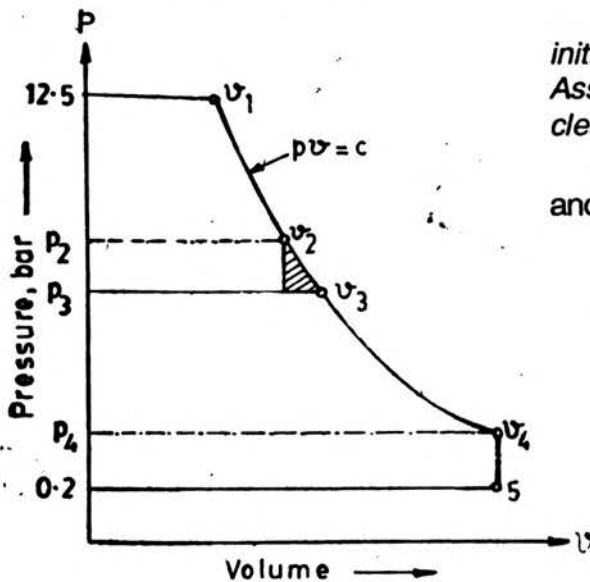


Fig. 2-9.

Estimate the ratio of work done and ratio of initial load on the pistons in H.P. and L.P. cylinders. Assume hyperbolic expansion. Neglect the effect of clearance and compression.

Referring to fig. 2-9, ratio of work done in H.P. and L.P. cylinders

$$= \frac{p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 v_2}{p_3 v_3 \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b v_4}$$

$$= \frac{p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 v_2}{p_1 v_1 \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b v_4} \quad (\text{as } p_3 v_3 = p_1 v_1)$$

Dividing throughout by v_2 , ratio of work done

$$= \frac{\rho_1 \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - \rho_3}{\rho_1 \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - \rho_b \times \frac{v_4}{v_2}} \quad \dots (1)$$

But, $\frac{v_1}{v_2}$ = cut-off ratio in H.P. cylinder = 0.4, $\frac{v_2}{v_4} = \frac{1}{\text{cylinder volume ratio}} = \frac{1}{3.6}$ and $\frac{v_4}{v_3}$ = expansion ratio in L.P. cylinder = $\frac{3}{2}$

Applying hyperbolic relation, between points of cut-off in H.P. and L.P. cylinders,

$$\rho_1 v_1 = \rho_3 v_3 \quad \text{i.e. } \rho_1 \times \frac{v_1}{v_2} \times v_2 = \rho_3 \times \frac{v_3}{v_4} \times v_4$$

$$\text{i.e. } 12.5 \times 0.4 \times v_2 = \rho_3 \times \frac{2}{3} \times v_4$$

$$\therefore \text{Receiver pressure, } \rho_3 = \frac{12.5 \times 0.4 \times 3}{2} \times \frac{v_2}{v_4} = \frac{12.5 \times 0.4 \times 3}{2} \times \frac{1}{3.6} = 2.08 \text{ bar}$$

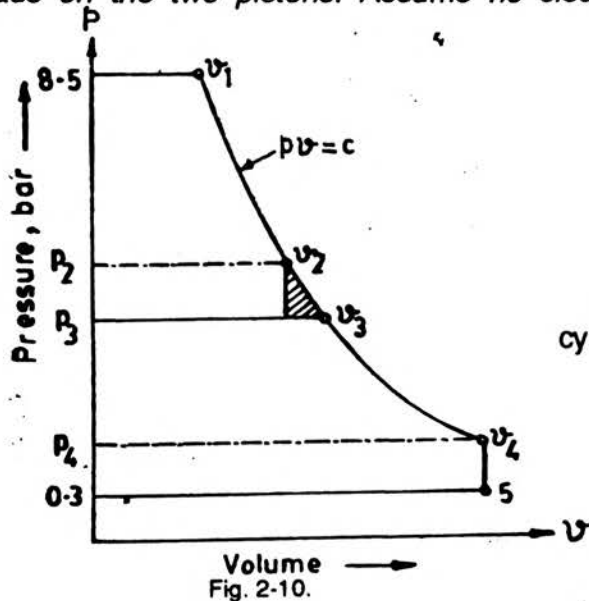
Substituting the values the eqn. (i), we have,

$$\left[\text{Ratio of work done in H.P. and L.P. cylinders} \right] = \frac{12.5 \times 0.4 \left[1 + \log_e \left(\frac{1}{0.4} \right) \right] - 2.08}{12.5 \times 0.4 \left[1 + \log_e \left(\frac{3}{2} \right) \right] - 0.2 \times 3.6} = 1.19$$

Ratio of initial loads on the H.P. and L.P. pistons is given by

$$\frac{(\rho_1 - \rho_3) \frac{\pi}{4} (d_1)^2}{(\rho_3 - \rho_b) \frac{\pi}{4} (d_2)^2} = \frac{(\rho_1 - \rho_3) \frac{\pi}{4} (d_1)^2 \times l}{(\rho_3 - \rho_b) \frac{\pi}{4} (d_2)^2 \times l} = \frac{(\rho_1 - \rho_3) \times v_2}{(\rho_3 - \rho_b) \times v_4} = \frac{12.5 - 2.8}{2.08 - 0.2} \times \frac{1}{3.6} = 1.54$$

Problem-4 : Estimate the dimensions of a compound steam engine to develop indicated power of 93.8 kW at 4 r.p.s. with following particulars : initial pressure 8.5 bar; back pressure 0.3 bar; allowable piston speed 2.5 m/sec.; cylinder volume ratio 3.5; overall diagram factor 0.85; cut-off in H.P. cylinder at 0.4 stroke. If the point of cut-off in the L.P. cylinder is at 0.53 stroke, determine the receiver pressure and compare the initial loads on the two pistons. Assume no clearance and hyperbolic expansion.



Referring to fig. 2-10, $\frac{v_1}{v_2} = 0.4$, $\frac{v_4}{v_2} = 3.5$.

$$\text{Total expansion ratio, } R = \frac{v_4}{v_1} = \frac{v_4}{v_2} \times \frac{v_2}{v_1} = \frac{3.5}{0.4} = 8.75$$

M.E.P. of the whole engine referred to L.P. cylinder is calculated by using eqn. (2.1),

$$\begin{aligned} \text{M.E.P.} &= f \left[\frac{\rho_1}{R} \left\{ 1 + \log_e (R) \right\} - \rho_b \right] \\ &= 0.85 \left[\frac{8.5}{8.75} \left\{ 1 + \log_e (8.75) \right\} - 0.3 \right] \\ &= 2.36 \text{ bar.} \end{aligned}$$

Using eqn. (2.2), indicated power = $\frac{2 \times 10^5 \times \text{M.E.P.} \times \frac{\pi}{4} (d_2)^2 \times l \times N}{1,000}$ kW

(where $2lN$ = average piston speed = 2.5 m/sec.)

i.e. $93.8 = \frac{10^5 \times 2.36 \times \frac{\pi}{4} (d_2)^2 \times 2.5}{1,000}$

$\therefore (d_2)^2 = 0.2025$ and $d_2 = \sqrt{0.2025} = 0.45$ m i.e. 45 cm (dia. of L.P. cylinder)

Cylinder volume ratio, $\frac{V_4}{V_2} = \left(\frac{d_2}{d_1}\right)^2$ as stroke of both pistons is same.

\therefore Diameter of H.P. cylinder, $d_1 = \frac{d_2}{\sqrt{\text{cylinder volume ratio}}} = \frac{45}{\sqrt{3.5}} = 24$ cm

Piston speed = $2 \times l \times N = 2.5$, i.e., $2 \times l \times 4 = 2.5$

Hence, $l = 0.3125$ m, i.e., 31.25 cm

Using hyperbolic relationships for points of cut-off in H.P and L.P. cylinders,

$p_3 V_3 = p_1 V_1$

i.e. $p_3 = p_1 \times \frac{V_1}{V_3} = p_1 \times \frac{V_1}{V_4} \times \frac{V_4}{V_3}$

$\therefore p_3 = 8.5 \times \frac{1}{8.75} \times \frac{1}{0.53} = 1.83$ bar

$\therefore \frac{\text{Initial load on high-pressure piston}}{\text{Initial load on low-pressure piston}} = \frac{p_1 - p_3}{p_3 - p_b} \times \left(\frac{d_1}{d_2}\right)^2 = \frac{8.5 - 1.83}{1.83 - 0.3} \times \frac{1}{3.5} = 1.24$

Problem-5 : A double-acting compound steam engine is required to give indicated power of 302 kW at 2.5 r.p.s with a steam supply at 12.5 bar and exhaust at 0.3 bar. Take the total number of expansions as 8.4, ratio of cylinder volumes 4.2 to 1, stroke equal to two-thirds of the L.P. cylinder diameter, overall diagram factor 0.66. Assume hyperbolic expansion and neglect effect of clearance.

Allowing for a pressure loss of 0.35 bar in the receiver between the two cylinders, find the cylinder diameters, common stroke and L.P. cut-off, if the initial loads on the two pistons are to be equal.

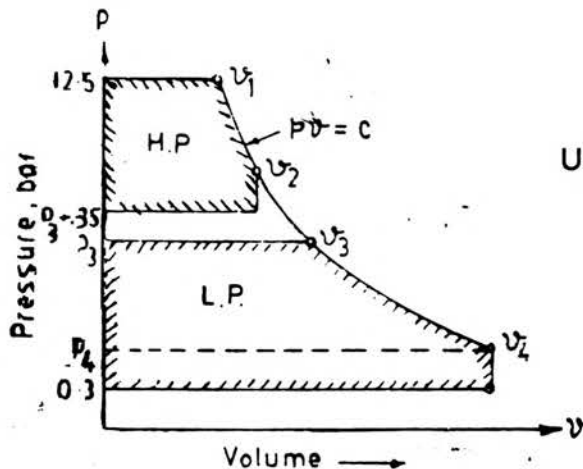


Fig. 2-11.

Referring to fig 2-11, $p_1 = 12.5$ bar,

$p_b = 0.3$ bar, $R = \frac{V_4}{V_1} = 8.4$, $\frac{V_4}{V_2} = 4.2$

Using eqn. (2.1), M.E.P. referred to L.P. cylinder

$= f \left[\frac{p_1}{R} (1 + \log_e R) - p_b \right]$ bar
 $= 0.66 \left[\frac{12.5}{8.4} (1 + \log_e 8.4) \right] - 0.3$
 $= 2.87$ bar.

Using eqn. (2.2), Indicated power

$= \frac{2 \times 10^5 \times \text{M.E.P.} \times \frac{\pi}{4} (d_2)^2 \times l \times N}{1,000}$ kW

$$\text{i.e., } 302 = \frac{2 \times 10^5 \times 2.87 \times \frac{\pi}{4} (d_2)^2 \times \frac{2}{3} d_2 \times 2.5}{1,000}$$

$$\therefore (d_2)^3 = 0.403 \text{ m}^3 \text{ and } d_2 = \sqrt[3]{0.403} = 0.738 \text{ m, i.e. } 73.8 \text{ cm (dia. of L.P. cylinder)}$$

$$\text{Diameter of H.P. cylinder, } d_1 = \frac{d_2}{\sqrt{\text{cylinder volume ratio}}} = \frac{0.738}{4.2} = 0.36 \text{ m, i.e. } 36 \text{ cm}$$

$$\text{Length of piston stroke, } l = \frac{2}{3} d_2 = \frac{2}{3} \times 0.738 = 0.49 \text{ m, i.e. } 49 \text{ cm.}$$

Referring to fig. 2-11, let p_3 = admission pressure of L.P. cylinder, then $(p_3 + 0.35)$ will be exhaust pressure of H.P. cylinder.

Since the initial loads on both pistons are same,

$$p_1 - (p_3 + 0.35) = (p_3 - p_b) \frac{v_4}{v_2}$$

$$\text{i.e., } 12.5 - (p_3 + 0.35) = (p_3 - 0.3) 4.2$$

$$\therefore p_3 = 2.58 \text{ bar}$$

As $p_1 v_1 = p_4 v_4$,

$$p_4 = p_1 \times \frac{v_1}{v_4} = 12.5 \times \frac{1}{8.4} = 1.49 \text{ bar.}$$

Now, $p_3 v_3 = p_4 v_4$

$$\therefore \text{Cut-off in L.P. cylinder, } \frac{v_3}{v_4} = \frac{p_4}{p_3} = \frac{1.49}{2.58} = 0.577 \text{ of stroke.}$$

Problem-6 : A double-acting compound steam engine receives steam at 10 bar and exhausts at 0.3 bar. The overall expansion ratio is 10 to 1 and expansion is hyperbolic. The engine runs at 200 r.p.m. with an overall diagram factor of 0.7 and a mechanical efficiency of 80% when developing brake power of 150 kW. Determine : (a) the swept volume of the L.P. cylinder, and (b) the swept volume of the H.P. cylinder for equal work to be done in the two cylinders. Neglect clearance and receiver losses.

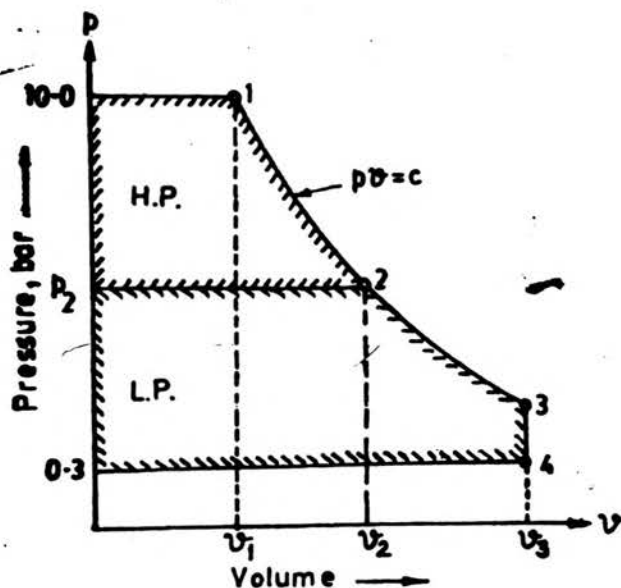


Fig. 2-12

(a) Referring to fig. 2-12,

The overall expansion ratio, $R = \frac{v_3}{v_1} = 10$ and overall diagram factor, $f = 0.7$.

$$N = \frac{200}{60} \text{ r.p.s.}$$

Using eqn. (2.1), M.E.P. referred to L.P. cylinder

$$= f \left[\frac{p_1}{R} (1 + \log_e R) - p_b \right]$$

$$= 0.7 \left[\frac{10}{10} (1 + \log_e 10) - 0.3 \right] = 2.1 \text{ bar}$$

Let v_3 = swept volume of L.P. cylinder in m^3 .
Total indicated power in kW

$$= \frac{150}{0.8} = \frac{10^5 \times 2.1 \times v_3 \times 2 \times 200/60}{1,000}$$

∴ $v_3 = 0.134 \text{ m}^3$ (swept volume of L.P. cylinder).

(b) For equal work to be done in the two cylinders, neglecting receiver losses, area of $p - v$ diagram of the H.P. cylinder = area of $p - v$ diagram of the L.P. cylinder.

$$\text{i.e. } p_1 v_1 + p_1 v_1 \log_e \left(\frac{v_2}{v_1} \right) - p_2 v_2 = p_2 v_2 + p_2 v_2 \log_e \left(\frac{v_3}{v_2} \right) - p_4 v_3$$

$$\therefore p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_2 v_2 = p_2 v_2 \left[1 + \log_e \left(\frac{v_3}{v_2} \right) \right] - p_4 v_3 \quad \dots(i)$$

Since the curve 1 - 2 - 3 is hyperbolic, $p_1 v_1 = p_2 v_2$

Dividing eqn. (i) by $p_1 v_1$, we have

$$\log_e \left(\frac{v_2}{v_1} \right) = 1 + \log_e \left(\frac{v_3}{v_2} \right) - \frac{p_4 v_3}{p_1 v_1}$$

$$\therefore \log_e \left(\frac{v_2}{v_1} \right) - \log_e \left(\frac{v_3}{v_2} \right) = 1 - \frac{0.3 \times 10}{10 \times 1} = 0.7$$

$$\therefore \log_e \left[\frac{v_2}{v_1} \times \frac{v_2}{v_3} \right] = 0.7 \quad \text{i.e. } \log_e \left[\frac{(v_2)^2}{0.0134 + 0.134} \right] = 0.7$$

$$\therefore 2.014 = \frac{(v_2)^2}{0.0134 \times 0.134} \quad (\text{as } \log_e 2.014 = 0.7)$$

∴ $(v_2)^2 = 0.00384$ and $v_2 = \sqrt{0.00384} = 0.062 \text{ m}^3$. (swept volume of H.P. cylinder)

Problem-7 : Find the ratio of diameters of the cylinders of a two-cylinder compound steam engine in order that the work done by each cylinder should be the same. Assume a hypothetical diagram, viz., expansion $pv = \text{constant}$, range of expansion 10 to 1 bar, exhaust at 0.35 bar and the stroke of each piston to be the same. Neglect clearance and receiver losses.

For equal work done in each cylinder, neglecting receiver losses, the $p - v$ area of the H.P. and L.P. cylinders must be equal.

Let volume of the L.P. cylinder be denoted by v_3 .

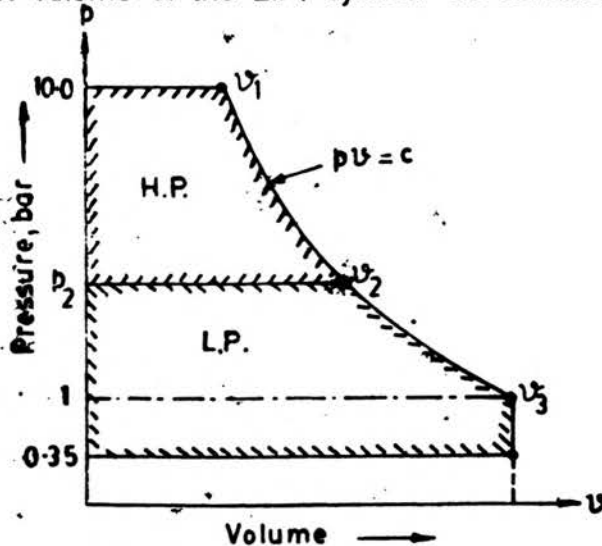


Fig. 2-13. Hypothetical indicator diagram.

Referring to fig. 2-13, $p_1 v_1 = p_3 v_3$

$$\therefore v_1 = \left(\frac{p_3}{p_1} \right) v_3 = \left(\frac{1}{10} \right) v_3$$

Work done in H.P. cylinder

$$= p_1 v_1 + p_1 v_1 \log_e \left(\frac{v_2}{v_1} \right) - p_2 v_2$$

$$= p_1 v_1 \log_e \left(\frac{v_2}{v_1} \right) \quad [\text{since } p_1 v_1 = p_2 v_2] \quad \dots (i)$$

Work done in L.P. cylinder

$$= p_2 v_2 + p_2 v_2 \log_e \left(\frac{v_3}{v_2} \right) - p_b v_3$$

$$= p_1 v_1 \left[1 + \log_e \left(\frac{v_3}{v_2} \right) \right] - p_b v_3 \quad \dots (ii)$$

Equating eqns. (i) & (ii) for equal work to be done in the two cylinders,

$$p_1 v_1 \log_e \left(\frac{v_2}{v_1} \right) = p_1 v_1 \left[1 + \log_e \left(\frac{v_3}{v_2} \right) \right] - p_b v_3$$

$$\therefore p_1 v_1 \left[1 + \log_e \left(\frac{v_3}{v_2} \times \frac{v_1}{v_2} \right) \right] = p_b v_3$$

Substituting for v_1 in terms of v_3 and numerical values of p_1 and p_b , we have,

$$10 \times \frac{1}{10} v_3 \left[1 + \log_e \left(v_3 \times \frac{1}{10} \cdot \frac{v_3}{(v_2)^2} \right) \right] = 0.35 v_3$$

$$\text{i.e. } 1 + \log_e \left(\frac{1}{10} \times \frac{(v_3)^2}{(v_2)^2} \right) = 0.35$$

$$\therefore 2 \log_e \left\{ \sqrt{\frac{1}{10}} \left(\frac{v_3}{v_2} \right) \right\} = -0.65 \quad \text{i.e. } \log_e \left\{ \sqrt{\frac{1}{10}} \left(\frac{v_3}{v_2} \right) \right\} = -0.325$$

$$\therefore \sqrt{\frac{1}{10}} \left(\frac{v_3}{v_2} \right) = \frac{1}{e^{0.325}} = \frac{1}{0.384} = 0.722$$

$$\frac{v_3}{v_2} = \sqrt{10} \times 0.722 = 2.288 \quad (\text{cylinder volume ratio}).$$

$$\text{Now, } \frac{v_3}{v_2} = \frac{\frac{\pi}{4} (d_2)^2 \times l}{\frac{\pi}{4} (d_1)^2 \times l} = 2.288$$

$$\therefore \frac{d_2}{d_1} = \sqrt{2.288} = 1.51 \quad (\text{ratio of diameters of cylinders})$$

Problem-8 : A two-cylinder compound, double-acting, steam engine is required to develop brake power of 58.75 kW at 6 r.p.s when supplied with steam at 18 bar and exhausting to a condenser at 0.18 bar. Cut-off ratio in both the cylinders is to be 0.4. The stroke length for both the cylinders is 25 cm. Estimate suitable cylinder diameters to develop equal work. Neglect clearance and assume hyperbolic expansion. Take mechanical efficiency as 85% and diagram factor for each cylinder as 0.8.

Referring to fig. 2-14 for equal work done in both the cylinders,

$$f \left[p_1 v_1 \left\{ 1 + \log_e \left(\frac{v_2}{v_1} \right) \right\} - p_3 v_3 \right] = f \left[p_3 v_3 \left\{ 1 + \log_e \left(\frac{v_4}{v_3} \right) \right\} - p_b v_4 \right]$$

Dividing throughout by v_2 ,

$$p_1 \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 = p_3 \frac{v_3}{v_2} \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b \frac{v_4}{v_2}$$

Substituting $p_3 v_3 = p_1 v_1$,

$$p_1 \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 = p_1 \frac{v_1}{v_2} \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b \frac{v_4}{v_2}$$

$$\text{i.e. } p_1 \frac{v_1}{v_2} \left[\log_e \left(\frac{v_2}{v_1} \right) - \log_e \left(\frac{v_4}{v_3} \right) \right] = p_3 - p_b \times \frac{v_4}{v_2}$$

$$\text{i.e. } 18 \times 0.4 \left[\log_e 2.5 - \log_e 2.5 \right] = p_3 - p_b \times \frac{v_4}{v_2}$$

$$\therefore p_3 - p_b \times \frac{v_4}{v_2} = 0 \quad \text{Hence, } p_3 = p_b \times \frac{v_4}{v_2} \quad \dots (i)$$

Considering points of cut-off in H.P. and L.P. cylinders on hyperbolic curve,

$$p_1 v_1 = p_3 v_3$$

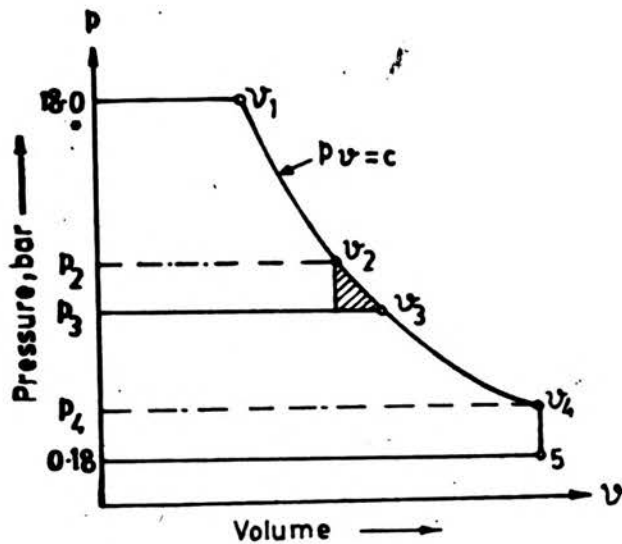


Fig. 2-14.

i.e., $18 \times 0.4v_2 = p_3 \times 0.4v_4$

$$\therefore \frac{v_4}{v_2} = \frac{18}{p_3}$$

Substituting value of $\frac{v_4}{v_2}$ in eqn. (i)

$$p_3 = p_b \times \frac{18}{p_3}$$

i.e. $(p_3)^2 = 0.18 \times 18$

$\therefore p_3 = 1.8$ bar.

Indicated power developed in H.P. cylinder

$$= \frac{58.75}{2 \times 0.85} = 34.56 \text{ kW}$$

\therefore Work done per stroke in the H.P. cylinder

$$= \frac{34.56}{6 \times 2} = 2.88 \text{ kJ or } 2,880 \text{ J per stroke.}$$

Work done per stroke in the H.P. cylinder

$$= 10^5 \times f \left[p_1 v_1 \left\{ 1 + \log_e \left(\frac{v_2}{v_1} \right) \right\} - p_3 v_2 \right]$$

$$= 10^5 \times f \times v_2 \left[p_1 \frac{v_1}{v_2} \left\{ 1 + \log_e \left(\frac{v_2}{v_1} \right) \right\} - p_3 \right]$$

i.e. $2,880 = 10^5 \times 0.8 \times v_2 \left[18 \times 0.4 \left\{ 1 + \log_e \left(\frac{1}{0.4} \right) \right\} - 1.8 \right]$

$\therefore v_2 = 0.0031 \text{ m}^3$.

But, $v_2 = \frac{\pi}{4} (d_1)^2 \times l$ i.e. $0.0031 = \frac{\pi}{4} (d_1)^2 \times \frac{25}{100}$

$\therefore (d_1)^2 = 0.0518$ and $d_1 = \sqrt{0.0518} = 0.125 \text{ m}$ i.e. **12.5 cm** (H.P. cylinder diameter).

As shown earlier, $\frac{v_4}{v_2} = \frac{18}{p_3}$

But, $\frac{v_4}{v_2} = \frac{\text{volume of L.P. cylinder}}{\text{volume of H.P. cylinder}} = \frac{\frac{\pi}{4} (d_2)^2 \times l}{\frac{\pi}{4} (d_1)^2 \times l} = \left(\frac{d_2}{d_1} \right)^2$

$\therefore \left(\frac{d_2}{d_1} \right)^2 = \frac{18}{p_3}$

$\therefore d_2 = d_1 \sqrt{\frac{18}{p_3}} = 12.5 \sqrt{\frac{18}{1.8}} = 39.53 \text{ cm}$ (L.P. cylinder diameter).

Problem-9 : A two-cylinder compound steam engine has an expansion ratio of 9 and the stroke of both cylinders is the same. The cut-off in the high-pressure cylinder takes place at half stroke. The engine is supplied with steam at 7 bar, and the condenser pressure is 0.15 bar. Assuming a common hyperbolic expansion curve for the two cylinders and neglecting effect of clearance and compression, find the percentage cut-off in the low-pressure cylinder and the receiver pressure so that the work shall be equally distributed between the cylinders.

Referring to fig. 2-15, for equal work done in the two cylinders,

$$p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 v_2 = p_3 v_3 \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b v_4$$

Since $p_1 v_1 = p_3 v_3$

$$p_1 v_1 \left[1 + \log_e \left(\frac{v_2}{v_1} \right) \right] - p_3 v_2 = p_1 v_1 \left[1 + \log_e \left(\frac{v_4}{v_3} \right) \right] - p_b v_4$$

Dividing by $p_1 v_1$ throughout, we have

$$1 + \log_e \left(\frac{v_2}{v_1} \right) - \frac{p_3 v_2}{p_1 v_1} = 1 + \log_e \left(\frac{v_4}{v_3} \right) - \frac{p_b v_4}{p_1 v_1}$$

$$\therefore \log_e \left(\frac{v_2}{v_1} \right) - \log_e \left(\frac{v_4}{v_3} \right) = \frac{p_3 v_2}{p_1 v_1} - \frac{p_b v_4}{p_1 v_1} \dots (i)$$

$$\text{Now, } \frac{p_3 v_2}{p_1 v_1} = \frac{p_3 v_2}{p_3 v_3} = \frac{v_2}{v_3} = \frac{v_2}{v_1} \times \frac{v_1}{v_3} = 2 \times \frac{v_1}{v_3}$$

$$\text{and } \frac{v_4}{v_3} = \frac{v_4}{v_1} \times \frac{v_1}{v_3} = 9 \times \frac{v_1}{v_3}$$

Substituting the values in eqn. (i),

$$\log_e (2) - \log_e \left(9 \frac{v_1}{v_3} \right) = 2 \left(\frac{v_1}{v_3} \right) - \frac{0.15 \times 9}{7}$$

$$\therefore \log_e \left(\frac{2}{9} \cdot \frac{v_3}{v_1} \right) = 2 \left(\frac{v_1}{v_3} \right) - 0.1925$$

$$2 \left(\frac{v_1}{v_3} \right) - \log_e \left(\frac{v_3}{v_1} \right) = \log_e \left(\frac{1}{4.5} \right) + 0.1925 \\ = -1.3116$$

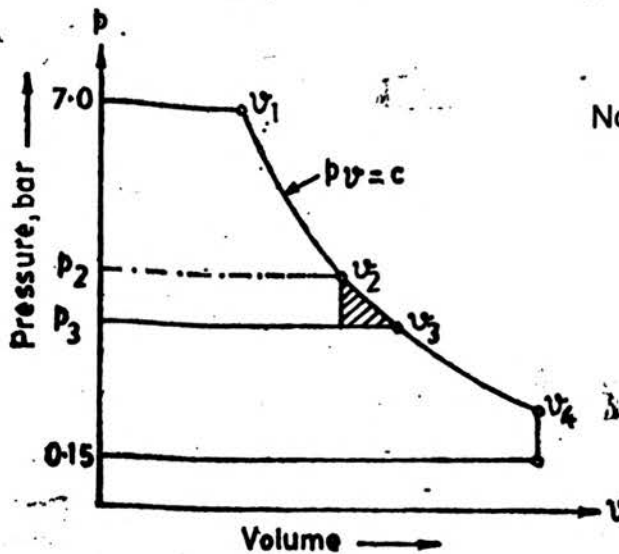


Fig. 2-15.

By trial and error, $\frac{v_3}{v_1} = 5.4$

Then, cut-off in L.P. cylinder, $\frac{v_3}{v_4} = \frac{v_3}{v_1} \times \frac{v_1}{v_4} = 5.4 \times \frac{1}{9} = 0.6$ or 60%

Now, $p_3 v_3 = p_1 v_1$

$$\therefore p_3 = p_1 \times \frac{v_1}{v_3} = 7 \times \frac{1}{5.4} = 1.296 \text{ bar (receiver pressure).}$$

Problem-10 : The following data refer to a triple expansion steam engine required to develop indicated power of 2,940 kW with a piston speed of 220 metres per minute :

Initial steam pressure 16 bar; exhaust pressure 0.15 bar; cylinder volume ratios 1 : 2.4 : 7.2; total ratio of expansion 18; overall diagram factor 0.6. Assuming equal initial loading on each piston, determine approximate values for : (a) cylinder diameters, (b) mean receiver pressures, and (c) cut-off points in each cylinder. Assume hyperbolic expansions with ideal receiver pressure conditions, and neglect clearance and compression effects.

Referring to fig. 2-16, let $v_2 = 1$ unit. As $v_2 : v_4 : v_6 :: 1 : 2.4 : 7.2$, then $v_4 = 2.4$ units and $v_6 = 7.2$ units.

$$\text{Now, } R = \frac{v_6}{v_1} \therefore v_1 = \frac{v_6}{R} = \frac{7.2}{18} = 0.4 \text{ unit.}$$

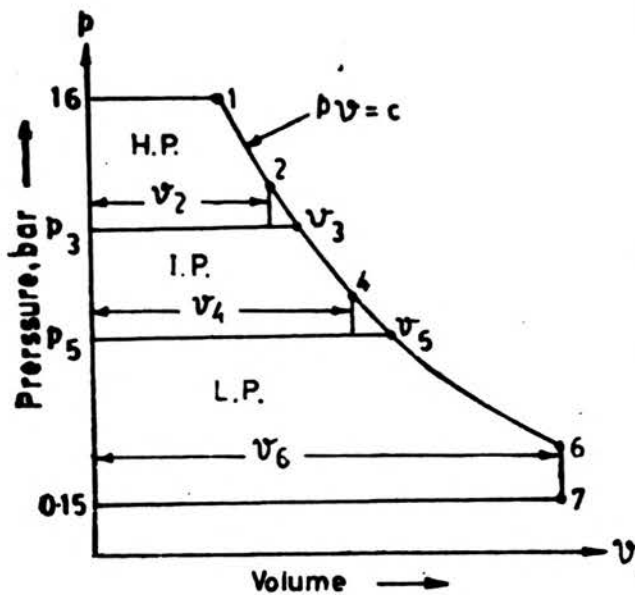


Fig. 2-16.

$$\therefore d_{H.P.} = \frac{d_{L.P.}}{\sqrt{\frac{v_6}{v_2}}} = \frac{227}{\sqrt{7.2}} = 84.6 \text{ cm. and } d_{L.P.} = \frac{d_{L.P.}}{\sqrt{\frac{v_6}{v_4}}} = \frac{227}{\sqrt{\frac{7.2}{2.4}}} = 131 \text{ cm.}$$

(a) M.E.P. referred to L.P. cylinder

$$= f \left[\frac{p_1}{R} (1 + \log_e R) - p_b \right] \text{ bar}$$

$$= 0.6 \left[\frac{16}{18} (1 + \log_e 18) - 0.15 \right]$$

$$= 1.98 \text{ bar}$$

Indicated power

$$= 2,940 = \frac{10^5 \times \text{M.E.P.} \times a_{L.P.} \times l \times N \times 2}{1,000} \text{ kW}$$

But, $2l/N =$ piston speed $= \frac{220}{60}$ m/sec.

$$\therefore a_{L.P.} = \frac{2,940 \times 1,000 \times 60}{10^5 \times 1.98 \times 220} = 4.05 \text{ m}^2$$

$$\therefore d_{L.P.} = \sqrt{\frac{4 \times 4.5}{\pi}} = 2.27 \text{ m or } 227 \text{ cm}$$

(b) For equal initial loads on each piston, $(p_1 - p_3) v_2 = (p_3 - p_5) v_4 = (p_5 - p_b) v_6$
 i.e. $(16 - p_3) 1 = (p_3 - p_5) 2.4 = (p_5 - 0.15) 7.2$

$$\therefore 16 - p_3 = 7.2 p_5 - 7.2 \times 0.15$$

$$\therefore p_3 = 17.08 - 7.2 p_5$$

... (i)

Now, $2.4 (p_3 - p_5) = 7.2 p_5 - 7.2 \times 0.15$

Substituting the value of p_3 from eqn.(i),

$$2.4 (17.08 - 7.2 p_5) - 2.4 p_5 = 7.2 p_5 - 1.08$$

$$40.992 - 17.28 p_5 - 2.4 p_5 = 7.2 p_5 - 1.08$$

$$\therefore p_5 = 1.562 \text{ bar (L.P. receiver pressure).}$$

Substituting the value of p_5 in eqn. (i),

$$p_3 = 17.08 - 7.5 p_5 = 17.08 - 7.2 \times 1.562 = 5.83 \text{ bar (I.P. receiver pressure).}$$

(c) H.P. cut-off $= \frac{v_1}{v_2} = \frac{0.4}{1} = 0.4$

Now, $p_1 v_1 = p_3 v_3$

i.e. $16 \times 0.4 = 5.83 \times v_3 \quad \therefore v_3 = 1.109 \text{ units.}$

I.P. cut-off $= \frac{v_3}{v_4} = \frac{1.109}{2.4} = 0.458$

Again, $p_1 v_1 = p_5 v_5$

i.e. $16 \times 0.4 = 1.562 \times v_5 \quad \therefore v_5 = 4.098 \text{ units.}$

L.P. cut-off $= \frac{v_5}{v_6} = \frac{4.098}{7.2} = 0.57$

2.6 Methods of Governing

There are three methods of governing compound steam engines. These are :
 ... *Throttle governing*—reducing the steam supply pressure in the H.P. cylinder,

... *Cut-off governing on H.P. cylinder*—varying the point of cut-off in the H.P. cylinder, and

... *Cut-off governing on L.P. cylinder*—varying the point of cut-off in the L.P. cylinder.

In Throttle Governing, the initial pressure in the H.P. cylinder is reduced by throttling the steam before entering the H.P. cylinder and the points of cut-off in both cylinders

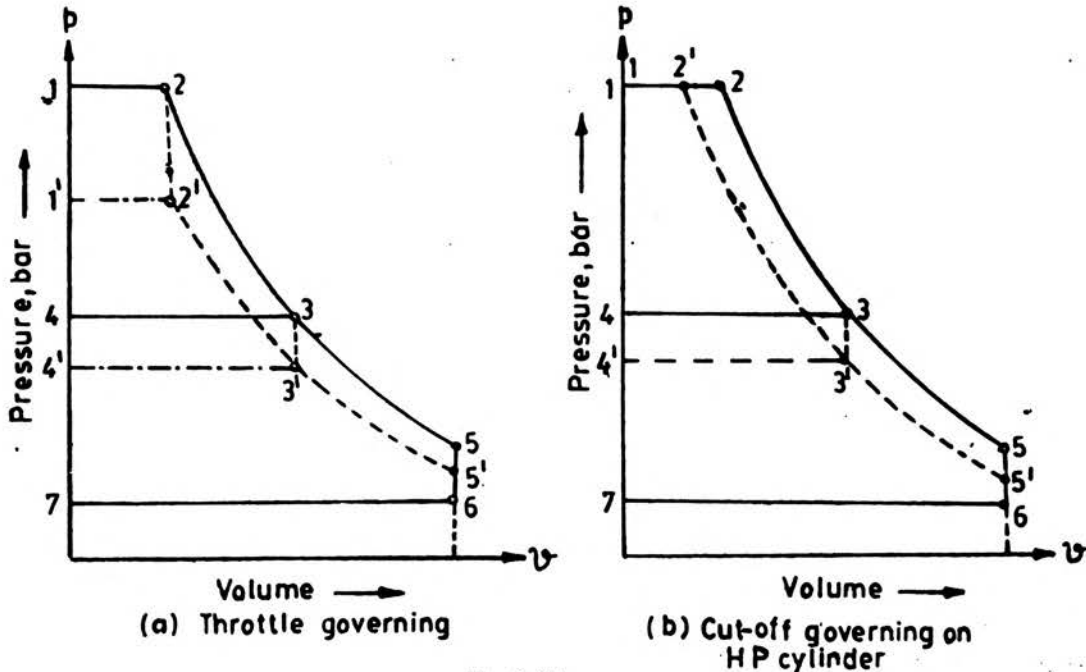


Fig. 2-17.

remain unaltered. The effect of this will be to reduce the admission pressure to the L.P. cylinder. Fig. 2-17(a) represents the hypothetical indicator diagram for a two-cylinder compound steam engine where area 1-2-3-4 is the H.P. cylinder diagram and area 4-3-5-6-7 is the L.P. cylinder diagram.

Let the steam supply pressure be reduced from p_1 to p_1' . The admission to the H.P. cylinder is represented by 1'-2'; the point of cut-off 2' must be vertically under 2, as the cut-off volume is the same. The new expansion curve will now be 2'-3'-5' as shown dotted. As the cylinder volume of the H.P. cylinder is the same, exhaust on H.P. cylinder will begin at 3', where 3' is vertically under 3. The new indicator diagram for L.P. cylinder is represented by 4'-3'-5'-6-7.

It may be noted from fig. 2-17(a) that the effect of throttle governing is to reduce the work done in both the cylinders, the greater reduction taking place in H.P. cylinder. Further, since the governing is by throttling, the steam consumption of the engine in kilograms per hour will follow Willan's law.

In Cut-off Governing on H.P. Cylinder, the point of cut-off in the H.P. cylinder is varied. Referring to fig 2-17(b), 1-2-3-4 is the H.P. indicator diagram area and 4-3-5-6-7 is the L.P. indicator diagram area at full load. With the decrease of load, the cut-off in the H.P. cylinder takes place earlier, say at point 2'. The expansion in H.P. cylinder is continued upto point 3' and from 3' to 5' in the L.P. cylinder. The exhaust pressure of the H.P. cylinder is reduced. The work done in H.P. cylinder is now given by the area 1-2'-3'-4', and the work done in the L.P. cylinder is represented by the area 4'-3'-5'-6-7.

It may be noted that the effect of cut-off governing on H.P. cylinder is to reduce the work done in L.P. cylinder, while there is very little change in the work done by the H.P. cylinder. This is because the reduction in H.P. work done due to early cut-off is compensated

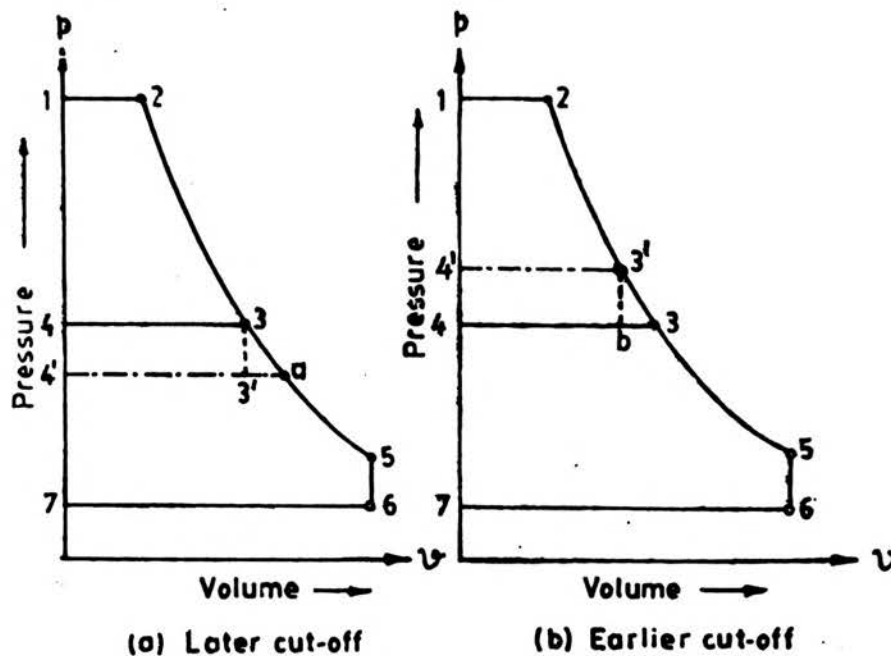


Fig. 2-18. Cut-off governing in L.P. cylinder.

by the decrease of exhaust pressure of H.P. cylinder.

From the efficiency point of view, power control by varying the point of cut-off is to be preferred to throttle governing; because the available pressure drop and hence enthalpy drop is not reduced.

Cut-off Governing on L.P. Cylinder makes very little reduction in the total work done by the engine. Referring to fig. 2-18 (a), let the action of governor cause later cut-off say at *a*. Thus, the exhaust pressure of the H.P. cylinder is reduced. As the volume of the H.P. cylinder is same as before, there must be a sudden pressure drop at release from 3 to 3' at constant volume. The area of the H.P. diagram is now 1-2-3-3'-4' and has correspondingly increased. The area of L.P. cylinder is now 4'-a-5-6-7 and has correspondingly reduced. This alters the ratio of work done in the two cylinders.

If the cut-off in the L.P. cylinder takes place at a point earlier than point 3 [fig. 2-18 (b)], the L.P. cylinder will now take in a small volume of steam; this will increase the exhaust pressure of the H.P. cylinder. But high pressure cylinder steam must expand down to 3, as the cylinder volume is the same; thus, the H.P. cylinder will release against a higher pressure than that in the cylinder. The exhaust stroke will tend to compress the steam back to 3' before exhausting along the line 3'-4'.

It may be noted that earlier cut-off in L.P. cylinder makes very little difference in the total work produced by the two cylinders together, but the work done by the H.P. cylinder is reduced, while the work done in the L.P. cylinder is increased.

Thus, it may be summarised that governing by controlling cut-off in the H.P. cylinder is the best, from the point of view of maintaining the efficiency of the engine at part loads. As seen earlier, its bad effect is to reduce the proportion of the work done in the L.P. cylinder at part loads. With condensing engines, at very light loads, this may cause the average pressure in the L.P. cylinder to fall below that necessary to overcome the back pressure and frictional resistances, thus reducing the efficiency of the engine.

To counteract this disparity of work, the cut-off in the L.P. cylinder should take place earlier so as to build up the exhaust pressure of H.P. (or admission pressure of L.P.), thereby increasing the L.P. work at the expense of that of the H.P. This variation in L.P.

cut-off will not affect steam consumption or the total work done.

Thus, it is advisable to operate cut-off governing on the H.P. and L.P. cylinders together to achieve the best results.

Tutorial-2

1. (a) What are the main objections to working the high pressure steam through large range of expansion in a single cylinder ?
(b) What do you mean by a compound steam engine ?
2. (a) Give reasons for compounding steam engines.
(b) State the advantages of a compound steam engine as compared to simple steam engine.
3. (a) Classify compound steam engines and state their main characteristics.
(b) Explain, with the help of sketches, the working of a receiver type compound steam engine.
4. Explain the following terms as applied to compound steam engines :
(i) cylinder volume ratio, (ii) total ratio of expansion, (iii) free or unresisted expansion, (iv) terminal drop, and (v) M.E.P. referred to L.P. cylinder.
5. What is meant by "M.E.P. referred to L.P. cylinder" ?

In a two-cylinder compound steam engine, the admission pressure of the H.P. cylinder is 7.5 bar and cut-off takes place at 0.6 stroke. The release pressure in the L.P. cylinder is 0.8 bar. The condenser pressure is 0.2 bar. If the initial loads on the two pistons are equal and expansion curve is assumed to be hyperbolic, estimate the ratio of cylinder volumes, the mean pressure in the receiver, and the point of cut-off in the L.P. cylinder.

[Ratio of cylinder volumes = 5.63; Mean pressure in receiver = 1.3 bar; Cut-off in L.P. cylinder = 0.615]

6. What are the differences between "cross-compounding" and "Woolfe-compounding" of a steam engine ? Explain this with the help of neat sketches.
7. Explain briefly the advantage of compounding in steam engines.

A compound double-acting steam engine develops brake power of 704 kW at 2 r.p.s. when taking in steam at 14 bar and exhausting it at 0.2 bar (20kPa). Cut-off in H.P. cylinder takes place at 0.5 of the stroke and the ratio of cylinder volumes is 3.5. Assuming a diagram factor of 0.75, mechanical efficiency of 80 per cent and piston speed of 3 metres per sec., calculate the H.P. and L.P. cylinder diameters and the stroke.

Find the fraction of stroke at which cut-off takes place in L.P. cylinder for equal initial loads on both the pistons. Assume hyperbolic expansion and neglect effect of clearance.

[Dia. of H.P. cylinder = 50 cm; Dia. of L.P. cylinder = 93.54 cm;
Length of piston stroke = 75 cm; Cut-off in L.P. cylinder = 0.61]

8. In a two-cylinder compound steam engine, the admission pressure of H.P. cylinder is 7.5 bar and cut-off takes place at 0.6 stroke. The release pressure in the L.P. cylinder is 0.8 bar and the condenser pressure is 0.2 bar. If the initial loads on the two pistons are equal and the curve of expansion is $p v^{1.2} = \text{constant}$, estimate the cylinder volume ratio, the mean pressure in the receiver, the point of cut-off in the L.P. cylinder, and the ratio of the work done in the two cylinders.

[Cylinder volume ratio = 3.87; Mean pressure in receiver = 1.7 bar;
Cut-off in L.P. cylinder = 0.534; Ratio of work done H.P./L.P. = 0.508]

9. Discuss the causes of loss of thermal efficiency in compound steam engines.

A compound, double-acting steam engine is required to develop indicated power of 370 kW at 2 r.p.s. The steam supply is at 8.5 bar and the condenser pressure is 0.3 bar. Cut-off in H.P. cylinder takes place at 0.4 of stroke, ratio of cylinder volumes is 3.5, piston speed is 2.5 metres per sec. and diagram factor is 0.85. If the cut-off in L.P. cylinder takes place at 0.475 of the stroke, determine the dimensions of the cylinders, and compare the initial loads on the two pistons. Assume hyperbolic expansion and neglect clearance.

[Stroke = 62.5 cm; Dia. of H.P. cylinder = 47.78 cm; Dia. of L.P. cylinder = 89.4 cm; Ratio of initial loads; H.P./L.P. = 1.052]

10. The following data refer to a double-acting compound steam engine required to give brake power of 299.4 kW at 3.33 r.p.s. (200 r.p.m.) with a mechanical efficiency of 80% :

Steam supply pressure, 15 bar; back pressure, 0.3 bar; cut-off in H.P. cylinder, at 0.4 stroke; total ratio of expansion, 10; piston speed, 200 metres/min; overall diagram factor, 0.75.

Assuming equal initial loading on each piston, determine : (i) the H.P. and L.P. cylinder diameters, (ii) the piston stroke, (iii) the receiver pressure, (iv) the release pressure in L.P. and H.P. cylinders; (v) the cut-off in L.P. cylinder, (vi) the mean effective pressure in H.P. cylinder (vii) the mean effective pressure

in L.P. cylinder, (viii) the overall mean effective pressure referred to L.P. cylinder considering overall diagram, (ix) the mean effective pressure of each cylinder referred to the L.P. cylinder, (x) the total mean effective pressure referred to L.P. cylinder, (xi) the percentage loss of work due to incomplete expansion in the H.P. cylinder, and (xii) the ratio of work done in the two cylinders.

Assume hyperbolic expansion and H.P. diagram factor = L.P. diagram factor = overall diagram factor. Neglect effect of clearance.

- [(i) $D_{H.P.} = 32$ cm, $D_{L.P.} = 64$ cm; (ii) 50 cm; (iii) Receiver pressure = 3.24 bar; (iv) Release pressures, L.P. = 1.5 bar, H.P. = 6 bar; (v) 0.463 of stroke; (vi) 6.19 bar (vii) 1.76 bar; (viii) 3.49 bar; (ix) H.P. m.e.p. = 1.55 bar, L.P. m.e.p. = 1.76 bar (x) 3.31 bar; (xi) 5.16%; (xii) Ratio of work done, $\frac{H.P.}{L.P.} = \frac{1}{1.13}$]

11. The following particulars relate to a non-condensing compound steam engine : H.P. cylinder bore, 40 cm; L.P. cylinder bore, 75 cm; stroke of each piston 100 cm; steam supply pressure; 15 bar; back pressure, 1.5 bar; cut-off in H.P. cylinder, 0.55 stroke; cut-off in L.P. cylinder; 0.35 stroke; speed; 3 r.p.s. Take a diagram factor of 0.65 for each cylinder, assume hyperbolic expansion and neglect effect of clearance. Estimate : (a) the pressure drop at release in H.P. cylinder, and (b) the indicated power of each cylinder.

[(a) 1.55 bar; (b) Indicated power of H.P. cylinder = 317.43 kW, Indicated power of L.P. cylinder = 572.56 kW]

12. Distinguish between Woolfe compound steam engines and receiver compound steam engines.

A compound steam engine is to develop indicated power of 93.75 kW at 1.83 r.p.s. Steam is supplied at 7.5 bar and condenser pressure is 0.2 bar. Assuming hyperbolic expansion and total expansion ratio of 15, a diagram factor of 0.7 and neglecting clearance and receiver losses, determine the diameters of the H.P. and L.P. cylinders so that they may develop equal power. Stroke of each piston is equal to L.P. cylinder diameter.

[Dia. of H.P. cylinder = 38.65 cm; Dia. of L.P. cylinder = 65.5 cm]

13. A compound steam engine receives steam at a pressure of 9 bar and exhausts into the condenser at 1 bar. The L.P. cylinder release pressure is 2 bar and the stroke of each piston is the same. Assuming hypothetical indicator diagram, find the ratio of cylinder diameters, if the work done in the two cylinders is equality shared. Neglect clearance and receiver losses.

[1.28]

14. Find the ratio of the diameters of the cylinders of a two-cylinder compound steam engine in order that the work done by each cylinder should be the same. Assume a hypothetical indicator diagram, viz $p v = \text{constant}$, range of expansion 9.5 to 2 bar and exhaust at 1 bar, and the stroke of each piston to be the same. Neglect clearance and receiver losses.

[1.304]

15. A compound steam engine is to develop indicated power of 120 kW at 2.3 r.p.s. The steam supply is at 8.5 bar and the condenser pressure is 0.3 bar. Assuming hyperbolic expansion and total ratio of expansion of 6, a diagram factor of 0.7, calculate the H.P. and L.P. cylinder diameters so that the power is equally divided between the two cylinders. Stroke of each piston may be taken equal to 1.2 times the diameter of L.P. cylinder. Assume no pressure drop at release in H.P. cylinder and neglect effect of clearance.

[Dia. of H.P. cylinder = 36.8 cm; Dia. of L.P. cylinder = 47.4 cm]

16. In a two-cylinder compound steam engine, the ratio of cylinder volumes is 5 and the total ratio of expansion is 10. The initial steam pressure is 10 bar and the back pressure is 0.4 bar. Assuming a common hyperbolic expansion curve for the two cylinders and equal distribution of work between the cylinders. compare the initial loads on the pistons. Neglect the effect of clearance and compression. [equal]

17. A triple-expansion steam engine is required to develop indicated power of 3,750 kW at 1.5 r.p.s. under the following conditions :

Pressure in H.P. steam chest	... 14 bar	:
Cut-off in H.P. cylinder	... 0.7 stroke	
Average piston speed	... 3.67 m per sec.	
Vacuum	... 700 mm of Hg	
Barometer	... 760 mm of Hg	

Using ratio of cylinder volumes of 1 : 3 : 7.5 and a diagram factor of 0.63, determine the dimensions of the cylinders. If the initial loads on the pistons are equal, estimate the mean receiver pressures for the engine. Assume hyperbolic expansion and neglect effect of clearance.

[Dia. of H.P. cylinder = 80 cm, Dia. of I.P. cylinder = 138.5 cm, Dia of L.P. cylinder = 219 cm; Length of stroke = 122 cm; Mean receiver pressure in 1st receiver = 4.5 bar, Mean receiver pressure in 2nd receiver = 1.345 bar]

18. Estimate the diameters of the cylinders for a quadruple expansion marine steam engine to develop indicated power of 9,000 kW with a piston, speed of 5 metres per second under the following conditions :
- Pressure in steam chest 16 bar; condenser pressure 0.15 bar; total ratio of expansion 14; overall diagram factor 0.65.
- Find also the point of cut-off in the H.P. cylinder. Assume hyperbolic expansion with ratio of cylinder volumes of 1 : 2.1 : 4.4 : 9. Neglect effect of clearance.
- [Dia. of H.P. cylinder = 99 cm; Dia. of 1st I.P. cylinder = 143.5 cm; Dia. of 2nd I.P. cylinder = 208 cm; Dia. of L.P. cylinder = 297 cm; Cut-off in H.P. cylinder = 0.643]
19. (a) What are the various methods of governing employed in compound steam engines ?
- (b) What will be the effect of the following on the distribution of power between the two cylinders of a compound steam engine.
- (i) Varying the cut-off in H.P. cylinder,
 - (ii) Varying the cut-off in L.P. cylinder, and
 - (iii) Throttling of inlet steam.
- Explain with the help of suitable diagrams wherever necessary.
20. What are the main factors to be considered in deciding the sizes of the cylinders in a compound steam engine ?
21. Explain briefly the effect on the distribution of the work between the two cylinders when governing is carried out
- (a) by throttling, and
 - (b) by cut-off.

3

STEAM ENGINE AND BOILER TRIALS

3.1 Steam Engine Trials

Engine trials are carried out for the purpose of comparing actual engine performance with theoretical or ideal performance. Trials are also carried out when the manufacturers have entered into agreement and guaranteed specified efficiency and maximum capacity (output) of the engine. The tests in this case are made to verify the guaranteed steam consumption per indicated power per hour or per brake power per hour under specified steam supply pressure and condenser vacuum.

For complete steam engine trial, it is necessary to measure losses in addition to the part of the heat converted into useful work and also to draw up a heat balance account. Such trials have been the direct cause of, and incentive to, the improvement in engines throughout the period of their development. This interest created a demand for authentic (trustworthy) records of engine performance which could only be satisfied by exhaustive trials carried out on steam plants. The measurements necessary to determine the thermal efficiency (brake and indicated) and to draw up complete heat balance sheet are :

- Indicated power (if possible),
- Brake power,
- Steam consumption in kilograms per hour,
- Pressure of steam supply at engine stop valve,
- Condition of steam supply at engine stop valve i.e. dryness fraction of steam if wet steam is used or temperature of steam if superheated steam is used,
- Temperature and pressure of exhaust steam,
- Quantity of condenser cooling or circulating water per hour, and
- Inlet and outlet temperatures of condenser cooling water.

When proper precautions are taken, it is possible to estimate the **indicated power** of a steam engine with great accuracy by taking indicator diagrams. In order to have the pressure inside the indicator cylinder same as the pressure inside the engine cylinder, the connecting pipe between the indicator and engine cylinder should be as short and straight as possible and of large bore. For double-acting steam engines, a separate indicator diagram should be taken for each end of the cylinder. Indicator diagrams taken from a double-acting cross-compound steam engine are shown in fig. 3-1. Before taking a diagram, the steam should be allowed to blow through freely in order to clear out condensed steam which may have collected in the pipes. Then the indicator cord is coupled up to the reducing gear. The pencil should be lightly pressed against the paper for about twenty seconds and the diagram is taken. The atmospheric line should then be drawn and the indicator cord uncoupled. The mean effective pressure is then calculated by measuring the area of the indicator diagram by means of a planimeter or by the

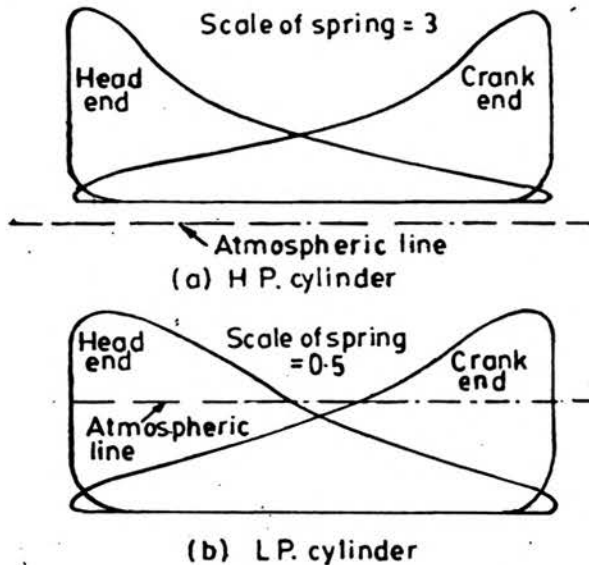


Fig. 3-1. Indicator diagrams from cross-compound steam engine.

while for large engines (marine engines) torsional (transmission) dynamometer may be used.

The **steam consumption** is best measured by condensing and weighing the exhaust steam. This is very easy in case of a condensing engine, fitted with a surface condenser. For a non-condensing engine, the steam consumption can only be measured by using a boiler solely to supply the steam to the engine under test; the steam consumption is estimated by deducting from the measured boiler feed, the sum of (i) the steam condensed in the steam pipes, (ii) the steam used for driving the feed pump, and (iii) the leakage of steam from the steam pipes.

When saturated steam (or wet steam) is used, its pressure and dryness fraction should be measured close to the engine side of the stop valve. The dryness fraction can be estimated by using a *combined separating and throttling calorimeter* if the steam is very wet, or by using a *throttling calorimeter* if the steam is nearly dry. In determining the dryness fraction of steam, great care should be taken in getting a proper sample of steam supplied to the engine. If superheated steam is used, the pressure and temperature of steam should be measured close to the engine side of the stop valve.

The pressure in the condenser is measured by taking vacuum gauge and the barometer readings. The vacuum gauge reading should be taken at every five minutes and the barometer readings at the beginning and at the end of the trial. The temperature of exhaust steam is measured by taking the temperature of water discharged from the air pump. The temperature of exhaust steam is also known as the temperature of condensate or hot-well temperature.

The quantity of condenser cooling water used per hour is measured by taking reading of the hook gauge to give the head of water in the water channel every five minutes. Then for a right-angled V-notch,

$$Q = 1.418 (h)^{5/2} \quad (\text{assuming } C_d = 0.6) \quad \dots (3.1)$$

where, Q = cubic metres of water flowing per second, and
 h = head of water over the notch in metres.

The quantity of condenser cooling water also may be measured by calibrated tanks. The inlet and outlet temperatures of condenser cooling water are measured by reading

method of mean ordinates.

The brake measures the work available for use external to the engine itself, and helps to assess the useful power available known as **brake power**. The type of dynamometer which should be used will depend upon the size of the engine under test. For comparatively small powers, an ordinary rope brake may be used with success, but for large powers, several alternatives are possible. A very convenient method is to couple up the engine directly to electric dynamo whose efficiency is known at all loads. The output from the dynamo is very easily measured and the brake power of the engine estimated from the known efficiency of the dynamo. If a dynamo is not available and the power is too large to be measured by a rope brake, a hydraulic brake may be used,

while for large engines (marine engines) torsional (transmission) dynamometer may be used.

the thermometers fitted in the inlet and outlet water pipes every ten minutes.

3.2 Heat Balance Sheet

In a trial of any heat engine, the distribution of the heat supplied per minute or per hour is required. This appears in the heat balance sheet or heat account sheet. *In order to complete a heat balance sheet for a steam engine, the engine should be tested over a period of time under conditions of constant load and constant steam supply. All the measurements listed earlier should be taken at regular interval of time. On the completion of the trial, the necessary data should be averaged out and a heat account sheet should be drawn up as follows :*

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	— Heat equivalent of brake power
			— Heat removed by condenser cooling water
			— Heat remaining in condensate
			— Heat remaining in jacket drain
			— Heat lost by radiation, leakage, error of measurement, etc. (by difference)
Total	Total

Note : *The heat equivalent of the friction power is not included in the above balance sheet on the right hand side because it is possible that some of the frictional heat will re-appear in the steam and eventually appear as heat removed from the condenser. Thus, the right hand side of the heat balance sheet should include the brake power and not the indicated power.*

Various items in heat balance sheet can be estimated as follows :

Heat supplied per minute (measured above 0°C) :

Let H_1 = Enthalpy in kJ per kg of steam at engine stop valve condition,

m_s = mass of steam supplied to the cylinder per min., and

m_j = mass of steam supplied to the cylinder jacket per min.

Then, gross heat supplied to the engine (measured above 0°C)

$$= (m_s + m_j) \times H_1 \text{ kJ/min.} \quad \dots(3.2)$$

Heat expenditure per minute :

(1) *Heat equivalent of brake power or heat converted into useful work :*

$$\text{Heat equivalent of brake power per min.} = \text{brake power} \times 60 \text{ kJ/min.} \quad \dots(3.3)$$

(2) *Heat remove by condenser cooling water /*

Let m_w = mass of condenser cooling water per min.; and

$t_2 - t_1$ = rise in temperature of condenser cooling water.

Then, heat removed by condenser cooling water per minute

$$= m_w \times 4.187 \times (t_2 - t_1) \text{ kJ/min.}$$

$$\text{(where 4.187 kJ/kg K is the specific heat of water (K))} \quad \dots(3.4)$$

(3) *Heat remaining in condensate or heat to hot-well (measured above 0°C) :*

Let h_2 = Enthalpy in kJ per kg of condensate (water) in the hot-well, and

m_s = mass of condensate per minute.

Then, heat remaining in condensate per min.

$$= m_s \times 4.187 \times (t_c - 0) \text{ kJ/min.} \quad \dots(3.5)$$

where, t_c is the temperature of condensate in hot-well in $^{\circ}\text{C}$

Heat rejected in exhaust steam per minute

= heat removed by condenser cooling water + heat to hot-well

$$= m_w \times 4.187 \times (t_2 - t_1) + m_s \times 4.187 \times (t_c - 0) \text{ kJ/min.} \quad \dots(3.6)$$

(4) *Heat remaining in jacket drain (measured above 0°C) :*

Let h_2 = Enthalpy in kJ per kg of water from jacket drain at the temperature measured, and

m_j = mass of steam supplied to jacket per min.

Then, heat remaining in jacket drain per min.

$$= m_j \times 4.187 \times (t_j - 0) \text{ kJ/min.} \quad \dots(3.7)$$

where, t_j is the temperature of condensed water in jacket drain in $^{\circ}\text{C}$.

(5) *Heat lost by radiation, leakage, error of observation, etc. (by difference) :*

This is obtained by the *difference* between gross heat supplied and the sum of items (1), (2), (3) and (4).

Problem-1 : *The following data was obtained during a test on a single-cylinder, double-acting steam engine having 20 cm cylinder diameter and 25 cm stroke ;*

M.E.P. (from indicator diagram)	... 250 kPa
Speed	... 5 r.p.s.
Effective radius of brake wheel	... 38 cm
Net brake load	... 1,360 newtons
Steam consumption	... 3.55 kg/min.
Steam supply pressure at engine stop valve	... 8 bar
Dryness fraction of steam at engine stop valve	... 0.97
Condenser cooling water	... 110 kg/min.
Temperature rise of condenser cooling water	... 14°C
Condensate temperature	... 40°C

Calculate : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the specific steam consumption in kg per kW-hr., (e) the brake thermal efficiency, and (f) the indicated thermal efficiency.

Draw up a heat balance sheet for the engine in kJ/min. and in percentages.

$$(a) \text{ Brake power} = (W - S) R 2 \pi N$$

$$= 1,360 \times 0.38 \times 2\pi \times 5 = 16,236 \text{ watts or } \mathbf{16.236 \text{ kW}}$$

$$(b) \text{ Indicated power} = 2 \times p_m \times a \times l \times N$$

$$= 2 \times (250 \times 10^3) \times (0.7854 \times (0.2)^2) \times 0.25 \times 5$$

$$= 19,635 \text{ watts or } \mathbf{19.635 \text{ kW}}$$

$$(c) \text{ Mechanical efficiency, } \eta_m = \frac{\text{Brake power}}{\text{Indicted power}} = \frac{16.236}{19.635} = 0.8268 \text{ or } \mathbf{82.68\%}$$

$$(d) \text{ Specific steam consumption} = \frac{3.55 \times 60}{16.236} = \mathbf{13.12 \text{ kg/kW-hr.}}$$

Gross heat supplied per minute :

At 8 bar, $h = 721.11 \text{ kJ/kg}$ and $L = 2,048 \text{ kJ/kg}$ (from steam tables).

Heat in steam supplied per min. measured above 0°C at engine stop valve
 $= m_s \times H_1 = m_s (h + xL) = 3.55 (721.11 + 0.97 \times 2,048) = 9,612 \text{ kJ/min.}$

Heat expenditure per minute :

- (1) Heat equivalent of brake power per minute = $16.236 \times 60 = 974.2 \text{ kJ}$
- (2) Heat removed by condenser cooling water per minute
 $= m_w \times (t_2 - t_1) \times 4.187 = 110 \times 14 \times 4.187 = 6,448 \text{ kJ}$
- (3) Heat to hot-well or heat remaining in condensate above 0°C per min.
 $= m_s \times (t_c - 0) \times 4.187 = 3.55 (40 - 0) \times 4.187 = 594.6 \text{ kJ}$
 (heat rejected in exhaust steam per min. = $6,448 + 594.6 = 7,042.6 \text{ kJ}$)
- (4) Heat lost by radiation, error, etc. per minute (by difference)
 $= 9,612 - (974.2 + 6,448 + 594.6) = 1,595.2 \text{ kJ}$

Heat balance sheet with 0°C as Datum

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	9,612	100	(1) Heat equivalent of brake power	794.2	10.13
			(2) Heat removed by condenser cooling water	6,448	67.08
			(3) Heat to hot-well	594.6	6.19
			(4) Heat lost by radiation error of measurements etc. (by difference)	1,595.2	16.60
Total	9,612	100	Total	9,612	100

(e) Net heat supplied per minute = gross heat supplied per min. – heat to hot-well per min.
 $= 9,612 - 594.6 = 9,017.4 \text{ kJ}$

Brake thermal efficiency, $\eta_b = \frac{\text{Heat equivalent of brake power per min.}}{\text{Net heat supplied per min.}}$
 $= \frac{974.2}{9,017.4} = 0.108 \text{ or } 10.8\%$

(f) Indicated thermal efficiency, $\eta_i = \frac{\text{Brake thermal efficiency}}{\text{Mechanical efficiency}}$
 $= \frac{0.108}{0.8268} = 0.1307 \text{ or } 13.07\%$

Problem-2 : The following readings were taken during a trial of a single-cylinder, double-acting steam engine having a cylinder diameter of 58 cm and stroke of 85 cm :

- Piston rod diameter .. 9 cm
- Speed .. 158 r.p.m.
- Mean effective pressure (cover end) .. 2.52 bar
- Mean effective pressure (crank end) .. 2.58 bar
- Brake power developed .. 224 kW
- Steam supply pressure .. 10 bar
- Steam supply temperature .. 234°C
- Condenser cooling water flow .. 1,700 kg/min.
- Inlet temperature of condenser cooling water .. 15°C

Outlet temperature of condenser cooling water	.. 28°C
Mass of condensate collected	.. 2,760 kg/hr.
Condensate temperature	.. 38°C
Condenser Vacuum	.. 637.5 mm Hg
Barometer reading	.. 750 mm Hg
K_p of superheated steam	.. 2.1 kJ/kg K

Calculate the mechanical and indicated thermal efficiencies of the engine and draw up a heat balance sheet in kJ/min. and in percentage. Also estimate the dryness fraction of the steam entering the condenser.

$$\text{Piston area (cover end)} = \frac{\pi}{4} (58)^2 = 2,640 \text{ cm}^2 = 0.264 \text{ m}^2$$

$$\text{Piston area (crank end)} = \frac{\pi}{4} (58^2 - 9^2) = 2,580 \text{ cm}^2 = 0.258 \text{ m}^2$$

$$\text{Indicated power}_{\text{engine}} = \text{indicated power}_{\text{cover}} + \text{indicated power}_{\text{crank}}$$

$$= \left[(2.52 \times 10^5) \times 0.264 \times 0.85 \times \frac{158}{60} \right] + \left[(2.58 \times 10^5) \times 0.258 \times 0.85 \times \frac{158}{60} \right]$$

$$= 2,97,900 \text{ watts} = 297.9 \text{ kW}$$

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{224}{297.9} = 0.7519 \text{ or } 75.19\%$$

At 10 bar, $H_s = 2,778.1$ kJ/kg, $t_s = 179.91^\circ\text{C}$ (from steam tables).

Enthalpy of 1 kg of superheated steam measured above 0°C ,

$$H_1 = H_s + K_p (t_{\text{sup}} - t_s) = 2,778.1 + 2.1 (234 - 179.91) = 2,891.7 \text{ kJ/kg.}$$

Enthalpy of 1 kg condensate (water) above 0°C ,

$$h_2 = (38 - 0) \times 4.187 = 159.1 \text{ kJ/kg.}$$

Net heat supplied (difference in enthalpy) per kg of steam

$$= H_1 - h_2 = 2,891.7 - 159.1 = 2,732.6 \text{ kJ/kg.}$$

$$\text{Net heat supplied per minute} = m_s (H_1 - h_2) = \frac{2760}{60} \times 2,732.6 \text{ kJ/min.}$$

$$\text{Indicated thermal efficiency} = \frac{\text{Heat equivalent of indicated power per min.}}{\text{Net heat supplied per min.}}$$

$$= \frac{\text{Indicated power} \times 60}{m_s (H_1 - h_2)} = \frac{297.9 \times 60}{\frac{2,760}{60} \times 2,732.6} = 0.1422 \text{ or } 14.22\%$$

Gross heat supplied per minute :

$$\text{Heat in steam supplied per minute} = m_s \times H_1 = \frac{2,760}{60} \times 2,891.7 = 1,33,018 \text{ kJ}$$

Heat expenditure per minute :

$$(1) \text{ Heat equivalent of brake power per min.} = 224 \times 60 = 13,440 \text{ kJ}$$

$$(2) \text{ Heat removed by condenser cooling water per min.}$$

$$= m_w \times (t_2 - t_1) \times 4.187 = 1,700 \times (28 - 15) \times 4.187 = 92,533 \text{ kJ}$$

$$(3) \text{ Heat remaining in condensate above } 0^\circ\text{C} \text{ per min.}$$

$$= m_s \times (t_c - 0) \times 4.187 = \frac{2,760}{60} \times (38 - 0) \times 4.187 = 7,319 \text{ kJ}$$

(4) Heat lost by radiation, error, etc. (by difference) per min.
 = 1,33,018 - (13,440 + 92,533 + 7,319) = 19,726 kJ.

Heat balance sheet with 0°C as Datum

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	1,33,018	100	(1) Heat equivalent of brake power	13,440	10.1
			(2) Heat removed by condenser cooling water	92,533	69.57
			(3) Heat remaining in condensate	7,319	5.50
			(4) Heat lost by radiation error of measurements etc. (by difference)	19,726	14.83
Total	1,33,018	100	Total	1,33,018	100

Condenser pressure = 750 - 637.5 = 112.5 mm of Hg = $\frac{112.5}{750} = 0.15$ bar.

At 0.15 bar, $L = 2,373.1$ kJ/kg and $t_s = 53.97^\circ\text{C}$ (from steam tables).

Using eqn. (1.2a),

heat lost by exhaust steam/hr. = heat gained by condenser cooling water/hr.

$$m_s [xL + (t_s - t_c) K] = m_w \times (t_2 - t_1) K$$

i.e. 2,760 [2,373.1x + (53.97 - 38) 4.187] = 1,700 × 60 × (28 - 15) 4.187

∴ x = 0.82 (dryness fraction of exhaust steam)

Problem-3 : The following data was obtained during a test on a single-cylinder, double-acting steam engine having 20 cm cylinder diameter and 25 cm stroke : Mean effective pressure from indicator diagram, 2.5 bar; speed, 5 r.p.s.; effective radius of brake wheel, 38 cm; net brake load, 1,340 newtons; steam consumption, 3.6 kg/min.; steam supply pressure at engine stop valve, 8 bar; dryness fraction of steam at engine stop valve, 0.9; condenser cooling water, 110 kg/min.; temperature rise of condenser cooling water, 14°C; condenser pressure, 0.1 bar (10 kPa); hot-well temperature, 40°C. Take specific heat of water as 4.187 kJ/kg K and calculate : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the brake thermal efficiency, (e) the indicated thermal efficiency, and (f) the brake power steam consumption in kg per kW-hour. Also draw up a heat balance sheet in kJ/min. and in percentages.

(a) Brake power = $(W - S) R 2 \pi N$

$$= 1,340 \times \frac{38}{100} \times 2 \times 3.14 \times 5 = 15,990 \text{ watts or } 15.99 \text{ kW}$$

(b) Indicated power = $p_m \times a \times l \times N \times 2$

$$= 10^5 \times 2.5 \times \frac{\pi}{4} \left(\frac{20}{100}\right)^2 \times \frac{25}{100} \times 5 \times 2 = 19,635 \text{ or } 19.635 \text{ kW}$$

(c) Mechanical efficiency, $\eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{15.99}{19.635} = 0.8143$ or 81.43%

(d) Gross heat supplied per minute :

At 8 bar, $h = 721.11$ kJ/kg, $L = 2,048$ kJ/kg (from steam tables).

Heat in steam supplied per min. measured above 0°C at engine stop valve

$$= m_s \times H_1 = m_s (h + xL) = 3.6 (721.11 + 0.9 \times 2,048) = 9,231.52 \text{ kJ/min.}$$

Heat expenditure per minute :

(1) Heat equivalent of brake power/min. = 15.99 × 60 = 959.4 kJ/min.

(2) Heat removed by condenser cooling water per min.

$$= m_w \times 4.187 \times (t_2 - t_1) = 110 \times 4.187 \times 14 = 6,447.98 \text{ kJ/min.}$$

(3) Heat to hot-well or heat remaining in condensate above 0°C per min.

$$= m_s \times 4.187 \times (t_c - 0) = 3.6 \times 4.187 (40 - 0) = 602.93 \text{ kJ/min.}$$

(4) Heat lost by radiation, error, etc. per min (by difference)

$$= 9,231.52 - (959.4 + 6,447.98 + 602.93) = 1,221.21 \text{ kJ/min.}$$

Net heat supplied per min. = Gross heat supplied/min. - heat to hot-well/min.

$$= 9,231.52 - 602.93 = 8,628.59 \text{ kJ/min.}$$

Brake thermal effi., $\eta_b = \frac{\text{Heat equivalent of brake power per min.}}{\text{Net heat supplied per min.}}$

$$= \frac{959.4}{8,628.59} = 0.1112 \text{ or } 11.12\%$$

(e) Indicated thermal efficiency, $\eta_i = \frac{\text{Heat equivalent of indicated power/min}}{\text{Net heat supplied per min}}$

$$= \frac{19.635 \times 60}{8,628.59} = 0.1366 \text{ or } 13.66\%$$

(f) Brake power steam consumption in kg/kW-hr. = $\frac{3.6 \times 60}{15.99} = 13.51 \text{ kg/kW-hr.}$

Heat balance sheet with 0°C as Datum

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	9,231.52	100	(1) Heat to brake power	959.4	10.40
			(2) Heat removed by condenser cooling	6,447.98	69.84
			(3) Heat to hot-well	602.93	6.53
			(4) Heat lost by radiation error etc. (by difference)	1,221.21	13.23
Total	9,231.52	100	Total	9,231.52	100

Problem-4 : During a test on a single-cylinder, double-acting jacketed steam engine, the following observations were made :

. Indicated power, 26 kW; brake power, 21; pressure of steam supplied, 6 bar, quality of steam supplied, 5% wet; mass of steam used in engine cylinder, 325 kg/hour; mass of steam used in Jacket, 30 kg/hour; condensate temperature, 40°C; temperature of jacket drain, 140°C; condenser cooling water, 10,200 kg/hour; temperature rise of condenser cooling water, 15°C. Taking specific heat of water as 4.187 kJ/kg K, draw up a heat balance sheet in kJ/min. and in percentage. Calculate the indicated thermal efficiency when the heat of jacket drain is not available to the boiler as feed water heat.

What will be the percentage improvement in the indicated thermal efficiency if the heat of the jacket drain is also available to the feed water ?

At 6 bar, $h = 670.56 \text{ kJ/kg}$, $L = 2,086.3 \text{ kJ/kg}$ (from steam tables).

Enthalpy of 1 kg of wet steam measured above 0°C,

$$H_1 = h + xL = 670.56 + 0.95 \times 2,086.3 = 2,652.54 \text{ kJ/kg.}$$

Total mass of steam supplied to engine per min.

= mass of steam used in engine cylinder/min. plus mass of steam used in jacket/min.

$$= \frac{m_s + m_j}{60} = \frac{325 + 30}{60} = \frac{355}{60} \text{ kg per min.}$$

Gross heat supplied per minute :

$$\text{Heat supplied to engine per min.} = \frac{m_s + m_j}{60} \times H_1 = \frac{355}{60} \times 2,652.54 = 15,694.2 \text{ kJ/min.}$$

Heat expenditure per minute :

(1) Heat equivalent of brake power per min. = $21 \times 60 = 1,260 \text{ kJ/min.}$

(2) Heat removed by condenser cooling water per min.

$$= m_w \times 4.187 \times (t_2 - t_1) = \frac{10,200}{60} \times 4.187 \times 15 = 10,676.85 \text{ kJ/min.}$$

(3) Heat remaining in condensate per min.

$$= m_s \times 4.187 \times (t_c - 0) = \frac{325}{60} \times 4.187 \times (40 - 0) = 907.2 \text{ kJ/min.}$$

(4) Heat remaining in jacket drain per min.

$$= m_j \times 4.187 \times (t_j - 0) = \frac{30}{60} \times 4.187 \times (140 - 0) = 293.09 \text{ kJ/min.}$$

(5) Heat lost by radiation, error, etc. per min. (by difference)

$$= 15,694.2 - (1,260 + 10,676.85 + 907.2 + 293.09) = 2,557.06 \text{ kJ/min.}$$

Heat balance sheet with 0°C as Datum

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	15,694.2	100	(1) Heat equivalent of brake power	1,260	8.03
			(2) Heat removed by condenser cooling water	10,676.85	68.04
			(3) Heat remaining in condensate	907.2	5.78
			(4) Heat remaining in jacket drain	293.09	1.86
			(5) Heat lost by radiation error etc. (by difference)	2,557.06	16.29
Total	15,694.2	100	Total	15,694.2	100

Indicated thermal efficiency when heat of condensate and heat of cylinder jacket drain is available to boiler as feed water heat

$$= \frac{\text{Heat equivalent of indicated power per min}}{\text{Net heat supplied per min.}}$$

$$= \frac{26 \times 60}{15,694.2 - (907.2 + 293.09)} = \frac{1,560}{14,493.93} = 0.1076 \text{ or } 10.76\%$$

Indicated thermal efficiency when heat of condensate is only available to boiler as

$$\text{feed water heat} = \frac{26 \times 60}{15,694.2 - 907.2} = \frac{1,560}{14,787} = 0.1055 \text{ or } 10.55\%$$

Percentage improvement in indicated thermal efficiency when heat of jacket drain is also available to boiler as feed water heat

$$= \frac{10.76 - 10.55}{10.55} = \frac{0.21}{10.55} = 0.019 \text{ or } 1.9\%$$

Problem-5 : The following observation were recorded during a trial on a jacketed double-acting compound steam engine supplied with dry saturated steam :

H.P. cylinder diameter	... 23 cm
L. P. cylinder diameter	... 40 cm
Stroke	... 58 cm
M.E.P. in H.P. cylinder	... 2.46 bar
M.E.P. in L.P. cylinder	... 1.39 bar
Average engine speed	... 92.4 r.p.m.
Brake-torque	... 4,150 N.m
Steam pressure during admission	... 6.5 bar
Receiver pressure	... 2.8 bar
Condenser vacuum	... 610 mm of Hg
Barometer reading	... 760 mm of Hg
Steam measured as discharged from air pump	... 8 kg/min.
Discharge from cylinder jacket drain	... 0.86 kg/min.
Discharge from receiver jacket drain	... 0.49 kg/min.
Mass of condenser cooling water	... 274 kg/min.
Temperature rise of condenser cooling water	... 15°C
Temperature of condensate	... 53°C

Draw up a heat balance account giving heat quantities in kJ per minute and in percentages. Estimate also the mechanical and brake thermal efficiencies of the engine.

$$\begin{aligned} \text{Indicated power of both cylinders (engine)} &= (p_{m1} a_1 + p_{m2} a_2) \times l \times N \times 2 \\ &= \left[(2.46 \times 10^5) \times 0.7854 \times (0.23)^2 + (1.39 \times 10^5) \times 0.7854 \times (0.4)^2 \right] \\ &\quad \times 0.58 \times \frac{92.4}{60} \times 2 = 49,462 \text{ watts} = 49.462 \text{ kW} \end{aligned}$$

$$\begin{aligned} \text{Brake power of the engine} &= \text{Torque} \times 2\pi N = 4,150 \times 2\pi \times \frac{92.4}{60} \\ &= 40,156 \text{ watts} = 40.156 \text{ kW} \end{aligned}$$

At 6.5 bar (from steam tables), enthalpy per kg of dry saturated steam $H_1 = 2,760$ kJ/kg.

Enthalpy per kg of condensate (water) from condenser,

$$h_2 = (53 - 0) 4.187 = 221.9 \text{ kJ/kg.}$$

Enthalpy per kg of condensate (water) from the receiver jacket drain at 2.8 bar is 551.48 kJ/kg (from steam tables), and

Enthalpy per kg of condensate (water) from cylinder jacket drain at 6.5 bar is 684.28 kJ/kg (from steam tables).

Total mass of steam supplied = 8 + 0.86 + 0.49 = 9.35 kg/min.

Gross heat supplied per minute :

$$\text{Heat in steam supplied} = 9.35 \times 2,760 = 25,806 \text{ kJ/min.}$$

Heat expenditure per minute :

$$(1) \text{ Heat equivalent of brake power} = 40.156 \times 60 = 2,409.5 \text{ kJ/min.}$$

$$(2) \text{ Heat removed by condenser cooling water} = 274 \times 15 \times 4.187 = 17,208.6 \text{ kJ/min.}$$

$$(3) \text{ Heat remaining in condensate above } 0^\circ\text{C} = 8(53 - 0) \times 4.187 = 1,775.3 \text{ kJ/min.}$$

$$(4) \text{ Heat remaining in cylinder jacket drain above } 0^\circ\text{C} = 0.86 \times 684.28 = 588.5 \text{ kJ/min.}$$

(5) Heat remaining in receiver jacket drain above 0°C = 0.49 × 551.48 = 270.2 kJ/min.

(6) Heat lost by radiation, error, etc. (by difference)

$$= 25,806 - (2,409.6 + 17,208.6 + 1,775.3 + 588.5 + 270.2) = 3,554 \text{ kJ/min.}$$

Heat Balance sheet with 0°C as Datum

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	25,806	100	(1) To useful work brake (power)	2,409.4	9.34
			(2) To condenser cooling water	17,208.6	66.68
			(3) To condensate	1,775.3	6.88
			(4) To cylinder jacket drain	588.5	2.28
			(5) To receiver jacket drain	270.2	1.05
			(6) To radiation error of measurement etc. (by difference)	3,554.0	13.77
Total	25,806	100	Total	25,806	100

$$\text{Mechanical efficiency } \eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{40.156}{49.462} = 0.8118 \text{ or } 81.18\%$$

Assuming that heat of receiver jacket and cylinder jacket drains and heat of condensate (from condenser) is available to the boiler feed water,

$$\text{Net heat supplied} = 25,808 - (270.2 + 588.5 + 1,775.3) = 23,172 \text{ kJ/min.}$$

$$\begin{aligned} \text{Brake thermal efficiency, } \eta_b &= \frac{\text{Heat equivalent of brake power per min}}{\text{Net heat supplied per min}} \\ &= \frac{2,409.4}{23,172} = 0.104 \text{ or } 10.4\% \end{aligned}$$

Problem-6 : A steam jacketed condensing steam engine working with dry saturated steam at an initial temperature of 123.27°C, develops brake power of 74 k.W. The air pump discharges 1,150 kg of water per hour to the hot-well at a temperature of 50°C. The condenser cooling water supplied per hour is 18,500 kg and its rise in temperature is 35°C. Neglecting radiation losses, find : (a) the heat received by the working steam from the jacket, and (b) assuming that the jacket to be supplied with boiler steam and only enthalpy of evaporation (latent heat) of the jacket steam to be given up to working steam, find the mass of cylinder jacket steam used per kilogram of cylinder feed.

(a) Absolute pressure of steam corresponding to the saturation temperature of 123.27°C is 2.2 bar, and enthalpy (H_1) of dry saturated steam at 2.2 bar is 2,711 kJ/kg. and enthalpy of evaporation $L_1 = 2,193.4$ kJ/kg (from steam tables).

Now, heat supplied by the boiler steam entering the cylinder per min.

$$= \frac{m_s}{60} \times H_1 = \frac{1,150}{60} \times 2,711 = 51,961 \text{ kJ/min.}$$

(1) Brake power heat equivalent = 74 × 60 = 4,440 kJ/min.

(2) Heat removed by condenser cooling water = $\frac{18,500}{60} \times 35 \times 4.187 = 45,185$ kJ/min.

(3) Heat to hot-well = $\frac{1,150}{60} (50 - 0) \times 4.187 = 4,013$ kJ/min.

∴ Heat rejected in exhaust steam = 45,185 + 4,013 = 49,198 kJ/min.

(4) Neglecting radiation losses, heat received by the working steam from the cylinder

jacket steam per min.

$$= \text{heat supplied by the working steam entering the cylinder} \textit{ minus} \\ \text{heat to brake power} + \text{heat rejected in exhaust steam} \\ = 51,961 - (4,440 + 49,198) = -1,677 \text{ kJ/min.}$$

Negative sign indicates that heat is received by the working steam from the cylinder jacket steam.

(b) Now assuming that the cylinder jacket to be supplied with boiler steam at 2.2 bar and *only enthalpy of evaporation (latent heat) of cylinder jacket steam to be given up to working steam,*

$$\text{Mass of jacket steam used per hour, } m_j = \frac{1,677 \times 60}{2,193.4} = 45.87 \text{ kg}$$

(2,193.4 kJ/kg is the enthalpy of evaporation of 1 kg of steam at 2.2 bar from steam tables),

\therefore Mass of cylinder jacket steam used per kg of the cylinder feed (or working steam),

$$m_j = \frac{45.87}{1,150} = 0.399 \text{ kg}$$

3.3 Steam Boiler Trials

Boiler trials are carried out to determine the thermal efficiency of the boiler and to draw up the heat balance account. Boiler trials are also carried out to verify the guaranteed maximum evaporative capacity of the boiler. For complete boiler trial, it is necessary to measure losses in addition to the heat utilised in raising steam. Such trials have been the direct cause of and incentive to the improvement of boilers. The measurements necessary to determine the thermal efficiency of a boiler and to draw up the heat balance account for a boiler are :

- Rate of fuel consumption, i.e. mass of fuel burned/hr.,
- Calorific value and chemical analysis of fuel after proper sampling,
- Rate of water evaporation, i.e. mass of water evaporated/hr.,
- Pressure of steam at the boiler stop valve,
- Condition of steam at the boiler stop valve, i.e. dryness fraction of steam if there is no superheater, or temperature of steam if there is a superheater,
- Feed water temperature,
- Flue gases temperature,
- Analysis of flue gases,
- Mass of ashes and determination of their calorific value after proper sampling, and
- Measurement of pressure, temperature and humidity of air.

To obtain the best results from a trial on the steam boiler, special attention must be paid to the method of stoking. The method of starting and stopping the trial, and the duration of the trial are also of great importance. The decisions regarding these, entirely depend upon the conditions under which the boiler has to work.

The following method of starting and stopping the trial will generally be found the most convenient :

The boiler should be kept running on load for some time in order to get settled down to working conditions. About fifteen minutes before the trial commences, the fire should be cleaned and all ashes and clinker removed. *Then, at the start of the trial the thickness*

of the fire, the steam pressure and the temperature of the flue gases should immediately be noted. The water level be marked by tying a piece of string around the gauge glass, and the feed pump should be stopped. At the end of the trial, the thickness of the fire, the steam pressure and the temperature of the flue gases should be same as that at the start of the trial. If the water levels of the feed water tank and boiler water tank and boiler water gauge glass is the same at the end as at the start of the trial, the working of result is much simplified.

The duration of the trial will depend chiefly upon the magnitude of the error likely to be made in judging the thickness and condition of the fire at the start and end of the trial, as compared to the mass of fuel fired during the trial. *The duration of trial should not, as a rule, be less than six hours.*

The following measurements and readings should be taken to determine the thermal efficiency and to draw up the heat balance account for the boiler :

1. The fuel should be weighted out in convenient lots of, from 20 to 400 kg depending upon the capacity of the boiler. This should be done by using two boxes. At the time of commencement of the trial, the first lot (box) should be emptied on the floor and stoking commenced. A complete record of time of emptying the boxes may be kept on a log sheet.

2. A sample should be taken from every lot of fuel weighted out and towards the end of the trial the samples should be broken up into small pieces and well mixed, and two representative samples be taken – one for the determination of its calorific value (by using the Bomb calorimeter) and the other for chemical analysis.

3. Some sort of volumetric measurement is used for measuring the feed water supplied to the boiler. Various methods may be adopted for the purpose. For small size boilers, one feed tank is used, while for large size boilers, two tanks may be used, each fitted with a gauge-glass and accurately calibrated. Just before the trial commences the feed pump should be stopped and the water level in the feed tank should be marked and recorded if only one feed tank is used. The feed pump is then started. The difference in water levels at the start and end of the trial gives the amount of feed water used. In case if two tanks are used, just before the commencement of the trial, No. 1 tank should be filled up, the boiler being fed from No. 2 tank. At the beginning of the trial the boiler is fed from No. 1 tank and No. 2 tank being filled up. The number of refilling of tanks depends totally on the size of the boiler and the duration of the trial.

4. Readings should be taken every five minutes of the steam pressure gauge.

5. Measurements regarding condition of steam at the boiler stop valve are done exactly in similar manner as for condition of steam at engine stop valve described earlier in this chapter under steam engine trial.

6. The temperature of feed water supplied to the boiler is measured at regular time interval of 10 minutes by means of ordinary mercury glass thermometer. For the purpose of calculation of feed water temperature, average reading of the temperature is considered.

7. The temperature of the flue gases is most accurately measured by a pyrometer. This should be placed at the bottom of the chimney and near the damper on the chimney side. Readings are taken at regular time interval of 10 minutes and average value is taken into account for the purpose of calculation of heat carried away by the flue gases.

8. The sample of flue gases should be taken just on the chimney side of the damper at the same place at which the temperature of flue gases is measured. When the boiler under test is fired by mechanical stokers, the flue gas sample may be drawn directly into the analysing apparatus, but when firing is by hand, continuous collection is necessary

to secure an average sample. When great accuracy is required, the flue gases should be collected over mercury, but distilled water which has been saturated with common salt, or water with a layer of oil on the top, will give results accurate enough for most purposes. The flue gas is conveniently analysed on the spot by means of Orsat apparatus described in chapter 7 of volume I.

9. The amount of ashes formed during the trial period is obtained by weighing the ashes formed in the ash pit at the end of trial. As in the case of fuel, representative sample of the ash is obtained and its calorific value is determined by using the Bomb calorimeter.

10. Average values of temperature and pressure of the boiler house are obtained by reading a thermometer and a barometer at regular time interval of 10 minutes. In order to estimate the humidity (moisture) in the air of the boiler house, readings of the dry and wet bulb thermometers are taken at regular time interval.

3.4 Efficiency of Boiler

The thermal efficiency of boiler is expressed by the ratio :

$$\frac{\text{Heat transferred to feed water in converting it into steam per kg of fuel}}{\text{Heat released by complete combustion of one kg of fuel}}$$

The available heat in one kilogram of fuel as fired will not be the calorific value of one kilogram of fuel, unless of course the fuel is dry. The moisture present in the fuel has to be evaporated and superheated to the temperature of the flue gases, and the amount of heat so utilised is lost. The effect of moisture in air supply may also have an appreciable effect on the performance of a boiler, as this moisture has to be heated, evaporated and superheated, the heat so utilised being lost in the flue gases. The amount of this heat may be estimated from the readings of the wet and dry bulb thermometers. *It will be found that the heat absorbed in superheating the water vapour (moisture) in the air is negligibly small and may be neglected.*

The actual available heat supplied to the boiler per kilogram of coal
 = calorific value of 1 kg of dry coal as fired - (heat absorbed by the moisture in 1 kg of dry coal as fired) - (heat absorbed by the moisture in the mass of air supplied per kg of coal).

The lower caloric value of fuel was formerly used in estimating the boiler thermal efficiency, but now the gross value (higher calorific value) is recommended.

3.5 Heat Balance Sheet for Boiler

The various items of the heat balance sheet for a boiler test are as follows :

Heat balance sheet per kg of coal fired

Heat supplied by 1 kg of coal	kJ	%	Heat expenditure per kg of coal	kJ	%
Heat supplied	(1) Heat utilized in steam formation
			(2) Heat carried away by products of combustion (dry flue gases)
			(3) Heat carried away by excess air
			(4) Heat lost in evaporating and superheating the moisture in the coal and the water vapour formed due to burning of hydrogen in coal
			(5) Heat lost by incomplete combustion

			(6) Heat lost by unburnt carbon in ash
			(7) Unmeasured losses such as those due to radiation, escape of unburnt hydrocarbons, superheating the moisture in air, losses in hot ashes, error of observation, etc. (by difference)
Total			Total

The effect of transferring the heat loss due to moisture in the coal from the debit (heat expenditure) side to the credit (heat supplied) side of the heat balance sheet is to raise the efficiency slightly, and for practical purpose the difference may be considered very small.

The method of estimating the various items of heat balance sheet is illustrated by the following problems :

Problem-7 : *The following data was obtained in a steam boiler trial :*

Feed water supplied per hour 690 kg at 28°C, steam produced 0.97 dry at 8 bar, coal fired per hour 91 kg of calorific value 27,200 kJ/kg, ash and unburnt coal collected from beneath the fire bars 7.5 kg/hour of calorific value 2,760 kJ/kg, mass of flue gases per kg of coal burnt 17.3 kg, temperature of flue gases 325°C, room temperature 17°C, and the specific heat of the flue gases 1.026 kJ/kg K.

Estimate : (a) the boiler efficiency (b) the percentage heat carried away by the flue gases, (c) the percentage heat loss in ashes, and (d) the percentage heat loss unaccounted for.

Explain what may have actually happened to the heat included under unaccounted losses.

(a) Heat supplied to the boiler per hour = $91 \times 27,200 = 24,75,200$ kJ/hr.

At 8 bar, $h = 721.1$ kJ/kg, $L = 2,048$ kJ/kg (from steam tables).

Enthalpy of wet steam, = $H_1 = h_1 + xL_1 = 721.1 + 0.97 \times 2,048 = 2,707.67$ kJ/kg.

Enthalpy of feed water at 28°C, $h_2 = 28 \times 4.187 = 117.24$ kJ/kg.

\therefore Heat utilised in steam formation per hour

$$= \text{mass of steam produced per hr.} \times (H_1 - h_2)$$

$$= 690 (2,707.67 - 117.24) = 17,87,396 \text{ kJ/hr.}$$

Boiler efficiency = $\frac{\text{Heat utilized in steam formation per hr.}}{\text{Heat supplied to the boiler per hr.}}$

$$= \frac{17,87,396}{24,75,200} = 0.7221 \text{ or } 72.21\%$$

(b) Heat carried away by the flue gases = $m_g \times K_p (t_g - t_r)$

where, m_g = mass of flue gases = 17.3 kg/kg of coal fired,

t_g = temperature of flue gases = 325°C,

t_r = room temperature = 17°C, and

K_p = specific heat of flue gases = 1.026 kJ/kg K.

\therefore Heat carried away by the flue gases

$$= 17.3 \times 1.026 (325 - 17) = 5,467 \text{ kJ/kg of coal.}$$

Hence, percentage heat carried away by the flue gases per kg of coal fired

$$= \frac{5,467}{27,200} \times 100 = 20.1\%$$

$$\begin{aligned} \text{(c) Percentage heat loss in ashes} &= \frac{\text{Heating value of ash in kJ/hr.}}{\text{Heat supplied to the boiler in kJ/hr.}} \\ &= \frac{7.5 \times 2,760}{24,75,200} \times 100 = 0.836\% \end{aligned}$$

$$\begin{aligned} \text{(d) Percentage heat loss unaccounted for (by difference)} \\ &= 100 - (72.21 + 20.1 + 0.836) = 6.854\% \end{aligned}$$

Heat loss unaccounted for, includes error of observation and unmeasured losses such as those due to radiation, escape of unburnt hydrocarbons, superheating of moisture in air and coal, loss in hot ashes, etc.

Problem-8 : *The following particulars refer to a boiler trial in which it was not convenient to measure the amount of water evaporated :*

Percentage analysis of dry coal on mass basis : C, 85.5; H₂, 3.9; O₂, 3.6; Ash 7

Percentage analysis of dry flue gases by volume : CO₂, 11.6; CO, 0.6; O₂, 8; N₂, 79.8

Percentage analysis of ash collected in ash pit : C, 15; Ash, 85

Higher C.V. of dry coal per kg ... 33,900 kJ

Moisture in coal as fired ... 2 %

Temperature of the flue gases ... 310°C

Temperature of the boiler room ... 24°C.

Mean specific heat of dry flue gases ... 1.026 kJ/kg K

Specific heat of air ... 0.997 kJ/kg K

Barometric pressure (atmospheric) ... 1 bar

Specific heat of superheated water vapour ... 2 kJ/kg K

Calorific value of carbon burnt to CO₂ per kg ... 34,125 kJ

Calorific value of carbon burnt to CO per kg ... 10,175 kJ

Assuming a radiation loss of 7 per cent, draw up a heat balance sheet for the boiler and determine its thermal efficiency.

Considering one kg of coal as fired,

Gross heat supplied = 0.98 × 33,900 = 33,222 kJ per kg.

Heat expenditure per kg of coal as fired :

(2) As one kg of coal fired contains 0.07 kg of ash, the mass of carbon in association

$$\text{with 0.07 kg of ash in the ash pit is} = 0.07 \times \frac{15}{85} = 0.0123 \text{ kg.}$$

∴ Mass of carbon taking part in combustion per kg of coal fired
= (0.855 × 0.98) × 0.0123 = 0.827 kg. i.e. 82.7% on mass basis,

$$\text{Mass of air supplied per kg of coal fired} = \frac{NC}{33(C_1 + C_2)}$$

where, N, C₁ and C₂ are percentages of Nitrogen, Carbon dioxide and Carbon monoxide by volume in flue gas, and C is the percentage of carbon in coal on mass basis.

$$\therefore \text{Mass of air supplied} = \frac{79.8 \times 82.7}{33(11.6 + 0.6)} = 16.4 \text{ kg per kg of coal fired.}$$

Total mass of flue gases per kg of coal fired = $16.4 + (1 - 0.07) = 17.33$ kg

Minimum quantity of air theoretically required per kg of coal

$$= (2.66C + 8H) \frac{100}{23} = (2.66 \times 0.827 + 8 \times 0.039 \times 0.98) 4.35 = 10.45 \text{ kg.}$$

\therefore Excess air supplied per kg of coal fired = $16.4 - 10.45 = 5.95$ kg.

Mass (m_s) of moisture and water vapour per kg of coal fired.

$$= 0.02 + (9 \times 0.039 \times 0.98) = 0.371 \text{ kg.}$$

Mass (m_g) of dry products of combustion per kg of coal fired

$$= 17.33 - 5.95 - 0.371 = 11.01 \text{ kg.}$$

\therefore Heat carried away by dry products of combustion = $m_g \times K_p \times (t_g - t_f)$

$$= 11.01 \times 1.026 (310 - 24) = 3,230.6 \text{ kJ/kg of coal fired.}$$

(3) Heat carried away by excess air

$$= 5.95 \times 0.997 (310 - 24) = 1,696.5 \text{ kJ/kg of coal fired.}$$

(4) Heat carried away by water vapour in the products of combustion =

$$m_s \times (H_{sup} - h_o)$$

$$= 0.371 [2,675.5 + 2(310 - 99.63) - 24 \times 4.187] = 1,112 \text{ kJ per kg of coal fired.}$$

(5) We next require the proportion of carbon burned to CO_2 and CO respectively.

In 11.6×44 parts of CO_2 on mass basis, there are $11.6 \times 44 \times \frac{12}{44} = 11.6 \times 12$ parts of carbon on mass basis.

\therefore In 0.6×28 parts of CO on mass basis, there are $0.6 \times 28 \times \frac{12}{28} = 0.6 \times 12$ parts of carbon on mass basis.

\therefore Proportion of carbon burnt to CO = $\frac{0.6 \times 12}{(11.6 + 0.6) 12} = 0.0492$

\therefore Mass of carbon burnt to CO in one kg of coal fired

$$= 0.855 \times 0.98 \times 0.0492 = 0.0412 \text{ kg}$$

Hence, heat lost through incomplete combustion per kg of coal fired

$$= 0.0412 (34,125 - 10,175) = 986.7 \text{ kJ per kg of coal fired}$$

(6) Heat carried away by unburnt carbon in the ash pit per kg of coal fired

$$= 0.0123 \times 34,125 = 419.7 \text{ kJ per kg of coal fired}$$

(7) Heat lost by radiation (assumed) = $0.07 \times 33,222 = 2,325.5$ kJ per kg of coal fired.

\therefore Total heat loss = $3,230.6 + 1,696.5 + 1,112.0 + 986.7 + 419.7 + 2,325.5$
 $= 9,771$ kJ/kg of coal fired

(1) Thus, heat utilized in steam formation per kg of coal fired (by difference)

$$= 33,222 - 9,771 = 23,451 \text{ kJ per kg of coal fired}$$

\therefore Thermal efficiency of the boiler

$$= \frac{\text{Heat utilized in steam formation per kg of coal}}{\text{Gross heat supplied per kg of coal}}$$

$$= \frac{23,451}{33,222} = 0.7059 \text{ or } 70.59\%$$

Heat balance sheet per kg of coal fired

Heat supplied by 1 kg of coal	kJ	Heat expenditure/kg of coal	kJ
Heat supplied	33,222	(1) Heat utilized in steam formation (by difference)	23,451.0
		(2) Heat carried away by dry products of combustion	3,230.6
		(3) Heat carried away by excess air	1,696.5
		(4) Heat lost in evaporating and superheated moisture in coal and water formed by combustion of hydrogen in coal	1,112.0
		(5) Heat lost through incomplete combustion	986.7
		(6) Heat carried away by unburnt carbon in ash pit	419.7
		(7) Heat lost by radiation, error, etc. (assumed)	2,325.5
Total	32,222	Total	33,222

Problem-9 : The following data was obtained during a trial of a steam boiler :

Feed water temperature, 75°C ; mass of feed water supplied per hour, 4,900 kg; steam pressure, 11 bar; dryness fraction of steam, 0.9; coal fired per hour, 490 kg; higher calorific value of 1 kg of dry coal, 35,600 kJ/kg; moisture in coal, 4% on mass basis; temperature of flue gases, 300°C ; boiler house temperature, 16°C ; barometric (atmospheric) pressure 1 bar; analysis of dry coal on mass basis, C = 89%; H_2 = 3%; ash = 4%; and other matter = 4%; analysis of flue gases by volume, CO_2 = 10.9%; CO = 1.1%; O_2 = 7% and N_2 = 81%. Take specific heat of dry flue gases as 1 kJ/kg K and K_p of superheated steam as 2 kJ/kg K. Draw the heat balance sheet for the boiler per kg of coal fired. What is the thermal efficiency of the boiler ?

Heat supplied per kg of coal = $(1 - 0.04) 35,600 = 34,176$ kJ per kg of coal

At 11 bar, $h = 781.34$ kJ/kg, $L = 2,000.4$ kJ/kg (from steam tables)

(1) Heat utilized per kg of steam at 11 bar and 0.9 dry = $H_1 - h_2$

$$= (h_1 + x_1 L_1) - h_2 = (781.34 + 0.9 \times 2,000.4) - 75 \times 4.187 = 2,267.67 \text{ kJ/kg.}$$

Hence, heat utilized per kg of coal fired

$$= \frac{4,900}{490} \times 2,267.67 = 22,676.7 \text{ kJ/kg of coal.}$$

(2) Mass of air supplied per kg of coal fired = $\frac{NC}{33(C_1 + C_2)}$

where, N , C_1 and C_2 are percentages of nitrogen, carbon dioxide and carbon monoxide by volume in flue gases, and C is the percentage of carbon in coal on mass basis.

$$\therefore \text{Mass of air supplied} = \frac{81 \times (89 \times 0.96)}{33(10.9 + 1.1)} = 17.7 \text{ kg/kg of coal fired}$$

Mass of dry flue gases per kg of coal fired, $m_g = 17.7 + (0.89 \times 0.96) = 18.57$ kg.

Thus, heat carried away by dry flue gases per kg of coal fired

$$= m_g \times K_p (t_g - t_r) = 18.57 \times 1 \times (300 - 16) = 5,273.88 \text{ kJ/kg of coal fired.}$$

(3) Mass of moisture in coal and water vapor formed due to combustion of hydrogen in coal per kg of coal fired

$$= m_s = m + (9 \text{ H}_2) = 0.04 + (9 \times 0.03) = 0.31 \text{ kg.}$$

Heat carried away by moisture in coal and water vapour formed in flue gases due to burning of hydrogen in the coal = $m_s (H_{\text{sup}} - h_0)$

where, m_s = mass of steam (0.31 kg) formed per kg of coal fired,

H_{sup} = enthalpy of 1 kg of superheated steam at the temperature of flue gases (300°C) and at atmospheric pressure of 1 bar, and

h_0 = Enthalpy (sensible heat) of 1 kg of water at the boiler room temperature (16°C).

Now, $m_s \times (H_{sup} - h_0) = m_s \{ [H_s + K_p (t_{sup} - t)] - h_0 \}$

= 0.31 [{2,675.5 + 2 (300 - 99.63)} - 16 x 4.187] = 932.87 kJ/kg of coal fired.

(4) Heat lost by radiation, error, etc. (by diff.)

= 34,176 - (22,676.7 + 5,273.88 + 932.87) = 5,292.55 kJ/kg of coal fired.

Heat balance sheet per kg of coal fired

Heat supplied by 1 kg of coal	kJ	Heat expenditure per kg of coal	kJ
Heat supplied	34,176	(1) Heat utilized in steam formation	22,676.7
		(2) Heat carried away by dry flue gases.	5,273.88
		(3) Heat lost in evaporating and superheating moisture in coal and water vapour formed due to combustion of hydrogen in coal.	932.87
		(4) Unmeasured losses such as those due to radiation, escape of unburnt hydrocarbons, losses in hot ashes error of observation etc. (by difference)	5,292.55
Total	34,176	Total	34,176

Thermal efficiency of the boiler = $\frac{\text{Heat utilized in steam formation per kg of coal}}{\text{Gross heat supplied per kg of coal}}$

= $\frac{22,676.7}{34,176} = 0.6638$ or **66.38%**

Tutorial - 3

1. Delete the phrase which is not applicable from the following statements :

- (i) Steam engines are generally single/double acting.
- (ii) Live steam from the boiler/exhaust steam is passed through cylinder jacket.
- (iii) Indicated power of a steam engine is estimated by taking indicator diagram/ using a dynamometer.
- (iv) Mean effective pressure on cover end and crank end of a double-acting steam engine are/are not the same.
- (v) Thermal efficiency of steam engine is/is not improved when heat of jacket drain is also available to boiler feed water in addition to heat of condensate.

[(i) Single, (i) Exhaust steam, (iii) Using a dynamometer, (iv) are, (v) is not]

2. Fill in the blanks to complete the following statements :

- (i) Brake specific steam consumption is expressed as _____ .
- (ii) The rate of steam consumption of a condensing engine is best measured by condensing the exhaust steam and _____ the condensate.
- (iii) Thermal efficiency of a boiler is defined as the ratio of heat utilized in steam formation per kg of coal and _____ .
- (iv) Boiler house instruments may be broadly divided into : (a) those which give information about performance, and (b) those which help — the performance.

[(kg/kW-hr, (ii) weighing, (iii) gross heat supplied per kg of coal, (iv) to control]

3. Indicate the correct phrase out of phrases given in the following :

- (i) Thermal efficiency of a well maintained boiler will be of the order of
(a) 20% (b) 40% (c) 60% (d) 75% (e). 90%
- (ii) Maximum energy loss in a boiler occurs due to

- (a) flue gases (b) ash content (c) radiation losses (d) incomplete combustion.
- (iii) The temperature of the flue gases is most accurately measured by
(a) a thermometer (b) a thermocouple (c) a pyrometer.
- (iv) The object of steam boiler trial is
(a) to estimate steam raising capacity of the boiler when working at a definite pressure.
(b) to determine the thermal efficiency of the boiler when working at a definite pressure.
(c) to draw up a heat balance sheet for the boiler.
(d) all the three objects mentioned above.
- [(i) e, (ii) a, (iii) c, (iv) d]
4. In a test on single-cylinder, double-acting condensing steam engine, the following observations were made :
Indicated power, 24 KW; brake power; 20 kW; steam supply at 5.5 bar with 12°C superheat; K_p of superheated steam, 2.3 kJ/kg K; condenser vacuum, 647.5 mm of Hg when barometer reads 760 mm of Hg; steam used per hour, 300 kg; temperature of condensate; 40°C; cooling water for condenser, 7,800 kg per hour; temperature rise of condenser cooling water, 20°C.
Estimate : (a) the indicated thermal efficiency and (b) the dryness fraction of steam entering the condenser. Assume that all the heat given up by steam in the condenser is gained by condenser cooling water.
Also draw up a heat balance sheet for the engine in kJ/min.

[(a) 11.02%; (b) 0.893]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat in steam supplied	13,903	(1) To brake power	1,200
		(2) To condenser cooling water	10,886.2
		(3) To condensate	837.4
		(4) To radiation etc.	979.4
Total	13,903	Total	13,903.0

5. The following data relate to a test on a compound double-acting steam engine :

Cylinder	Cylinder diameter cm	Stroke cm	M.E.P. bar	Inlet steam		Exhaust steam pressure bar
				Pressure bar	Temperature °C	
H. P.	21	15	3	14	205	—
L. P.	33	15	0.9	—	—	0.5

Steam consumption 9.3 kg/min.; condensate temperature, 66°C; speed 550 r.p.m.; brake torque, 750 N.m; condenser cooling water flow, 470 kg/min.; temperature rise of condenser cooling water, 10°C.

Ignoring piston rod areas, calculate : (a) the indicated power, (b) the mechanical efficiency, and (c) the efficiency ratio on the indicated power basis. Also draw up a heat balance sheet in kJ/minute.

[49.743 kW; (b) 86.84% (c) 56.72%

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat in steam supplied	26,140.9	(1) To brake power	2,591.8
		(2) To condenser cooling water	19,678.9
		(3) To condensate	2,570.0
		(4) To radiation etc.	1,300.2
Total	26,140.9	Total	26,140.9

6. The following observation were made during a trial of jacketed steam engine :

Supply steam pressure	... 13 bar
Dryness fraction of steam supplied	... 0.98
Mass of steam supplied to the cylinder	... 20 kg/min.
Mass of steam supplied to the cylinder jacket	... 2.5 kg/min.
Brake power	... 130 kW
Indicated power	... 160 kW
Condenser cooling water	... 450 kg/min.
Rise in temperature of condenser cooling water	... 22°C

Temperature of condensate ... 50°C
 Temperature of cylinder jacket drain ... 185°C

Draw up a heat balance sheet in kJ/min. and determine the indicated thermal efficiency of the engine assuming that the heat of condensate and cylinder jacket drain is available to the feed water.

[$\eta_i = 17.23\%$]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat in steam supplied	61,834	(1) To Brake power	7,800
		(2) To condenser cooling water	41,451
		(3) To condensate	4,187
		(4) To cylinder jacket drain	1,937
		(5) To radiation etc.	6,459
Total	61,834	Total	61,834

7. During a test on a Jacketed steam engine, the following observations were made :

Indicated power, 90 KW; brake power, 72 kW; pressure of steam supplied 14 bar; quality of steam supplied 10% wet; mass of steam used in the engine cylinder 900 kg/ hour; mass of steam used in Jacket, 100 kg/hour; condensate temperature, 60°C; cooling water for condenser, 15,500 kg/hour; inlet temperature of cooling water for condenser, 26°C; outlet temperature of cooling water for condenser, 55°C. Taking specific heat of water as 4.187 kJ/kg K, draw a heat balance sheet in kJ per minute and on percentage basis. Calculate the indicated thermal efficiency when the heat of Jacket drain is not available to the boiler feed water. What will be the percentage improvement in the indicated thermal efficiency if the heat of the Jacket drain is also available to the feed water ?

[13.68%; 3.65%

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat in steam supplied	43,233.83	100	(1) To brake power	4,320	9.99
			(2) To condenser cooling water	31,360	72.54
			(3) To condensate	3,768.3	8.72
			(4) To Jacket drains	1,383.83	3.2
			(5) To radiation error etc.	2,401.7	5.55
Total	43,233.83	100	Total	43,233.83	100

8. The following observations were made in a trial on a jacketed, double-acting compound steam engine supplied with dry saturated steam at 11 bar :

Cylinder	Piston stroke	Cylinder diameter	Mean effective pressure
H. P.	60 cm	23 cm	2.8 bar
L.P.	60 cm	40 cm	1.1 bar

Speed, 87.5 r.p.m.; Brake torque, 4,300 N.m; Water from cylinder jacket drain, 1 kg/minute; Condensate, 8.5 kg/minute; Temperature of condensate, 45°C; Condenser cooling water, 160 kg/minute; Rise in temperature of condenser cooling water, 30°C.

Calculate : the brake power, the indicated power, the brake thermal efficiency, and indicated thermal efficiency assuming the heat of condensate and cylinder jacket drain is available to the feed water, and make out a heat balance sheet in kJ/min.

[Brake power = 39.4 kW; Indicated power = 44.548 kW;
 $\eta_b = 9.83\%$; $\eta_i = 11.12\%$

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat in steam supplied	26,426	(1) To brake power (useful work)	2,364
		(2) To condenser cooling water	20,098
		(3) To condensate	1,601.5
		(4) To cylinder jacket drain	781.34
		(5) To radiation, etc.	1,581.16
Total	26,426	Total	26,426

9. The following are the average readings taken during a trial on a double-acting steam engine :

Stroke, 30 cm; Cylinder diameter, 21.6 cm; Mean speed, 123.9 r.p.m.; Area of indicator diagram, 6.52 cm²; Length of indicator diagram, 6.6 cm; Strength of indicator spring, 85 kPa per cm; Coal consumption, 0.252 kg/min.; Calorific value of coal, 35,200 kJ per kg; Condenser cooling water, 46.2 kg/min.; Rise in temperature of condenser cooling water, 28.9°C; Load on brake, 457 newtons; Spring balance reading, 66 newtons; Radius of brake wheel, 60 cm; Steam used per min., 2.51 kg; Steam pressure, 3 bar; steam supply, dry saturated; Condenser pressure, 0.2 bar.

Calculate the indicated power, brake power, mechanical and indicated thermal efficiencies of the engine, and overall efficiency of the steam plant from coal to brake. Draw up a percentage heat balance sheet for the engine cylinder.

[indicated power = 3.814 kW; Brake power = 3.044 kW,
 $\eta_m = 78.76\%$; $\eta_l = 3.68\%$; 2.06%

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ.	%
Heat in steam supplied	6,840.5	%	(1) To Brake power	182.6	2.67
			(2) To condensing cooling water	5,590.4	81.72
			(3) To condensate	626.4	9.16
			(4) To radiation etc.	441.1	6.45
Total	6,840.5	100	Total	6,840.5	100

10. What are the purposes of steam engine trials ? What measurements are necessary in engine trials to determine the thermal efficiency and to draw up heat balance sheet ? Draw up a typical heat balance sheet on percentage basis of an average steam engine.

11. The following observations and deductions are taken from a report of a trial of a boiler plant, consisting of six Lancashire boilers and an economiser :

Calorific value of coal per kg., 30,000 kJ; mass of feed water per kg of dry coal, 9.1 kg; Equivalent evaporation from and at 100°C per kg of dry coal, 9.6 kg; Temperature of feed water to economiser, 12°C; Temperature of feed water to boiler, 105°C; Air temperature, 13°C; Temperature of flue gases entering the economiser, 370°C; Mass of flue gases entering economiser 18.2 kg per kg of dry coal; Mean specific heat of the flue gases, 1.05 kJ/kg K.

Find : (a) the efficiency of the boilers alone, (b) the efficiency of the economiser alone, and (c) the efficiency of the whole boiler plant.

[(a) 72.23%; (b) 51.94% (c) 84.04%]

12. In a boiler trial the following quantities were obtained : Coal burned per hour, 48 kg; Calorific value of coal, 31,200 kJ/kg. Feed water per hour, 387 kg; Temperature of feed water, 20°C; Pressure of steam, 8.5 bar; Dryness fraction of steam, 0.99; Ash and unburnt coal collected from beneath fire bars, 4 kg/hour of the calorific value 2,850 kJ/kg; Mass of flue gases per kg of coal burned, 17.3 kg; Temperature of flue gases; 340°C; Room temperature, 16°C; Specific heat of flue gases, 1.026 kJ/kg K.

Estimate : (a) the thermal efficiency of the boiler, (b) the percentage heat carried away by the flue gases, (c) the percentage heat loss in ashes, and (d) the percentage heat loss unaccounted for.

[(a) 68.93%; (b) 18.43% (c) 0.76% (d) 11.88%]

13. In a boiler trial, 445 kg of coal were consumed per hour. The mass of water evaporated per hour was 4,150 kg. The steam pressure was 10 bar and dryness fraction of steam was 0.98. The coal contained 4 per cent of moisture on mass basis. The feed water temperature was 50°C. Calorific value of one kilogram of dry coal was 35,000 kJ. The boiler house temperature was 15°C and the temperature of the chimney gases was 280°C. Take specific heat of dry flue gases as 1.005 kJ/kg K and K_p of superheated steam as 2 kJ/kg K.

Analysis of dry coal on mass basis : C = 86%; H_2 = 4% ash = 5%; and other matter = 5%.

Analysis of dry flue gases by volume :

CO_2 = 10.4%; CO = 1.2%; O_2 = 9.1%; and N_2 = 79.3 (by difference)

Determine the thermal efficiency of the boiler and draw up a heat balance sheet for the boiler per kg of coal fired on percentage basis.

[70.18%

Heat supplied by 1 kg of coal	kJ	%	Heat expenditure per kg of coal	kJ	%
Heat supplied	33,600	100	(1) To steam	23,580	70.18
			(2) To dry flue gases	4,774	14.21
			(3) To water vapour in flue gases	654	1.95
			(4) To radiation unmeasured losses etc. (By difference)	4,592	13.66
Total	33,600	100	Total	33,600	100

14. The following particulars relate to a boiler trial in which it was not convenient to measure the amount of water evaporated :

Percentage analysis of dry coal on mass basis : C, 83; H₂, 5; O₂, 4; Ash, etc., 8

Percentage analysis of dry flue gases by volume :

CO₂, 10.1; CO, 0.3; O₂, 9.3 and N₂ (by difference), 80.3

Percentage analysis of ash collected in ash pit : C, 14; ash, 86

Higher calorific value of dry coal per kg ... 34,000 kJ

Moisture in coal as burned ... 2%

Temperature of the flue gases ... 330°C

Temperature of boiler room ... 17°C;

Mean specific heat of flue gases ... 1.005 kJ/kg K

Specific heat of air ... 1 kJ/kg K

Specific heat of superheated water vapour ... 2 kJ/kg K

Calorific value of C burnt to CO₂ per kg ... 33,830 kJ

Calorific value of C burnt to CO per kg ... 10,130 kJ

Assuming radiation loss of 6%, draw up a percentage heat balance sheet for the boiler and determine its thermal efficiency.

[Thermal efficiency = 68.68%

Heat supplied by 1 kg of dry coal	kJ	%	Heat expenditure per kg of coal	kJ	%
Heat supplied	33,320	100	(1) Steam (by difference)	22,884	68.68
			(2) Dry flue gases	3,605	10.82
			(3) Water vapour in flue gases	1,413	4.24
			(4) Excess air	2,423	7.27
			(5) Incomplete combustion	556	1.67
			(6) Unburnt carbon in ash	440	1.32
			(7) Radiation (assumed)	1,999	6.00
Total	33,320	100	Total	33,320	100

4

MODERN STEAM GENERATORS

4.1 Introduction

Boiler is a container into which water is fed, and by the application of heat, it is evaporated into steam. In early designs, the boiler was a simple shell with a feed pipe and steam outlet, mounted on a brick setting. Fuel was burnt on a grate within the setting and the heat so released was directed over the lower shell surface before most of it went out.

Soon the designers realized that heating a single shell is inefficient and it was necessary to bring more of water into close contact with heat. One way, is to direct flue gases through tubes in the boiler shell. Such a 'fire-tube design' not only increases the heating surface but also distributes area of steam formation more uniformly.

Second way is water-tube design. It consists of one or more relatively small drums with number of tubes in which water-steam mixture circulates. Heat flows from flue gases outside tubes to the mixture. Thus sub-division of pressure parts make possible construction of large capacity and high pressure boilers.

Fire-tube boilers and simple water-tube boilers are described in detail in chapter-8 of volume-I. Fire-tube boilers are limited to a maximum design working pressure of 25 bar and steam generating capacity of 25 tonnes per hour. Conventional water-tubes boilers work upto steam pressures of about 70 bar and 250°C superheat with a steam generating capacity of 40 tonnes per hour.

Shell or fire-tube boilers are cheaper than water-tube boilers but they are suitable for low pressures and low output. There is no such limit to water-tube boilers. Water-tube boiler can be erected at site from easily transportable parts. They are flexible from constructional point of view. They are capable of quick steam generation and their constructional design can be varied to suit a wide range of situations. Furnace of a water-tube boiler is not limited to cylindrical form. Therefore, water-tube boilers are generally preferred for high pressure high duty performance.

The present-day demand for higher power outputs from the thermal power plants requires high pressure high duty boilers. A high pressure boiler is much more than an assembly of certain components like burners, superheaters, air heaters, etc. The functions of these components are inter-related. The quality of coal and the operating conditions have a great influence on the types of components to be selected and more than that, they influence the philosophy underlying the general design. For increasing the steam pressure and the rate of steam generation of a boiler, forced circulation of water and/or steam and radiant heat transfer from the furnace to the water were considered essential.

4.2 Water-Tube Boilers

Introduction of water-tube boilers dates back to the eighteenth century. The last twenty five years have been a period of significant change in their design and construction.

Horizontal water-tube boilers with vertical or slightly inclined sectional headers having a longitudinal or transverse drum (fig.4-1), were quite popular during first quarter of the twentieth century. Now-a-days they are not built, as they can not cope up with high pressure and high duties demanded from modern boilers.

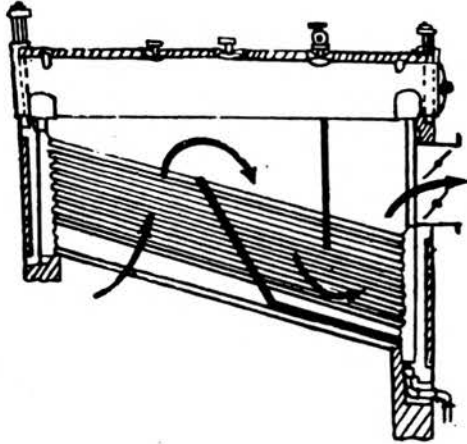


Fig. 4-1. Horizontal water-tube boiler.

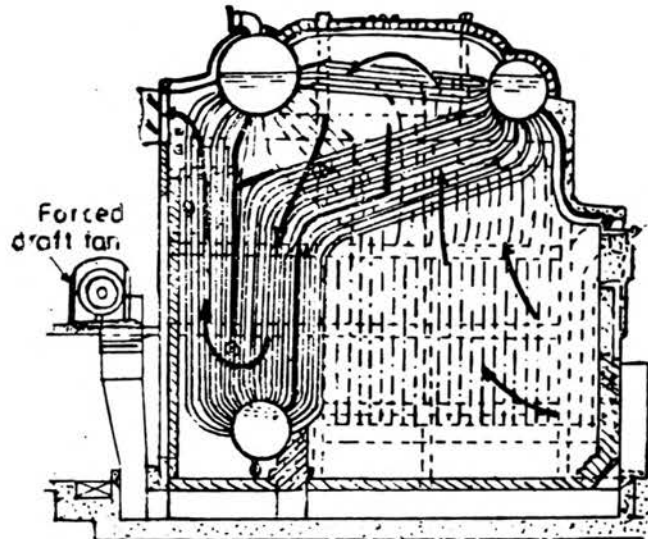


Fig. 4-2. Bent tube boiler.

Bent tube boilers (fig. 4-2) are more flexible in construction. Where head room is limited, they can be made wide and low or narrow, and high where floor space is limited. Thus, their overall dimensions can be adjusted according to the space available.

As the demand for large capacity and high pressure boilers grew, the demand for more active furnace cooling methods increased. Water cooled furnace walls were developed because of increasing rate of heat transfer in furnace proper. Water from drum is supplied to lower header as shown in fig. 4-2. Steam is actively generated in walls, to rise to upper drum where it separates from boiler water.

In a simple water-tube circuit, steam bubbles are formed on the heated side. The resulting steam-water mixture weighs less than cooler water on the unheated side and thus free convective currents (circulation) are established. In drum, steam bubbles rise to water surface and steam is generated in this manner. Free circulative currents are affected by two factors :

- Difference in density between water and steam-water mixture, and
- Frictional forces opposing circulation.

At a higher pressure, the effect of first factor reduces and thus forced circulation is inevitable. Also the forced circulation increases the rate of heat transfer thus permitting higher rate of steam generation and reduction in overall size of the boiler. Thus, large capacity boilers are possible. Even recently, designers have gone one step further to increase the boiler capacity by adopting once-through boiler. It consists of a single tube (no drum) into which goes feed water and out of which comes saturated or superheated steam. In actual units (boilers), the theoretical single circuit becomes a number of parallel circuits. At pressures below critical, a "once through unit" may have a separator to deliver saturated steam to the superheater and to return collected moisture to the feed pump suction.

The "once through" cycle is, of course, ideally suited for pressures above critical point where water turns to steam without actually boiling. At critical pressure, the density of

water and steam is same and hence natural convective flow does not take place at critical pressure. Thus, the use of natural circulation is limited to sub-critical boilers upto about 140 bar boiler pressure and use of forced circulation becomes essential for higher pressures. High water velocities rather than high gas velocities are suitable, as a smaller quantity of fluid is dealt with and increase in pressure can be more easily attained than gas. Hence, the tubes of smaller diameter may be used for a boiler of a given output.

If the flow takes place through one continuous tube, large pressure drop takes place due to friction. This can be reduced by arranging the flow to pass through parallel system of tubing.

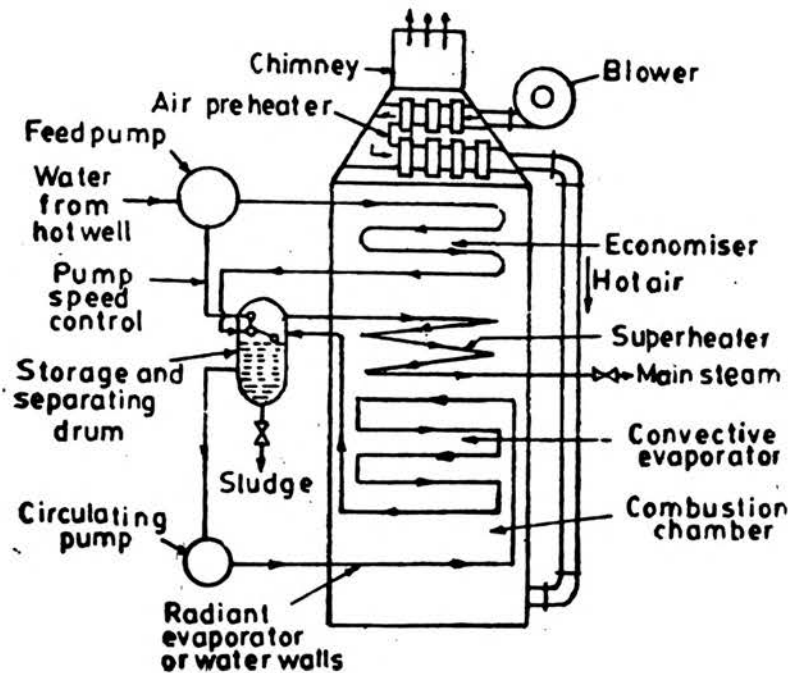


Fig. 4-3. La Mont boiler.

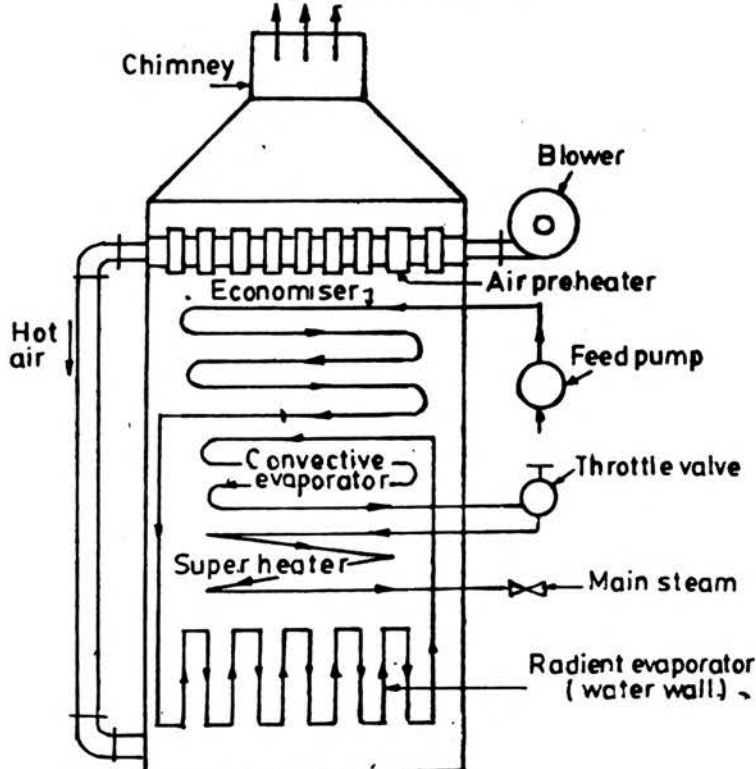


Fig. 4-4. Benson boiler.

The best examples of high pressure boilers are :

- La Mont boiler,
- Benson boiler,
- Loeffler boiler,
- Schmidt-Hartmann boiler, and
- Velox boiler.

LaMont boiler is a first forced circulation boiler introduced by LaMont in 1925. This boiler is of the water-tube type and is used in Europe and America. Water circulation and schematic location of different components of the boiler are shown in fig. 4-3.

This boiler incorporates water circulation in tubes surrounded by gases. Water is supplied through an economiser to a separating and storage drum which contains a feed regulator that controls the speed of the feed pump. Most of the sensible heat is supplied to the feed water passing through the economiser. From the drum a centrifugal pump circulates about 8 to 10 times the quantity of water evaporated. This large quantity of water circulated prevents the tubes from being overheated. The circulating pump passes water first to radiant evaporator or water wall (of which the sides for the combustion chamber are composed). Then steam and water

pass to convective evaporator and again to the drum. From the drum the released steam then passes to the superheater.

This boiler is capable of generating 40 to 50 tonnes of superheated steam per hour at about 500°C and 120 to 130 bar pressure.

This boiler has the advantage of flexibility of design, compactness and small size of drum. It generally resembles a natural circulation boiler. Formation and attachment of bubbles on the inner surfaces of the heating tubes of LaMount boiler reduces the heat flow and steam generation as it offers high thermal resistance than water film. Mark Benson argued that if the boiler pressure was raised to the critical value (220.9 bar), the steam and water would have the same density, and therefore danger of bubble formation can be eliminated

Fig-4-4 shows the layout sketch of a **Benson boiler**. Benson boilers are drumless or "once through" type. Feed water is pumped through the economiser, radiant and convective evaporators, and superheater. The boiler pressure is critical pressure and hence water turns to steam directly without actually boiling.

If distilled water is not used, heavy deposits of salt occur in the transformation zone from water into steam. To avoid this difficulty, the evaporator is flushed out after every 4,000 working hours to remove salt. Because of the reduced value of entropy at the critical pressure, the steam rapidly becomes wet when it is expanded in a turbine, thereby causing erosion of the blading. To obviate erosion and to provide a more moderate working pressure, the steam is throttled to a pressure of about 150 bar.

From the figure, it appears that the boiler consists of a single tube of great length, but actually it consists of many parallel circuits which yield a thermal efficiency of about 90%. Benson boilers of 150 tonnes of steam per hour generating capacity at 50 MPa (500 bar) pressure and 650°C temperature have been constructed and are in use. The main advantages of the Benson boiler are :

- Absence of drums reduces the total weight of boiler and hence low cost of transport,
- The boiler can be erected easily and quickly,
- Operation is economical, and
- Quick starting and can reach full capacity operation within 10 minutes from start.

Loeffler Boiler uses circulation of steam instead of water. Thus, the difficulty experienced in La Mont boiler, a deposition of salt and sediment in boiler tubes, is avoided. This boiler has advantages of forced circulation and indirect heating. In this boiler, steam is used as heat carrying and heat absorbing medium.

A line diagram of Loeffler boiler is shown in fig. 4-5. This boiler has economiser and superheater units in the path of gases from furnace to chimney. The

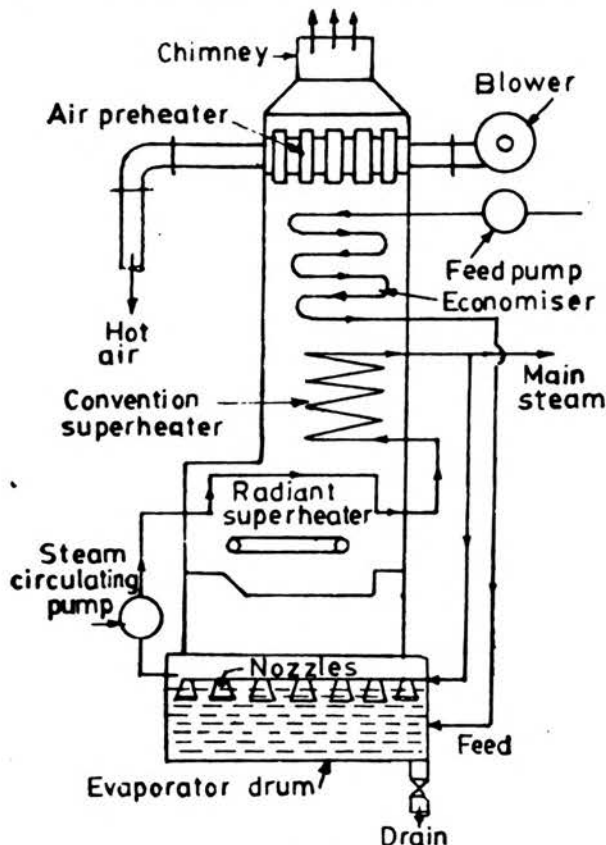


Fig. 4-5. Loeffler boiler.

evaporator drum is outside the boiler. A portion of main superheated steam (about 35%) is tapped off for external use, whilst the remainder passes on to the evaporator drum, where, by giving up its superheat to water coming from economiser, steam is generated equal to the steam tapped off. The steam circulating pump draws the saturated steam from the evaporator drum and is passed through the radiant and convective superheaters.

The nozzles distributing the superheated steam throughout the water in the drum are of special design to avoid priming and noise. This boiler can carry higher salt concentrations than any other type and is more compact than indirectly heated natural circulation boilers. These qualities make this boiler fit for land or sea transport power generation. Loeffle boilers of generating capacity 90 tonnes per hour and pressure 125 bar are in use.

Like Loeffler boiler, **Schmidt-Hartmann Boiler** is also high pressure indirectly heated boiler.

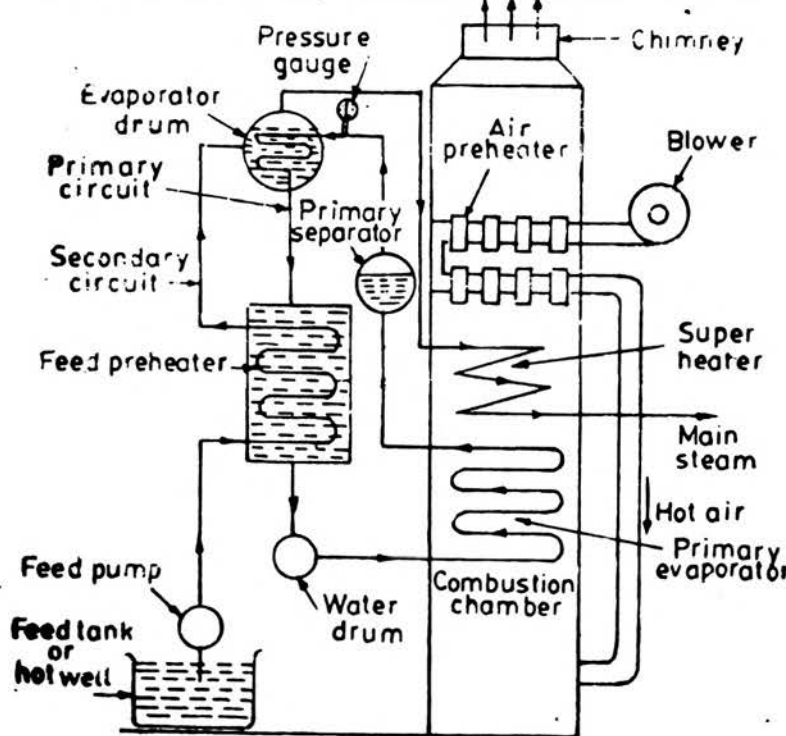


Fig. 4-6. Schmidt-Hartmann boiler.

The arrangement of the boiler components is shown in fig. 4-6. This boiler is very similar to an electric transformer. Two pressures are used to effect an interchange of energy. In the primary circuit, steam at pressure 100 bar is produced from distilled water. This steam is passed through submerged heating coil, located in the evaporator drum. The high pressure steam of primary circuit possesses sufficient thermal heat to produce steam at pressure 60 bar with a heat transfer rate of $2,900 \text{ watts/m}^2\text{°C}$. This main steam is passed through a superheater placed in the uptake and then to the application point. The condensate of high pressure primary circuit steam is circulated through the

water drum where feed water is heated to its saturation temperature.

In the primary circuit, natural circulation is used which is sufficient to produce the desired rate of heat transfer in conjunction with high gas velocities. In this way, circulating velocities of 0.5 to 0.8 metre per second for thermo-siphon head of about 2.5 to 10 metres are possible.

As a safeguard against leakage or the safety valve lifting, a combined pressure gauge and thermometer are fitted to the primary circuit. An arrangement is provided for making distilled water of the primary circuit. Main advantages of the Schmidt-Hartmann boiler are :

- ... Due to distilled water in the primary circuit, there is rare chance of overheating or burning of the high heated components as there is no danger of salt deposition.
- ... There is no chance of interruption to the circulation either by rust or other material, due to use of distilled water in the primary circuit.
- ... Feed water is external to the heating coil and hence it is easy to brush off salt deposits, just by removing the heating coil from the evaporator drum.
- ... Due to high thermal and water capacity, wide fluctuations of load are allowed without undue priming or abnormal increase in the primary pressure.

... The absence of water risers in the drum, and the moderate temperature difference across the heating coil, allows evaporation to proceed without priming.

When the velocity of gas exceeds the velocity of sound, the heat is transferred from the gas at a much greater rate than the rate achieved with subsonic flow. This fact is used in the **Velox Boiler** to achieve the large amount of heat transfer from the given surface area.

In the velox boiler, air is compressed to 2.5 bar pressure by an air compressor run by a gas turbine, before supplying to the combustion chamber as shown in fig. 4-7. The

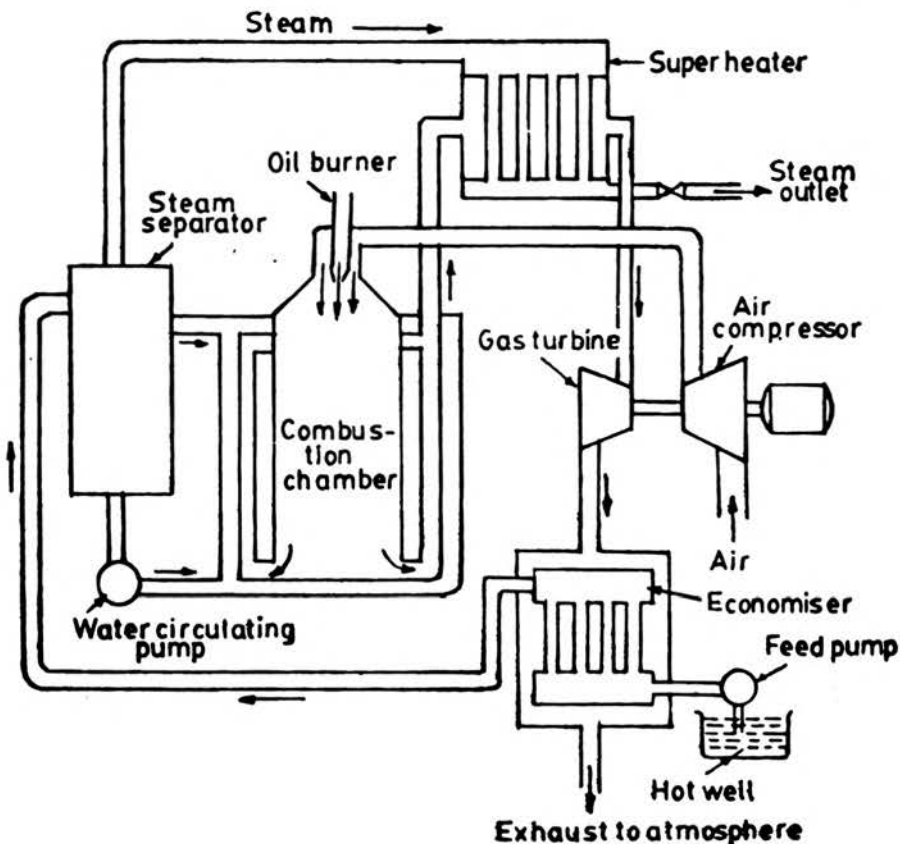


Fig. 4-7. Velox boiler.

object of this compression is to secure a supersonic velocity of the gases passing through the combustion chamber and gas tubes. As a result of this high rate of heat release (32 to 40 million kJ per m^3 of combustion chamber volume) is achieved and hence this boiler is a very compact one. The steam generating capacity of this boiler is limited to about 100 tonnes per hour because large power (brake power of about 4,400 kW) is required to run the air compressor at this output.

Fuel and air are injected downwards into a vertical combustion chamber which consists of annulus gas tubes and annulus water tubes (fire tube principle). On reaching the bottom of the combustion chamber, the products of combustion are deflected upwards into the evaporator tubes which consist of an outer annulus through which 10 to 20 times the water evaporated is circulated at a high velocity (this prevents the overheating of the metal walls). This way heat is transferred from gases to the water at a very high rate. The mixture of water and steam thus formed then passes into a separator from which the separated steam passes to the superheater and finally to the application point. The water removed from steam in the separator is again passed through the water tubes along with preheated feed water coming from economiser. The gases coming out from

the evaporator tubes are first passed over the superheater tubes and are then led to the gas turbine. The power output of the gas turbine is supplied to drive the compressor, and the exhaust gases coming out from the gas turbine are passed through the economiser before going to the atmosphere.

Advantages of Velox boiler over similar boiler are :

- ... Very high rate of heat transfer,
- ... Compact steam generating unit of great flexibility,
- ... Capable of quick starting, even though the separating drum has a storage capacity of about one-eighth of the maximum hourly output,
- ... Low excess air is required as compressed air is used and the problem of draught is simplified,
- ... The control is entirely automatic, and
- ... A thermal efficiency of about 90 to 95% is maintained over a wide range of load.

4.3 Materials of Construction

Modern boilers consist of steel tubes of various dimensions, shape and thickness. The material used is of great importance as it has to withstand high temperatures and pressures. Low-carbon steel, is used in most water-tube boilers operating between 270°C and 400°C. Medium-carbon steel, with 0.35% maximum of carbon, permits higher stresses than low-carbon steel at temperature upto 500°C

For superheater tubes, alloy steels are required as they have to resist temperatures above 500°C. These may contain chromium, chromium-molybdenum and chromium-nickel. They may be of ferric structure or, for the highest temperature at which modern boilers operate, of an austenitic structure.

Steamless tubes or electric-resistance welded tubes are used in water-tube boilers. Electric-resistance welded tubes are becoming increasingly popular for most applications, except for high pressures where wall thickness makes the use of steamless tubes more practical.

4.4 Advantages of High pressure Boilers

The principal advantages of high pressure boilers are as under :

- ... Greater freedom for disposing of the heating surfaces and hence greater evaporation for a given size.
- ... Reduction in the number of drums required.
- ... Smaller bore tubes, and therefore lighter tubes.
- ... Lighter for a given output.
- ... Rapid changes of load can be met without the use of complicated, or delicate control devices.
- ... Using external supply of power, a very rapid start from cold is possible. Hence the boiler is suitable for carrying peak loads, or for stand-by purposes in hydraulic stations.
- ... Tendency of scale formation is eliminated due to high velocity of water through the tubes.
- ... Due to uniform heating of all parts, the danger of overheating is reduced and thermal stress problem is simplified.

Against above advantages, the high cost of the pumping equipment, the power required to run the pumps, and safety of the boiler are some of the limitations to be kept in mind.

4.5 Arrangement of Heating Surfaces

Commonly used furnace layout for pulverised fuel boilers is shown in fig. 4-8. Three

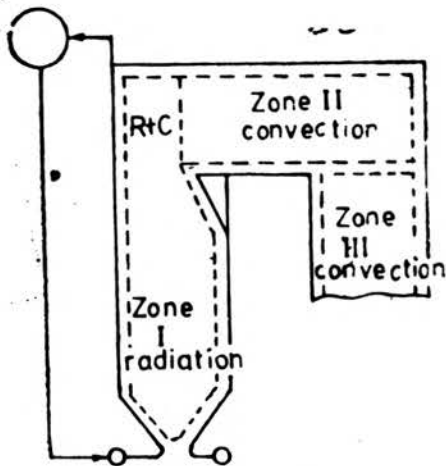


Fig. 4-8.

zones are seen in the elevation of a boiler in this figure. In zone I, the heat transfer is by a radiation and it includes the space marked R + C which can receive heat by convection as well as radiation if suitable heat transfer surface is introduced into the path. In zone II and III, the main mode of heat transfer is convection. Gas temperatures are high in zone II. Clear cut demarcation cannot be made even though zone III is essentially treated as a low temperature zone. In order to achieve the complete combustion of fuel, opportunity must be given to the combustible constituents of the fuel to interact with oxygen. This opportunity, however, decreases rapidly as the reaction proceeds towards completion. In order to ensure complete combustion of fuel, it becomes essential to supply excess air. The supply of excess air and turbulence

assume a greater significance in the final stages of combustion. The presence of excess air reduces the boiler efficiency. With increased surface contact between fuel and air and with created turbulence, the demand of excess air is very limited in a pulverised fuel fired furnace relative to stoker fired furnace.

A very high flame temperature is produced by hot turbulent air coupled with low excess air. At this temperature, ash is always in a molten stage. Metal temperature of all heat transfer surfaces are less than the *ash fusion temperature*. In order to reduce the possibility of this molten ash solidifying on tube surfaces, the use of convective heat transfer should be avoided as long as the gas temperature are higher than ash fusion temperature. Till then the heat transfer must be by radiation only. Zone I has radiant heat transfer surfaces. The furnace exit gas temperature has a profound bearing on the safe operation of the unit. This should be as high as possible so as to give a good temperature potential for the heat transfer surfaces to be located in convective zone, but at the same time, it should be lower than ash fusion temperature to avoid slag deposition. Roughly about 50 per cent of the heat liberated during combustion is absorbed in the radiant zone. This value increases with a fall in ash fusion temperature, rise in air temperature or fall in excess air. If a low ash-fusion-temperature coal is fired in a furnace designed for high ash-fusion-temperature coal, operational and maintenance problems are created. During radiation heat transfer, the tube metal temperature should be as low as feasible in order to have a smaller furnace. Relative to superheaters, evaporators offer lower metal temperature and hence the evaporator is the most appropriate component to be located in the radiant zone.

Slagging should not be ignored in zone II, where the gas temperatures are fairly high and convection is the mode of heat transfer. Sometimes panels and platens are located before this zone II so as to bring down the gas temperature to a safer level. These panels and platens can be evaporator or superheater. Panels are heat transfer surfaces at a considerably greater distance from each other. Panels permit large radiant heat absorption. Platens are heat transfer surfaces which are closer to each other. In platens, heat transfer takes place by convection and radiation.

Because of higher metal temperatures, superheater elements are more expensive than evaporator surfaces and hence it is desirable to put limit to superheater surfaces. Thus, this high temperature convection zone (zone II) is the most appropriate zone for superheaters.

In zone III, the gas temperatures are relatively low. Location of superheaters in this zone makes them expensive due to lower temperature potentials. So the heat recovery units like economiser and air heater are most appropriate for this temperature convection zone.

With increase in operating pressure, the superheat temperature increases. Usually beyond 100 bar pressure, reheat becomes necessary. The proportion of the heat generated in the furnace and absorbed in the various components like economiser, evaporator, superheater and reheater are listed in table 4-1.

Table 4-1

Pressure bar	Temperature °C	Approximate % of energy distribution			
		Economiser and Preheater	Evaporator	Superheater	Reheater
63	480	12	64	24	–
90	510	9.9	62.1	28	–
130	540	2.7	55.5	28.3	13.5
170	565	3.6	48.7	34.4	13.3

From table-4-1, it is seen that the major parameters which influence the orientation of heat transfer surfaces are pressure and temperature. In support of this fact, features of four representative boilers working in power plants in India are discussed below. The particulars of the operating of the four high pressure boilers are given in table 4.2.

Table 4-2

Power plant	Pressure bar	Temperature °C	Rate of steam generation tonnes/hr.	Output MW
Bokaro (two boilers)	62.5	485	136	25 + 25 = 50
Ramagundam	90	515	280	66
Chandrapura	135	540	435	140
Trombay	178.5	570	480	150

4.5.1 Bokaro Plant : The operating conditions are comparable with the data given

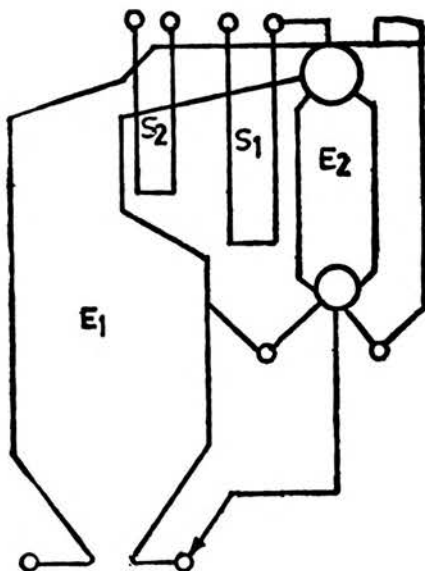


Fig. 4-9. Layout of Bokaro Plant.

at no.1 of the table 4-1. The arrangement of components is shown in fig. 4.9. The superheater requires about 24 per cent energy where as the evaporator needs 64 per cent. Therefore, the entire furnace is water cooled (E_1) and remaining part of the evaporator (E_2) is kept in the latter portion of zone II as tubes between two drums. For these operating conditions two drum construction is conventional. However, it is possible to have water cooled platens and panels instead of two drum arrangement but this arrangement if used excessively, can lead to lower gas temperature at zone II, which may be undesirable for superheater. The superheater in this boiler is a convective heat transfer type. Baffles are provided to increase the gas velocities.

4.5.2 Ramagundam Plant : The arrangement of components of this boiler is shown in fig. 4-10. Relative to the Bokaro plant, the evaporator duty is only slightly lower. Entire furnace walls (E_1) and convective tubes

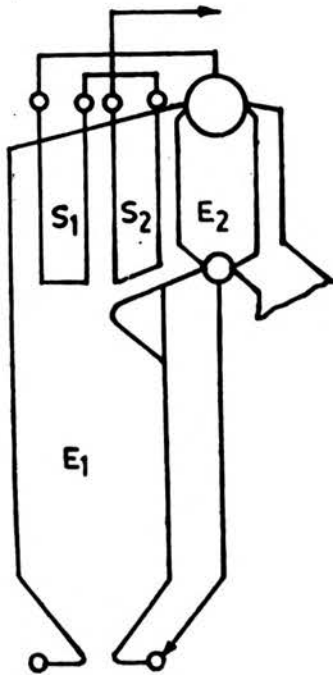


Fig. 4-10. Layout of Ramagundam plant.

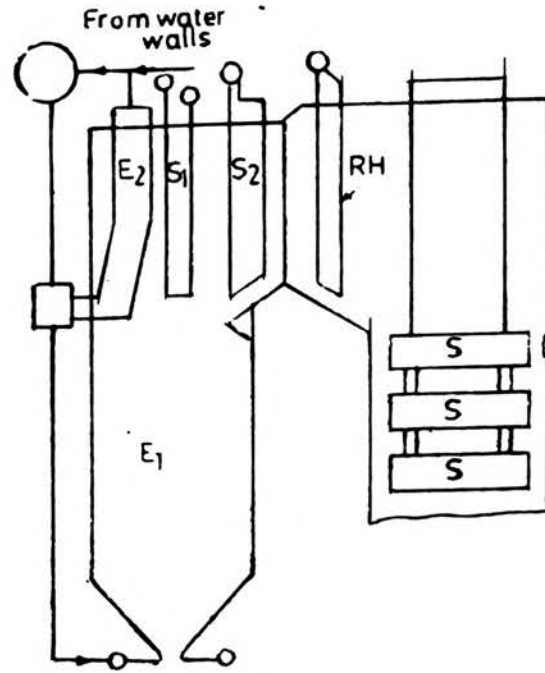


Fig. 4-11. Layout of Chandrapura plant.

between two drums (E_2) form the evaporator surface. The low temperature section of the superheater (S_1) is introduced as widely spaced platens and the finishing stage of the superheater (S_2) is away from the flame. There are no baffles in the gas flow path.

4.5.3 Chandrapura Plant : The arrangement of components of this boiler is shown in fig. 4-11. This boiler has significantly higher operating conditions, and reheat of the steam is adopted. As the evaporator duty is considerably reduced, it is not necessary to locate the evaporator in zone II as was needed for Bokaro and Ramagundam boiler plants. The furnace walls are totally covered with evaporator (E_1) and a small portion of the evaporator is placed as radiant platens (E_2) near the upper front wall. All the space of zone II is occupied by the superheater and reheater which need nearly 42 percent of the energy. A part of the superheater (S_1) is platen. The reheater (RH) and a part of superheater (S_2) is pendant. The bulk of the superheater is located as three horizontal banks (S) in the rear pass. It is interesting to note that the space occupied by the two drum arrangement of Bokaro and Ramagundam boiler plants is now utilised by the superheater.

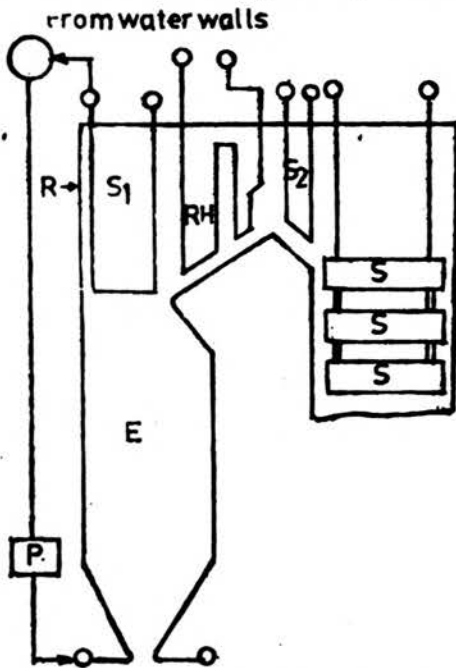


Fig. 4-12. Layout of Trombay plant.

4.5.4 Trombay Plant : This boiler has higher pressure and temperature as compared to boilers discussed above. The arrangement of components of this boiler is shown in fig. 4-12.

In this boiler evaporator duty is decreased and superheater and reheater duty is increased. All the superheater and reheater elements cannot be located in zone II, and on other side the evaporator does not need all the energy available in zone I. Therefore, some of the superheater elements must be located in zone I. The evaporator is located in the furnace wall (E). In the upper front wall some of the radiant

surface (R) is actually reheater. Widely spaced panels (S_1) and platens (S_2) are superheater elements. The reheaters are located between the superheater elements. The unit has controlled circulation with pump (P). The rear pass or zone III consists of horizontal banks (S) of superheater elements. The important point to be noted is that superheater elements have entered zone I in a big way.

Tutorial - 4

1. What do you understand by high pressure high duty boilers ?
2. Explain general features of water-tube boilers.
3. What are the trends observed in the design, construction and operation of modern steam generators ?
4. Describe giving illustrations, the development which has taken place in water-tube boilers to attain higher operating pressure and higher steaming capacity.
5. Explain arrangement of components and working of La Mont boiler.
6. Sketch a layout and explain arrangement of components and working of Benson boiler.
7. Explain the construction and working of Loeffler boiler.
8. Sketch a layout, and explain arrangement of components and working of Schmidt Hartmann boiler.
9. What are the main advantages of Schmidt-Hartmann boiler ?
10. Sketch a layout, and explain arrangement of components and working of Velox boiler.
11. What are the main advantages of Velox boiler ?
12. Discuss the materials of construction of modern high pressure boilers.
13. What are the advantages of high pressure boilers ?
14. Sketch a layout for a pulverised fuel boiler, showing important zones and explain efficient use of heat transfer surfaces.
15. Discuss most suitable arrangement of superheaters and reheaters in modern high pressure boilers.
16. Sketch a layout for pulverised fuel boiler and divide it into three zones according to intensity of temperature. In this layout, show radiation heat transfer and convection heat transfer surfaces and hence discuss suitable location of evaporators, superheaters, economisers, and reheaters.
17. Give particulars and general arrangement of components of the boilers at the following power plants :
(i) Bokaro, (ii) Ramagundum, (iii) Chandrapura, and (iv) Trombay.
18. Delete the phrase which is not applicable to complete the following statements :
(i) Water-tube/fire-tube principle is preferred for high pressure boilers.
(ii) In all modern high pressure boilers, the water circulation is maintained with the help of a pump/natural circulation due to density difference.
(iii) At critical pressure, the density of water and steam is different/same.
(iv) In Loeffler/La Mont boiler, heat of steam is used for evaporation of water.
(v) La Mont/Benson boiler is drumless.
(vi) Natural water circulation by convection in water tube boilers, increases/decreases with increase in pressure.
(vii) When the velocity of gas exceeds the velocity of sound, the heat transferred from the gas is at much greater/smaller rate than the rate achieved with subsonic flow.
(viii) Gas turbine and air compressor unit is provided in La Mont/Velox boiler.
(ix) High temperature zone in a boiler is suitable for superheater/economiser.
[Delete : (i) fire-tube, (ii) natural circulation due to density difference, (iii) different, (iv) La Mont
(v) La Mont, (vi) increases, (vii) smaller, (viii) La Mont, (ix) economiser.]
19. Fill in the blanks in the following statements :
(i) _____ and _____ are indirectly heated boilers.
(ii) _____ boiler is drumless and once through type.
(iii) Maximum energy loss in a boiler occurs due to _____ .
(iv) A supercritical boiler is one that operates above the pressure and temperature of _____ bar and _____ °C.
(v) In _____ boiler, two pressures are used to effect interchange of energy.
[(i) Loeffler, Schmidt-Hartmann; (ii) Benson; (iii) flue gases; (iv) 220.9 and 374.14 (v) Schmidt-Hartmann.]
20. Indicate the correct answer by selecting correct phrase in each of the following statements :

- (i) Benson boiler has
 - (a) no drum, (b) one drum, (c) two drums, (d) three drums.
- (ii) In a boiler, whose walls are lined with water tubes, transference of heat to tubes is mainly by
 - (a) convection, (b) conduction, (c) radiation, (d) combination of above modes.
- (iii) Indirect heating and evaporation of water, is the underlying thermodynamic principle of
 - (a) La Mont boiler, (b) Benson boiler, (c) Loeffler boiler, (d) Velox boiler.
- (iv) Velocity of gases exceeds the velocity of sound in
 - (a) La Mont boiler, (b) Benson boiler, (c) Loeffler boiler, (d) Velox boiler.
- (v) Due to high velocity of water through tubes, the tendency of scale formation is
 - (a) increased, (b) eliminated, (c) not affected.

[(i) a (ii) c, (iii) c, (iv) d, (v) b]

AIR-STANDARD CYCLES

5.1 Introduction

A heat engine cycle is a series of thermodynamic processes through which a working fluid (working substance) passes in a certain sequence. At the completion of the cycle, the working fluid returns to its original condition, i.e., the working fluid at the end of the cycle has the same pressure, volume, temperature and internal energy that it had at the beginning of the cycle. Somewhere during every cycle, heat is received by the working fluid. It is, then, the object of the cycle to convert as much of this heat energy as possible into useful work. The heat energy which is not converted, is rejected by the working fluid during some process of the cycle.

5.2 Heat Engine

Any machine designed to carry out a thermodynamic cycle, and thus converts heat energy supplied to it into mechanical energy, is called a *heat engine*. Hence, the cycle it operates on is known as a *heat engine cycle*. Heat engine is generally made up of a piston and cylinder, together with the following main elements :

- (i) a *hot body*, serving as a source of heat which is received during the cycle,
- (ii) a *cold body*, whose function is to receive the heat rejected during the cycle, and
- (iii) a *working fluid (working substance)*, which receives heat directly from the hot body, rejects to the cold body, does external work on the piston during expansion, and have work done upon it by the piston during compression. The working substance may be steam, air, or mixture of fuel and air.

5.2.1 Types of heat engines : Heat engines may be of the following types :

- (i) *Steam engine, and steam turbine*, in which the working fluid (working substance) is steam,
- (ii) *Hot air engine*, in which the working fluid is air, and
- (iii) *Internal combustion engine, and gas turbine*, in which the working fluid is a mixture of gases and air, or products of combustion of fuel oil and air.

The cycles which will be presented in this chapter are ideal cycles which will apply to the last two types of heat engines, i.e., hot air engines and internal combustion engines including gas turbines. The ideal cycles which apply to the first type (i.e., steam engine) are described in chapter 9 of Volume I.

5.2.2 Available work of cycle : As stated above, it is evident that the function of any heat engine cycle is to receive heat from some external source -- the hot body, and transform as much of this heat as possible into mechanical energy. The amount of heat which is transformed into mechanical energy is known as *available energy* of the cycle. It is equal to the difference between the heat received during the cycle from the hot body and the heat rejected during the cycle to the cold body, in the absence of any other losses. This statement is of course a direct consequence of the law of conservation of energy.

Let Q = available energy for doing work per cycle in heat units,

Q_1 = heat received during each cycle from the hot body in heat units, and

Q_2 = heat rejected during each cycle to the cold body in heat units.

Then, $Q = Q_1 - Q_2$ in heat units. ... (5.1)

Every cycle contains thermodynamic processes involving both expansion and compression processes. During the former (expansion), work is done upon the piston by the gas while during the latter (compression), work is done on the gas by the piston. The difference between the work done by the gas and the work done on the gas during the complete cycle is called the *net available work* of the cycle. It is necessarily equal to the available energy for doing work per cycle.

If W = net work done during the cycle in heat units,

then, $W = Q = Q_1 - Q_2$ in heat units. ... (5.2)

5.2.3 Efficiency of a cycle : The thermal efficiency of a heat engine cycle is defined as the ratio of the available heat energy of the cycle for doing work to the heat received during the cycle from the hot body. It is usually denoted by the letter η (eta),

i.e., Efficiency, $\eta = \frac{\text{Heat equivalent of the net work done per cycle}}{\text{Heat received during the cycle from the hot body}}$... (5.3)

Thus, using eqns. (5.1) and (5.2), $\eta = \frac{Q}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{W}{Q_1}$

The definition of efficiency given above is applicable to any type of heat engine cycle.

Hence, the expression for the efficiency given by the eqn. (5.3) is known as *theoretical or ideal thermal efficiency* of the cycle, as it does not take into account any practical losses which do occur in the actual running of the engine.

5.2.4 Air-standard efficiency of a cycle : In order to compare the thermal efficiency of actual internal combustion engine cycles, the engineer needs some standard to serve as a yard-stick. The yard-stick used is the theoretical thermal efficiency of the engine working on ideal cycle, using air as the working fluid. The theoretical thermal efficiency of the ideal cycle is known as the *air-standard efficiency*, since it is worked out on the basis of the working fluid being air throughout the cycle, i.e., the effect of calorific value of fuel used is eliminated, and the heat is supplied by bringing a hot body in contact with the end of the cylinder. Thermal efficiency of the ideal cycle can be worked out before the engine is constructed and hence indicates the maximum approachable efficiency of the completed engine. Should the actual indicated thermal efficiency of the completed engine not closely approach this efficiency (air-standard efficiency), alterations and improvements may be made to bring about the desired result. It may be noted that actual engine can never give thermal efficiency as high as the air-standard efficiency when operated on the same cycle as air engine. Actual indicated thermal efficiency of a well designed and well constructed internal combustion engine, when properly operated, should be at least two-third of air-standard efficiency.

5.3 Thermodynamic Reversibility

In chapter 2 of volume 1, we have defined eight thermodynamic process; any one of these processes which can be operated in a reverse direction is known as *reversible* process. The factors which make a process irreversible are : (i) temperature difference required for heat to flow, and (ii) fluid friction. Thus, for an operation to be thermodynamically reversible, following conditions should be satisfied :

(i) The temperature of the hot body supplying the heat must at any instant be the same as that of working fluid which receives the heat. If the source of heat is at a

higher temperature than the working fluid, heat will be transferred to the latter (working fluid), but when the process is reversed, heat must flow from the working fluid back into the hot body, which is at a higher temperature. This is contrary to the second law of thermodynamics. It follows that the operation could not be reversed. Thus, for an operation to be thermodynamically reversible, there cannot be temperature difference between the hot body and the working fluid during the transfer of heat. Flow of heat to or from the working fluid without finite temperature drop implies perfect exchange or infinitely slow process. This is known as *external reversibility*.

(ii) Friction between the fluid and the walls of the container and viscous friction of the fluid can never be eliminated in a thermodynamic process. The energy lost in overcoming the frictional forces is regenerated into heat. When any thermodynamic process is reversed, say from expansion to compression, friction effect cannot be reversed, i.e., friction heat cannot be absorbed back into the fluid. Thus, for an operation to be thermodynamically reversible, fluid friction must be absent. This is known as *internal reversibility*.

It will be seen from the above that the concept of thermodynamic reversibility is purely hypothetical, because the transfer of heat becomes less as the condition of reversibility is approached, and fluid friction can never be completely eliminated. Thus any thermodynamic process can be reversed, if external and internal reversibility is assumed. It may be noted that the basic requirement for throttling process is friction. Thus, throttling process is not reversible. A frictionless adiabatic operation is reversible. All other thermodynamic processes are reversible if external and internal reversibility is assumed.

5.3.1 Reversible cycle : For thermodynamic cycle to be reversible, it must consist of reversible processes only. When a cycle is reversed, all the processes are performed in the reversed direction. A heat engine cycle takes heat from the hot body and rejects portion of it to a cold body, and converts remaining quantity of heat into mechanical work. When this cycle is reversed, heat will be absorbed from the cold body and rejected to a hot body. This will necessitate external work to be supplied. This reversed cycle is known as *heat pump or refrigerating machine*.

A reversible cycle should not be confused with a mechanically reversible engine. Steam engine can be made to revolve in a reversed direction by mechanically altering the valve settings but this does not reverse the cycle on which the engine works. A reversed engine merely rotates in the opposite direction, but a reversed cycle converts a power producing engine into a heat pump or refrigerator.

It may be noted that an engine working on reversible cycle is the most efficient engine.

5.4 Ideal Heat Engine Cycle

There are number of ideal heat engine cycles made up of some of the following processes in which

- (a) heat is taken in or rejected at constant temperature (isothermal compression or expansion),
- (b) heat is taken in or rejected at constant pressure,
- (c) heat is taken in or rejected at constant volume, and
- (d) compression and expansion are frictionless adiabatic (isentropic).

Only five principal ideal heat engine cycles will be described in this chapter which may be summarised as follows :

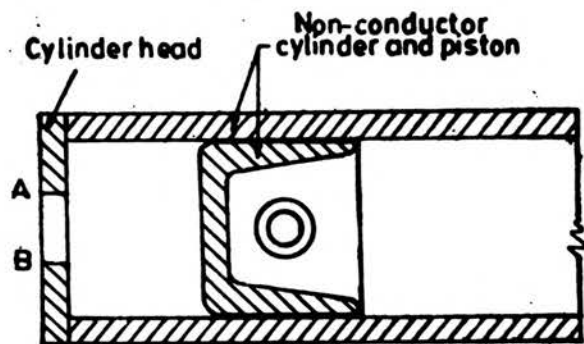
- (i) *The constant temperature cycle :* Here, heat is taken in and rejected at constant temperature, and compression and expansion being frictionless adiabatic or isentropic.

This cycle is known as *Carnot cycle*.

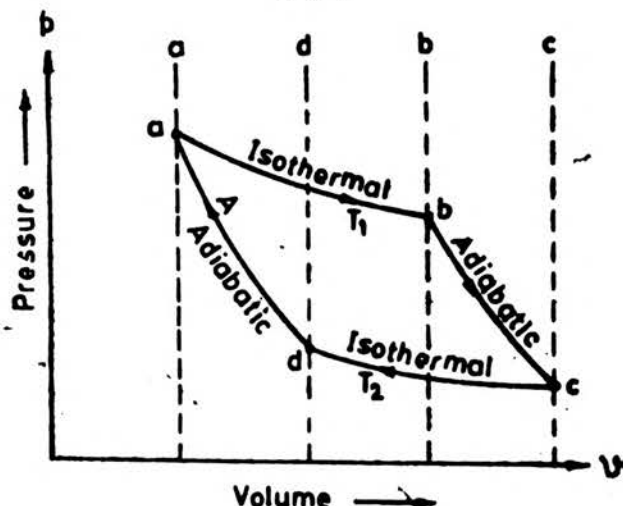
- (ii) *The constant volume cycle* : In this, heat is taken in and rejected at constant volume, and compression and expansion being frictionless adiabatic. This cycle is known as *Otto cycle*.
- (iii) *The modified constant pressure cycle* : In this, heat is taken in at constant pressure and rejected at constant volume, and compression and expansion being frictionless adiabatic. This cycle is known as *Diesel cycle*.
- (iv) *The dual-combustion or mixed cycle* : Here, heat is partly taken in at constant volume and then at constant pressure and heat is rejected at constant volume, and compression and expansion being frictionless adiabatic. This cycle is sometimes known as *semi-Diesel cycle*.
- (v) *The (true) constant pressure cycle* : In this, heat is taken in and rejected at constant pressure, and compression and expansion being frictionless adiabatic. This cycle is known as *Joule cycle*.

5.5 Carnot Cycle

This cycle was brought out in 1824 by a French engineer named Sadi Carnot. Although its limitations are such that no heat engine has ever been constructed to use it. This cycle theoretically permits the conversion of the maximum quantity of heat energy into mechanical energy, as being a reversible cycle. In other words, it gives the maximum efficiency that is possible to obtain in a heat engine. Hence, its usefulness lies in the comparison which it affords with other heat engines, giving as it does under the conditions, the maximum efficiency that they would like to approach.



(a)



(b)

Fig. 5-1. p-v diagram of Carnot cycle.

An engine operating on this ideal cycle, would require a cylinder and a piston of perfectly non-conducting material, a cylinder head that will conduct heat perfectly, and three other elements that can be brought into contact with the conducting cylinder head at AB as shown in fig. 5-1, as occasion demands. These three elements are : (a) the *hot body*, always at temperature T_1 , the source of heat energy supplied to the working fluid (air), (b) the *non-conducting cover* to fit the cylinder at AB, and (c) the *cold body*, maintained at a temperature T_2 , the minimum temperature of the cycle. The element (cold body) receives the heat that is rejected from the working fluid (air).

Consider one kg of air at temperature T_1 as the working fluid in the engine cylinder. Let point a (fig. 5-1b) represent the state of the working fluid as regards pressure p_a and volume v_a at absolute temperature T_1 .

Isothermal expansion : At point a, the body at temperature T_1 is brought in contact with the cylinder head at AB and heat is supplied at temperature T_1 to the working fluid (air). This causes the air to expand

isothermally along the curve $a-b$ from volume v_a to v_b until point b is reached. This point is the end of isothermal expansion. The temperature through this process ab has been maintained constant at T_1 . As the air expands, it forces the piston outward thus doing work on the piston.

Adiabatic expansion : At point b , the hot body is removed and replaced by the non-conductor cover. Since all the elements of the engine which are now in contact with the working fluid (air) are non-conductors, no heat can be added or abstracted from the air. The air now expands adiabatically along curve $b-c$, doing work on the piston at the expense of its internal energy. Consequently the temperature falls from T_1 to T_2 and the volume increases from v_b to v_c . At point c , the piston is at the end of the outward stroke.

Isothermal compression : At point c , the non-conducting cover is removed and the cold body at temperature T_2 is brought in contact with the conducting cylinder head at AB. The piston now moves inward compressing the air isothermally along the curve $c-d$ from volume v_c to v_d , until point d is reached. During this compression, the heat which is rejected by the air goes into the cold body. This makes the isothermal compression at constant temperature T_2 possible.

Adiabatic compression : At point d the cold body is removed and the non-conducting cover again takes the position at AB. The air is now adiabatically compressed along the curve $d-a$, until it reaches the starting point a of the cycle, where it resumes its initial conditions of temperature, pressure and volume, and the piston is returned to the end of the stroke.

Since no transfer of heat occurs during both adiabatic operations, then by the law of conservation of energy, the difference between the heat received and heat rejected must be equal to the net work done. Now for any non-flow thermodynamic process,

Heat added = work done + change in internal energy.

Since during isothermal expansion process $a-b$ the temperature does not change neither will the internal energy change.

Heat added (supplied) during operation $a-b$ = work done

$$= p_a v_a \log_e \left(\frac{v_b}{v_a} \right) = RT_1 \log_e (r_1) \text{ per kg of air}$$

where r_1 = isothermal expansion ratio $\frac{v_b}{v_a}$.

Heat rejected during operation $c-d$

$$= p_c v_c \log_e \left(\frac{v_c}{v_d} \right) = RT_2 \log_e (r_2) \text{ per kg of air}$$

where r_2 = isothermal compression ratio $\frac{v_c}{v_d}$.

As stated above, the net work done per kg of air is the difference between the heat supplied and heat rejected.

$$\begin{aligned} \therefore \text{Net work done} &= RT_1 \log_e (r_1) - RT_2 \log_e (r_2) \\ &= R [T_1 \log_e (r_1) - T_2 \log_e (r_2)] \text{ per kg of air} \end{aligned} \quad \dots(5.4)$$

Using temperature and volume relationship for adiabatic process and considering adiabatic expansion $b-c$ (fig. 5-1),

$$\frac{\text{Higher temperature}}{\text{Lower temperature}} = (r)^{\gamma - 1} \quad \text{i.e.} \quad \frac{T_b}{T_c} = \left(\frac{v_c}{v_b} \right)^{\gamma - 1}$$

Since, $T_a = T_b = T_1$ and $T_c = T_d = T_2$

$$\therefore \frac{T_1}{T_2} = \left(\frac{v_c}{v_b} \right)^{\gamma - 1} \quad \dots(i)$$

Similarly, for adiabatic compression $d \rightarrow a$, $\frac{T_a}{T_d} = \left(\frac{v_d}{v_a} \right)^{\gamma - 1}$

Since, $T_a = T_1$ and $T_d = T_2$,

$$\therefore \frac{T_1}{T_2} = \left(\frac{v_d}{v_a} \right)^{\gamma - 1} \quad \dots(ii)$$

From (i) and (ii), $\frac{v_c}{v_b} = \frac{v_d}{v_a}$ or $\frac{v_c}{v_d} = \frac{v_b}{v_a}$ i.e., $r_2 = r_1$

From eqn. (5.4), Net work done = $R [T_1 \log_e (r_1) - T_2 \log_e (r_2)]$ per kg of air
 $= R(T_1 - T_2) \log_e (r_1)$ per kg of air. $\dots(5.5)$

$$\begin{aligned} \text{But, Efficiency} &= \frac{\text{Net work done per kg of air}}{\text{Heat supplied per kg of air}} \\ &= \frac{R(T_1 - T_2) \log_e (r_1)}{R T_1 \log_e (r_1)} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1} \quad \dots(5.6) \end{aligned}$$

Problem - 1 : While undergoing a Carnot cycle, the working fluid receives heat at a temperature of 317°C and rejects heat at a temperature of 22°C . Find the theoretical efficiency of the cycle. If the engine working on this cycle absorbs $2,100 \text{ kJ/min}$. from the hot body, calculate the net work done in kJ per sec . and the theoretical power of the engine.

Referring to fig. 5-1, $T_1 = 317 + 273 = 590 \text{ K}$; $T_2 = 22 + 273 = 295 \text{ K}$

Using eqn. (5.6),

Theoretical efficiency of the Carnot cycle, $\eta = 1 - \frac{T_2}{T_1} = 1 - \frac{295}{590} = 0.5$ i.e. **50%**

$$\text{Efficiency} = \frac{\text{Work done per minute}}{\text{Heat supplied per minute}}$$

$$\text{i.e., } 0.5 = \frac{\text{Work done per minute}}{2,100}$$

\therefore Work one per minute = $0.5 \times 2,100 = 1,050 \text{ kJ/min}$. or **17.5 kJ/sec**

Since, one $\text{kW} = 1 \text{ kJ/sec}$,

Theoretical power of the engine = **17.5 kW**.

5.6 Otto Cycle (Constant Volume Cycle)

This ideal heat engine cycle was proposed in 1862 by Beau de Rochas. In 1876 Dr. Otto designed an engine to operate on this cycle. The Otto engine immediately became so successful from a commercial stand point, that its name was affixed to the cycle used by it. This cycle is in use in all gas and petrol engines together with many types of oil engines.

The ideal $p - v$ and $T - \phi$ diagrams of this cycle are shown in fig 5-2. In working out the air-standard efficiency of the cycle, the following assumptions are made :

(i) The working fluid (working substance) in the engine cylinder is air, and it behaves as a perfect gas, i.e., it obeys the gas laws and has constant specific heats.

(ii) The air is compressed adiabatically (without friction) according to law

$p v^\gamma = C$ (where $\gamma = \frac{K_p}{K_v}$) in the engine cylinder during the compression stroke.

(iii) The heat is supplied to the air at constant volume by bringing a hot body in contact with the end of the engine cylinder.

(iv) The air expands in the engine cylinder adiabatically (without friction) during the expansion stroke.

(v) The heat is rejected from the air at constant volume by bringing a cold body in contact with the end of the engine cylinder.

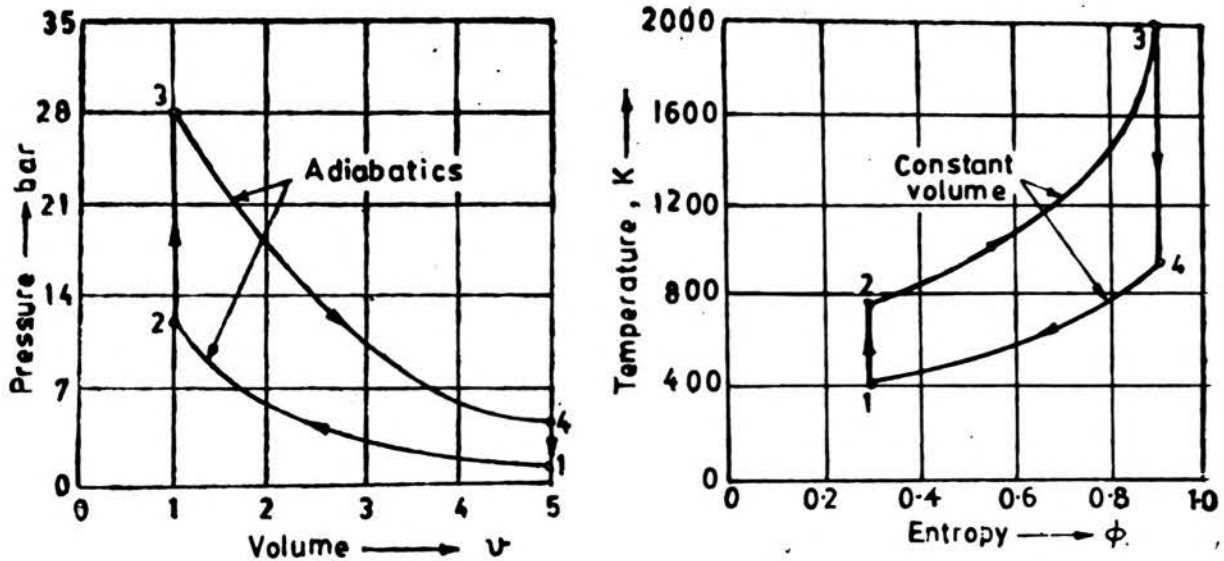


Fig. 5-2. p - v and T - ϕ diagrams of Otto cycle.

Consider one kilogram of air in the engine cylinder at point (1). This air is compressed adiabatically to point (2), at which condition the hot body is placed in contact with the end of the cylinder. Heat is now supplied at constant volume, and temperature and pressure rise; this operation is represented by (2-3). The hot body is then removed and the air expands adiabatically to point (4). During this process, work is done on the piston. At point (4), the cold body is placed at the end of the cylinder. Heat is now rejected at constant volume, resulting in drop of temperature and pressure. This operation is represented by (4-1). The cold body is then removed after the air is brought to its original state (condition). The cycle is thus completed.

The cycle consists of two constant volume processes and two frictionless adiabatic processes. The heat is supplied during constant volume process (2-3) and rejected during constant volume process (4-1). There is no exchange of heat during the two frictionless adiabatics (1-2) and (3-4).

Heat supplied during constant volume operation (2-3) = $K_v (T_3 - T_2)$ heat units/kg of air.

Heat rejected during constant volume operation (4-1) = $K_v (T_4 - T_1)$ heat units/kg of air.

Net work done = Heat supplied - Heat rejected

$$= K_v (T_3 - T_2) - K_v (T_4 - T_1) \text{ heat units per kg of air.}$$

Efficiency, $\eta = \frac{\text{Net work done per kg of air}}{\text{Heat supplied per kg of air}}$

$$= \frac{K_v (T_3 - T_2) - K_v (T_4 - T_1)}{K_v (T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \quad \dots (5.7)$$

Considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \quad (\text{where } r = \text{ratio of compression})$$

or $\frac{\text{Higher temperature } (T_2)}{\text{Lower temperature } (T_1)} = (r)^{\gamma-1}$

$$\therefore T_2 = T_1 (r)^{\gamma-1} \quad \dots (i)$$

Again considering adiabatic expansion (3-4),

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \quad (\text{as } v_4 = v_1 \text{ and } v_3 = v_2)$$

or $\frac{\text{Higher temperature } (T_3)}{\text{Lower temperature } (T_4)} = (r)^{\gamma-1}$

$$\therefore T_3 = T_4 (r)^{\gamma-1} \quad \dots (ii)$$

Substituting values of T_2 and T_3 from (i) and (ii) in eqn. (5.7), we get,

$$\begin{aligned} \text{Air-standard efficiency (A. S. E.)} &= 1 - \frac{T_4 - T_1}{T_4 (r)^{\gamma-1} - T_1 (r)^{\gamma-1}} \\ &= 1 - \frac{T_4 - T_1}{(r)^{\gamma-1} (T_4 - T_1)} = 1 - \frac{1}{(r)^{\gamma-1}} \end{aligned} \quad \dots (5.8)$$

From eqn. (5.8), it is seen that the air-standard efficiency of I.C. engines working on Otto cycle is a function of the compression ratio (r) only. The following table gives the value of air-standard efficiency for various ratios of compression :

Ratio of compression, r	2.0	3.0	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
Percentage air-standard efficiency ($\gamma = 1.4$)	24.51	35.42	42.56	45.21	47.47	49.44	51.16	52.70	54.00	55.34	56.46

The air-standard efficiency expression, $\eta = 1 - \frac{1}{(r)^{\gamma-1}}$ can also be expressed in terms of temperatures T_1 and T_2 .

From (i), $\frac{T_2}{T_1} = (r)^{\gamma-1}$

$$\therefore \text{Air-standard efficiency} = 1 - \frac{1}{\frac{T_2}{T_1}} = 1 - \frac{T_1}{T_2} = \frac{T_2 - T_1}{T_2} \quad \dots (5.9)$$

From eqn. (5.9), it should be observed that T_2 is not the highest temperature of the cycle, and therefore the efficiency is less than Carnot, which, for the temperature range obtaining, would be $1 - \frac{T_1}{T_3}$ or $\frac{T_3 - T_1}{T_3}$ where T_3 is the highest temperature of the Otto cycle.

The eqn. (5.8) shows, that higher thermal efficiency can be obtained with higher compression ratio, and smaller the difference between T_3 and T_2 , the more closely is the Carnot efficiency approached, but at the expense of reduction in net work done per kg of air.

Problem - 2 : In an ideal Otto cycle engine, the temperature and pressure at the beginning of compression are 43°C and 100 kPa respectively and the temperature at the

end of adiabatic compression is 323°C . If the temperature at the end of constant volume heat addition is $1,500^{\circ}\text{C}$, calculate : (a) the compression ratio, (b) the air-standard efficiency, and (c) the temperature and pressure at the end of adiabatic expansion. Assume γ as 1.4 for air.

(a) Referring to fig. 5-3, $p_1 = 100$ kPa,

$$T_1 = 43 + 273 = 316 \text{ K};$$

$$T_2 = 323 + 273 = 596 \text{ K};$$

$$T_3 = 1,500 + 273 = 1,773 \text{ K}.$$

Referring to fig. 5-3 and considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1}$$

$$\text{i.e. } (r)^{\gamma-1} = \frac{T_2}{T_1}$$

\therefore Compression ratio,

$$r = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} = \left(\frac{596}{316}\right)^{\frac{1}{1.4-1}}$$

$$= (1.886)^{2.5} = 4.87$$

(b) Using eqn. (5.8), Air-standard efficiency (A.S.E.)

$$= 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(4.87)^{0.4}}$$

$$= 1 - \frac{1}{1.884} = 0.4695 \text{ or } 46.95\%$$

Alternatively, using eqn. (5.9),

Fig. 5-3. p - v diagram of Otto cycle.

$$\text{Air-standard efficiency } 1 - \frac{T_1}{T_2} = 1 - \frac{316}{596} = 0.4695 \text{ or } 46.95\% \text{ (same as before)}$$

$$(c) \text{ Now, } \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r)^{\gamma} = (4.87)^{1.4} = 9.173$$

$$\therefore p_2 = p_1 \times 9.173 = 100 \times 9.173 = 917.3 \text{ kPa}$$

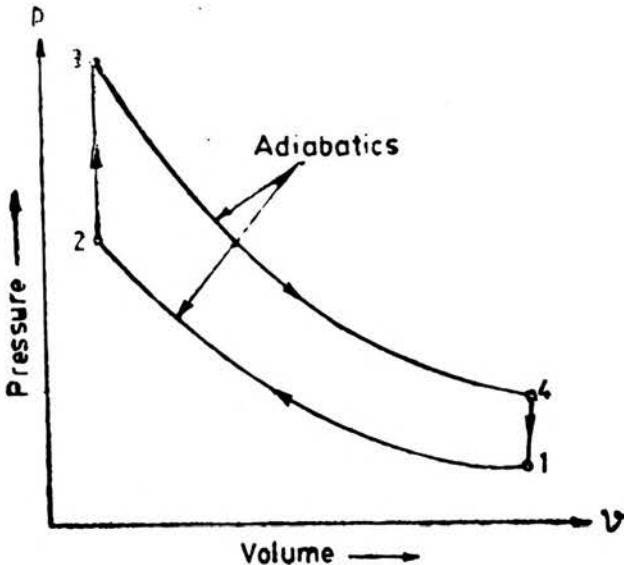
$$\text{From constant volume heat addition (2-3), } \frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$$

$$\text{Hence, as } v_2 = v_3, \frac{p_3}{p_2} = \frac{T_3}{T_2}$$

$$\therefore p_3 = p_2 \times \frac{T_3}{T_2} = 917.3 \times \frac{(1,500 + 273)}{(323 + 273)} = 2,729 \text{ kPa}$$

Considering adiabatic expansion (3-4) of fig. 5-3, $p_3 v_3^{\gamma} = p_4 v_4^{\gamma}$

$$\therefore p_4 = \frac{p_3}{\left(\frac{v_4}{v_3}\right)^{\gamma}} = \frac{p_3}{(r)^{\gamma}} = \frac{2,729}{(4.87)^{1.4}} = \frac{2,729}{9.173} = 297.5 \text{ kPa}$$



Considering adiabatic expansion (3-4) of fig. 5-3, $\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1}$

$$\therefore T_4 = \frac{T_3}{\left(\frac{v_4}{v_3}\right)^{\gamma-1}} = \frac{T_3}{(r)^{\gamma-1}} = \frac{1,773}{(4.87)^{0.4}} = \frac{1,773}{1.884} = 941\text{K or } t_4 = 668^\circ\text{C}$$

Problem - 3 : In an engine working on the ideal Otto cycle, the pressure and temperature at the beginning of compression are 100 kPa and 40°C respectively. If the air-standard efficiency of the engine is 50%, determine; (i) the compression ratio, and (ii) the pressure and temperature at the end of adiabatic compression. Assume γ for air as 1.4.

Referring to fig. 5.3, $p_1 = 100$ kPa, $T_1 = 40 + 273 = 313$ K.

(i) Using eqn. (5.8),

$$\text{Air-standard efficiency} = 1 - \frac{1}{(r)^{\gamma-1}} \quad (\text{where } r = \text{compression ratio})$$

$$\text{i.e. } 0.5 = 1 - \frac{1}{(r)^{0.4}}$$

$$\therefore (r)^{0.4} = 2$$

$$\text{Taking logs of both the sides, } \log r = \frac{\log 2}{0.4} = \frac{0.301}{0.4} = 0.7525$$

$$\therefore r = 5.656 \quad (\text{compression ratio})$$

(ii) Referring to fig. 5-3 and considering adiabatic compression (1-2), $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma$

$$\therefore p_2 = p_1 \times \left(\frac{v_1}{v_2}\right)^\gamma = p_1 \times (r)^\gamma$$

$$= 100 \times (5.656)^{1.4} = 1,131 \text{ kPa}$$

Again referring to fig. 5-3 and considering adiabatic compression (1-2), $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$

$$\begin{aligned} \therefore T_2 &= T_1 \times \left(\frac{v_1}{v_2}\right)^{\gamma-1} = T_1 \times (r)^{\gamma-1} \\ &= 313 \times (5.656)^{0.4} = 626 \text{ K or } t_2 \\ &= 353^\circ\text{C} \end{aligned}$$

Problem - 4 : An engine working on the ideal Otto cycle, has a clearance volume of 0.03 m³ and swept volume of 0.12 m³. The pressure and temperature at the beginning of compression are 100 kPa and 100°C respectively. If the pressure at the end of constant volume heat addition is 2,500 kPa, calculate : (a) the air-standard efficiency of the cycle, and (b) the temperatures at the salient (key) points of the cycle. Assume $\gamma = 1.4$ for air.

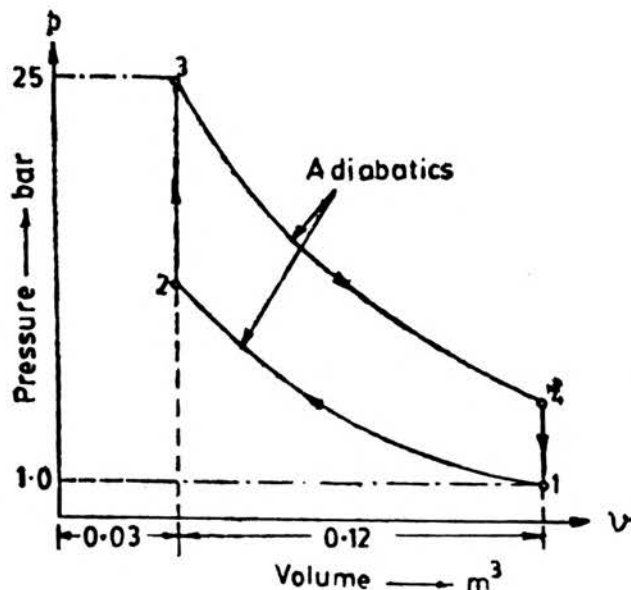


Fig. 5-4. p - v diagram of Otto cycle.

(a) Referring to fig. 5-4, $p_1 = 1$ bar; $p_3 = 25$ bar;
 $T_1 = 100 + 273 = 373$ K; $v_2 = 0.03$ m³; $v_1 - v_2 = 0.12$ m³.
 Compression ratio,

$$r = \frac{v_1}{v_2} = \frac{\text{clearance volume} + \text{swept volume}}{\text{clearance volume}}$$

$$= \frac{0.03 + 0.12}{0.03} = 5$$

Using eqn. (5.8), Air-standard efficiency

$$= 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(5)^{0.4}}$$

$$= 0.475 \text{ or } 47.5\%$$

(b) Referring to fig. 5-4 and considering adiabatic compression (1-2),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma$$

$$\therefore p_2 = p_1 \times \left(\frac{v_1}{v_2}\right)^\gamma = p_1 \times (r)^\gamma = 1 \times (5)^{1.4} = 9.52 \text{ bar}$$

Again from adiabatic compression (1-2), $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1}$

$$\therefore T_2 = T_1 \times (r)^{\gamma-1} = 373 \times (5)^{0.4} = 710 \text{ K or } t_2 = 437^\circ\text{C}$$

From constant volume heat addition (2-3), $\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$

$$\text{Hence, as } v_2 = v_3, \frac{T_3}{T_2} = \frac{p_3}{p_2}$$

$$\therefore T_3 = T_2 \times \frac{p_3}{p_2} = 710 \times \frac{25}{9.52} = 1,865 \text{ K or } t_3 = 1,592^\circ\text{C}$$

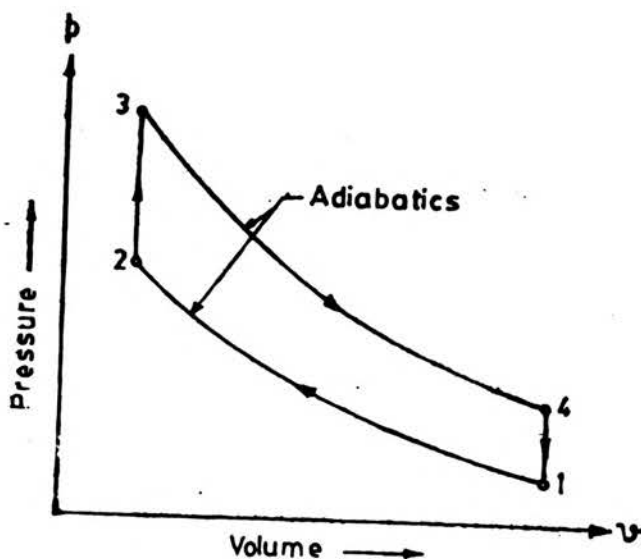


Fig. 5.5. p - v diagram of Otto cycle.

Considering adiabatic expansion (3-4),

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1}$$

$$\therefore T_4 = \frac{T_3}{\left(\frac{v_4}{v_3}\right)^{\gamma-1}} = \frac{T_3}{(r)^{\gamma-1}}$$

$$= \frac{1,865}{(5)^{0.4}} = 980 \text{ K or } t_4 = 707^\circ\text{C}$$

Problem - 5 : An air engine works on the ideal cycle in which heat is received and rejected at constant volume. The pressure and temperature at the beginning of compression are 100 kPa and 40°C respectively. The pressure at the end of adiabatic compression is 15 times that at start. If the temperature reached at the end of constant volume heat addition is 1,947°C, find : (a)

the heat supplied per kg of air, (b) the air-standard efficiency, (c) the work done per kg of air, and (d) the pressure and temperature at the end of adiabatic expansion. Take $K_v = 0.7165 \text{ kJ/kg K}$ and $\gamma = 1.4$ for air.

Referring to fig. 5.5, $p_1 = 100 \text{ kPa}$;

$p_2 = 15p_1 = 15 \times 100 = 1,500 \text{ kPa}$;

$T_1 = 40 + 273 = 313 \text{ K}$;

$T_3 = 1,947 + 273 = 2,220 \text{ K}$; and $\gamma = 1.4$

(a) Considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{i.e.} \quad T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore T_2 = 313 \times \left(\frac{1,500}{100}\right)^{\frac{0.4}{1.4}} = 313 \times 2.169 = 678.9 \text{ K}$$

Heat supplied per kg of air = $K_v (T_3 - T_2)$

$$= 0.7165 (2,200 - 678.9) = 1,104.2 \text{ kJ/kg of air}$$

(b) Using eqn. (5.9), Air-standard efficiency = $1 - \frac{T_1}{T_2} = 1 - \frac{313}{678.9} = 0.535$ or **53.5%**

(c) Now, Air-standard efficiency (A.S.E.) = $\frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}}$

$$\therefore \text{Work done per kg of air} = \text{A.S.E.} \times \text{Heat supplied per kg of air} \\ = 0.535 \times 1,104.2 = 590.75 \text{ kJ/kg of air.}$$

(d) From constant volume heat addition (2-3), $\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$

Hence as $v_2 = v_3$, $\frac{p_3}{p_2} = \frac{T_3}{T_2}$ i.e. $p_3 = p_2 \times \frac{T_3}{T_2}$

$$\therefore p_3 = 1,500 \times \frac{2,220}{678.9} = 4,948 \text{ kPa}$$

From adiabatic compression (1-2) and adiabatic expansion (3 - 4),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma \quad \text{and} \quad \frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^\gamma \quad \therefore \frac{p_2}{p_1} = \frac{p_3}{p_4} \quad (\text{as } v_1 = v_4 \text{ and } v_2 = v_3)$$

$$\therefore p_4 = \frac{p_1}{p_2} \times p_3 = \frac{1}{15} \times 4,948 = 329.6 \text{ kPa.}$$

From constant volume heat rejection (4-1), $\frac{p_4 v_4}{T_4} = \frac{p_1 v_1}{T_1}$

Hence, as $v_4 = v_1$, $\frac{p_4}{T_4} = \frac{p_1}{T_1}$

$$\therefore T_4 = \frac{p_4}{p_1} \times T_1 = \frac{329.6}{100} \times 313 = 1,031.6 \text{ K or } t_4 = 758.6^\circ\text{C}$$

5.7 Diesel Cycle (Constant Pressure Cycle)

Internal combustion engines of today, operate on heat engine cycles which approximate either the ideal Otto cycle or the ideal Diesel cycle. In 1897, Dr. Rudolph Diesel constructed

the first successful Diesel engine. This engine was designed to operate on a new heat engine cycle devised by him and hence is known as *Diesel cycle*. Diesel engines have been used to a considerable extent in stationary marine and locomotive practice.

The ideal $p - v$ and $T - \phi$ diagrams of the cycle are shown in fig. 5-6.

In working out the air-standard efficiency of this cycle, the following assumptions are made :

- (i) The working fluid in the engine cylinder is air and it behaves as a perfect gas, i.e., it obeys the gas laws and has constant specific heats.
- (ii) The air is compressed adiabatically (without friction) in the engine cylinder during the compression stroke.
- (iii) Heat is supplied to the air at constant pressure by bringing a hot body in contact with the end of the cylinder.
- (iv) The air expands in the engine cylinder adiabatically (without friction) during expansion stroke.
- (v) Heat is abstracted from the working substance at constant volume by bringing a cold body in contact with the end of the cylinder.

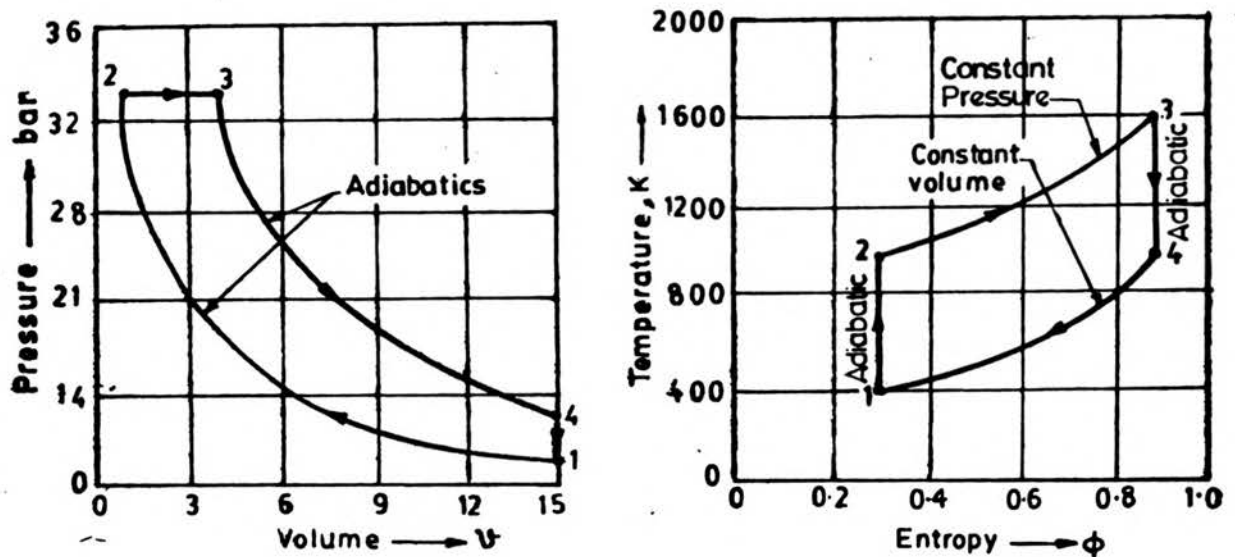


Fig. 5.6. $p-v$ and $T-\phi$ diagrams of Diesel cycle.

Imagine the cylinder to contain 1 kg of air at point (1). This air is compressed adiabatically to point (2) by the piston during its inward stroke. The air now occupies the clearance volume. The heat is then supplied at constant pressure by bringing a hot body in contact with the end of the cylinder. At point (3) the hot body is removed and the supply of heat is stopped. This point is known as the point of *cut-off*. The air now expands adiabatically to point (4). During this process, work is done on the piston. The air now occupies the whole cylinder volume. The cold body is then placed at the end of the cylinder and heat is abstracted from the working substance at constant volume until the pressure falls to point (1). This operation is represented by (4-1). The cold body is removed after the air is brought to its original condition (1). The cycle is thus completed.

The cycle consists of two adiabatic processes, one constant pressure process and one constant volume process. Heat is supplied during constant pressure process (2-3)

and heat is rejected during constant volume process (4-1). There is no exchange of heat during adiabatic processes (1-2) and (3-4)

Heat supplied during constant pressure process (2-3) = $K_p (T_3 - T_2)$ heat units per kg of air.

Heat rejected during constant volume process (4-1) = $K_v (T_4 - T_1)$ heat units per kg of air.

Net work done by the air = Heat supplied - Heat rejected

= $K_p (T_3 - T_2) - K_v (T_4 - T_1)$ heat units per kg of air.

A.S. efficiency, $\eta = \frac{\text{Net work done per kg of air}}{\text{Heat supplied per kg of air}}$

$$\begin{aligned} &= \frac{k_p (T_3 - T_2) - k_v (T_4 - T_1)}{k_p (T_3 - T_2)} \\ &= 1 - \frac{k_v}{k_p} \times \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1}{\gamma} \times \frac{T_4 - T_1}{T_3 - T_2} \end{aligned} \quad \dots(5.10)$$

Let compression ratio, $\frac{v_1}{v_2} = r$ and cut-off ratio, $\frac{v_3}{v_2} = \rho$

Then, expansion ratio, $\frac{v_4}{v_3} = \frac{v_4}{v_2} \times \frac{v_2}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3} = \frac{r}{\rho}$...(5.11)

From constant pressure heat addition (2-3), $\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$

Hence, as $p_2 = p_3$, $\frac{T_3}{T_2} = \frac{v_3}{v_2} = \rho \therefore T_2 = \frac{T_3}{\rho}$...(i)

From adiabatic compression (1-2), $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1}$

$\therefore T_1 = \frac{T_2}{(r)^{\gamma-1}}$...(ii)

Substituting value of T_2 from (i) in (ii), we get, $T_1 = \frac{T_3}{\rho (r)^{\gamma-1}}$...(iii)

From adiabatic expansion (3-4), $\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r}{\rho}\right)^{\gamma-1}$

$\therefore T_4 = \frac{T_3 (\rho)^{\gamma-1}}{(r)^{\gamma-1}}$ or $T_4 = \frac{T_3 (\rho)^\gamma}{\rho (r)^{\gamma-1}}$...(iv)

Substituting values of T_2 , T_1 and T_4 from (i), (iii) and (iv) in eqn. (5.10), we get,

$$\begin{aligned} \text{Air-standard efficiency} &= 1 - \frac{1}{\gamma} \times \frac{\frac{T_3 (\rho)^\gamma}{\rho (r)^{\gamma-1}} - \frac{T_3}{\rho}}{T_3 - \frac{T_3}{\rho}} \\ &= 1 - \frac{1}{\gamma} \times \frac{1}{(r)^{\gamma-1}} \left\{ \frac{\frac{T_3 (\rho)^\gamma}{\rho} - \frac{T_3}{\rho}}{T_3 - \frac{T_3}{\rho}} \right\} \end{aligned}$$

$$\begin{aligned}
 \text{A.S.E.} &= 1 - \frac{1}{\gamma} \times \frac{1}{(r)^{\gamma-1}} \left\{ \frac{(\rho)^{\gamma} - 1}{\rho} \right\} \\
 &= 1 - \frac{1}{\gamma} \times \frac{1}{(r)^{\gamma-1}} \left\{ \frac{(\rho)^{\gamma} - 1}{\rho - 1} \right\} \\
 &= 1 - \frac{1}{(r)^{\gamma-1}} \times \left\{ \frac{(\rho)^{\gamma} - 1}{\gamma(\rho - 1)} \right\} \quad \dots(5.12)
 \end{aligned}$$

This expression is the air-standard efficiency of the Diesel cycle. It will be noted from the expression (5.12), that the air-standard efficiency of the Diesel cycle depends upon the value of r and ρ . The efficiency increases as r is increased and decreases as ρ is increased.

The factor $\left[\frac{(\rho)^{\gamma} - 1}{\gamma(\rho - 1)} \right]$ depends upon the value of cut-off ratio and is greater than unity; hence the air-standard efficiency of the Diesel cycle for a given compression ratio is less than $1 - \left(\frac{1}{r} \right)^{\gamma-1}$, which is the efficiency of Otto cycle.

Problem - 6 : An engine working on Diesel cycle has a compression ratio of 15 and cut-off takes place at 5% of the stroke. Find the air-standard efficiency. Assume value of $\gamma = 1.4$ for air.

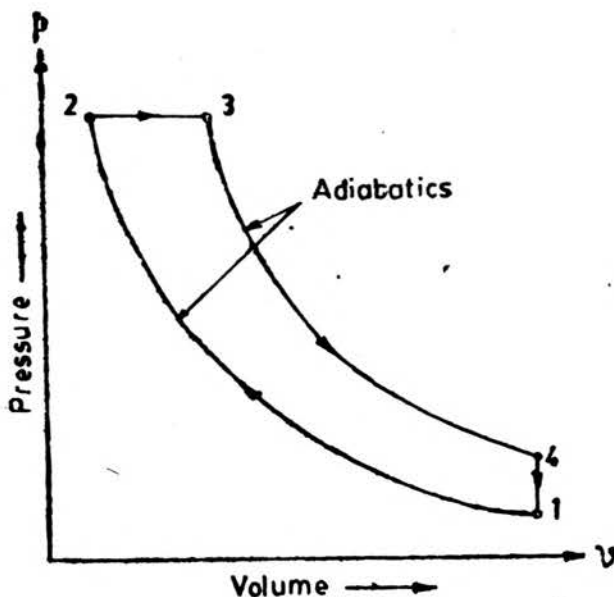


Fig. 5-7. p - v diagram of Diesel cycle.

Referring to fig. 5-7,

Compression ratio, r

$$= \frac{v_1}{v_2} = 15 \quad \therefore v_1 = 15v_2$$

Now, stroke volume

$$= v_1 - v_2 = 15v_2 - v_2 = 14v_2$$

$v_3 = 5\%$ of stroke volume + clearance volume.

$$\therefore v_3 = \left(\frac{5}{100} \times 14v_2 \right) + v_2 = 1.7v_2$$

$$\text{Cut-off ratio, } \rho = \frac{v_3}{v_2} = \frac{1.7v_2}{v_2} = 1.7$$

Using eqn. (5.12), Air-standard efficiency

$$\begin{aligned}
 &= 1 - \frac{1}{(r)^{\gamma-1}} \left\{ \frac{(\rho)^{\gamma} - 1}{\gamma(\rho - 1)} \right\} \\
 &= 1 - \frac{1}{(15)^{0.4}} \times \left[\frac{(1.7)^{1.4} - 1}{1.4(1.7 - 1)} \right] \\
 &= 0.625 \text{ or } 62.5\%
 \end{aligned}$$

Problem - 7 : An air engine works on the following cycle: Air is taken in at atmospheric pressure of 110 kPa and temperature of 16°C , and is compressed adiabatically, the pressure at the end of the stroke being 3,500 kPa. Heat is taken in at constant pressure, the expansion afterwards takes place adiabatically, the ratio of expansion being 5. The air is exhausted at the end of the

stroke, the heat is assumed to be rejected at constant volume. Find the ideal thermal efficiency. Take the specific heats of air as $k_p = 1.0035$ kJ/kg K and $k_v = 0.7165$ kJ/kg K.

$$\text{Here, } \gamma = \frac{K_p}{K_v} = \frac{1.0035}{0.7165} = 1.4 \text{ and } \frac{\gamma - 1}{\gamma} = \frac{0.4}{1.4} = 0.286, \frac{1}{\gamma} = \frac{1}{1.4} = 0.715;$$

$$p_1 = 110 \text{ kPa; } p_2 = p_3 = 3,500 \text{ kPa; } T_1 = 273 + 16 = 289 \text{ K.}$$

Referring to fig. 5-8 and considering adiabatic compression (1-2),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2} \right)^\gamma = (r)^\gamma$$

$$\begin{aligned} \therefore r &= \left(\frac{p_2}{p_1} \right)^{\frac{1}{\gamma}} = \left(\frac{3,500}{110} \right)^{0.715} \\ &= 11.8 \text{ (compression ratio)} \end{aligned}$$

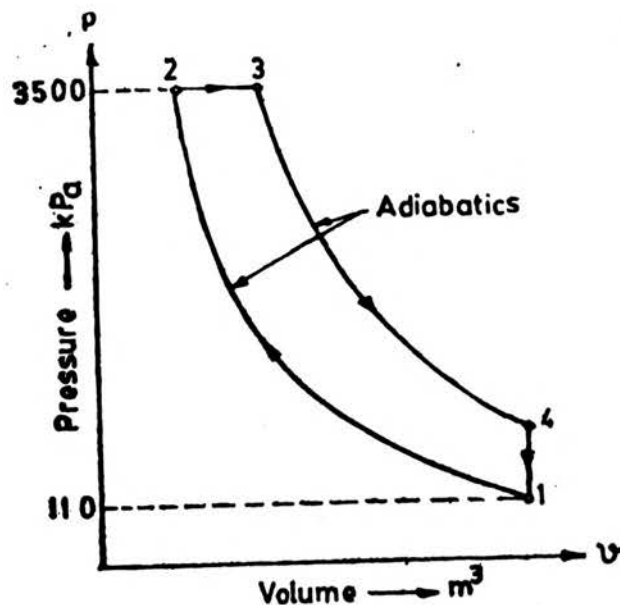


Fig. 5-8. p - v diagram of Diesel cycle.

$$\text{Hence, as } p_2 = p_3, \frac{T_3}{T_2} = \frac{v_3}{v_2}$$

$$\therefore T_3 = \frac{v_3}{v_2} \times T_2 = 2.36 \times 778 = 1,838 \text{ K.}$$

$$\text{Considering adiabatic expansion (3-4), } \frac{T_4}{T_3} = \left(\frac{v_3}{v_4} \right)^{\gamma - 1}$$

$$\therefore T_4 = T_3 \times \left(\frac{v_3}{v_4} \right)^{\gamma - 1} = 1,838 \times \left(\frac{1}{5} \right)^{0.4} = 1,838 \times 0.524 = 960 \text{ K.}$$

Now, heat supplied per kg of air

$$= k_p (T_3 - T_2) = 1.0035 (1,838 - 778) = 1,063.71 \text{ kJ per kg of air, and heat rejected per kg of air} = k_v (T_4 - T_1) = 0.7165 (960 - 289) = 480.7 \text{ kJ per kg of air.}$$

Hence, heat converted to work or work done per kg of air

$$= k_p (T_3 - T_2) - k_v (T_4 - T_1) = 1,063.71 - 480.7 = 583.01 \text{ kJ/kg of air.}$$

Also the ratio of expansion, $\frac{v_4}{v_3} = 5$ (given).

$$\text{Using eqn. (5.11), cut-off ratio } \left(\frac{v_3}{v_2} \right) =$$

$$\rho = \frac{\text{Ratio of compression}}{\text{Ratio of expansion}}$$

$$= \frac{11.8}{5} = 2.36$$

Considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{3,500}{110} \right)^{0.286} = 2.69$$

$$\therefore T_2 = T_1 \times 2.69 = 289 \times 2.69 = 778 \text{ K.}$$

From constant pressure heat addition

$$(2-3), \frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$$

Ideal thermal efficiency of Air standard efficiency

$$= \frac{\text{work done per kg. of air}}{\text{heat supplied per kg. of air}} = \frac{583.01}{1,063.7} = 0.5483 \text{ or } 54.83\%$$

Alternatively, using eqn. (5.12), the ideal thermal efficiency or air-standard efficiency.

$$\text{A.S.E.} = 1 - \frac{1}{(r)^{\gamma-1}} \times \left[\frac{(\rho)^{\gamma} - 1}{\gamma(\rho - 1)} \right]$$

Here, ratio of compression, $r = 11.8$, cut-off ratio, $\rho = 2.36$, $\gamma = 1.4$.

On substitution of values in eqn. (5.12), we get,

$$\begin{aligned} \text{Air standard efficiency} &= 1 - \frac{1}{(11.8)^{1.4-1}} \times \left[\frac{(2.36)^{1.4} - 1}{1.4(2.36 - 1)} \right] \\ &= 1 - 0.371 \left[\frac{3.32 - 1}{1.904} \right] \\ &= 1 - 0.4517 = 0.5483 \text{ i.e. } 54.83\% \text{ (same as before)} \end{aligned}$$

Problem - 8 : The following data relate to a theoretical Diesel cycle, using air as the working fluid :

Pressure at the end of suction stroke	...100 kPa
Temperature at the end of suction stroke	...30°C
Temperature at the end of constant pressure heat addition	...1,500°C
Compression ratio	...16
Specific heat of air at constant pressure	...1.005 kJ/Kg K
Specific heat of air at constant volume	...0.7115 kJ/kg K

Find : (a) the percentage of stroke at which cut-off takes place, (b) the temperature at the end of expansion stroke, and (c) the ideal thermal efficiency.

Here, $p_1 = 100 \text{ kPa}$; $t_1 = 30^\circ \text{C}$; $t_3 = 1,500^\circ \text{C}$; $\frac{v_1}{v_2} = r = 16$; $\gamma = \frac{k_p}{k_v} = \frac{1.005}{0.7115} = 1.41$

(a) Referring to fig. 5-9 and considering adiabatic compression 1-2,

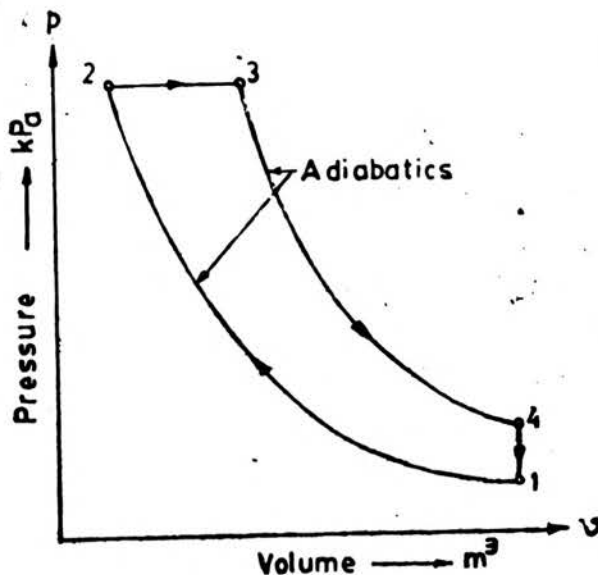


Fig. 5-9. p - v diagram of Diesel cycle

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2} \right)^{\gamma-1} = (r)^{\gamma-1} \text{ or}$$

$$T_2 = T_1 \times (r)^{\gamma-1}$$

$$\begin{aligned} \therefore T_2 &= (30 + 273) \times (16)^{0.41} \\ &= 303 \times 3.117 = 945 \text{ K.} \end{aligned}$$

From constant pressure heat addition (2-3),

$$\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$$

$$\text{Hence, as } p_2 = p_3, \frac{v_3}{v_2} = \frac{T_3}{T_2}$$

$$\therefore \frac{v_3}{v_2} = \frac{(1,500 + 273)}{945} = 1.876 \text{ or } v_3 = 1.876 v_2$$

Now percentage of the stroke at which cut-off takes place

$$= \frac{\text{Volume at cut-off} - \text{Clearance volume}}{\text{Stroke volume}} \times 100$$

$$= \frac{v_3 - v_2}{v_1 - v_2} \times 100 = \frac{1.876v_2 - v_2}{16v_2 - v_2} \times 100 = 5.84\%$$

(b) $\frac{v_4}{v_3} = \frac{v_4}{v_2} \times \frac{v_2}{v_3} = \frac{v_1}{v_2} \times \frac{v_2}{v_3}$
 (as $v_4 = v_1$)

∴ Expansion ratio, $\frac{v_4}{v_3} = 16 \times \frac{1}{1.876} = 8.529$

From adiabatic expansion (3 - 4),

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} \quad \therefore T_4 = \frac{T_3}{\left(\frac{v_4}{v_3}\right)^{\gamma-1}}$$

∴ $T_4 = \frac{1,773}{(8.529)^{0.41}} = \frac{1,773}{2.408} = 736 \text{ K or } T_4 = 463^\circ\text{C}$

(c) Heat supplied = $kp (T_3 - T_2) = 1.005 (1,773 - 945) = 832.14 \text{ kJ/kg of air.}$

Heat rejected = $kv (T_4 - T_1) = 0.7115 (736 - 303) = 308.07 \text{ kJ/kg of air.}$

∴ Heat converted to work or work done = $832.14 - 308.07 = 524.07 \text{ kJ/kg of air.}$

Now, ideal thermal eff.] = $\frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}} = \frac{524.07}{832.14} = 0.6286 \text{ or } 62.86 \%$
 or A.S. efficiency

5.8 Dual-combustion Cycle

This cycle is known as a dual-combustion or mixed cycle because the heat is taken in partly at constant volume and partly at constant pressure. This cycle is used in modern high speed oil engines and is a combination of Otto and Diesel cycles. Engines working on this cycle are sometimes called *Semi-Diesel* engines.

The ideal $p - v$ and $T - \phi$ diagrams for this cycle are shown in fig. 5-10. In working out the air-standard efficiency of this cycle the following assumptions are made :

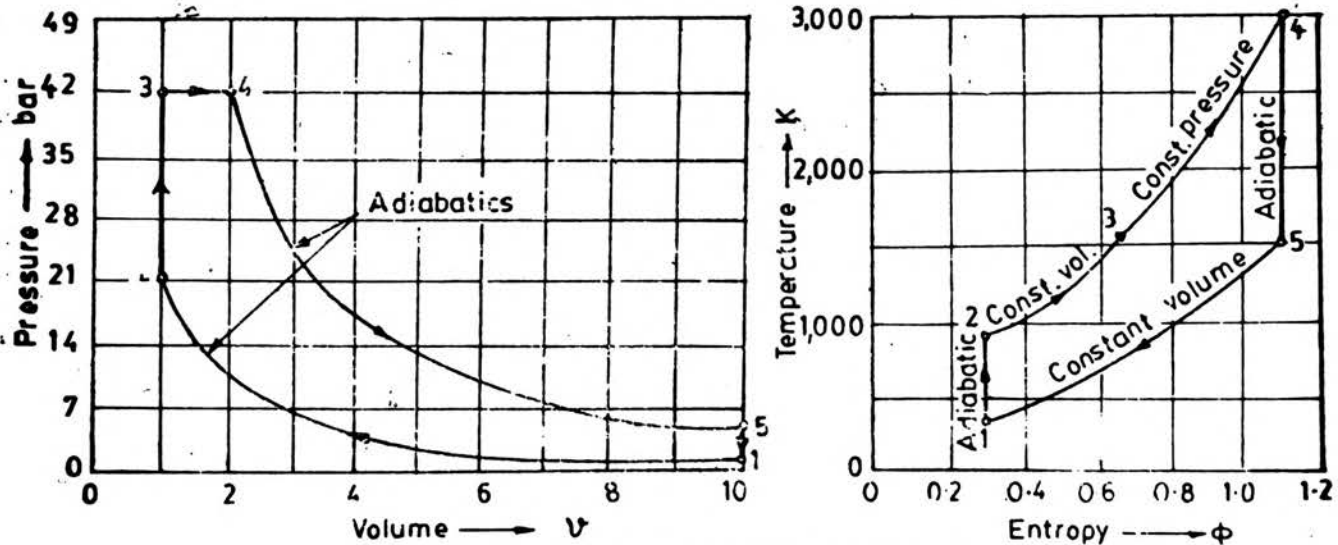


Fig. 5-10. $p-v$ and $T-\phi$ diagrams of dual-combustion cycle.

(i) The working fluid in the engine cylinder is air and behaves as a perfect gas, i.e., it obeys gas laws and has constant specific heats.

(ii) The working fluid is compressed in the engine cylinder adiabatically (without friction) during compression stroke.

(iii) Heat is partly supplied at constant volume to the working fluid by bringing a hot body in contact with the end of the cylinder. The source of heat is still maintained while the piston moves outward during the working stroke and the remaining heat is supplied to the working fluid at constant pressure. The hot body is then removed after the first portion of the working stroke is completed.

(iv) The working fluid expands in the engine cylinder adiabatically (without friction) during the expansion stroke.

(v) Heat is abstracted from the working fluid (substance) at constant volume by bringing a cold body in contact with the end of the cylinder.

Imagine the cylinder to contain one kg of air at point (1). This air is compressed adiabatically to point (2) by the piston during its inward stroke. The air now occupies the clearance volume. The heat is then supplied at constant volume by bringing a hot body in contact with the end of the cylinder. This operation is represented by line (2-3). The hot body is still maintained at the end of the cylinder during first portion of the working stroke and heat is supplied at constant pressure. This operation is represented by line (3-4). The hot body is then removed and the supply of heat is stopped at point 4. The air now expands adiabatically to point (5). During this process work is done on the piston. The air now occupies the whole cylinder volume. The cold body is then placed at the end of the cylinder and heat is abstracted from the working fluid at constant volume until the pressure falls to point (1). This operation is represented by line (5-1). The cold body is then removed and air is brought to its original condition (1). The cycle is thus completed.

This cycle consists of two adiabatic processes, two constant volume processes and one constant pressure process. Heat is supplied during constant volume process (2-3) and constant pressure process (3-4). Heat is rejected during constant volume process (5-1). There is no exchange of heat during the adiabatics (1-2) and (4-5). Referring to fig. 5-10,

Heat supplied = $k_v (T_3 - T_2) + k_p (T_4 - T_3)$ heat units/kg of air and

Heat rejected = $k_v (T_5 - T_1)$ heat units/kg of air.

∴ Net work done = $k_v (T_3 - T_2) + k_p (T_4 - T_3) - k_v (T_5 - T_1)$ heat units/kg of air.

$$\text{A.S.E.} = \frac{\text{Work done/kg of air}}{\text{Heat supplied/kg of air}}$$

$$= \frac{k_v(T_3 - T_2) + k_p(T_4 - T_3) - k_v(T_5 - T_1)}{k_v(T_3 - T_2) + k_p(T_4 - T_3)}$$

$$= 1 - \frac{k_v(T_5 - T_1)}{k_v(T_3 - T_2) + k_p(T_4 - T_3)}$$

$$= 1 - \frac{(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)}$$

...(5.13)

Let compression ratio, $\frac{v_1}{v_2} = r$ and cut-off ratio, $\frac{v_4}{v_3} = \rho$,

Then, expansion ratio, $\frac{v_5}{v_4} = \frac{v_1}{v_2} \times \frac{v_3}{v_4} = \frac{r}{\rho}$ (as $v_5 = v_1$ and $v_2 = v_3$)

Let pressure ratio or explosion ratio, $\frac{p_3}{p_2} = \beta$

From constant volume heat addition (2-3), $\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$

Hence, as $v_2 = v_3$, $\frac{T_3}{T_2} = \frac{p_3}{p_2} = \beta \quad \therefore T_2 = \frac{T_3}{\beta}$... (i)

From constant pressure heat addition (3-4), $\frac{T_4}{T_3} = \frac{v_4}{v_3} = \rho$ or $T_4 = \rho T_3$... (ii)

From adiabatic compression (1-2), $\frac{T_2}{T_1} = (r)^{\gamma-1}$ or $T_1 = \frac{T_2}{(r)^{\gamma-1}}$... (iii)

Substituting value of T_2 from (i) in (iii), we get, $T_1 = \frac{T_3}{\beta (r)^{\gamma-1}}$... (iv)

From adiabatic expansion (4-5), $\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r}{\rho}\right)^{\gamma-1}$

$\therefore T_5 = \frac{T_4}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = \frac{T_4 (\rho)^{\gamma-1}}{(r)^{\gamma-1}}$... (v)

Substituting value of T_4 from (ii) in (v), we get,

$T_5 = \frac{\rho T_3 (\rho)^{\gamma-1}}{(r)^{\gamma-1}} = \frac{\rho^\gamma T_3}{(r)^{\gamma-1}}$... (vi)

Substituting values of T_2 , T_4 , T_1 and T_5 from (i), (ii), (iv) and (vi) in eqn. (5.13), we get,

$$\begin{aligned} \text{Air-standard efficiency (A.S.E.)} &= 1 - \frac{\frac{\rho^\gamma T_3}{(r)^{\gamma-1}} - \frac{T_3}{\beta (r)^{\gamma-1}}}{\left(T_3 - \frac{T_3}{\beta}\right) + \gamma (\rho T_3 - T_3)} \\ &= 1 - \frac{\frac{\rho^\gamma}{(r)^{\gamma-1}} - \frac{1}{\beta (r)^{\gamma-1}}}{\left(1 - \frac{1}{\beta}\right) + \gamma (\rho - 1)} \\ &= 1 - \frac{1}{(r)^{\gamma-1}} \left\{ \frac{\left(\rho^\gamma - \frac{1}{\beta}\right)}{\left(1 - \frac{1}{\beta}\right) + \gamma (\rho - 1)} \right\} \\ &= 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{\beta (\rho)^\gamma - 1}{(\beta - 1) + \beta \gamma (\rho - 1)} \right] \end{aligned} \quad \dots(5.14)$$

This is the air-standard efficiency of the dual-combustion cycle.

Now, in eqn. (5.14), if pressure ratio, $\beta = 1$, i.e. $p_3 = p_2$,

$$\text{A.S.E.} = 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{(\rho)^\gamma - 1}{\gamma (\rho - 1)} \right]$$

which is the expression of the A.S.E. of the Diesel cycle.

Again, in eqn. (5.14), if cut-off ratio, $\rho = 1$, i.e. $v_3 = v_4$,

$$\begin{aligned} \text{A.S.E.} &= 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{(\beta-1)}{(\beta-1) + \beta\gamma(0)} \right] \\ &= 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{\beta-1}{\beta-1} \right] = 1 - \frac{1}{(r)^{\gamma-1}} \end{aligned}$$

which is the expression of the Air-standard efficiency (A.S.E.) of the Otto cycle.

Problem - 9 : An oil engine working on the dual-combustion cycle has a cylinder diameter of 25 cm and stroke of 36 cm. The clearance volume is 1,600 cm³ and cut-off takes place at 5 per cent of the stroke. The explosion pressure ratio is 1.4. Find the air-standard efficiency of the engine. Assume $\gamma = 1.4$ for air.

$$\text{Stroke volume, } v_s = \frac{\pi}{4} d^2 \times l = \frac{\pi}{4} (25)^2 \times 36 = 17,600 \text{ cm}^3,$$

$$\text{Clearance volume, } v_c = 1,600 \text{ cm}^3 \text{ (given).}$$

$$\text{Compression ratio, } r = \frac{v_c + v_s}{v_c} = \frac{1,600 + 17,600}{1,600} = 12$$

$$\text{Cut-off ratio, } \rho = \frac{v_c + (0.05 v_s)}{v_c} = \frac{1,600 + (0.05 \times 17,600)}{1,600} = 1.55$$

$$\text{Explosion pressure ratio, } \beta = 1.4 \text{ (given).}$$

$$\begin{aligned} \text{Using eqn. (5.14), Air-standard efficiency} &= 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{\beta(\rho)^{\gamma} - 1}{(\beta-1) + \beta\gamma(\rho-1)} \right] \\ &= 1 - \frac{1}{(12)^{0.4}} \left[\frac{1.4(1.55)^{1.4} - 1}{(1.4-1) + 1.4 \times 1.4(1.55-1)} \right] \\ &= 1 - \frac{1}{2.7} \left[\frac{2.68 - 1}{0.4 + 1.08} \right] = 0.605 \text{ or } 60.5\% \end{aligned}$$

Problem - 10 : In an engine working on the ideal dual-combustion cycle, the temperature and pressure at the beginning of compression are 100°C and 100 kPa respectively. The compression ratio is 10:1. If the maximum pressure is limited to 7,000 kPa and 1,675 kJ of heat is supplied per kg of air, determine the temperatures at salient (key) points of the cycle and the air standard efficiency of the engine. Assume $k_p = 1.01 \text{ kJ/kg K}$ and $k_v = 0.716 \text{ kJ/kg K}$ for air.

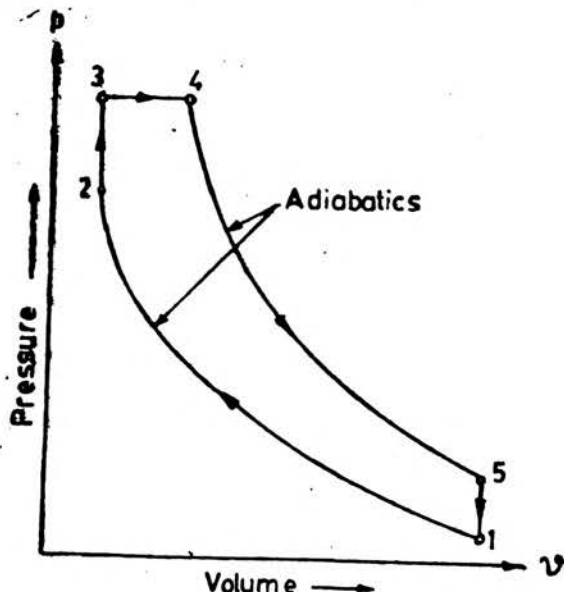


Fig. 5.11. p-v diagram of dual-combustion cycle.

$$\begin{aligned} \text{Here, } p_1 &= 100 \text{ kPa;} \\ T_1 &= 100 + 273 = 373\text{K;} \\ p_3 &= p_4 = 7,000 \text{ kPa;} \\ r &= \frac{v_1}{v_2} = 10; \gamma = \frac{k_p}{k_v} = \frac{1.01}{0.716} = 1.41 \end{aligned}$$

Referring to fig 5-11, and considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2} \right)^{\gamma-1} = (r)^{\gamma-1}$$

$$\therefore T_2 = T_1 \times (r)^{\gamma-1}$$

$$= 373 \times (10)^{0.41} = 958.8 \text{ K or } t_2 = 685.8^\circ\text{C}.$$

$$\text{Again, } \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma \text{ or } p_2 = p_1 \left(\frac{v_1}{v_2}\right)^\gamma$$

$$\therefore p_2 = p_1 (r)^\gamma = 100 \times (10)^{1.41} = 2,570 \text{ kPa}$$

$$\text{From constant volume heat addition (2-3), } \frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$$

$$\text{Hence, as } v_2 = v_3, \frac{T_3}{T_2} = \frac{p_3}{p_2} \text{ or } T_3 = T_2 \times \frac{p_3}{p_2}$$

$$\therefore T_3 = 958.8 \times \frac{7,000}{2,570} = 2,612 \text{ K or } t_3 = 2,339^\circ\text{C}$$

$$\text{Heat added at constant volume (2-3) per kg. of air} \\ = K_v (T_3 - T_2) = 0.716 (2,612 - 958.8) = 1,183.7 \text{ kJ/kg of air.}$$

$$\therefore \text{Heat added at constant pressure (3-4) per kg of air} \\ = 1,675 - 1,183.7 = 491.3 \text{ kJ per kg of air} = k_p (T_4 - T_3)$$

$$\text{i.e., } T_4 = T_3 + \frac{491.3}{k_p} = 2,612 + \frac{491.3}{1.01} = 3,098 \text{ K or } t_4 = 2,825^\circ\text{C}.$$

$$\text{From constant pressure heat addition (3-4), } \frac{p_3 v_3}{T_3} = \frac{p_4 v_4}{T_4}$$

$$\text{Hence, as } p_3 = p_4, \frac{v_4}{v_3} = \frac{T_4}{T_3} = \frac{3,098}{2,612} = 1.186 = \rho \text{ (cut-off ratio).}$$

$$\text{From adiabatic expansion (4-5), } \frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1} = \left(\frac{r}{\rho}\right)^{\gamma-1}$$

$$\therefore T_5 = \frac{T_4}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = \frac{3,098}{\left(\frac{10}{1.186}\right)^{0.41}} = 1,293 \text{ K or } t_5 = 1,020^\circ\text{C}$$

$$\text{Heat rejected at constant volume (5-1) per kg of air} \\ = K_v (T_5 - T_1) = 0.716 (1,293 - 373) = 658.72 \text{ kJ/kg of air}$$

$$\therefore \text{Work done per kg of air} = \text{Heat supplied per kg of air} - \text{Heat rejected per kg of air}$$

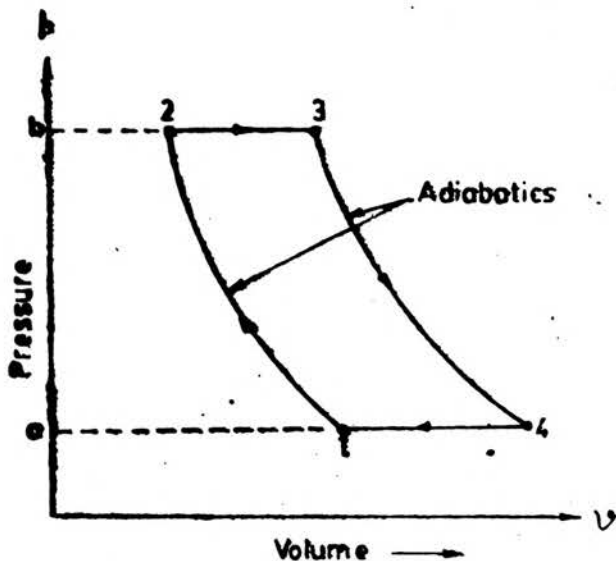
$$= 1,675 - 658.72 = 1,016.28 \text{ kJ/kg of air.}$$

$$\therefore \text{Air-standard efficiency} = \frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}} = \frac{1,016.28}{1,675} = 0.6067 \text{ or } 60.67\%$$

5.9 Joule Cycle

In 1851, Dr. Joule proposed to use a cycle in which heat was received and rejected at constant pressure and called this cycle as constant pressure cycle. This cycle is used in gas turbine plant of the constant pressure type. In the year 1873, Mr. Brayton used Joule air cycle in open cycle constant pressure gas turbine plants and hence Joule cycle is also known as Brayton cycle. This Joule (or Brayton) cycle, used in gas turbine plants, is described in Volume III.

The cycle, consisting of two constant pressure processes (2-3) and (4-1), and two adiabatic processes (3-4) and (1-2), was suggested by Joule for use in hot air engine. The engine consists of an expansion cylinder, a compression cylinder, a heating chamber maintained at temperature T_3 by means of a heater, and cooling chamber

Fig. 5-12. p - v diagram of Joule cycle.

this operation, work is done on the piston, provides excess work and also drives the compressor. This operation is represented by (3-4). During the return stroke of the piston of the expansion cylinder, the low temperature air at T_4 is delivered through a valve to a cooling chamber where it is further cooled at constant pressure from T_4 to T_1 . The cycle is thus completed.

Work done by the air on the piston of the expansion cylinder is given by the area b -3-4- a ; the work done on the air in the compression cylinder is given by the area b -2-1- a . Hence, the net work done per cycle is represented by the area 1-2-3-4. Heat supplied per kg of air = $k_p (T_3 - T_2)$ heat units. Heat rejected per kg of air

$$= k_p (T_4 - T_1) \text{ heat units.}$$

Since there is no exchange of heat during the frictionless adiabatic processes,

Net work done = Heat supplied - Heat rejected

$$= k_p (T_3 - T_2) - k_p (T_4 - T_1) \text{ heat units per kg of air.}$$

$$\begin{aligned} \text{Air-standard efficiency} &= \frac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}} \\ &= \frac{k_p (T_3 - T_2) - k_p (T_4 - T_1)}{k_p (T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \quad \dots 5.15 \end{aligned}$$

Now, since the adiabatic expansion and adiabatic compression both take place between the same terminal pressure (same pressure ratio), the ratio of compression and expansion are equal. Calling this common ratio r , we have,

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

$$\text{Similarly, } \frac{T_3}{T_4} = (r)^{\gamma-1} \quad \text{or} \quad T_3 = T_4 (r)^{\gamma-1}$$

Substituting the values of T_2 and T_3 in the eqn. (5-15) we get,

$$\begin{aligned} \text{Air-standard efficiency} &= 1 - \frac{T_4 - T_1}{T_4 (r)^{\gamma-1} - T_1 (r)^{\gamma-1}} \\ &= 1 - \frac{1}{(r)^{\gamma-1}} \quad \dots (5.16) \end{aligned}$$

maintained at temperature T_1 by means of cooling water.

The cycle of operation is as follows :

Cold air from cooling chamber at temperature T_1 is drawn in the compression cylinder. This operation is represented by (a-1) on the p - v diagram as shown in fig. 5-12. The air is now compressed adiabatically to temperature T_2 and delivered through a valve to a heater where it is heated from temperature T_2 to T_3 at constant pressure, increasing its volume from v_2 to v_3 . This hot air passes into the working or expansion cylinder where it is allowed to expand adiabatically until its pressure and temperature fall to p_4 and T_4 respectively. During

The efficiency expression, $1 - \frac{1}{(r)^{\gamma-1}}$ can also be expressed in terms of temperatures T_3 and T_4

$$\text{We know that } \frac{T_3}{T_4} = (r)^{\gamma-1} \text{ or } \frac{T_4}{T_3} = \frac{1}{(r)^{\gamma-1}}$$

Substituting the value of $\frac{1}{(r)^{\gamma-1}}$ in the eqn. (5-15), we get,

$$\text{Air-standard efficiency} = 1 - \frac{T_4}{T_3} = \frac{T_3 - T_4}{T_3} \quad \dots(5-17a)$$

Now, though T_3 is the maximum temperature, T_4 is greater than the minimum temperature T_1 , so that efficiency of this cycle is less than Carnot cycle efficiency when operating between same limits of maximum and minimum temperatures.

The expression of air-standard efficiency, as given in eqn. 5-16, is in terms of volume ratio $\left(r = \frac{v_1}{v_2}\right)$. It can also be given in terms of pressure ratio $\left(r_p = \frac{p_2}{p_1}\right)$ as under :

$$\text{Air-standard efficiency} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \quad \dots (5-17b)$$

$$\text{as } \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ or } \frac{T_2}{T_1} = (r)^{\gamma-1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

Although no engine was constructed to work on this cycle, the reversed cycle, i.e., Joule air engine reversed in direction was extensively used in refrigeration for a number of years. The reversed Joule cycle, known as *Bell-Coleman cycle*, is described in volume III.

Problem – 11 : A gas turbine working on Joule cycle takes in air at an atmospheric pressure of 110 kPa and 20°C. The air is compressed adiabatically to a pressure of 300 kPa in the compressor. Heat is then added at constant pressure in combustion chamber and then expanded adiabatically to atmospheric pressure in the turbine. If the maximum temperature is limited to 550°C, find the air-standard efficiency of the cycle. Assume $k_p = 0.9965$ kJ/kg and $\gamma = 1.4$ for air.

Referring to fig. 5-12, $p_1 = 110$ kPa; $T_1 = 20 + 273 = 293$ K;

$p_2 = p_3 = 300$ kPa; $T_3 = 550 + 273 = 823$ K.

$$\frac{p_3}{p_4} = (r)^\gamma \text{ and } \frac{p_2}{p_1} = (r)^\gamma$$

$$\therefore \frac{p_3}{p_4} = \frac{p_2}{p_1} = \frac{300}{110} \quad (\text{ratio of expansion} = \text{ratio of compression})$$

$$\text{From adiabatic expansion (3 - 4), } \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{300}{110}\right)^{\frac{0.4}{1.4}} = 1.332$$

$$\therefore T_4 = \frac{T_3}{1.332} = \frac{823}{1.332} = 617.7\text{K}$$

$$\text{Using eqn. (5.17a), Air-standard efficiency} = \frac{T_3 - T_4}{T_3} = \frac{823 - 617.7}{823} = 0.25 \text{ or } 25\%$$

5.10 Mean Effective Pressure (M.E.P.)

The mean effective pressure (M.E.P.) of a cycle or heat engine is the average net pressure in newtons per unit area that operates on the piston throughout its stroke. It is then the average height of the p - v diagram of the cycle or indicator diagram of any actual engine.

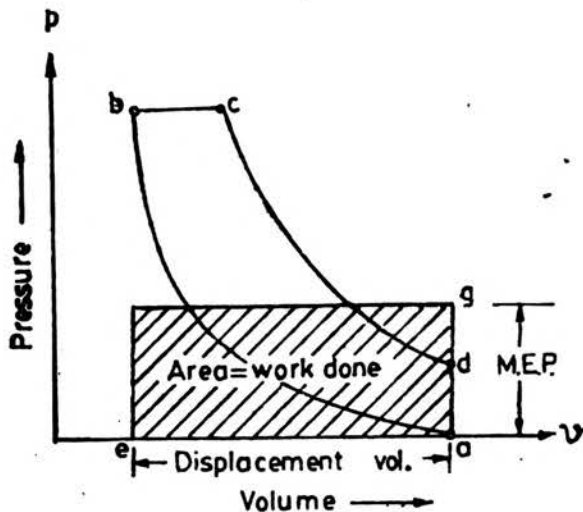


Fig. 5-13. Explanation of mean effective pressure.

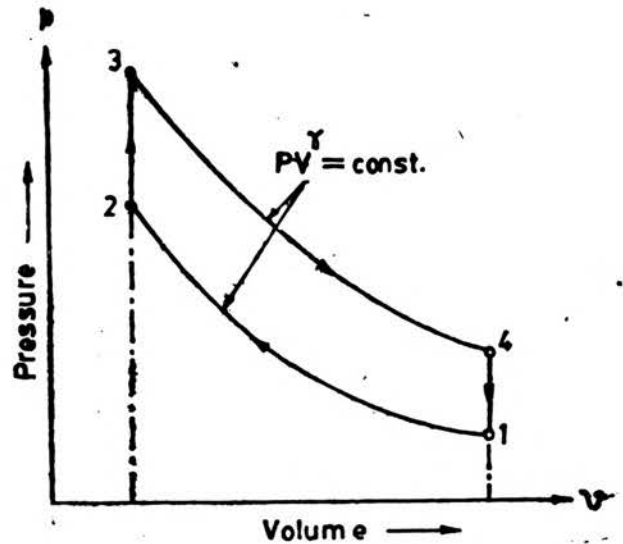


Fig. 5-14. p - v diagram of Otto cycle.

In fig. 5-13, a - b - c - d shows the p - v diagram and a - e - f - g the equivalent rectangular diagram. The area of the rectangle a - e - f - g is equal to area of indicator diagram a - b - c - d . Since the area of the p - v diagram is equal to the net work of the cycle in kJ, it is evident that,

$$\begin{aligned} \text{M.E.P.} &= \frac{\text{Work done per cycle in kJ}}{\text{Displacement volume in m}^3} \text{ kPa or kN/m}^2 \\ &= \frac{\text{Work done per cycle in kJ}}{v_a - v_e \text{ in m}^3} \text{ kPa or kN/m}^2 \end{aligned} \quad \dots(5.18)$$

where, v_a = total cylinder volume in m^3 ,

v_e = clearance volume in m^3 , and

$v_a - v_e$ = piston displacement volume in m^3 .

The mean effective pressure of the ideal cycles used in modern internal combustion engines is obtained as follows :

5.10.1 Otto cycle : The p - v diagram of the ideal Otto cycle is shown in fig. 5-14.

Work done per cycle = Area 1-2-3-4

= area under adiabatic expansion (3-4) minus area under adiabatic compression (1-2)

$$= \left[\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ}$$

where, pressure are in kPa and volumes in m^3 .

$$\text{Ideal M.E.P.} = \frac{\text{Work done per cycle in kJ}}{v_1 - v_2 \text{ in m}^3} \text{ kPa or kN/m}^2 \quad \dots(5.19)$$

where, $v_1 - v_2$ = piston displacement volume in m^3 .

Problem - 12 : Show that the ideal M.E.P. of the Otto cycle is given by

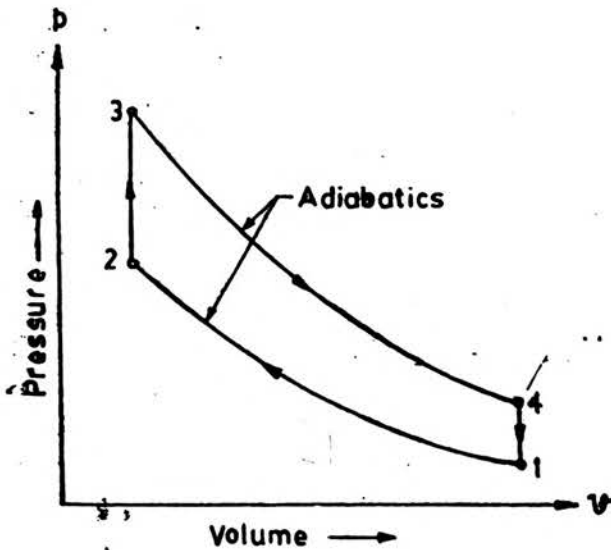


Fig. 5-15. p-v diagram of Otto cycle.

$$\frac{p_1 r (\beta - 1) (r^{\gamma-1} - 1)}{(r - 1) (\gamma - 1)}$$

where, p_1 = pressure at the beginning of compression,

r = compression ratio, and

β = ratio of maximum pressure to compression pressure.

Referring to fig. 5-15,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r)^{\gamma} \quad \therefore p_2 = p_1 (r)^{\gamma}$$

$$\text{Now, } \frac{p_3}{p_2} = \beta \quad \therefore p_3 = p_2 \beta = p_1 (r)^{\gamma} \beta$$

$$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = (r)^{\gamma}$$

$$\therefore p_4 = \frac{p_3}{(r)^{\gamma}} = \frac{p_1 r^{\gamma} \beta}{(r)^{\gamma}} = p_1 \beta$$

$$\text{Now, } \frac{v_1}{v_2} = r \quad \therefore v_1 = r v_2 = v_4$$

$$v_3 = v_2$$

Work done per cycle = area 1-2-3-4

= area under (3-4) minus area under (2-1)

$$= \frac{p_3 v_3 - p_4 v_4}{(\gamma - 1)} - \frac{p_2 v_2 - p_1 v_1}{(\gamma - 1)}$$

$$\text{M.E.P.} = \frac{\text{area of the diagram}}{\text{length of diagram}} = \frac{\text{Work done per cycle}}{(v_1 - v_2)}$$

$$= \frac{1}{(v_1 - v_2)} \left[\frac{(p_3 v_3 - p_4 v_4) - (p_2 v_2 - p_1 v_1)}{(\gamma - 1)} \right]$$

$$= \frac{1}{v_2 (r - 1)} \left[\frac{(p_1 (r)^{\gamma} \beta v_2 - p_1 \beta v_2 r) - (p_1 (r)^{\gamma} v_2 - p_1 r v_2)}{(\gamma - 1)} \right]$$

$$= \frac{1}{(r - 1)} \left[\frac{(p_1 (r)^{\gamma} \beta - p_1 \beta r) - (p_1 (r)^{\gamma} - p_1 r)}{(\gamma - 1)} \right]$$

$$= \frac{p_1 r \left[\beta \left((r)^{\gamma-1} - 1 \right) - \left((r)^{\gamma-1} - 1 \right) \right]}{(r - 1) (\gamma - 1)}$$

$$= \frac{p_1 r (\beta - 1) (r^{\gamma-1} - 1)}{(r - 1) (\gamma - 1)}$$

Problem - 13 : In an ideal Otto cycle the charge taken in is assumed to be air at a temperature of 20°C and a pressure of 110 kPa. If the clearance volume is 25 per

cent of the swept volume and the temperature at the end of the constant volume heat addition is $1,440^{\circ}\text{C}$, find the ideal mean effective pressure in kPa. Take $\gamma = 1.4$ for air.

Referring to fig. 5-16.

$$p_1 = 110 \text{ kPa}; t_1 = 20^{\circ}\text{C}; t_3 = 1,440^{\circ}\text{C}.$$

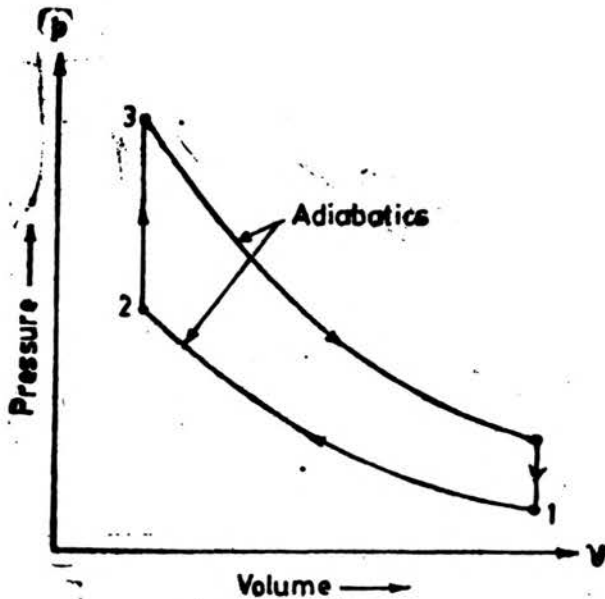


Fig. 5.16. p - v diagram of Otto cycle.

Swept volume = $v_1 - v_2$ and clearance volume = v_2 .

Now, $v_2 = 0.25 (v_1 - v_2)$ i.e.,

$$1.25 v_2 = 0.25 v_1$$

$$\therefore \frac{v_1}{v_2} = 5 \text{ (Compression ratio)}$$

Considering adiabatic compression (1-2),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r)^{\gamma} = (5)^{1.4}$$

$$\therefore p_2 = p_1 \times (5)^{1.4} = 110 \times 9.518 = 1,047 \text{ kPa}$$

Considering adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} = (5)^{1.4-1}$$

$$\therefore T_2 = T_1 \times (5)^{0.4} = 293 \times 1.905 = 558 \text{ K}$$

$$\text{From constant volume process (2-3), } \frac{p_3 v_3}{T_3} = \frac{p_2 v_2}{T_2}$$

$$\text{Hence, as } v_3 = v_2, \frac{p_3}{p_2} = \frac{T_3}{T_2}$$

$$\therefore p_3 = p_2 \times \frac{T_3}{T_2} = 1,047 \times \frac{(1,440 + 273)}{558} = 3,214.17 \text{ kPa}$$

$$\text{Considering adiabatic expansions (3-4), } \frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^{\gamma} = (r)^{\gamma}$$

$$\therefore p_4 = \frac{p_3}{(r)^{\gamma}} = \frac{3,214.17}{(5)^{1.4}} = \frac{3,214.17}{9.518} = 337.69 \text{ kPa}$$

$$\begin{aligned} \text{Work done per cycle} &= \left[\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ} \\ &= \frac{3,214.17 v_2 - 337.69 \times 5v_2}{1.4 - 1} - \frac{1,047 v_2 - 110 \times 5v_2}{1.4 - 1} \\ &= v_2 \times 2,580.42 \text{ kJ} \end{aligned}$$

$$\text{Ideal M.E.P.} = \frac{\text{Work done per cycle in kJ}}{(v_1 - v_2) \text{ in m}^3} \text{ kPa}$$

$$= \frac{v_2 \times 2,580.42}{v_2 \times 4} = 645.1 \text{ kPa}$$

Problem - 14 : An engine working on the ideal Otto cycle has a clearance volume

of 0.03 m^3 and swept volume of 0.12 m^3 . If the heat supplied at constant volume is 145 kJ per cycle, calculate the ideal mean effective pressure in kPa . Take $\gamma = 1.4$ for air.

$$\text{Compression ratio, } r = \frac{\text{Clearance volume} + \text{Swept volume}}{\text{Clearance volume}} = \frac{0.03 + 0.12}{0.03} = 5$$

$$\text{Air-standard efficiency} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(5)^{0.4}} = 0.475 \text{ or } 47.5\%$$

$$\text{Again, air-standard efficiency} = \frac{\text{Work done per cycle in kJ}}{\text{Heat supplied per cycle in kJ}}$$

$$\text{i.e., } 0.475 = \frac{\text{Work done per cycle in kJ}}{145}$$

$$\therefore \text{Work done per cycle} = 145 \times 0.475 = 68.88 \text{ kJ}$$

$$\text{Ideal M.E.P.} = \frac{\text{Work done per cycle in kJ}}{\text{Swept volume in m}^3} = \frac{68.88}{0.12} = 574 \text{ kPa}$$

5.10.2 Diesel cycle : The $p-v$ diagram of the ideal Diesel cycle is shown in fig. 5-17.

Work done per cycle = Area 1-2-3-4

= area under (2-3) + area under (3-4) minus area under (2-1)

$$= \left[p_2 (v_3 - v_2) + \left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} \right) - \left[\left(\frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right) \right] \right] \text{ kJ}$$

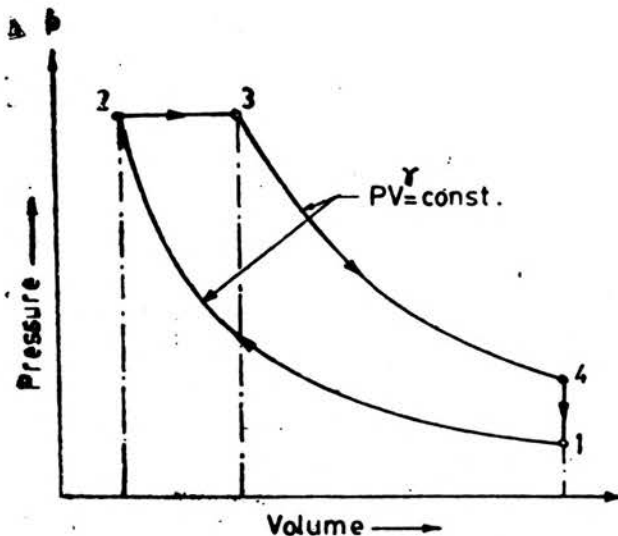


Fig. 5-17. $p-v$ diagram of ideal Diesel cycle for air.

Referring to fig. 5-17, $p_1 = 108 \text{ kPa}$, $T_1 = 30 + 273 = 303 \text{ K}$.

$$(i) \quad \gamma = \frac{k_p}{k_v} \text{ i.e. } 1.4 = \frac{k_p}{0.718} \quad \therefore k_p = 1.4 \times 0.718 = 1.005 \text{ kJ/kg K}$$

$$\text{Let clearance volume} = v_2; \frac{v_1}{v_2} = r = 14 \quad \therefore v_1 = 14 v_2 = v_4$$

$$\text{Again, } v_3 = v_2 + \frac{5}{100} (v_1 - v_2) = v_2 + \frac{5}{100} (13 v_2) = 1.65 v_2; \frac{v_3}{v_2} = \rho = 1.65$$

where, pressures are in kPa and volumes in m^3 .

Ideal M.E.P.

$$= \frac{\text{Work done per cycle in kJ}}{(v_1 - v_2) \text{ in m}^3} \text{ kPa} \quad \dots(5.20)$$

where, $v_1 - v_2 =$ piston in m^3 displacement volume in m^3 .

Problem - 15 : A Diesel engine, working on an ideal cycle, has a compression ratio of 14 and takes in a charge of air at a pressure of 108 kPa and temperature of 30°C . If the cut-off takes place at 5 per cent of the stroke, find : (i) the ideal thermal efficiency, and (ii) the ideal mean effective pressure of the cycle in kPa .

Take $\gamma = 1.4$ and $k_v = 0.718 \text{ kJ/kg K}$.

From adiabatic compression (1-2), $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma$

$$\therefore p_2 = p_1 \times \left(\frac{v_1}{v_2}\right)^\gamma = p_1 \times (r)^\gamma = 108 \times (14)^{1.4} = 108 \times 40.23 = 4,345 \text{ kPa}$$

From adiabatic compression (1-2), $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1}$

$$\therefore T_2 = T_1 (r)^{\gamma-1} = 303 \times (14)^{0.4} = 871 \text{ K}$$

From constant pressure process (2-3), $\frac{p_3 v_3}{T_3} = \frac{p_2 v_2}{T_2}$

Hence, as $p_3 = p_2$, $\frac{T_3}{T_2} = \frac{v_3}{v_2} = 1.65$

$$\therefore T_3 = T_2 \times 1.65 = 871 \times 1.65 = 1,437 \text{ K}$$

From adiabatic expansion (3-4), $\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^\gamma = \left(\frac{14 v_2}{1.65 v_2}\right)^{1.4} = (8.485)^{1.4} = 19.95$

$$\therefore p_4 = \frac{p_3}{19.95} = \frac{4,345}{19.95} = 217.8 \text{ kPa}$$

Considering adiabatic expansion (3-4), $\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{r}{\rho}\right)^{\gamma-1}$

$$\therefore T_4 = \frac{T_3}{\left(\frac{r}{\rho}\right)^{\gamma-1}} = \frac{1,437}{\left(\frac{14}{1.65}\right)^{0.4}} = \frac{1,437}{2.355} = 610 \text{ K}$$

Heat supplied per kg of air = $k_p (T_3 - T_2) = 1.005 (1,437 - 871) = 868.8 \text{ kJ/kg of air}$

Heat rejected per kg of air = $k_v (T_4 - T_1) = 0.718 (610 - 303) = 220.4 \text{ kJ/kg of air}$

$$\begin{aligned} \text{Ideal thermal efficiency} &= \frac{\text{Heat converted into work in kJ per kg of air}}{\text{Heat supplied in kJ per kg of air}} \\ &= \frac{868.8 - 220.4}{868.8} = 0.6125 \text{ or } 61.25\% \end{aligned}$$

(ii) Referring to fig. 5-17, work done per cycle = Area 1-2-3-4
= area under (2-3) plus area under (3-4) minus area under (2-1)

$$\begin{aligned} \text{Work done/cycle} &= \left[p_2 (v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ} \\ &= v_2 \left[4,345 \times 0.65 + \frac{4,345 \times 1.65 - 217.8 \times 14}{1.4 - 1} - \frac{4,345 \times 1 - 108 \times 14}{1.4 - 1} \right] \\ &= v_2 \times 6,041.75 \text{ kJ} \end{aligned}$$

$$\begin{aligned} \text{Using eqn. (5.20), Ideal M.E.P.} &= \frac{\text{Work done per cycle in kJ}}{\text{stroke volume } (v_1 - v_2) \text{ in m}^3} \text{ kPa} \\ &= \frac{v_2 \times 6,041.75}{v_2 \times 13} = 464.75 \text{ kPa} \end{aligned}$$

Problem – 16: In an ideal Diesel cycle, the pressure and temperature at the commencement of the compression stroke are 95 kPa and 15°C. The pressure at the end of the adiabatic

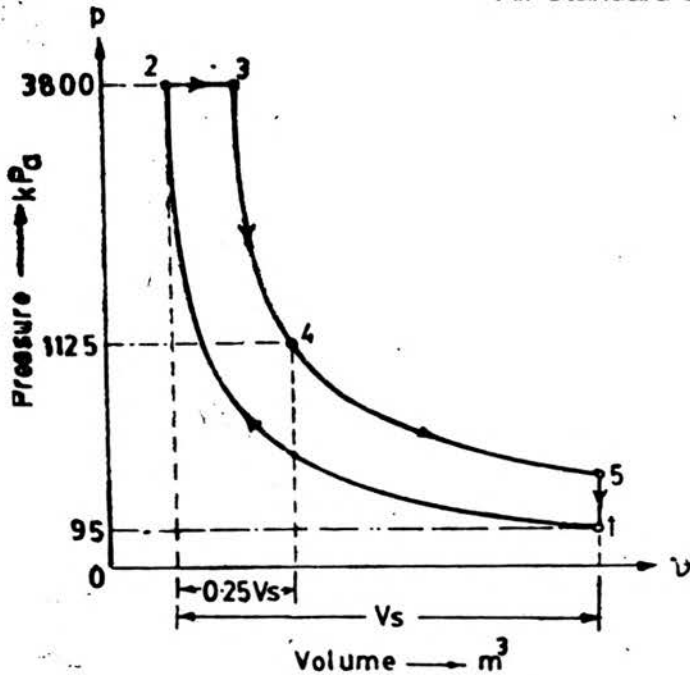


Fig. 5-18. p-v diagram of ideal Diesel cycle

$$\therefore k_p = 1.4 \times 0.718 = 1.005 \text{ kJ/kg K}$$

Referring to fig. 5-18 and considering adiabatic compression (1-2),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2} \right)^\gamma = (r)^\gamma$$

$$\therefore r = \left(\frac{p_2}{p_1} \right)^{\frac{1}{\gamma}} = \left(\frac{3,800}{95} \right)^{\frac{1}{1.4}} = 14$$

Let the clearance volume be unity then $v_1 = 14$, $v_2 = 1$, $v_5 = 14$, and stroke volume = $v_1 - v_2 = 13$

25% of the stroke volume = $0.25 \times 13 = 3.25$. Hence, $v_4 = 3.25 + 1 = 4.25$

Considering points 3 and 4 adiabatic expansion (3-4), $p_3 v_3^\gamma = p_4 v_4^\gamma$

$$\therefore \frac{p_3}{p_4} = \left(\frac{v_4}{v_3} \right)^\gamma \quad \text{or} \quad \frac{v_4}{v_3} = \left(\frac{p_3}{p_4} \right)^{\frac{1}{\gamma}}$$

$$\therefore v_3 = \frac{v_4}{\left(\frac{p_3}{p_4} \right)^{\frac{1}{\gamma}}} = \frac{4.25}{\left(\frac{3,800}{95} \right)^{\frac{1}{1.4}}} = \frac{4.25}{2.382} = 1.78$$

Considering adiabatic compression (1-2), $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2} \right)^{\gamma-1} = (r)^{\gamma-1}$

$$\therefore T_2 = T_1 \times (r)^{\gamma-1} = 288 \times (14)^{1.4-1} = 827\text{K}$$

From constant pressure heat addition (2-3), $\frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$

$$\text{Hence, as } p_2 = p_3, \frac{T_3}{T_2} = \frac{v_3}{v_2}$$

end of the adiabatic compression is 3,800 kPa, and it is 1,125 kPa when the piston displaces 25% of the stroke volume. Find :

- the maximum temperature reached in the cycle,
- the ideal thermal efficiency,
- the ideal m.e.p., and
- approximate percentage of expansion stroke at which cut-off takes place.

Take $\gamma = 1.4$ and $k_v = 0.718 \text{ kJ/kg K}$ for air.

Referring to fig. 5-18,

$$p_1 = 95 \text{ kPa}; t_1 = 15^\circ\text{C};$$

$$p_2 = p_3 = 3,800 \text{ kPa};$$

$$p_4 = 1,125 \text{ kPa}.$$

$$(a) \gamma = \frac{k_p}{k_v} \text{ i.e., } 1.4 = \frac{k_p}{0.718}$$

$$\therefore T_3 = T_2 \times \frac{v_3}{v_2} = 827 \times \frac{1.78}{1} = 1,472\text{K or } T_3 = 1,199^\circ\text{C}$$

$$(b) \text{ Considering adiabatic expansions (3-5), } \frac{T_3}{T_5} = \left(\frac{v_5}{v_3}\right)^{\gamma-1}$$

$$\therefore T_5 = \frac{T_3}{\left(\frac{v_5}{v_3}\right)^{\gamma-1}} = \frac{1,472}{\left(\frac{14}{1.78}\right)^{0.4}} = 645 \text{ K}$$

Heat supplied per kg of air = $k_p (T_3 - T_2) = 1.005 (1,472 - 827) = 648.2 \text{ kJ/kg of air}$

Heat rejected per kg of air = $k_v (T_5 - T_1) = 0.718 (645 - 288) = 256.2 \text{ kJ/kg of air}$

Ideal thermal efficiency = $\frac{\text{Heat converted work in kJ per kg of air}}{\text{Heat supplied in kJ per kg of air}}$

$$= \frac{648.2 - 256.2}{648.2} = 0.6047 \text{ or } 60.47\%$$

$$(c) \text{ From adiabatic expansion (3-5), } \frac{p_3}{p_5} = \left(\frac{v_5}{v_3}\right)^\gamma$$

$$\therefore p_5 = \frac{p_3}{\left(\frac{v_5}{v_3}\right)^\gamma} = \frac{3,800}{\left(\frac{14}{1.78}\right)^{1.4}} = \frac{3,800}{17.95} = 212 \text{ kPa}$$

Referring to fig. 5-18, ideal work done per cycle = Area 1-2-3-5

= area under (2-3) plus area under (3-5) minus area under (2-1)

$$= \left[p_2 (v_3 - v_2) + \frac{p_3 v_3 - p_5 v_5}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ}$$

$$= v_2 \left[3,800 (1.78 - 1) + \frac{3,800 \times 1.78 - 212 \times 14}{0.4} - \frac{3,800 \times 1 - 95 \times 14}{0.4} \right]$$

$$= v_2 \times 6,029 \text{ kJ}$$

Using eqn. (5.20), Ideal M.E.P. = $\frac{\text{Work done per cycle in kJ}}{\text{Stroke volume } (v_1 - v_2) \text{ in m}^3} \text{ kPa}$

$$= \frac{v_2 \times 6,029}{v_2 \times 13} = 463.77 \text{ kPa}$$

(d) Percentage of expansion stroke at which cut-off takes place

$$= \frac{\text{volume at cut-off - clearance volume}}{\text{stroke volume}} \times 100$$

$$= \frac{v_3 - v_2}{v_1 - v_2} \times 100 = \frac{1.78 - 1}{13} \times 100 = 6\%$$

Problem - 17 : Show that ideal M.E.P. of the Diesel cycle is given by ;

$$\frac{p_1 (r)^\gamma \left[\gamma (r - 1) - (r)^{1-\gamma} (r^\gamma - 1) \right]}{(\gamma - 1) (r - 1)}$$

where, p_1 = Pressure at the beginning of the compression,

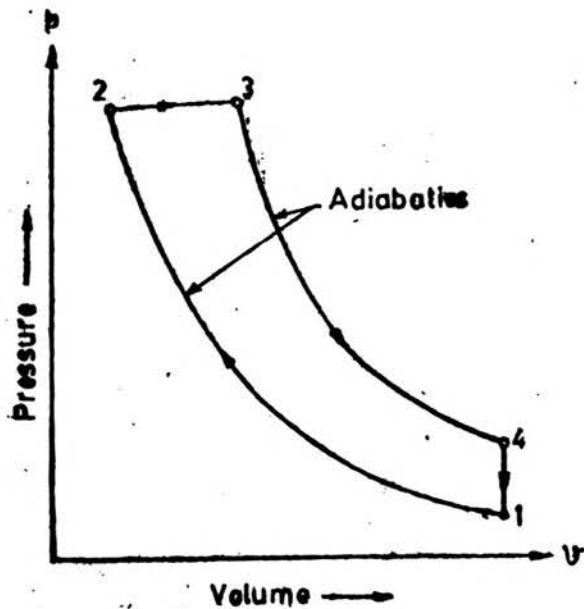


Fig. 5-19. p-v diagram of ideal Diesel cycle

r_1 = Compression ratio, and ρ = Cut-off ratio.

Referring to fig. 5-19,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^\gamma = (r)^\gamma$$

$$\therefore p_2 = p_1 (r)^\gamma = p_3$$

$$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^\gamma = \left(\frac{r}{\rho}\right)^\gamma$$

$$\therefore p_4 = \frac{p_3 (\rho)^\gamma}{(r)^\gamma}$$

$$\frac{v_1}{v_2} = r \quad \therefore v_1 = r v_2 = v_4$$

$$\frac{v_3}{v_2} = \rho \quad \therefore v_3 = \rho v_2$$

Work done per cycle = Area 1-2-3-4
= area under (2-3) plus area under (3-4) minus area under (2-1)

$$= p_3 (v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1}$$

$$\text{Ideal M.E.P.} = \frac{\text{Area of diagram}}{\text{Length of diagram}} = \frac{\text{Work done per cycle}}{v_1 - v_2}$$

$$= \frac{1}{v_1 - v_2} \left[p_3 (v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right]$$

$$= \frac{1}{v_2 (r - 1)} \left[p_3 (\rho v_2 - v_2) + \frac{p_3 \rho v_2 - p_4 v_2 r}{\gamma - 1} - \frac{p_2 v_2 - p_1 r v_2}{\gamma - 1} \right]$$

$$= \frac{1}{r - 1} \left[p_3 (\rho - 1) + \frac{p_3 \rho - p_4 r}{\gamma - 1} - \frac{p_2 - p_1 r}{\gamma - 1} \right]$$

$$= \frac{1}{(r - 1)} \left[\frac{p_3 (\rho - 1) (\gamma - 1) + (p_3 \rho - p_4 r) - (p_2 - p_1 r)}{\gamma - 1} \right]$$

$$= \frac{1}{(r - 1)} \left[\frac{p_3 (\rho - 1) (\gamma - 1) + (p_3 \rho - p_3 \rho^\gamma r^{1-\gamma}) - (p_2 - p_2 r^{1-\gamma})}{\gamma - 1} \right]$$

$$= \frac{1}{(r - 1)} \left[\frac{p_3 (\rho^\gamma - \rho - \gamma + 1 + \rho - \rho^\gamma r^{1-\gamma} - 1 + r^{1-\gamma})}{(\gamma - 1)} \right]$$

$$= \frac{1}{(r - 1)} \left[\frac{p_3 (\rho^\gamma - \gamma - \rho^\gamma r^{1-\gamma} + r^{1-\gamma})}{(\gamma - 1)} \right]$$

$$= \frac{p_3 [\gamma (\rho - 1) - r^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1) (r - 1)}$$

$$= \frac{p_1 (r)^\gamma [\gamma (\rho - 1) - (r)^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1) (r - 1)}$$

Problem - 18 : The mean effective pressure of ideal Diesel cycle engine is 6.1 bar.

If the pressure at the beginning of compression is 1 bar and the compression ratio is 13, determine the cut-off ratio. Assume $\gamma = 1.4$ for air.

Using the expression of ideal M.E.P. of diesel cycle derived earlier (vide problem-17) Ideal M.E.P.

$$= \frac{p_1 (r)^\gamma [\gamma (\rho - 1) - (r)^{1-\gamma} (\rho^\gamma - 1)]}{(\gamma - 1) (r - 1)}$$

where, p_1 = Pressure at the beginning of compression,
 r = Compression ratio, and ρ = Cut-off ratio.

Referring to fig. 5-19, $p_1 = 1$ bar; $\frac{v_1}{v_2} = r = 13$; ideal M.E.P. = 6.1 bar; $\gamma = 1.4$

On substitution of the values, we get

$$\begin{aligned} 6.1 &= \frac{1 \times (13)^{1.4} [1.4 (\rho - 1) - (13)^{-0.4} (\rho^{1.4} - 1)]}{(1.4 - 1) (13 - 1)} \\ &= \frac{36.27 [1.4 (\rho - 1) - 0.3584 (\rho^{1.4} - 1)]}{4.8} \end{aligned}$$

$$\text{or } \frac{6.1 \times 4.8}{36.27} = 1.4 \rho - 1.4 - 0.3584 \rho^{1.4} + 0.3548$$

$$\text{or } 1.8493 = 1.4 \rho - 0.3584 \rho^{1.4}$$

$$\text{or } 1.4 \rho - 0.3584 \rho^{1.4} - 1.8493 = 0$$

By trial and error method putting $\rho = 2$, we get

$$(1.4 \times 2) - 0.3584 (2)^{1.4} - 1.8493 = 0 \text{ or } 2.8 - 0.9458 - 1.8493 \approx 0$$

Thus cut-off ratio, $\rho \approx 2$

5.10.3 Dual-Combustion cycle : The $p - v$ diagram of ideal cycle is shown in fig. 5-20. Work done per cycle = Area 1-2-3-4-5 = area under (3-4) + area under (4-5) minus area under (2-1).

$$= \left[p_3 (v_4 - v_3) + \frac{p_4 v_4 - p_5 v_5}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ}$$

where pressures are in kPa and volumes in m^3 .

$$\therefore \text{Ideal M.E.P.} = \frac{\text{Work done per cycle in kJ}}{(v_1 - v_2) \text{ is } \text{m}^3} \text{ kPa} \quad \dots(5.21)$$

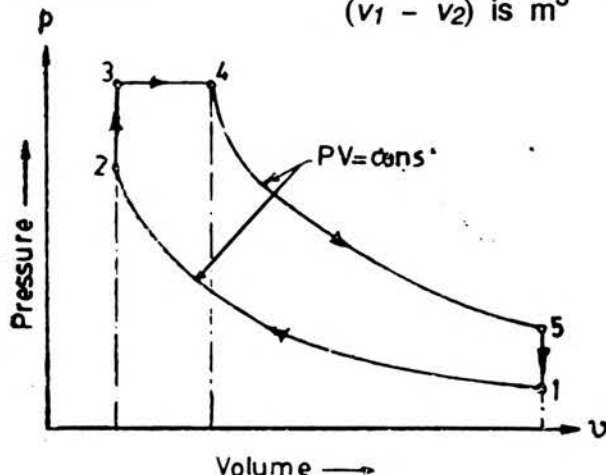


Fig. 5-20. $p-v$ diagram of ideal dual-combustion cycle

where $(v_1 - v_2)$ = piston displacement volume in m^3 .

Problem - 19 : A compression-ignition engine, working on the dual-combustion cycle, has a compression ratio of 10 and two-thirds of the total heat supplied is taken in at constant volume and the remainder at constant pressure. The maximum pressure in the cycle is 4200 kPa and the pressure and temperature at the beginning of compression are 105 kPa and 303 K respectively. Assuming the working fluid to be air

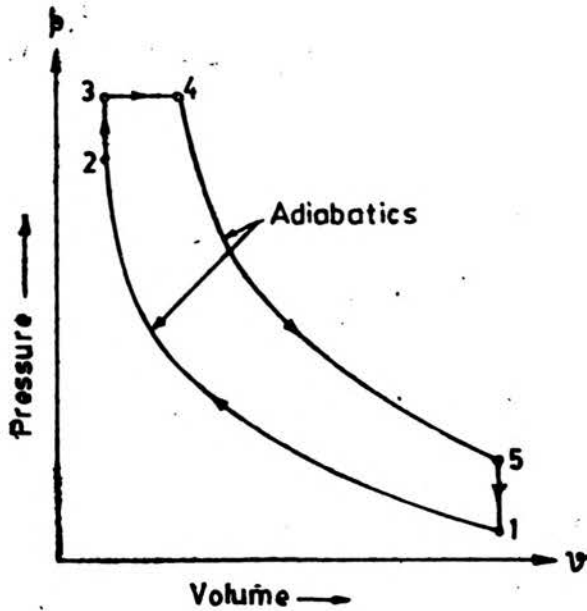


Fig. 5-21. p-v diagram of ideal dual-combustion cycle

throughout the cycle, find the ideal mean effective pressure of the cycle in kPa. Assume $k_p = 1.0035$ kJ/kg K, $k_v = 0.7165$ kJ/kg K and $\gamma = 1.4$ for air.

Referring to fig. 5-21,

$$p_3 = p_4 = 4,200 \text{ kPa}; p_1 = 105 \text{ kPa};$$

$$T_1 = 303 \text{ K}; \frac{v_1}{v_2} = r = 10$$

Then, $v_1 = 10 v_2 = r$ and swept volume = $v_1 - v_2 = 9 v_2$.

From adiabatic compression (1-2),

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1}$$

$$\therefore T_2 = T_1 \times (r)^{\gamma-1} = 303 \times (10)^{0.4} = 761.1 \text{ K}$$

From adiabatic compression (1-2),

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r)^{\gamma}$$

$$\therefore p_2 = p_1 \times (r)^{\gamma} = 105 \times (10)^{1.4} = 2,637.6 \text{ kPa}$$

$$\text{From constant volume heat addition (2-3), } \frac{p_2 v_2}{T_2} = \frac{p_3 v_3}{T_3}$$

$$\text{Hence, as } v_2 = v_3, \frac{p_3}{p_2} = \frac{T_3}{T_2} \text{ i.e., } \frac{4,200}{2,637.6} = \frac{T_3}{761.1} \therefore T_3 = 1,211.9 \text{ K}$$

Heat supplied at constant volume per kg of air

$$= k_v (T_3 - T_2) = 0.7165 (1,211.9 - 761.1) = 323 \text{ kJ/kg of air}$$

Heat supplied at constant pressure per kg of air

$$= k_p (T_4 - T_3) = \frac{1}{2} \times 323 = 161.5 \text{ kJ/kg of air.}$$

$$\therefore 161.5 = k_p (T_4 - T_3) = 1.0035 (T_4 - 1,211.9)$$

$$\therefore T_4 = 1,372.8 \text{ K}$$

$$\text{From constant pressure heat addition (3-4), } \frac{p_3 v_3}{T_3} = \frac{p_4 v_4}{T_4}$$

$$\text{Hence, as } p_3 = p_4, \frac{v_4}{v_3} = \frac{T_4}{T_3} = \frac{1,372.8}{1,211.9} = 1.132 = \rho \text{ (cut-off ratio)}$$

$$\therefore v_4 = 1.132 v_3 = 1.132 v_2$$

$$\text{From adiabatic expansion (4-5), } \frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma}$$

$$\therefore p_5 = \frac{p_4}{\left(\frac{v_5}{v_4}\right)^{\gamma}} = \frac{4,200}{\left(\frac{r}{\rho}\right)^{\gamma}} = \frac{4,200}{\left(\frac{10}{1.132}\right)^{1.4}} = 199.14 \text{ kPa}$$

Work done per cycle = area 1 - 2 - 3 - 4 - 5

= area under (3-4) plus area under (4-5) minus area under (2-1).

$$= \left[p_3 (v_4 - v_3) + \frac{p_4 v_4 - p_5 v_5}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right] \text{ kJ}$$

$$= v_2 \left[4,200 \times 0.132 + \frac{4,200 \times 1.132 - 199.14 \times 10}{0.4} - \frac{2,637.6 \times 1 - 105 \times 10}{0.4} \right]$$

$$= 3,507.6 v_2 \text{ kJ}$$

$$\text{Ideal M.E.P.} = \frac{\text{Work done per cycle in kJ}}{\text{Swept volume in m}^3 (v_1 - v_2)} \text{ kPa}$$

$$= \frac{3,507.6 v_2}{9 v_2} = 389.73 \text{ kPa.}$$

5.11 Actual Cycle Thermal Efficiency

Most of the internal combustion engines of today are designed to follow as closely as possible the Ideal Otto and Diesel cycles. That they can not follow exactly these ideal cycles and hence operate with thermal efficiencies lower than those of ideal cycle is evident when one thinks of some of the practical limitations that are involved. For instance, both ideal cycles use an adiabatic compression and an adiabatic expansion. The adiabatic process requires that no heat be added or rejected throughout its duration and hence requires that the working substance be surrounded by a material that is a perfect non-conductor of heat. The cast iron cylinders of actual engines are conductors of heat and hence the process carried on within them can only approach or approximate the adiabatic. For this reason combined with others, an expansion or compression carried out in the actual engine follows a polytropic process whose value of index n in the equation $p v^n = \text{constant}$ is about 1.35 instead of 1.4 as in a ideal adiabatic (isentropic) process.

The combustion in the actual Otto cycle engines cannot be instantaneous as it is in the ideal Otto cycle. This results in a sloping combustion line on $p - v$ diagram (indicator diagram), that tends to decrease the area of the diagram and the effect of this is to produce a lower actual thermal efficiency. Similarly, in the actual Diesel engine cycle, there is a similar tendency in that the combustion line instead of being maintained horizontal as in the ideal Diesel cycle, tends to slope downward.

The efficiency of the ideal cycle is known as the air-standard efficiency, since it is worked out, on the basis of the working substance being air throughout the cycle. Therefore, the specific heats for air are used. It is evident that this is in variance with the actual cycle. In the actual engine, the working substance is not air but may contain a proportion of gases whose specific heats at constant pressure and constant volume do not bear the same relation as in the case of air.

In the ideal cycle, the specific heat of working substance (air) is considered constant throughout the whole range of temperature. The specific heat of any gas varies with temperature. Therefore, in actual engine the temperature and pressure to which the working substance will be raised, will be lower than would be the case with constant specific heats. Therefore, the area of indicator diagram will be less and the effect of this is to produce lower actual thermal efficiency.

The total effect of all these differences is to produce thermal efficiency of the actual cycle much less than the ideal or air-standard efficiency.

Tutorial - 5

1. Delete the phrase which is not applicable in the following statements :

- (i) The air-standard efficiency of a Diesel cycle having fixed compression ratio will decrease / increase with increase in cut-off ratio.
- (ii) The air-standard efficiency of an I.C. engine decreases / increases with increase in compression ratio.
- (iii) In a Diesel cycle, the ratio of volume of cylinder at the point of cut-off and clearance volume is known as cut-off ratio/compression ratio.
- (iv) In a Diesel cycle, heat is rejected at constant volume/constant pressure.
- (v) Indicated thermal efficiency of well designed and well constructed I.C. engine, when properly operated, will be about one-third/two-third of air-standard efficiency.
- (vi) The most efficient engine is that which works on a reversible/an irreversible cycle.

[(i) increase, (ii) decreases, (iii) compression ratio,
(iv) constant pressure, (v) one-third, (vi) an irreversible]

2. Fill in the blanks in the following statements :

- (i) For thermodynamic cycle to be reversible it must consist of _____ processes only.
- (ii) In Otto cycle, heat is added at constant _____.
- (iii) In Diesel cycle, heat is added at constant _____.
- (iv) The theoretical thermal efficiency of the ideal cycle using air as the working fluid is known as _____.
- (v) Joule cycle is used in gas turbine plant of _____ type.
- (vi) The mean effective pressure of a cycle or heat engine is the _____ pressure in newtons per unit area that operates on the piston throughout its stroke.
- (vii) Cut-off ratio of a Diesel cycle is _____ than unity.
- (viii) Dual-combustion cycle is used in _____ engines.

[(i) reversible, (ii) volume, (iii) pressure, (iv) air-standard efficiency, (v) the constant pressure, (vi) average net, (vii) greater, (viii) modern high speed oil]

3. Indicate the correct answer by choosing correct phrase out of the following statements:

- (i) Air-standard efficiency of Otto cycle is expressed as

$$(a) \quad 1 - \left(\frac{1}{r}\right)^{\frac{\gamma}{\gamma-1}} \quad (b) \quad 1 - \left(\frac{1}{r}\right)^{\gamma+1}$$

$$(c) \quad 1 - \left(\frac{1}{r}\right)^{\gamma-1} \quad (d) \quad 1 - \left(\frac{1}{r}\right)^{\frac{\gamma-1}{\gamma}}$$

where, r is the compression ratio and γ is the ratio of the specific heats of air.

- (ii) For the same compression ratio,
 - (a) both Otto cycle and Diesel cycle are equally efficient.
 - (b) Efficiency of Otto cycle is more than that of the Diesel cycle.
 - (c) Efficiency of Diesel cycle is more than that of the Otto cycle.
- (iii) In an engine working on ideal Otto cycle, combustion takes place
 - (a) at constant pressure.
 - (b) at constant volume.
 - (c) partly at constant volume and partly at constant pressure.
 - (d) at constant temperature.
- (iv) Dual-combustion cycle is also known as
 - (a) Otto cycle. (b) Diesel cycle. (c) Semi-Diesel cycle. (d) Joule cycle.
- (v) Thermal efficiency of the Carnot cycle is
 - (a) $\frac{T_1 - T_2}{T_2}$ (b) $\frac{T_1 - T_2}{T_1}$
 - (c) $\frac{T_2}{T_1 - T_2}$ (d) $\frac{T_1}{T_1 - T_2}$
- (vi) In an air-standard cycle,

- (a) all processes are reversible.
 (b) all processes are irreversible.
 (c) two processes are reversible and other processes are irreversible.
 (d) reversibility and irreversibility is not important.
- (vii) Efficiency of a Diesel cycle
 (a) increases as compression ratio is increased and decreases as cut-off ratio is increased.
 (b) increases with increase in both compression ratio and cut-off ratio.
 (c) decreases with increase in both compression ratio and cut-off ratio.
 (d) decreases as compression ratio is increased and increases as cut-off ratio is increased.
- (viii) Joule cycle consists of
 (a) two isentropic and two isothermal operations.
 (b) two isentropic and two constant pressure operations.
 (c) two isentropic and two constant volume operations.
 (d) two isothermal and two constant pressure operations.
- (ix) Compression ratio of an I.C. engine is the ratio of
 (a) the volume of air in the cylinder before compression stroke and volume of air in the cylinder after compression stroke.
 (b) volume displaced by piston per stroke and clearance volume.
 (c) pressure after compression and pressure before compression.
 (d) temperature after compression and temperature before compression.
- (x) Joule cycle is used in :
 (a) petrol. engine.
 (b) diesel engine.
 (c) constant pressure type gas turbine plant.
 (d) gas engine.

[(i) c, (ii) b, (iii) b, (iv) c, (v) b, (vi) a, (vii) a, (viii) b,;
 (ix) a, (x) c]

4. The temperature limits for a Carnot cycle using air as working fluid are 420°C and 10°C . Calculate the efficiency of the cycle and the ratio of adiabatic expansion. Take $\gamma = 1.4$ for air
 [59.16%; 9.38 : 1]
5. Define the term 'Air-Standard Efficiency' as applied to an internal combustion engine.
 A petrol engine working on Otto cycle has a cylinder diameter of 10 cm and stroke of 15 cm. The clearance volume is 250 cm^3 . Find the ideal thermal efficiency (air-standard efficiency) of the engine. Take $\gamma = 1.4$ for air.
 [50.17%]
6. (a) Obtain an expression for the air-standard efficiency of an internal combustion engine working on the Otto cycle in terms of the ratio of compression r and the ratio of the specific heats of air γ .
 In an engine working on the ideal Otto cycle, the temperatures at the beginning and the end of adiabatic compression are 100°C and 473°C respectively. Find the compression ratio and the air-standard efficiency of the engine. Take $\gamma = 1.4$ for air.
 [$r = 5.656$; 50%]
- (b) Establish an expression for the air-standard efficiency of an engine working on the Otto cycle. If an engine working on this cycle and using air as the working fluid has its compression ratio raised from 5 to 6, find the percentage increase in ideal thermal efficiency. Take $\gamma = 1.4$ for air.
 [7.75%]
7. Show that the efficiency of an air engine working on the constant volume cycle is given by $1 - \left(\frac{1}{r}\right)^{\gamma - 1}$ where, r is the compression ratio and γ is the ratio of the specific heats of air.
8. Describe the constant volume cycle for an air engine.
 Calculate the air-standard efficiency (theoretical thermal efficiency) of this cycle when the pressure at the end of compression is 15 times that at the start.
 If in the above case the initial temperature of air is 40°C and maximum temperature is $1,677^{\circ}\text{C}$, find : (i) the heat supplied per kg of air, and (ii) the work done per kg of air. Take $\gamma = 1.4$ and $k_v = 0.7165\text{ kJ/kg}$

K for air.

[53.88%; (i) 910.6 kJ/kg; (ii) 490.6 kJ/kg]

9. In an ideal Otto cycle engine the compression and expansion follow the adiabatic law with the value of γ as 1.4. The pressure, temperature, and volume at the beginning of the compression are 100 kPa, 40°C and 0.03 m³ respectively. The pressure at the end of compression is 750 kPa and that at the end of constant volume heat addition is 1,900 kPa. Calculate the temperatures at the end of (i) adiabatic compression, (ii) constant volume heat addition, and (iii) adiabatic expansion. Also find the compression ratio and the air-standard efficiency of the engine. Take $k_v = 0.7165$ kJ/kg K for air.

Sketch the pressure - volume and temperature - entropy diagrams for the cycle.

[(i) 284°C; (ii) 1,138°C; (iii) 519.7°C; 4.224; 43.82%]

10. (a) Sketch the ideal indicator diagrams for the Otto, Diesel and dual-combustion cycles.

A Diesel engine has a cylinder diameter of 17 cm and a stroke of 25 cm. The clearance volume is 450 cm³ and cut-off takes place at 6% of the stroke. Find the air-standard efficiency of the engine. Take $\gamma = 1.4$ for air.

[60.1%]

- (b) Obtain the formula for the ideal efficiency of the Diesel cycle in terms of the volume ratios, assuming constant specific heats.

Find the percentage loss in the ideal thermal efficiency of a Diesel cycle engine with compression ratio of 15, by delaying the cut-off from 5 per cent to 10 per cent of the stroke. Take $\gamma = 1.4$ for air.

[6.6%]

11. Derive an expression for the thermal efficiency of an internal combustion engine working on the ideal Diesel cycle.

12. A Diesel engine working on the ideal cycle draws in air at a pressure of 110 kPa and temperature of 288 K. The air is compressed adiabatically to 3.5 MPa (3,500 kPa). Heat is taken in at constant pressure and expansion takes place adiabatically, the ratio of expansion being 5. The air is exhausted at the end of the stroke at constant volume. Calculate : (i) the temperatures at the salient (key) points of the cycle, (ii) the heat received per kg of working fluid, (iii) the heat rejected per kg of working fluid, (iv) the work done per kg of working fluid, and (v) the ideal thermal efficiency. Take $k_p = 1.0035$ kJ/kg K, $k_v = 0.7165$ kJ/kg K and $\gamma = 1.4$ for air.

[(i) 500.9°C, 1,559.6°C; 689.5°C; (ii) 1,062.41 kJ/kg; (iii) 483.28 kJ/kg; (iv) 579.13 kJ/kg; (v) 54.51%]

13. Describe the ideal air cycle for the Diesel engine receiving heat at constant pressure and rejecting heat at constant volume. Show that the efficiency of this cycle is less than that of the constant volume cycle for the same compression ratio.

14. (a) In an ideal Diesel cycle the temperatures at the beginning and end of compression are 57°C and 603°C respectively, whilst those at the beginning and end of expansion are 1,950°C and 870°C respectively. Determine per kg of working fluid for which $R = 0.287$ kJ/kg K and $\gamma = 1.4$, (a) the heat received in kJ, (b) the heat rejected in kJ, (c) the work done in kJ, and (d) the ideal thermal efficiency.

If the compression ratio is 14:1 and the pressure at the beginning of compression is 100 kPa, determine the maximum pressure in the cycle.

[(a) 1,343 kJ/kg; (b) 583 kJ/kg; (c) 760 kJ/kg; (d) 56.59%; 4,023 kPa]

- (b) Sketch the pressure-volume and temperature-entropy diagrams for the ideal Diesel cycle and describe the sequence of operations.

In an ideal Diesel cycle, the temperatures at the beginning and end of compression are 32°C and 615°C respectively. If the temperature at the end of constant pressure heat addition is 1,780°C, determine : (a) the value of the compression ratio, (b) the percentage of the working stroke at which cut-off takes place, and (c) the ideal thermal efficiency. Assume $\gamma = 1.4$ and $k_p = 0.997$ kJ/kg K for air.

[(a) $r = 14.4$; (b) 9.78%; (c) 58.19%]

15. Show that the efficiency of an air engine working on the Diesel cycle may be expressed as :

$$1 - \frac{1}{(r)^{\gamma-1}} \cdot \frac{1}{\gamma} \cdot \frac{(\rho)^{\gamma} - 1}{\rho - 1}$$

where, r is the compression ratio, ρ is the cut-off ratio, and γ is the ratio of the specific heats of air.

16. Derive an expression for the air-standard efficiency of an oil engine working on the Diesel cycle, stating clearly the assumptions made.

State the reasons why the actual thermal efficiency of an internal combustion engine is lower than its air-standard efficiency.

17. Show that in an engine working on the dual-combustion cycle and using air as the working fluid, the

air-standard efficiency is given by the expression :

$$1 - \frac{1}{(r)^{\gamma-1} \left[\frac{\beta (\rho)^{\gamma} - 1}{(\beta - 1) + \beta \gamma (\rho - 1)} \right]}$$

where, r = compression ratio, β = explosion ratio, ρ = cut-off ratio, and γ = ratio of specific heats of air.

18. An oil engine working on the dual-combustion cycle has a cylinder diameter of 20 cm and a stroke of 40 cm. The compression ratio is 13.5 and the explosion or pressure ratio obtained from indicator diagram is 1.42. From the indicator diagram it was found that cut-off occurred at 5.1% of the stroke. Find the air-standard efficiency of the engine. Assume $\gamma = 1.4$ for air.

[61.58%]

19. A high speed Diesel engine working on ideal dual-combustion cycle, takes in air at a pressure of 100 kPa and the temperature of 50°C and compresses it adiabatically to $1/14$ th of its original volume. At the end of the compression the heat is added in such a manner that during the first stage, the pressure increases at constant volume to twice the pressure of the adiabatic compression, and during the second stage following the constant volume heat addition, the volume is increased twice the clearance volume at constant pressure. The air is then allowed to expand adiabatically to the end of the stroke where it is exhausted, heat being rejected at constant volume. Calculate (i) the temperatures at the salient (key) points of the cycle, and (ii) the ideal thermal efficiency. Take $k_p = 0.9923$ kJ/kg K and $k_v = 0.7076$ kJ/kg K for air.

Sketch the pressure-volume and temperature-entropy diagrams for the cycle.

[(i) 655.3°C, 1,583.6°C, 3,440.2°C, 1,432°C, (ii) 60.87%]

20. In a compression-ignition engine working on ideal dual-combustion cycle, the pressure and temperature at the beginning of compression are 1 bar and 127°C respectively. The pressure at the end of compression is 30 bar and the maximum pressure of the cycle is 50 bar. During combustion, half of the heat is added at constant volume and half at constant pressure. Both the compression and expansion curves are adiabatic and heat is rejected at constant volume. Calculate the temperatures at the salient (key) points of the cycle and the ideal thermal efficiency. Take $k_p = 0.9965$ kJ/kg K and $k_v = 0.7118$ kJ/kg K for air throughout the cycle.

[783.8°C, 1,488°C, 1,991°C, 674.3°C; 61.14%]

21. A high speed Diesel engine working on the ideal dual-combustion cycle has compression ratio of 11. The pressure and temperature before compression are 100 kPa and 90°C respectively. If the maximum pressure in the cycle is 5,000 kPa and the constant pressure heat addition continues for $1/20$ th of the stroke, find the work done per kg of air and the ideal thermal efficiency. Take $k_p = 0.9965$ kJ/kg K and $k_v = 0.7118$ kJ/kg K for air.

Sketch the pressure-volume and temperature-entropy diagrams for the cycle.

[787 kJ/kg; 59.52%]

22. Calculate the air-standard efficiency of a gas turbine plant working on Joule cycle between 103 kPa and 412 kPa. If minimum and maximum temperatures in the cycle are 27°C and 527°C respectively, find temperatures after isentropic compression and after isentropic expansion. Take $\gamma = 1.4$ for air.

[32.7%, 172.5°C, 265.72°C]

23. Show that the ideal M.E.P. of the Otto cycle is given by the expression :

$$\frac{p_1 r (\beta - 1) (r^{\gamma-1} - 1)}{(r - 1) (\gamma - 1)}$$

where, p_1 = pressure at the beginning of compression, r = compression ratio, and β = ratio of maximum pressure to compression pressure.

24. A petrol engine with supply pressure and temperature of 100 kPa and 40°C respectively and working on ideal Otto cycle has a compression ratio of 5.8. Heat supplied at constant volume per kilogram of charge is 586 kJ. Find the pressures and temperatures at the salient (key) points of the ideal cycle, if the compression and expansion law is $p v^{1.4} = \text{constant}$. Calculate also the theoretical mean effective pressure. Take $k_v = 0.712$ kJ/kg K for air.

[1,171 kPa, 2,696 kPa; 230.2 kPa; 359.26°C, 1,182.62°C, 447.6°C; M.E.P. = 401 kPa]

25. Show that the ideal M.E.P. of the Diesel cycle may be expressed as

$$\frac{p_1 (r)^{\gamma} \left[\gamma (\rho - 1) - (r)^{1-\gamma} (\rho^{\gamma} - 1) \right]}{(\gamma - 1) (r - 1)}$$

where, p_1 = pressure at the beginning of compression,
 r = compression ratio, and ρ = cut-off ratio.

26. An air engine works on ideal cycle in which heat is received at constant pressure and rejected at constant volume. The pressure and temperature at the beginning of the compression stroke are 100 kPa and 15°C

respectively. The compression ratio is 15.3 and expansion ratio is 7. If the law of adiabatic compression and expansion is $p\nu^{1.4} = \text{constant}$, calculate : (i) the ideal thermal efficiency, and (ii) the ideal mean effective pressure of the cycle. Take $k_p = 0.994 \text{ kJ/kg K}$ and $k_v = 0.709 \text{ kJ/kg K}$ for air.

[(i) 59.87% ; (ii) M.E.P. = 791.3 kPa]

27. An air engine working on ideal cycle in which heat is received at constant pressure and rejected at constant volume. The pressure and temperature at the beginning of compression stroke are 100 kPa and 40°C respectively. The compression ratio is 13 and cut-off ratio is 2. If the compression and expansion curves are adiabatic, calculate the ideal mean effective pressure of the cycle and its ideal thermal efficiency.

Take $k_p = 1.0035 \text{ kJ/kg K}$, $k_v = 0.7165 \text{ kJ/kg K}$ and $\gamma = 1.4$ for air.

[M.E.P. = 613.68 kPa ; 58.08%]

28. A Diesel engine works on the ideal cycle with a compression ratio of 14 and with cut-off taking place at 10% of the stroke. The pressure at the beginning of compression is 100 kPa. Calculate the ideal thermal efficiency and ideal mean effective pressure of the cycle. Take $\gamma = 1.4$ for air.

[57.86% ; M.E.P. = 813.58 kPa]

29. The compression ratio of an engine working on the dual-combustion cycle is 9 to 1 and the maximum pressure is 3,900 kPa. The temperature at the beginning of compression is 95°C and at the end of expansion is 545°C. Considering the ideal cycle with air as the working fluid and assuming that the pressure at the beginning of compression is 100 kPa, find (a) the ideal thermal efficiency, and (b) the ideal mean effective pressure of the cycle. Take $k_p = 0.9965 \text{ kJ/kg K}$ and $k_v = 0.7118 \text{ kJ/kg K}$ for air.

[(a) 57.98% ; (b) M.E.P. = 475.6 kPa]

6

INTERNAL COMBUSTION ENGINES

6.1 Introduction

Heat engines may be divided into two main classes, according to where combustion of fuel takes place. In one class, the combustion of fuel takes place entirely outside the working cylinder. Such engines may be called *external combustion engines*. The most common examples of this class are steam engines and steam turbines, where the working substance is steam. In an external combustion engine the power is produced in two stages. The energy released from the fuel is first utilized to evaporate water in a boiler and then the steam so produced is made to act on the piston of the steam engine or on the blades of the steam turbine, producing external power. If the combustion of fuel takes place inside the engine cylinder, so that the products of combustion directly act on the piston or blades, we have the engines of the second class — the so called *internal combustion engines*. Diesel, gas and petrol engines and gas turbines are the common examples of this type, where the working substance is products of combustion.

6.2 Advantages of Internal Combustion Engines over External Combustion Engines

Internal combustion engines have certain advantages over the external combustion engines. In steam engine plant, the heat of combustion generated in the boiler furnace passes through the shell or tubes of the boiler to the water on the other side, thus generating the steam. In most modern steam generators in which the boiler is equipped with superheater, economiser, etc., about 20% of the heat is wasted during the process by radiation and by loss up the chimney. The proportion of the total heat going to an engine which can be converted into work depends upon the range of temperature of working substance, and in a steam engine this range is small, not exceeding 150°C when saturated steam is used, and about 280°C when superheated steam is used.

Consequently, a steam plant (steam turbine or steam engine) not only loses much of its heat up the chimney, but also is able to convert only a small part of heat that goes to the engine into work. In the best modern steam engines and steam turbines, only about 20 to 30 per cent of the heat going to the engines is converted into work i.e. about 15 to 25 per cent of the heat of combustion of fuel is converted into work in the modern best steam plants, i.e. the overall efficiency of the modern steam plants is about 15 to 25 per cent. The ordinary steam engine does not convert into work more than 8 to 10 per cent of heat of combustion of fuel. *The steam plant after shut down requires considerable time and fuel before the plant can again be put in operation.* If the boiler is kept running (so as to maintain steam pressure while the engine is not working), a considerable amount of fuel will be wasted. *I. C. engine can be started and stopped within a few minutes.*

In an internal combustion engine, where the fuel is a gas or volatile fuel (petrol), there is no apparatus corresponding to a boiler, and no loss corresponding to the boiler losses. If the fuel is coal, it is usually converted into gas before it is used in an internal combustion engine; this necessitates the use of a gas producer, in which some of the heat will be lost, though not as much as in the case of the boiler.

In an internal combustion engine, the charge (air mixed with combustible gas or

vaporized liquid in correct proportion) is drawn into the cylinder by the piston. The mixture after being compressed into clearance space is ignited by an electric spark, so that the explosive combustion takes place while the volume of the charge is nearly constant. The heat thus internally developed gives the working medium a high temperature and pressure, and then expansion of the gas occurs and work is done as the piston advances. When the expansion is complete, gases are cleared from the engine cylinder in order to make way for fresh charge.

Considering thermodynamically, internal combustion engines have the advantage over the steam engines and steam turbines, that the working medium takes in heat (by its own combustion) at a very high temperature. In the combustion of the charge, a temperature of about 2,200°C is reached. The full thermodynamic advantage of a high temperature could not be reached in practice, for the cylinder walls if allowed to reach this high temperature would soon be destroyed. Lubrication of piston would also be impossible hence the cylinder is generally water jacketed to keep the cylinder walls and other engine parts cool. *With large internal combustion engine the difficulty is to keep the cylinder and the piston cool, while with steam engine the cylinder should be kept hot to reduce the losses due to condensation of steam.* The average temperature at which the heat is received in an internal combustion engine is far above that at which heat is received by the working medium of a steam engine or steam turbine. On the other hand, the internal combustion engines do not discharge heat at as low a temperature as do steam engines and steam turbines. But the actual working range of temperature is so large that an I.C. engine converts into work two or three times more of heat energy of the fuel than is realised by any steam engine or steam turbine. *A good I.C. engine will convert about 35 to 40 per cent of energy of fuel into work, the best steam engine will convert not more than 20 per cent and best steam turbine will convert not more than 30 per cent of the heat of combustion of fuel into work.*

High efficiency and absence of auxiliary apparatus such as furnaces, boilers, condensers, make the I.C. engines relatively light and compact for its output. In addition to these advantages, the I.C. engine has become one of the most reliable devices serving mankind. I.C. engines are almost main source of power for aircraft, road vehicles and tractors. Of the new locomotives ordered now-a-days in England and America, over 90 per cent are powered (driven) by Diesel engines. I.C. engines are very useful in marine service where space is of great importance.

6.3 Development of I.C. Engines

Around 1878, many experimental I.C. engines were constructed. The first really successful engine did not appear, however, until 1879, when a German engineer, Dr. Otto, built famous Otto gas engine. The operating cycle of this engine was based upon principles first laid down in 1860 by a French engineers named Bea de Rochas. The majority of modern I.C. engines operate according to these principles.

The development of the well known Diesel engine began around 1893 by Rudolf Diesel. Although this engine differs in many important respects from the Otto engine, the operating cycle of modern high speed engines is thermodynamically very similar to the Otto cycle.

6.4 Classification of I.C. Engines

Internal combustion engines may be classified according to :

- (i) *Cycle of operation (the number of strokes required to complete the cycle) — two - stroke cycle engine and four-stroke cycle engine.*
- (ii) *Cycle of combustion - Otto cycle engine (combustion at constant volume), Diesel cycle engine (combustion at constant pressure), and dual-combustion cycle*

engine or semi-Diesel cycle engine (combustion partly at constant volume and partly at constant pressure).

- (iii) *Arrangement of cylinders* - horizontal engine, vertical engine, V-type engine, radial engine, etc.
- (iv) *Number of cylinders* - single cylinder engine and multi-cylinder engine.
- (v) *Action of products of combustion upon the piston* - single-acting engine and double-acting engine.
- (vi) *Speed of the engine* - low speed, medium speed and high speed engine.
- (vii) *Type of fuel* - Diesel oil engine; petrol engine, gas engine, light oil (kerosene) engine.
- (viii) *Method of igniting fuel* - spark ignition and compression ignition (C.I.) engine.
- (ix) *Method of cooling the cylinder* - air cooled engine and water cooled engine.
- (x) *Method of governing the engine* - hit and miss, quality and quantity governed engine.
- (xi) *Method of fuel supply to the engine cylinder* - carburettor engine, air injection engine and solid or airless injection engine.
- (xii) *Suction pressure* - naturally aspirated engine and supercharged engine.
- (xiii) *Their uses* - stationary engine, portable engine, marine engine, automobile engine, tractor engine, aero-engine, etc.

For example, an engine may be described as an oil engine, but it can be more properly described as : 20 brake power, Diesel two-stroke cycle, horizontal, single cylinder, single-acting, high speed, solid injection, compression ignition, water cooled, quality governed, naturally aspirated, stationary engine.

6.5 Requirements of I.C. Engines

In any I.C. engine the following requirements must be met :

- (i) The charge of fuel and air in correct proportion must be supplied to the engine.
- (ii) The fuel and air or air only must be compressed either before or after the mixing takes place.
- (iii) The compressed mixture must be ignited, and the resulting expansion of combustion products is used to drive the engine mechanism.
- (iv) The combustion products must be cleared from the engine cylinder when their expansion is complete, in order to make room for the fresh charge to enter the cylinder.

Cycles of operation : Two methods are used to carry out the above mentioned processes in an I.C. engine, namely, the four-stroke cycle and two-stroke cycle. If an engine requires four strokes of the piston or two revolutions of the flywheel to complete the cycle, it is termed a *four-stroke cycle engine*. If on the other hand, the cycle (all the processes) is completed in two strokes of the piston or in one revolution of the flywheel, then the engine is termed a *two-stroke cycle engine*. Majority of the I.C. engines operate on the four-stroke cycle. For detailed description of two-stroke and four-stroke cycle engines, refer volume I, Chapter 10.

Cycles of combustion : Engines which draw in mixture of fuel and air during the suction stroke and ignite the compressed mixture by means of a timed electric spark or small hot spot and burn the mixture while the piston remains close to the top dead centre (constant volume burning), are called *Otto cycle engines*. Otto cycle or constant volume cycle engines may be two-stroke or four-stroke. Gas, petrol, light and heavy oil engines

use this cycle. This cycle is very popular in two-stroke petrol and oil engines.

In *Diesel cycle engines*, only air is drawn in and compressed to pressure of about 35 bar by the piston during the compression stroke, the fuel oil being pumped in the cylinder when the compression is complete. In this way the fuel is fired by coming in contact with the high pressure hot air. A *Diesel engine needs no spark plug or a separate ignition equipment*. Diesel cycle is known as constant pressure cycle because the burning of the fuel takes place at constant pressure. Diesel cycle is much used in heavy oil engines. Diesel cycle may be two-stroke or four—stroke.

In *dual-combustion cycle engine*, only air is drawn in the cylinder during the suction stroke. This air is then compressed into a hot combustion chamber or hot bulb at the end of the cylinder during the compression stroke, to a pressure of about 28 bar. The heat of the compressed air together with the heat of the hot combustion chamber is sufficient to ignite the fuel. The fuel is injected or sprayed into the hot combustion chamber just before the end of the compression stroke where it immediately ignites. The injection of fuel is continued during the first part of the working stroke until the point of cut-off is reached, which is regulated by the governor.

The burning of fuel at first takes place at constant volume and continues to burn at constant pressure during the first part of the working stroke. The cycle is known as dual combustion cycle or Mixed cycle because the heat is taken in partly at constant volume and partly at constant pressure. Dual-combustion cycle engines may be two-stroke or four-stroke. Engines working on this cycle are sometimes known as *semi-Diesel engines*. This cycle is much used in heavy oil engines. Modern high speed Diesel engines operate on this cycle. For detailed description of Otto cycle and Diesel cycle engines, refer volume 1, chapter X.

6.6 Scavenging Methods in Two-stroke Cycle Engines

The clearing or sweeping out of the exhaust gases from the combustion chamber of the cylinder is termed *scavenging*.

It is necessary that the cylinder should not have any trace of the burnt (exhaust) gases because they may mix with the fresh incoming charge and reduce its strength. Power will be lost if the fresh charge is diluted by the exhaust gases.

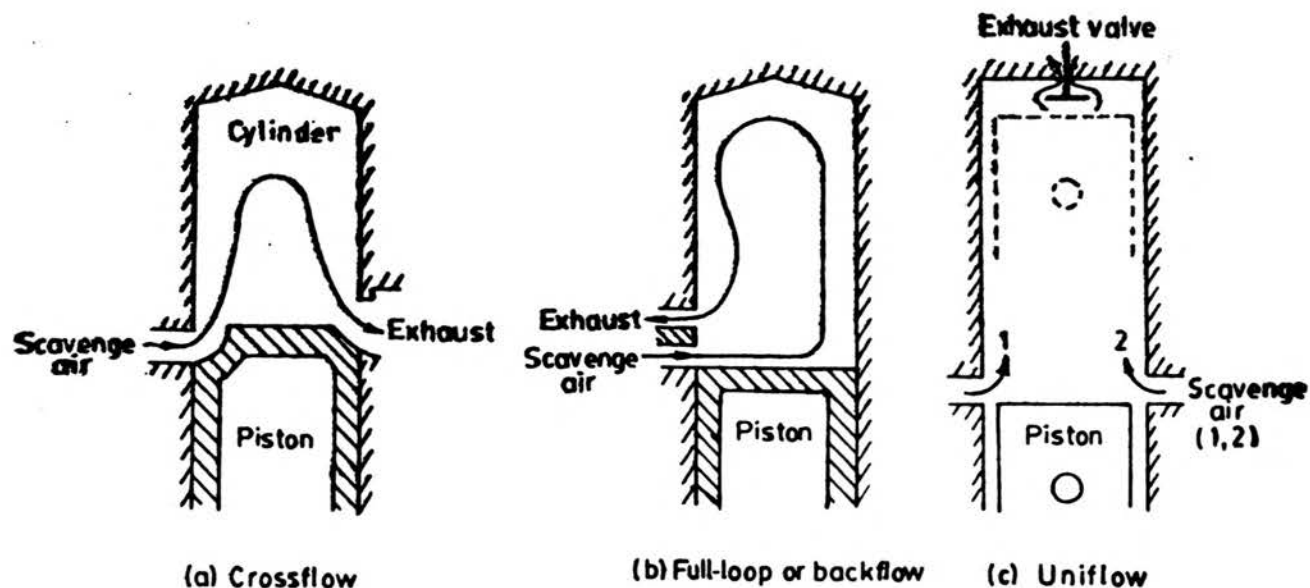


Fig. 6-1. Different methods of scavenging in two-stroke cycle engines.

In a four-stroke cycle engine the exhaust gases are pushed out of the cylinder by the incoming piston, but in a two-stroke cycle engine scavenging is necessary, since the piston does not help in forcing out the burnt gases from the cylinder. In a two-stroke cycle engine the scavenging is carried out with the help of incoming fresh charge (fuel-air mixture) or air which is partially compressed before it is admitted to the cylinder. The fresh charge or air, being at a higher pressure than exhaust gases, pushes out the gases through the exhaust passages. This is possible because admission of the fresh charge and removal of the exhaust gases are taking place at the same time in a two-stroke cycle engine. In order to prevent the fresh charge of fuel-air mixture from entering the exhaust ports and passing out through it, the charge is deflected upwards by the deflector or baffle provided on the crown of the piston [fig. 6-1 (a)]. The possibility of losing some of the entering charge which consists of fuel and air is only with two-stroke Otto cycle engines. In a two-stroke Diesel cycle engine, air alone is admitted into the cylinder which helps in the scavenging work.

The scavenging methods in two-stroke cycle engines are :

6.6.1 Crossflow scavenging : The admission (or scavenge) ports are provided on the sides of the cylinder and the exhaust ports are kept on the opposite cylinder wall. The charge or air entering through the scavenge ports is directed upwards which pushes out the exhaust gases through oppositely situated exhaust ports as shown in fig. 6-1 (a).

6.6.2 Full-loop or backflow scavenging : The exhaust ports and scavenge ports are provided on the same side of the cylinder wall and the exhaust ports are situated just above the scavenge ports as shown in fig 6-1(b). This method is particularly suitable for double—acting C. engines.

6.6.3 Uniflow scavenging : Scavenging ports are provided on one side of the cylinder wall, and exhaust valves are kept in the cylinder head for the removal of the exhaust gases. Here the scavenge air and the exhaust gases move in the same upward direction as shown in fig. 6-1 (c). The mixture or air requires to be compressed before it is admitted to cylinder so that it will help in scavenging the cylinder.

The scavenging air or mixture is produced in the following three ways :

(i) *Crankcase compression* : A mixture of vaporized fuel and air, or air alone in case of a Diesel engine, is compressed in the gas-tight crank case of the engine, by the back of the piston during its working or outward stroke. The compressed mixture or air then enters the cylinder through scavenge or transfer port. This method is known as crankcase scavenging and is used in small size engines.

(ii) *Cylinder compression* : The mixture or air is compressed at the non-working end of the cylinder by the back of the engine piston during its outward stroke. In such a case the non-working end of the cylinder is also a closed one with a suction valve provided at this end to admit the mixture of air and fuel or air for compression. The partially compressed charge is then admitted to the working end of the cylinder through the transfer port.

(iii) *Separate compression* : A separate compressor, either driven from the engine crank shaft or using outside power, may be provided to supply compressed air to the cylinder. For an Otto cycle engine (gas engine), separate valves may be provided for the admission of compressed air and gas. In such a case the air enters first, which will scavenge the cylinder. The air is then followed by gas and more air for combustion. In this method the question of loss of power due to some of the fresh mixture going out along with the exhaust gases does not arise, since only compressed air is used for scavenging. This method is the best and used in all large engines.

6.7 Handling of Volatile Liquid (Petrol) Fuels

The term *carburation* is applied to the process of vaporizing liquid hydrocarbon fuels. Spirit fuels such as petrol, benzol and alcohol, vaporize slightly at atmospheric conditions. Hence, the engine suction is sufficient to vaporize these fuels and no preheating is necessary. In case of oil fuels, such as light oil and paraffin, vaporization of the liquid is done by preheating in a vaporizer with the help of exhaust gases.

The apparatus used for vaporizing petrol and other spirit fuels is called a *carburettor*; in this apparatus no heating is necessary. Carburettors are constructed chiefly for gasoline fuels, though some are made for volatile fuels such as kerosene or alcohol. Fundamentally the carburettor must fulfil the following main functions :

- (i) To maintain a small reserve of petrol at a constant head.
- (ii) To give correct proportion of petrol to air at all speeds and according to varying load requirements of the engine.
- (iii) To vaporize the petrol and to produce homogeneous air-fuel mixture.
- (iv) To supply a comparatively richer mixture during slow running or idling periods and at starting.
- (v) To supply a special rich mixture when the engine is to be accelerated by opening the throttle valve suddenly.

6.7.1 Simple carburettor : The simplest design of a carburettor is simple carburettor which consists of a single jet situated in the centre of the choke tube and to which fuel is supplied at a constant level from a float chamber as shown in fig. 6-2.

Petrol enters the float chamber through a filter and a valve. The level of petrol is maintained constant at correct height by the float. When the correct level is reached, the float rises and forces the needle valve downwards and shuts off the petrol supply.

The suction of the engine causes air supply to rush through the choke which is shaped as a venturi cone. The choke tube surrounding the top of the jet is reduced in diameter, so as to increase the velocity of air at this point and reduce its pressure (pressure will be less than atmospheric). Atmospheric pressure exists on the top of the float chamber which is produced by an air vent hole. The difference between the atmospheric pressure and the pressure around

the top of the jet causes the petrol to flow into air stream at the throat of the choke tube and gets vaporized. Tip of the fuel jet is placed higher (about 1.5 mm) than the normal level of petrol in the float chamber, in order to avoid leakage of petrol when there is no air flow or when the engine is at rest.

A carburettor of this type (fig. 6-2), would give a rich mixture as the engine speed increases and weak mixture as the engine speed decreases. Assume that the throat of the choke tube and jet have been so designed as to permit the passage of fifteen parts of air and one part of petrol by weight under certain conditions of suction. A mixture of proper proportion will be drawn into

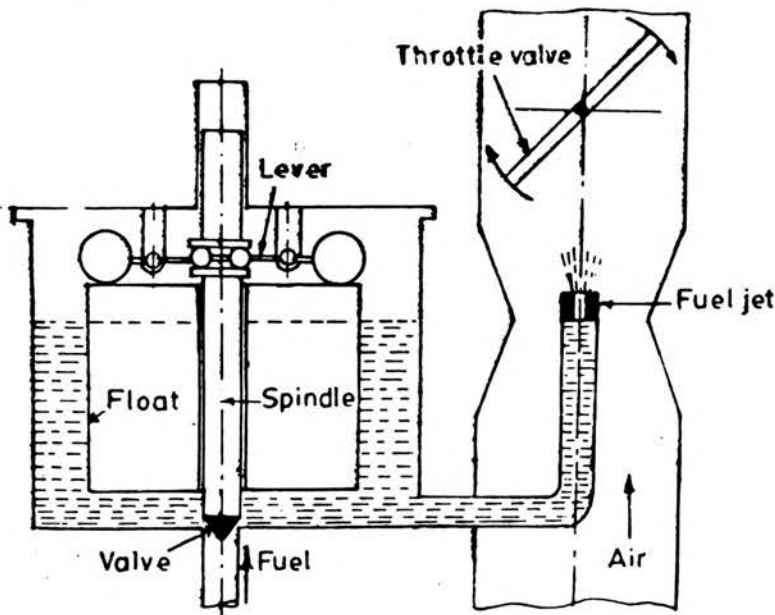


Fig. 6-2. Single jet (simple) Carburettor.

the engine cylinder. It is natural to suppose that as the speed of the engine increases, flow of petrol and air will increase in the same proportion. Such, however, is not case. Petrol is more responsive to suction than air. The laws governing the flow of liquids from a jet and air through the venturi cone are not the same, for one is a liquid and the other is a gas. Consequently, as the engine speed increases, the flow of petrol into engine cylinder increases faster than the flow of air, the mixture becoming too rich at the high speeds. Thus in a given example, if the velocity of air past (just over) the jet be doubled, the flow of the petrol will be increased by about $2\frac{1}{2}$ times.

6.7.2 Zenith carburettor : Many different devices have been used for balancing or

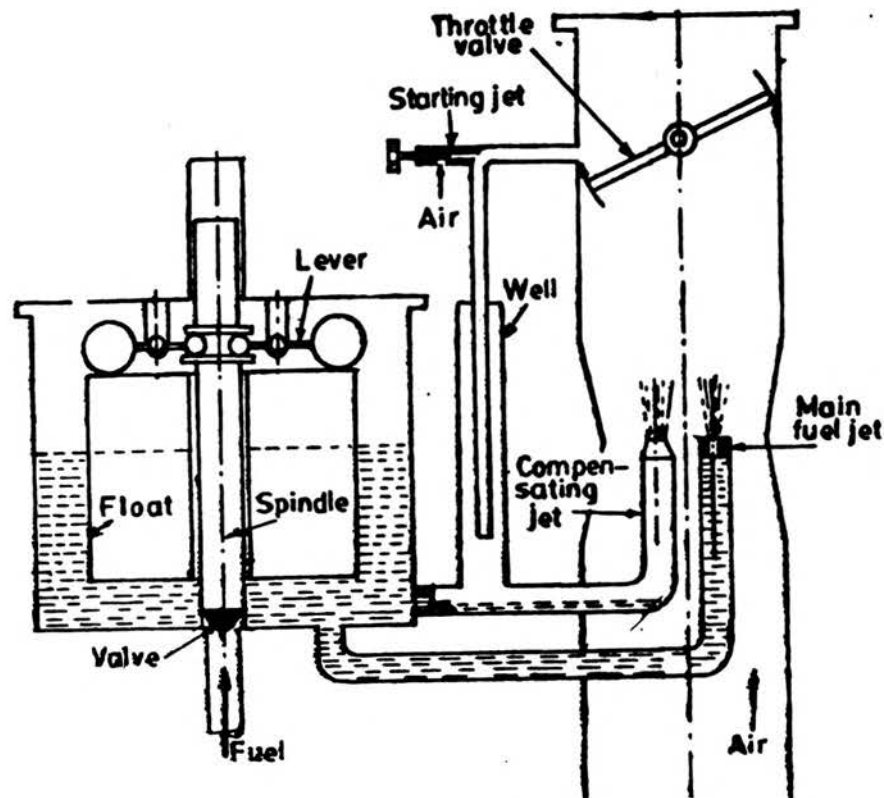


Fig. 6-3. Diagrammatic sketch of Zenith carburettor.

compensating the action of the single jet, so as to secure a constant mixture strength. One of the simplest and most satisfactory of these devices is the use of two jets, the main fuel jet and the compensating jet, shown in fig. 6-3.

The compensation is effected by means of an additional jet called the *compensating jet*. Fixed amount of petrol from the float chamber is permitted to flow by gravity through the metering jet into the well, open to atmosphere. The supply to the well is not affected by the suction of the engine, because suction is destroyed by the open well. As the engine speed increases, more air is drawn through the carburettor, while the amount of petrol drawn through the compensating jet remains the same and therefore the mixture grows leaner (weaker) and leaner. By combining this compensating device with single jet, we secure the compound nozzle giving us a constant mixture strength.

In addition to giving a correct proportion of petrol and air at all speeds, a carburettor should also provide a suitable mixture for *starting and slow running or idling*. The provision is also made in the Zenith carburettor by providing a separate starting jet which automatically comes into action for slow running or for starting, when the throttle valve is only slightly open. In that case, air velocity in the choke tube is not sufficient to operate the main

jet but in the contracted passage around the slightly open throttle valve, there is sufficient air velocity to operate the starting jet through a by-pass near the edge of throttle. As the engine speed increases and throttle valve is opened a certain amount, the air velocity at the mouth of the by-pass is not sufficient to operate the starting jet and it automatically goes out of action. The air supply to the starting jet is controlled by a pointed screw, so that any strength of mixture can be adjusted for starting.

Provision is also made in the Zenith carburettor for supplying 100% excess of petrol for 3 or 4 cycles when the engine is *accelerated* and the throttle is suddenly opened. When the engine is running with throttle full open, the well fed by the gravity is normally dry but when idling or slow running, the well fills up to nearly the level in the float chamber. So as soon as the throttle is opened, the sudden depression caused by the inflow of air to the induction system, draws in the whole contents of the well in the induction system and provides momentarily an over-rich mixture. Fig. 6-3 shows diagrammatically the Zenith carburettor with a compensating jet, and idling and starting jet.

6.8 Handling of Heavy Fuel Oils (Methods of Fuel Injection)

The purpose of fuel injection device is to inject the exact quantity of fuel in the engine cylinder at the proper moment of the working cycle and in a state most suitable for combustion. The main requirements which the injection system must, therefore fulfil are: (a) accurate metering of small quantity of fuel oil needed to develop the desired power, (b) correctly timing the beginning and end of fuel injection period. Injection should begin at the required moment so that the maximum power and fuel economy may result. Early injection delays ignition because the temperature of the charge at the instant is not high. Excessive delay in injection results in poor fuel economy, reduction of power, smoky exhaust and noisy operation of the engine, (c) control of rate of fuel injection, (d) atomisation of the fuel to facilitate proper combustion and (e) uniform distribution of fuel in the combustion space.

The fuel injection equipment, therefore, consists of the following units :

- (i) a governor to regulate the fuel oil supply according to load,
- (ii) a fuel pump or injection pump to deliver fuel oil under pressure,
- (iii) an injection nozzle (valve) or atomiser to inject fuel oil into the cylinder in a finely atomised state, and
- (iv) an air compressor in the case of air injection engines.

6.8.1 Fuel oil injection methods : Engines working on constant pressure (Diesel) or dual-combustion (Semi-Diesel) cycle both of which require pure air for compression, must have external source of forcing the oil into the cylinder. For these types of engines, there are two distinct methods of fuel injection - *air injection* and *airless injection*. The latter method is known under different names such as *mechanical injection* or *solid injection*.

6.8.2 Air injection : In this method of fuel injection, the fuel is injected through the nozzle (valve) by means of compressed air of a much higher pressure than that produced in the engine cylinder at the end of compression stroke. A measured quantity of fuel oil is pumped into an annular space provided in the bottom of injection valve and an air pressure of about 60 bar is applied to it. When the injection valve is opened by the cam and rocker lever arrangement, the fuel is driven into the combustion space at the high velocity by the high pressure air. The high pressure air is supplied from storage air bottles, which are kept charged (filled) by the air compressor driven by the engine itself.

Fig. 6-4 shows a mechanically operated air injection valve. The valve consists of a plain spindle *a* with a conical seating and held against its seating by a very stiff spring (not shown). Immediately surrounding the valve spindle is a light and long atomising tube

b which also rests on a coned seating *d*, but the seating is fluted or grooved in order to permit the passage of fuel and air past it. Above the seating are a number of perforated discs *e* through which fuel and air are driven. The long atomising tube *b* is placed in the valve body *g* which fits into an opening in the cylinder head *h*

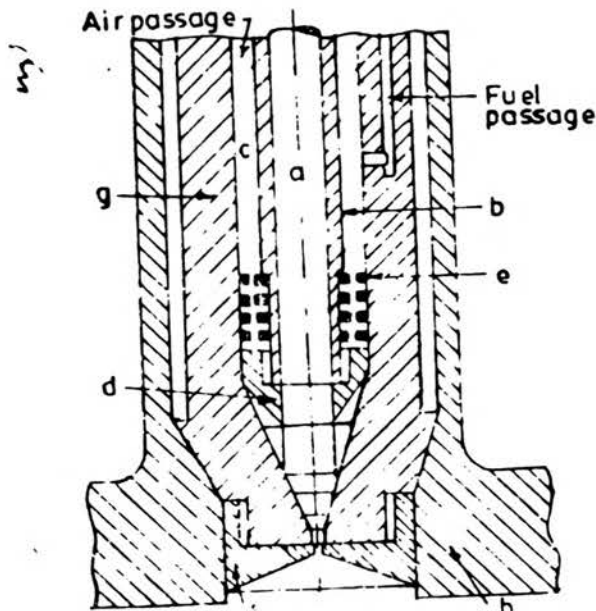


Fig. 6-4. Mechanically operated air injection valve.

The oil is forced through small openings (holes) in the discs *e* and broken in small particles. The fuel-air mixture is injected into the combustion chamber space through a central orifice in flame plate *f*. The flame plate has one central opening through which oil and air pass at a very high velocity, by means of which the fuel is atomised and distributed evenly in the combustion chamber.

6.8.3 Solid injection : Solid injection is also termed as *airless or mechanical injection*. This method employs mechanically operated fuel pump which meters out correctly the

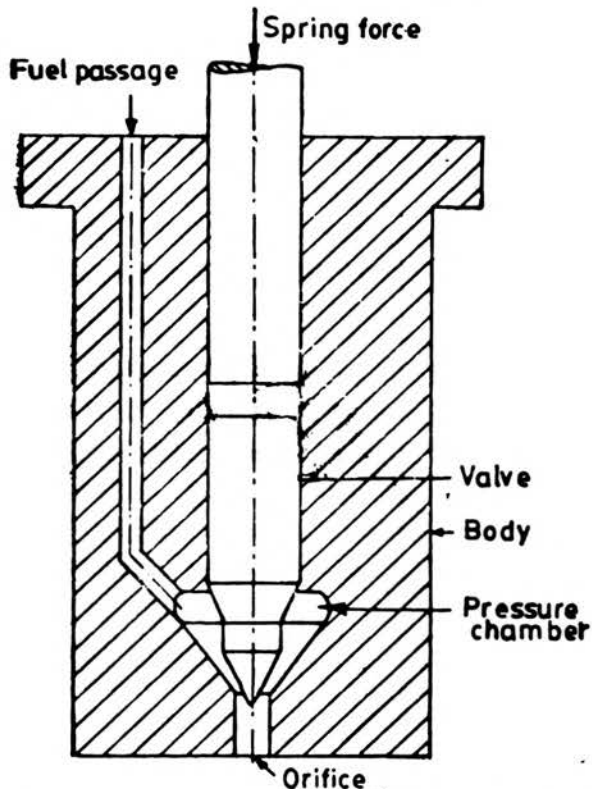


Fig. 6-5. Sectional view of a solid injection Bosch fuel nozzle.

quantity of fuel required for the working stroke, and to inject it through a fuel injection nozzle under high pressure with a view to atomise it or break into very small particles and to inject the fuel particles at a high velocity into the mass of compressed air in the combustion space. The fuel injection pressure varies from 100 to 125 bar and in some cases even more than this. The desired pressure is produced by the fuel pump of the plunger type, shown in fig. 6-19.

Figs. 6-5 and 6-18 show section of a solid injection Bosch fuel spray valve or atomiser. The nozzle is usually built with one or more holes through which the fuel sprays into the cylinder at high velocity. The holes in the nozzle body are carefully drilled to direct the spray in the cylinder most advantageously. Immediately behind the hole or holes, is a nozzle valve which is held on its seat by a very stiff adjustable spring force. The pressure at which the nozzle valve will lift, depends upon the amount of compression placed upon

the spring which is adjustable by means of adjusting screw (fig 6-18). The nozzle valve is usually set by a set-screw to open at 100 to 125 bar pressure. A feeler pin passes through the centre of compression screw, which enables the functioning of the nozzle valve to be felt while the engine is running-slight knock indicating that the nozzle is in operation. Any slight leakage of fuel that may accumulate above the valve, can be taken away to a drain tank by means of a pipe connected to leak-off nipple (fig 6-18).

The valve remains sealed (closed) until the motion of the fuel pump plunger on the delivery stroke builds up a sufficient pressure in the injection line. When the high pressure fuel oil in the pressure chamber (shown clearly in fig 6-5) overcomes the spring force, the valve is lifted off its seat. As soon as the communication to the engine cylinder or combustion chamber is reached, the pressure is suddenly released and the valve under the action of spring comes back to its seat, pushing the oil in front of it through one or more holes with a very high velocity.

The fuel pump (fig 6-19) has two main functions to perform - it must start the injection at the proper crank angle, late in the compression stroke, and it must force the oil through the nozzle and into the cylinder the exact quantity of fuel needed to develop the requisite (required) power.

6.9 Comparison between Solid Injection and Air Injection

Following are the advantages of solid injection :

- (i) The quantity of fuel is metered out correctly.
- (ii) The system possess simplicity and it is very suitable for high duty engines.

Following are the disadvantages of solid injection :

- (i) Penetration of fuel is not as perfect as with air injection.
- (ii) Brake mean effective pressure obtained is not as high as that obtained with air injection.
- (iii) Fuel and pipe line some times give injection timings troubles on light loads. This is due to the elasticity of the fuel and pipe lines.

Following are the advantages of air injection :

- (i) More power is obtained with the same cylinder size. This is possible because more fuel can be burned by the additional air available with injection air.
- (ii) Combustion takes place at approximately constant pressure as a result of better atomisation and penetration of the fuel.
- (iii) It is possible to control effectively the rate of admission of the fuel by varying the pressure of the injection air.
- (iv) Extreme accuracy is not required in the manufacture of the fuel pump.

Following are the disadvantages of air injection:

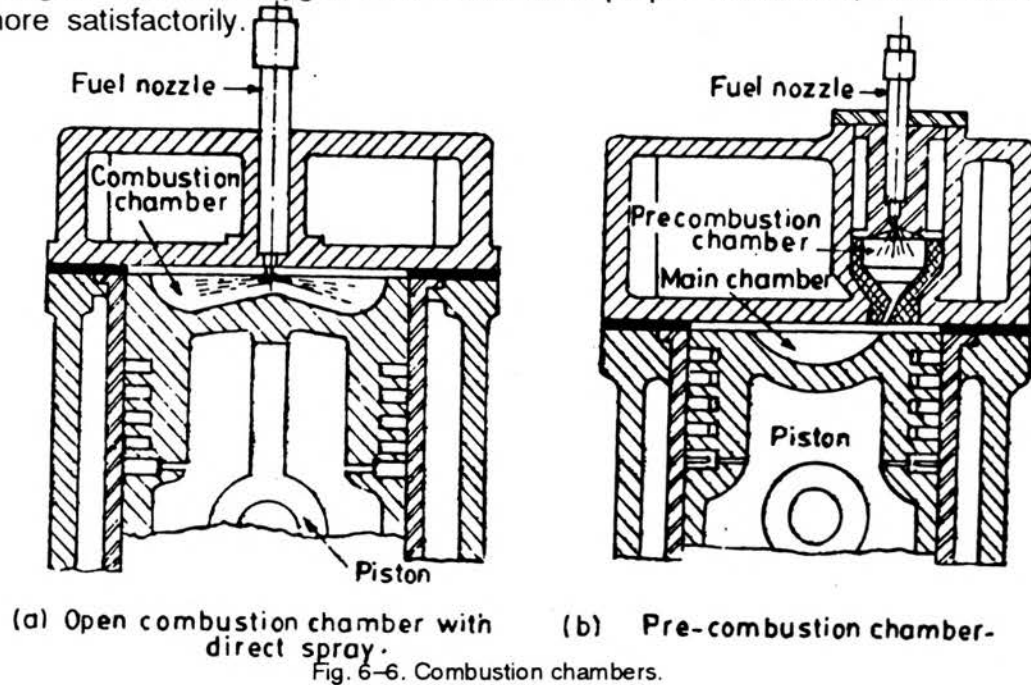
- (i) It is necessary to have a compressor for keeping the air bottles charged, and large power is absorbed in driving it (compressor).
- (ii) The weight of the machinery is increased and the system is expensive and complicated.

The mechanical injection (or solid injection), which has become possible due to the precise manufacture of fuel injection pump and nozzle, is now widely used and has driven out the air injection system from the field.

6.10 Compression-Ignition Combustion Chambers

For efficient combustion of a fuel it is necessary that each particle of atomised fuel

enjected into the cylinder head shall find the necessary amount of heated air to complete its combustion. This object can be achieved by mixing thoroughly, by some means, the atomised fuel and air necessary for combustion. This mixing of the fuel particles with air is known as *turbulence*. Turbulence increases the flame velocity and is roughly proportional to gas velocity and therefore to the engine speed. It also accelerates chemical action by intimate mixing of fuel and oxygen molecules. With proper turbulence, weak mixture can be burnt more satisfactorily.



A high relative velocity between the air and fuel is necessary to have rapid combustion of the fuel. This object is fulfilled in some cases (i) by having combustion chamber so designed as to give sufficient turbulence to the air to bring practically all its oxygen into the vicinity of the fuel stream from the injection nozzle. The fuel is injected directly into the combustion chamber, (ii) by having a separate combustion chamber, known as *pre-combustion chamber*, into which the fuel is injected and, (iii) by having combustion chambers with an *air cell* or *energy cell*.

The first method is known as *open chamber* or *direct method* (fig. 6-6a). The fuel is sprayed directly in the combustion chamber at high pressure and velocity, so that it can penetrate the mass of compressed air. The fuel distributes itself throughout the air in the cylinder. The distribution and mixing of fuel with air is effected chiefly by the fuel injector (nozzle). In some cases the deflector is provided on the inlet valve to give swirl to the air during the suction stroke. This gives higher velocity to air, and the swirling air sets up a certain amount of turbulence. The rate of pressure rise is moderate and on account of this engine runs smoothly. Direct injection makes possible the starting of the engine from cold. *Open chambers are suitable for moderate and low speed engines.*

In the second method, the fuel is injected into an *auxiliary* or *pre-combustion chamber* (fig. 6-6b) and not directly into the main combustion chamber. The fuel burns in two stages, partly in the pre-combustion chamber and partly in the main combustion space. The rise in pressure in the pre-combustion chamber, forces at high velocity the products of combustion and the remaining fuel into the main combustion chamber.

There is, however, a loss of heat in this type of combustion chamber and since combustion virtually is taking place in two stages, the thermal efficiency obtained is not the same as is given by the direct injection. It has the advantage that inferior fuels can

be burned at moderate injection pressures. However, starting from cold is more difficult than the direct injection type.

6.11 Ignition Methods

The energy of the fuel of I.C. engine is locked up in fuel in the form of chemical energy. Some means have to be employed whereby this energy can be released and

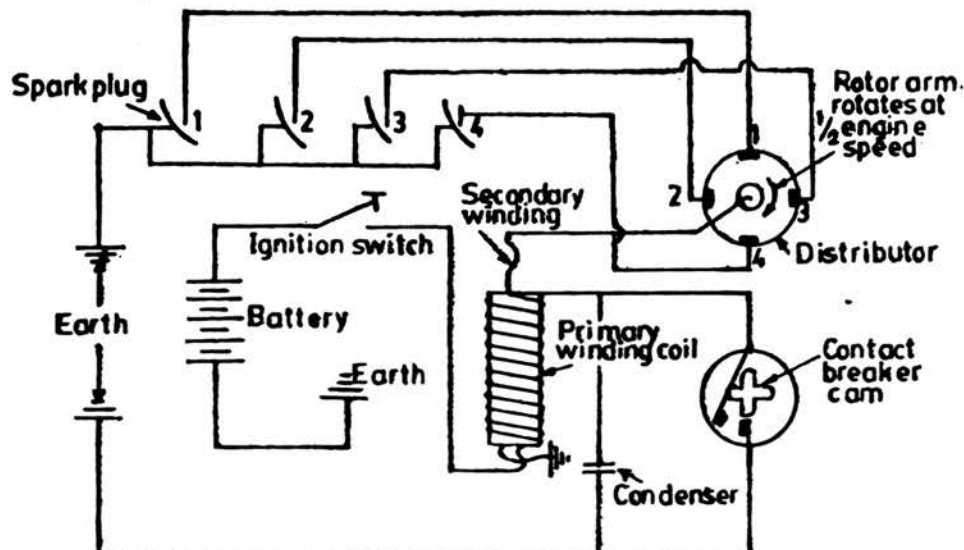


Fig. 6-7. Battery and coil ignition of four cylinder, four-stroke cycle petrol engine.

made available to run the engine. In addition to the fuel for the purpose of combustion, two things are necessary – the oxygen supplied in air and some means for igniting the fuel. All petrol and gas engines use electric spark ignition. Diesel engines, on the other hand, use the heat of compression alone to ignite the fuel. Before the successful introduction of spark ignition, petrol and gas engines used hot-tube ignition.

Electric spark ignition is used practically on all gas and petrol engines. The function of an electrical ignition system is to produce high voltage spark and to deliver it to the spark plugs at regular intervals and at the correct time with respect to the piston position. A high tension voltage of about 15,000 volts is normally required to ignite the mixture or air and fuel vapour in the cylinder. The high tension voltage supply is obtained from a magneto system or a battery and coil ignition system.

The ignition is performed by the supply of a high voltage of 15,000 volts across the electrodes of a spark plug placed in the combustion space of the cylinder. Air and gas separate the two electrodes of the spark gap and offer large resistance to the flow of electric current. When heavy voltage is applied to the two electrodes of the spark plug which are separated by air, the air is subjected to electric strain. The strain increases with the increasing voltage till a point is reached when the voltage is sufficient to break down the resistance of the air. As soon as the resistance of the air breaks down, the high voltage spark jumps across the gap between the two electrodes and ignites the air-fuel mixture.

6.11.1 Battery and coil ignition : The ignition circuit for a four-cylinder petrol engine working on the four-stroke cycle is shown in fig. 6-7. The primary circuit consists of a 12 volt *battery*, ignition switch, primary winding of the coil and contact breaker. The secondary circuit consists of the secondary winding of the coil, distributor and four spark plugs. The *primary winding* of the coil consists of a comparatively few turns of coarse (thick) wire around the iron core. Around this is wrapped the *secondary winding* consisting

of a large number of turns of very fine wire. The *contact breaker* is worked by a contact breaker cam fitted on a shaft driven by the engine.

To begin with, the *ignition switch* is put on and the engine is cranked. When the contacts touch, the current flows from the battery through the switch, through the primary winding of the *induction coil*, to the contact breaker points and return to the battery through the ground (earth). A *condenser* is connected across the terminals of the breaker points. This prevents excessive arcing at the breaker points and thereby prolongs the life of contact breaker points. Immediately after this the moving contact breaker cam break opens the contact. The breaking of the primary circuit causes a change of magnetic field and this induces a very high voltage in the secondary winding coil. The ratio of number of turns of secondary to primary has been so adjusted to give a voltage of about 15,000 volts across secondary terminals. This high tension voltage passes to the distributor and thence to the individual spark plugs which are screwed in the cylinder head. The high tension voltage is applied across the spark plug gap (approximately 1 mm). Due to high voltage, the spark jumps across the spark plug gap causing ignition of fuel-air mixture.

In multi-cylinder engines there being more than one cylinder, a high tension voltage has to be applied in turn to various spark plugs. For this purpose the *distributor* is used. The high tension voltage is brought to a rotating terminal (known as rotor arm) which in moving, contacts a definite order (i.e., 1-3-4-2) with various points in the distributor i.e. (1-3-4-2) as shown in fig. 6-7.

Both the rotor arm and contact breaker get their motion through the same mechanism driven by the engine. Every instant when the rotor arm is on one of the contacts, at that time the contact breaker cam must break the contact and therefore, the motion of the rotor arm has to be synchronized with that of the contact breakers. For this purpose the contact breaker mechanism is housed in the body of the distributor. The contact breaker cam must have as many projections as there are number of cylinders. The speed of the rotor arm and the contact breaker cam must be half the speed of the engine in case of four-stroke cycle engines.

From the distributor 4 wires are connected to 4 spark plugs. The induction coil is grounded on the high tension circuit and plugs are grounded by mounting them in the engine metal. As the spark or current is conducted from the high tension lead of the coil to the centre of the distributor and then from any four points of the distributor to the plug electrodes, the spark will jump over the air gap at the plug points and return through the ground to the coil.

It takes some time after the occurrence of the spark for the fuel-air mixture to ignite and release heat, so that arrangements must be made for spark to occur before the top dead centre. Such an operation is called *advancing the spark*. In variable speed engines, the angle turned through by the crank ignition delay period varies with the engine speed, and it is, therefore, necessary to have a device to increase the angle of advance as the speed increases. This is usually carried out automatically by a small centrifugal governor which alters the angular position of the cam operating the contact breaker.

In *magneto ignition system*, the magneto may consist of magnets rotating in fixed coils or coils rotating in fixed magnets. In this system, no battery is required.

6.11.2 Compression ignition : The original Diesel engines were heavy and of slow speed. The modern high speed and light Diesel engines differ in design so materially from the original Diesel engines that the term *compression-ignition engine* is employed to distinguish the modern Diesel units from the old types.

The compression-ignition (C.I.) engine does not rely upon a spark from an external source for ignition but utilises the high temperature produced at the end of the compression stroke to produce ignition of the fuel.

The C.I. engine is fed with air alone during the suction stroke. Consequently absence of fuel during compression, enables to obtain much higher compression ratio ranging from 12 to 20. The higher compression of the air results in higher temperature at the end of the compression, which is sufficient to ignite the fuel. The fuel is injected in the form of a fine spray into the hot air which is compressed to a pressure much more higher than that in petrol or gas engine.

The C.I. engines are cold starting engines but much trouble is experienced in starting them in cold weather. To overcome

this, some engines are provided with hot tube in the cylinder head, which can be heated to warm the combustion chamber of the cylinder, while some engines are fitted with electric heated plugs for starting purposes. The heating is not necessary once the engine starts and obtains general running temperature conditions.

6.11.3 Hot tube ignition : In this form of ignition, a porcelain tube is provided in the combustion chamber of the engine cylinder as shown in fig. 6-8.

Before starting the engine, the tube is heated to a red hot by a burner or blow lamp. The required temperature for the ignition of the charge is attained partly by the heat of the hot tube. Once the engine is started, the tube will be kept hot by the combustion of the fuel in the cylinder. The burner or blow lamp, therefore, is not required after starting the engine. The electric spark ignition was used on the earliest gas engines which

was then replaced by the hot tube. The hot tube ignition is used in gas and light oil engines.

6.11.4 Hot bulb ignition : In this system a chamber of bulb shape is attached to the cylinder head as shown in fig. 6-9. This chamber is unjacketed and is heated by a blow lamp before starting the engine. The fuel is injected into the hot combustion chamber at the end of the compression stroke and ignition takes place partly due to heat of the compressed charge of air and partly due to heat of the hot bulb.

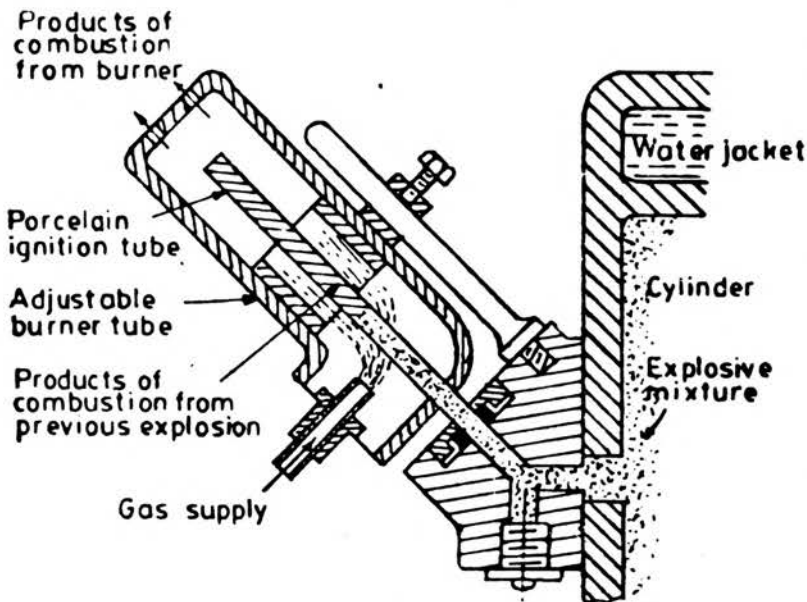


Fig. 6-8. Hot tube ignition.

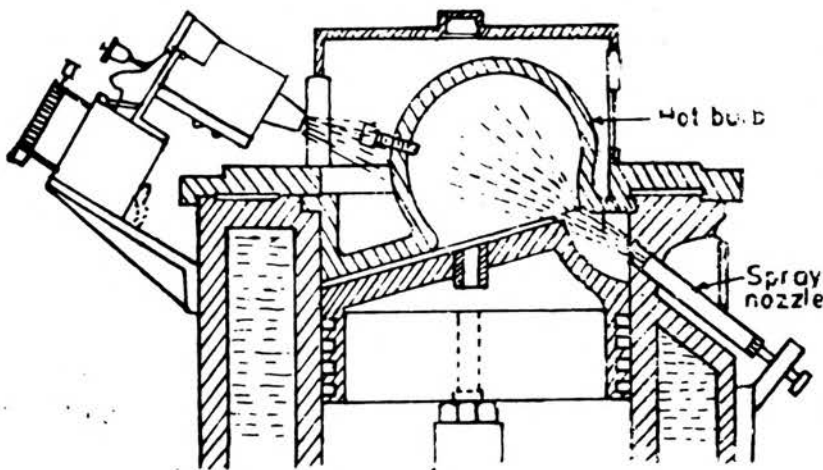


Fig. 6-9. Hot bulb ignition.

The blow lamp is removed after the engine takes up its speed. The ignition then goes on due to the combined effect of compression heat and the heat retained by the combustion chamber from the previous cycle. This method of ignition is used in semi-Diesel engines, where heavy oils can be successfully dealt with. The system is also known as *surface ignition* or *hot combustion chamber ignition*.

6.11.5 Ignition lag or Delay period : It is the time taken to heat the fuel particles, turn them into vapour, and start combustion after the ignition is begun or initiated. This means there is a time lag between the first ignition of the fuel and the beginning of the actual combustion process which also means that there is a time interval in the process of chemical reaction which prepares the molecules of the fuel to ignite.

This time interval is found to occur with all fuels. The length of the delay period depends upon various factors, such as pressure, temperature, rate of fuel injection and the nature of the fuel. The delay period may spread over about 10 degrees movement of the crank in case of compression-ignition engines.

6.12 Methods of Cooling I.C. Engine Cylinders

Very high temperature is developed in the cylinder of an I.C. engine as a result of the combustion taking place inside the cylinder. It is, therefore, necessary to carry away some of the heat from the cylinder to avoid injury to the metal of the cylinder and piston. If the cylinder is not cooled, the seizure (jamming) of piston in the cylinder would occur as a result the piston and its rings becoming too hot; also it would not be possible to lubricate the piston since the heat would burn any lubricant that may be used. The walls of the cylinder must be cooled so that the charge may be compressed without danger of pre-ignition and the temperature must be maintained within fairly close limits to achieve the desired compression ratio and therefore, maximum power. Too much cooling, on the other hand will reduce the thermal efficiency of the engine and cause waste of fuel due to improper vaporization of the fuel. Heat equivalent of about 80 per cent of the brake power developed has to be extracted away through the cylinder walls. The object of cooling is achieved by the use of any of the methods, namely direct or *air cooling*, and indirect or *water cooling*.

6.12.1 Air cooling : This is the simplest method in which the heat is taken away by the air flowing over and around the cylinder. In this method, cooling fins are cast on the cylinder head and cylinder barrel with the object of providing additional conductive and radiating surface as illustrated in fig 6-10. The cooling fins or circumferential flanges are arranged so that they are perpendicular to the axis of the cylinder. The current of air for cooling the fins may be obtained either from a fan driven by the engine or by movement of the engine itself, as in the case of motor cycle engines, automobile engines, or aero-plane engines.

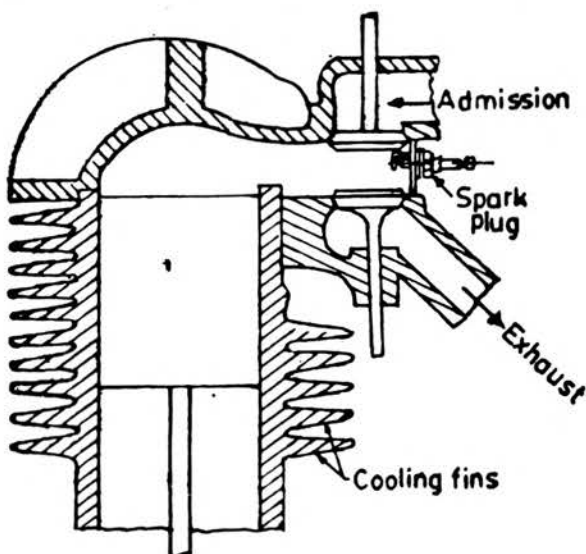


Fig. 6-10. Section of an air-cooled cylinder

6.12.2 Water cooling : In this method, the advantage of superior conductive and convective properties of water is taken. The cylinder is provided with an annular space called *water jacket* (fig. 6-11) through which water is circulated continuously. The water jacket should cover the entire length of the piston stroke to avoid unequal expansion in the cylinder bore and burning of lubrication oil. The water space should be wide in large cylinders, and cleaning doors should be

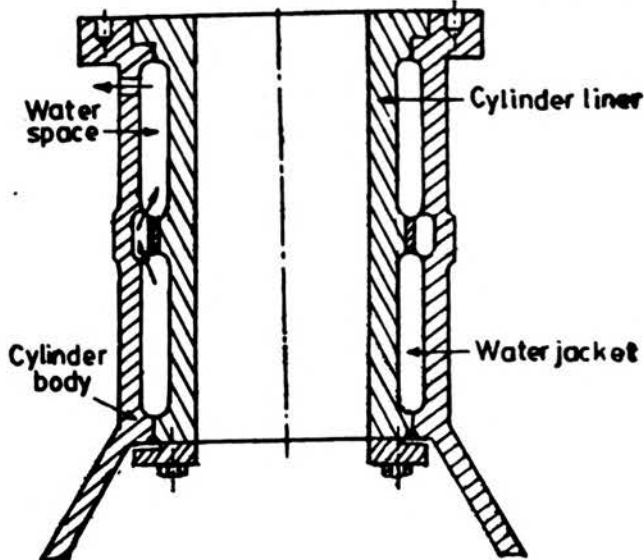


Fig. 6-11. Section of a water cooled cylinder.

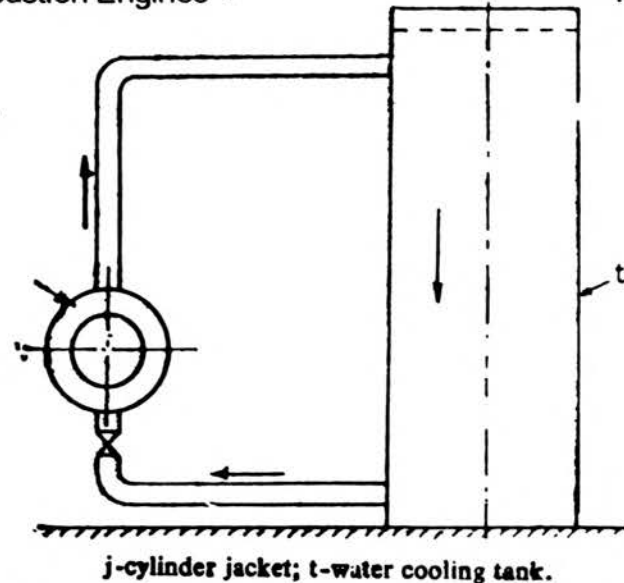
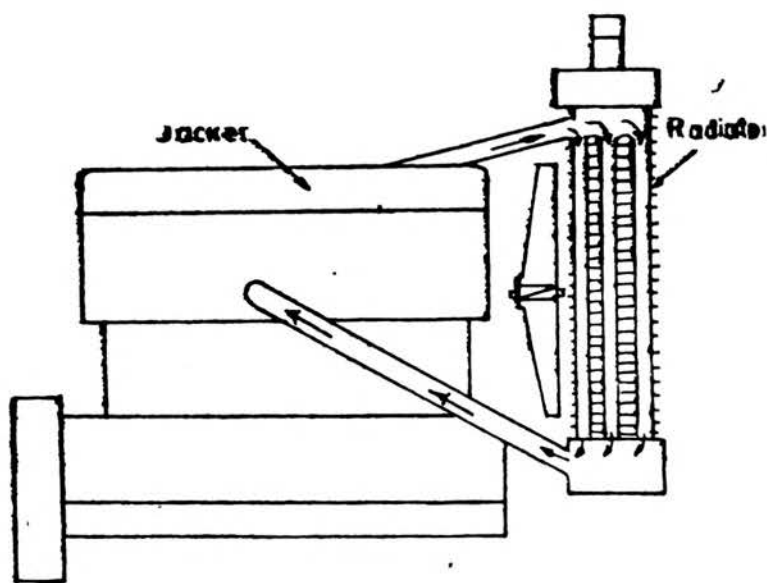


Fig. 6-12. Gravity or thermo-siphon system of circulation provided for cleaning water jacket. The life of the cylinder can be increased if the cleaning is done at the intervals of four to eight weeks.

There are two methods of obtaining the circulation of water in use, namely *gravity circulation* and *forced circulation*. Gravity circulation, also called *thermo-siphon circulation*, is based on the fact that when water is heated its density decreases and it tends to rise, the colder particles sinking in the place of rising ones. Circulation is obtained if the water is heated at one point and cooled at another.

Fig. 6-12 shows the gravity circulation system for a small horizontal engine. Water is heated in cylinder jacket *j* and flows to the tank *t* where it is cooled by radiation, gradually descending to the bottom, and flows back to the engine.

Fig. 6-13 shows the gravity circulation (thermo-siphon circulation) as applied to an automobile engine.



To obtain proper water circulation, the connection between the engine cylinder jacket and the radiator should have very small resistance to the water flow and be wide, short and have as few bends as possible. When the temperature difference is small, the circulation of water is slow, as at light loads. At heavy loads, water in jackets may boil. This system is used only in small engines where simplicity is of great importance.

The circulation of water in the thermo-siphon system is slow. It is, therefore, necessary

to use a pump to maintain circulation of cooling water in large and medium size engines. The pump should be of the centrifugal type, as it allows the free passing of water through it, if it stops working for any reason.

The advantage of forced circulation is the ease of controlling the jacket temperature by regulating the opening of the valve between pump and the engine. Fig. 6-14 shows a pump circulation system in an automobile engine with a centrifugal pump and automatic temperature control by a by-pass with valve operated by a thermostat element.

Water cooling is more effective than air cooling, for the heat conducted away from a surface surrounded by water is about one hundred times more than that conducted away from a similar surface surrounded by air. But the rate of heat flow depends entirely on the motion of air and water. The temperature of water leaving the jacket should be about 65°C for best economy. Too low a temperature leads to loss of efficiency and too high a temperature interferes with the lubrication of the cylinder.

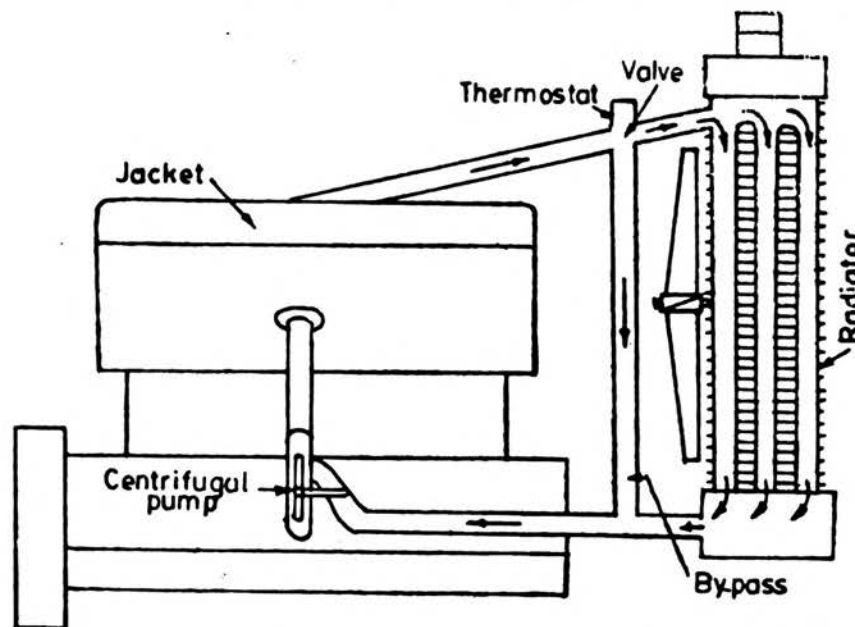


Fig. 6-14. Forced circulation system with thermostat control of automobile engine.

In large Diesel and gas engines, it is usual to cool the pistons and exhaust valves with water to prevent pre-ignition. For this purpose special pipes are provided to circulate the water through the hollow pistons and exhaust valves.

6.13 Merits and demerits of the Cooling Systems

The advantages of air cooling system are :

- (i) Simplicity and lightness of the system,
- (ii) Radiator is not required,
- (iii) No danger from freezing of water in cold climate,
- (iv) Reduction in warming up period,
- (v) Unit cylinder construction is used in case of multi-cylinder air-cooled engines (i.e., separate cylinder block is used for each cylinder). Hence, in multi-cylinder air-cooled engine, only damaged cylinder can be replaced while in multi-cylinder water cooled engine, whole cylinder block has to be replaced,
- (vi) Less influenced by damage. A small hole in radiator or tubing means break down of working of water-cooled engine while the loss of several fins of air-cooled engine, practically can continue its operation. This is very important when used for military purpose, and
- (vii) Easy maintenance.

The disadvantages of air cooling system are :

- (i) Air cooling being not so effective as water cooling, the resulting higher working temperature of the cylinder limits the compression ratio to a lower value than with water cooling.
- (ii) The fan, if used for air circulation, absorbs about 5 per cent of the engine power.
- (iii) The different portions of the cylinder are not uniformly cooled; the front portion being cooled more than the rear (back) portion. This results in some degree of distortion (twisting out) of shape.
- (iv) The lubricating oil of air-cooled engine runs hotter and so an oil cooler may be required.
- (v) The air-cooled engine is more prone (inclined) to distortion and so running clearances have to be increased which gives rise to noisier running. In addition, the fins are liable to resonate (give noise) while water cooling (requiring a jacket) helps to damp out the noise. Thus, the air-cooled engine is noisier than the water cooled engine.

The advantages of water cooling system are :

- (i) Cooling is more efficient and thus higher compression ratio is permitted.
- (ii) Uniform cooling is attained.

The disadvantages of water cooling system are :

- (i) With thermo-siphon system, large quantity of cooling water is required owing to slow rate of circulation.
- (ii) Necessary radiator installation and its maintenance is required.
- (iii) Water freezing causes trouble in very cold weather.
- (iv) Water circulating pump consumes power.

6.14 Methods of Governing I.C. Engines

The purpose of governor is to keep the engine running at a desired speed regardless of the changes in the load carried by the engine. If the load on the engine decreases, the speed of the engine will begin to increase, if the fuel supply is not decreased. As the speed of the engine increases, the centrifugal force on the rotating weights of the governor also increases and moves the control sleeve, together with the fuel regulating mechanism, in the direction of less fuel supply thereby the speed is brought to the rated value. If on the other hand, the load on the engine increases, the engine will begin to slow down because the fuel supply is not sufficient for the increased load. As the speed of the engine decreases, the centrifugal force on the rotating weights on the governor will also decrease and will move the control sleeve, together with the fuel regulating mechanism, in the direction of more fuel supply.

The methods of governing I.C. engines are :

- (i) Completely cutting-off the fuel supply for one or more cycles – This is called *hit and miss method*.
- (ii) Varying the supply of fuel to the cylinder per cycle – This is called *quality method* because the ratio of fuel to air or quality of mixture is altered.
- (iii) Varying the supply of air as well as the supply of fuel, the ratio of air to fuel is kept approximately constant so that quality of mixture remains approximately constant but quantity of fuel-air mixture supplied to the cylinder in each cycle is varied – This is called *quantity method*.

- (iv) Combination of the quality and quantity methods – This is called *combination method*.

6.14.1 Hit and miss method of governing : The system as the name implies consists in omitting an explosion occasionally when the speed rises above the mean speed. The lesser the load on the engine, the greater is the number of explosions omitted. The usual method of missing an explosion is to omit the opening of the gas valve in the case of gas engines, and putting the plunger of the fuel oil pump out of action in case of oil engines, so that no fuel is admitted and the engine performs an idle stroke.

With hit and miss method of governing, there is a working stroke for every cycle under condition of maximum load. At lighter loads, when the speed increases, the governor mechanism acts to prevent admission of the charge of fuel occasionally and there is no explosion, causing the engine to miss. This loss of power decreases the speed of the engine; the governor mechanism opens the inlet valve, an explosion or hit occurs and the engine receives the power stroke. With this method of governing, the engine operates either under condition of maximum efficiency or does not fire at all. Hit and miss governing gives better economy at light loads than other methods. The great disadvantage of this method is the absence of turning effort on the crankshaft during the idle cycle, necessitating a very heavy flywheel to avoid considerable variation of speed. This method may be used for engines which do not require close speed regulation and with small size engines of less than 40 brake power.

Fig. 6-15 shows the principle of hit and miss governing as applied to gas engines.

The cam C on the cam shaft lifts the end A of the lever ABD once in two revolutions, and the knife edge J opens the gas valve unless the speed is above normal.

When the speed exceeds the normal, the governor weights fly further outwards and lift the end F of the lever FGH which moves the knife edge J to the right thus causing it to miss the opening of the gas valve. As applied to oil engines the mechanism is same, but it is the plunger of the fuel pump, instead of gas valve, which is put out of action.

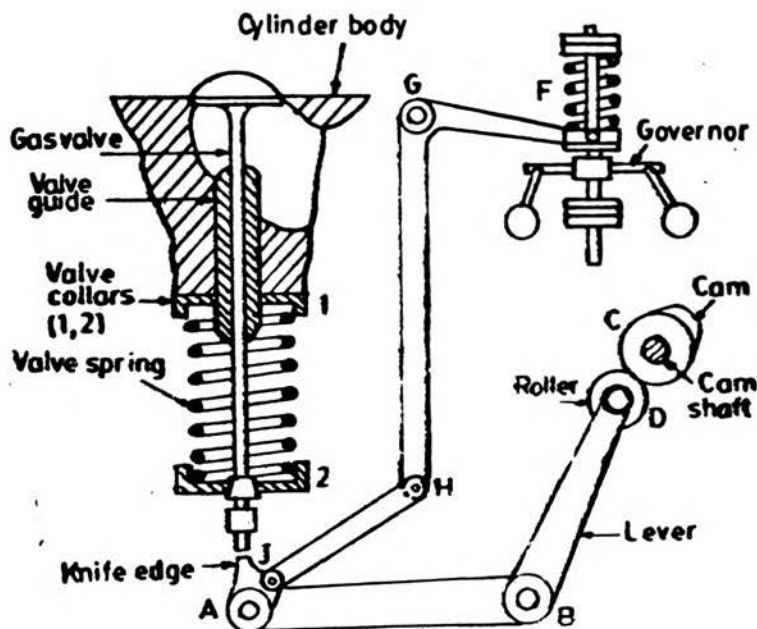


Fig. 6-15. Hit and miss governing as applied to gas engines.

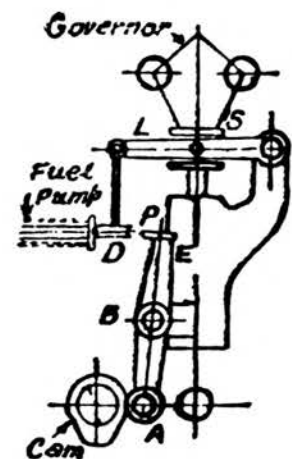


Fig. 6-16. Hit and miss as applied to oil engines.

Fig. 6-16 shows the principle of hit and miss governing as applied to oil engines.

The cam on the cam shaft operates the end A of the lever ABE once in two revolutions and the pecker piece P strikes the distance piece D hung against the outer end of the fuel pump plunger. When the speed of the governor exceeds a certain limit, due to reduction in load, the governor raises the distance piece D so that the pecker piece P misses it. In this way the fuel pump is put out of action until the speed is reduced sufficiently and D drops back in position.

6.14.2 Quality method of governing : As applied to gas engines, the quality governing is effected by reducing the quantity of gas supply to the engine. This is done by varying the lift of the gas valve. Another simple method is to have a throttle valve operated by the governor in the gas passage leading to the admission valve of the gas engine.

As applied to oil engines, the quality governing is effected by varying the amount of fuel oil entering the engine cylinder per cycle. This is done by :

(i) Altering the stroke of the fuel pump plunger under the action of the governor and so varying the oil supply to suit the load on the engine.

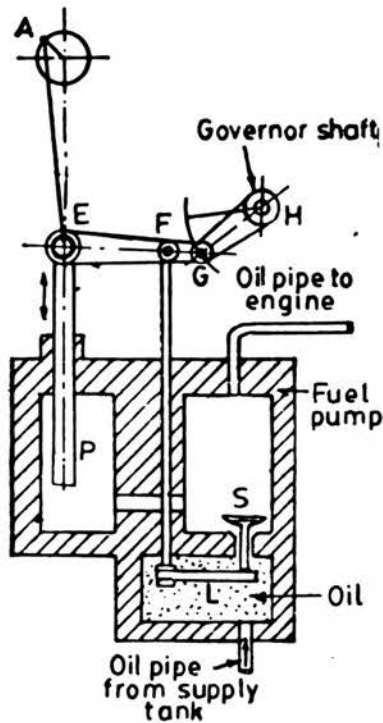


Fig. 6-17. Quality governing by spill method as applied to oil engines.

(ii) Having a control valve on the delivery side of the fuel pump which opens under the control of the governor after a part of delivery stroke has been performed. Here, oil is delivered during the first part of the delivery stroke and returned to the suction side during the remainder part of the delivery stroke.

(iii) Keeping the suction valve of the fuel pump held open by levers under the control of the governor, during the first part of the delivery stroke. Hence the oil is returned to the fuel pump during the first part of the delivery stroke and delivered to the injection nozzle during the remainder part of the delivery stroke. At light loads this suction valve may be kept open for almost the whole delivery stroke. This is the general practice in Diesel engines. This method is generally known as *spill method*, since the oil is "spilled" (fall) back to the fuel pump from oil sump (tank). The general principle of the simple spill valve will be understood by reference to fig. 6-17.

A is a rotating shaft which drives the pump plunger P. EG represents a lever which may oscillate about either E or G. F is a point between E and G, to which is connected a light spindle with lever L. The end of the lever L is under the stem of the suction valve S.

The shaft H is under the control of the governor. It will be evident that as F moves up under the control of governor, the suction valve will seat (close) late, with the result that less of the fuel pump stroke will be effective and a less amount of oil will reach the spray nozzle. The fuel pump plunger stroke is generally much longer than is necessary to deliver the full amount of oil needed at full engine load.

(iv) By altering the angular position of the helical groove of the fuel pump plunger relative to the suction port and thereby varying the effective stroke (part of the stroke for which oil is delivered) of the plunger. *This is a general practice in modern solid injection, compression-ignition, high speed engines.*

The principle of governing will be understood with reference to fig. 6-19(a). Fuel oil flows to the fuel pump under gravity when the fuel pump plunger P uncovers the suction

ports B and C on the downward stroke. The space above the plunger is filled with oil at the beginning of the upward stroke. During the first part of the upward or delivery stroke, a small quantity of oil is forced back into the suction space, until the plunger closes both the suction port holes B and C. From then on, the fuel is put under pressure and pump plunger begins to force it through the delivery valve and fuel line into the atomiser (Fig. 6-18a).

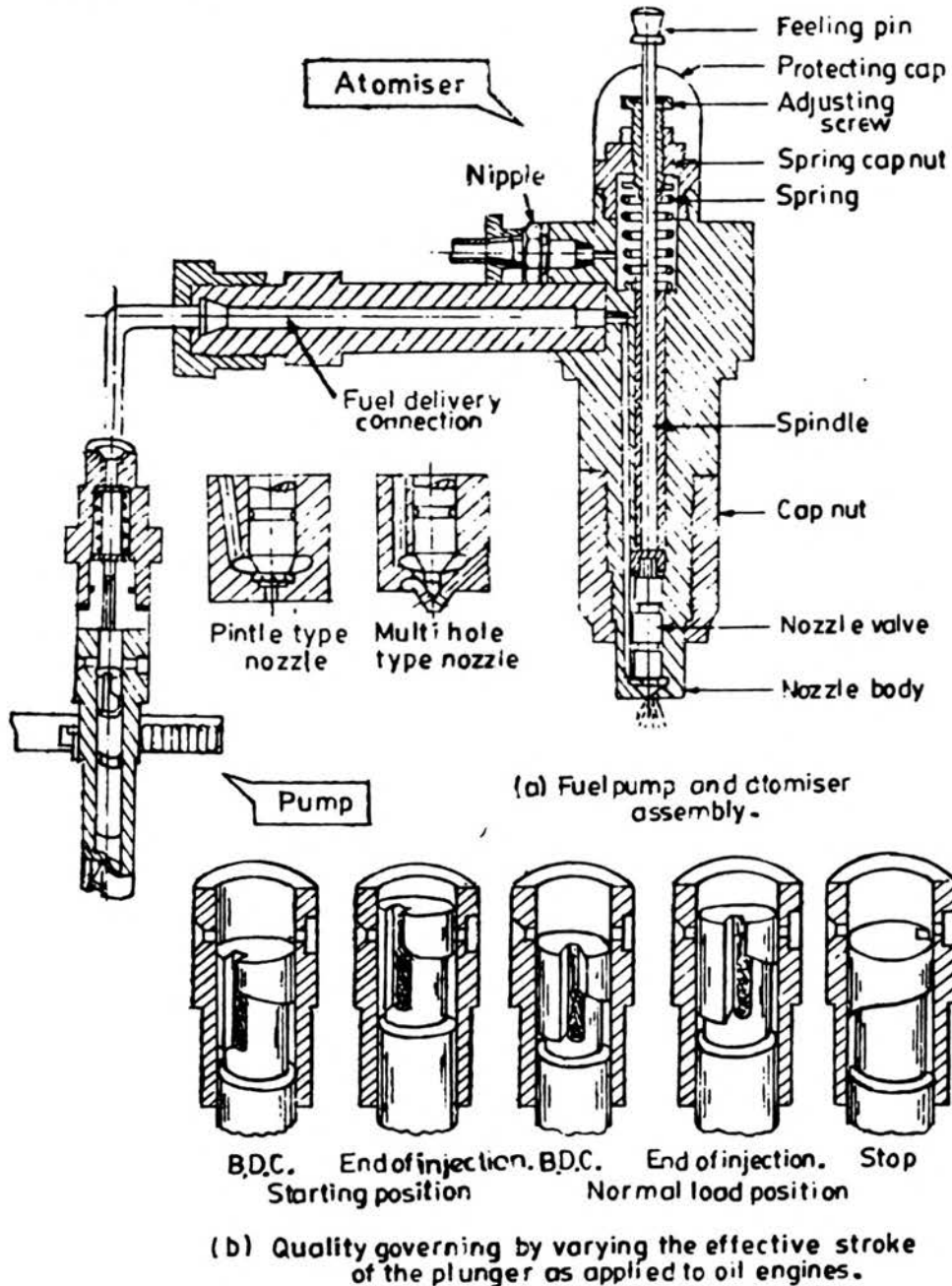


Fig. 6-18.

Delivery begins as soon as the plunger has covered the port holes on the way up and ends as the sloping edge E of the helical groove D opens the port hole C (Fig. 6-19a) on the right hand side and permits the fuel to escape from the pressure space above the plunger and the port hole C, to the suction space. The pressure is then relieved and the delivery stops. The plunger P is rotated by the rack shown in fig. 6-19(b). The toothed rack is moved in or out by the governor. Thus by rotating the plunger, i.e., by altering

the angular position of the helical groove D of the fuel pump plunger, relative to the suction port C, the length of the effective stroke for which oil is delivered is varied and hence the amount of fuel delivered to the engines is also varied.

In the two views (starting position) at the left [fig. 6-18(b)], the plunger is shown in the position for maximum delivery, in which the edge of the helical groove does not

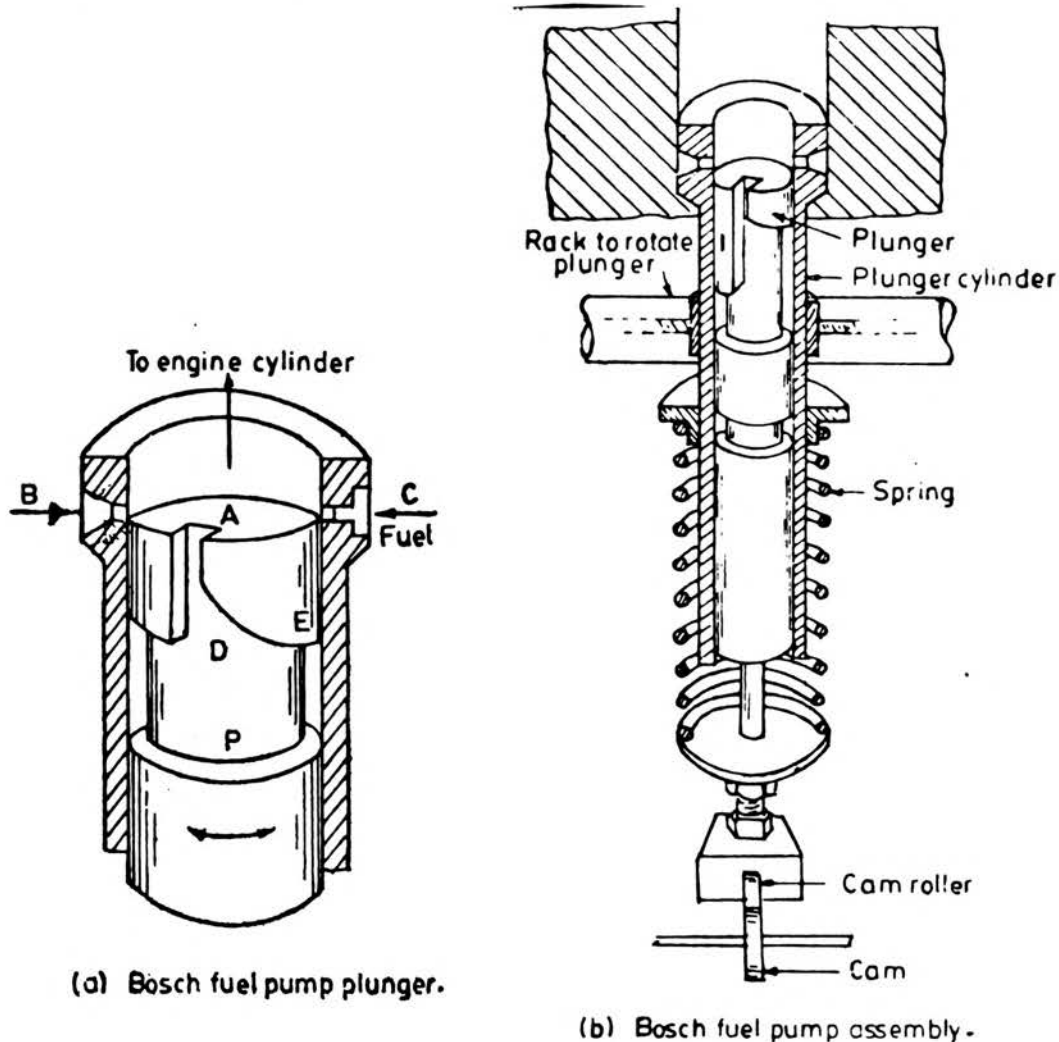


Fig. 6-19.

open the port hole on the right hand side at all. The next two views show the position of the plunger for medium delivery of fuel (normal load position), and the one at right (stop position) shows the position when no fuel is delivered.

In quality governing there being no restriction to the amount of air admitted, the same mass of charge is taken into the cylinder; hence pressure reached at the end of the compression is the same. Theoretically the thermal efficiency is, however, unchanged being dependent on the compression ratio. At light loads, the efficiency generally drops because of the difficulty of getting ignition and rapid combustion with weak mixture. Its chief advantage is a mechanical one, where high speeds are used; as at high speed the inertia of the reciprocating parts becomes considerable, and unless met by a constant compression pressure, the engine does not run smoothly.

Quality governing is chiefly used where the engine is to be worked at or near the full load.

6.14.3 Quantity method of governing : Quantity governing may be accomplished by varying the amount of mixture entering the cylinder, while the proportion of fuel to air and number of working cycles are constant.

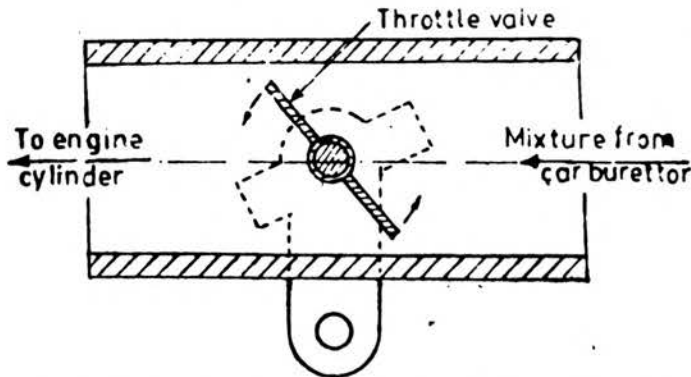


Fig. 6-20. Quantity governing throttle control as applied to petrol engines.

It is applied to petrol engines by having a throttle valve in the pipe leading from the carburettor to the engine cylinder (fig. 6-20). The automobile engine is hand governed by a quantity (throttle) control of the charge entering the cylinders, the proportion of petrol to air remaining the same for a given carburettor adjustment.

It is applied to gas engine in various ways. The gas and air supplied may each be throttled by separate valves in the gas and air passages leading to the admission valve or after passing through the mixing valve, the air and gas together may be throttled by a single valve just before reaching the admission valve. Another method is to regulate the lift of the admission valve. This method of governing is illustrated in fig.6-21. The cam on the cam-shaft moves one end of the valve through a fixed

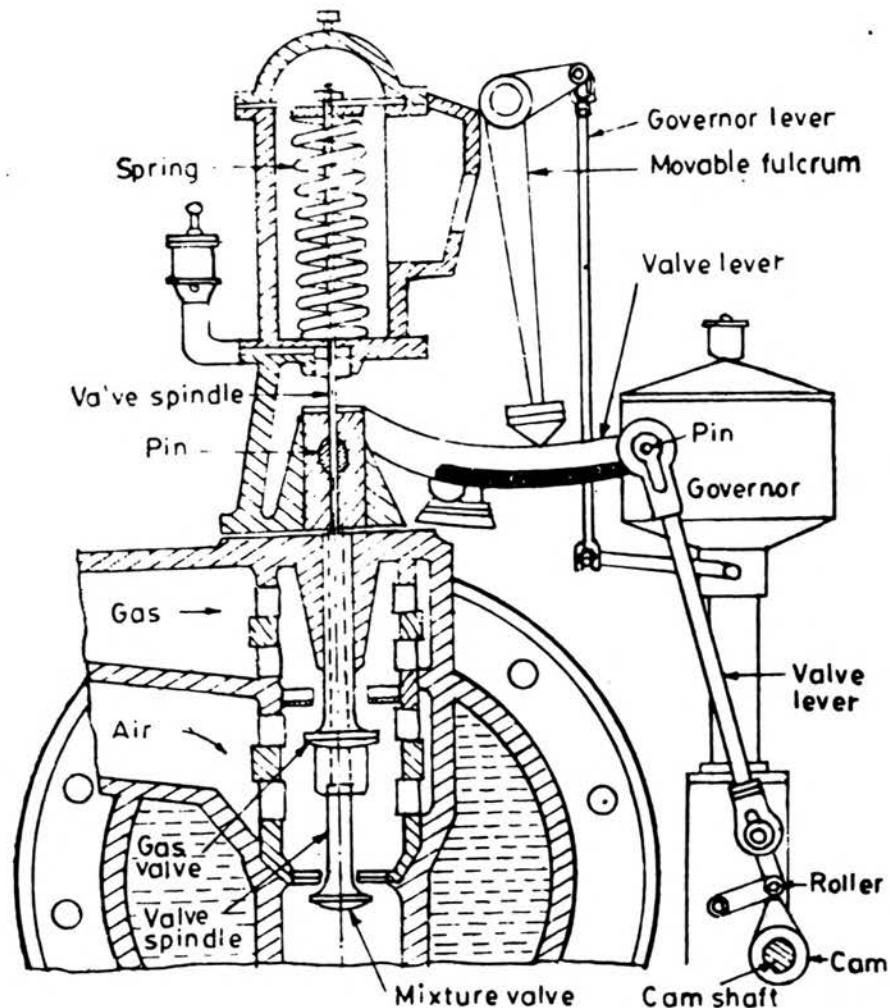


Fig. 6-21. Quantity governing as applied to gas engines by varying the lift of the admission valve.

distance, but the distance moved by the other end of the valve lever, which opens the admission and gas valve, depends upon the position of the movable fulcrum. The fulcrum is not fixed but is moved by the governor through governor levers to a position suitable for the load on the engine. Thus, the lift (opening) of the admission valve, suitable for the load, is regulated by changing the position of the movable fulcrum.

The efficiency of internal combustion engine chiefly depends upon having a high compression pressure. The mixture drawn into the cylinder in this system of governing (quantity governing) is less than the full charge and the pressure at the end of compression is reduced and the efficiency is, therefore, slightly less. An advantage of this system is that the mixture being of constant composition, there is little trouble in igniting the mixture even with no load. The combustion of the mixture is less rapid at low compression and, therefore, the ignition should be a little earlier at light loads. In some engines, the governor advances the spark as well reduces the quantity of mixture at light loads. Advancing the spark means the ignition takes place when the crank is on the top dead centre. If the spark is advanced too far, complete ignition may take place before the crank reaches the top dead centre and cause a back explosion.

6.14.4 Combination method of governing : The governing of an engine may be obtained by combining two or more of the above methods. For instance, quality or quantity governing at high loads has been successfully combined with hit and miss governing at low loads. Also quality governing at high loads is used with quantity governing at low loads. The latter system is economical and gives close governing.

6.15 Highest Useful Compression Ratio

Compression ratio is the ratio of the volume of the cylinder at the beginning of the compression stroke to the volume at the end of the compression stroke, or

$$\text{Compression ratio, } r = \frac{\text{Volume swept by the piston} + \text{Clearance volume}}{\text{Clearance volume}}$$

From thermodynamic considerations, it has been found that the ideal thermal efficiency (air-standard efficiency) of an engine running on any cycle improves as the compression ratio is increased. Fig. 6-22 is a graph showing variation of air-standard efficiency of an engine working on the Otto cycle with compression ratio. It is also found that the mean effective pressure on the engine piston increases with the increase in compression ratio. A higher compression ratio also causes an acceleration in the rate of combustion. The

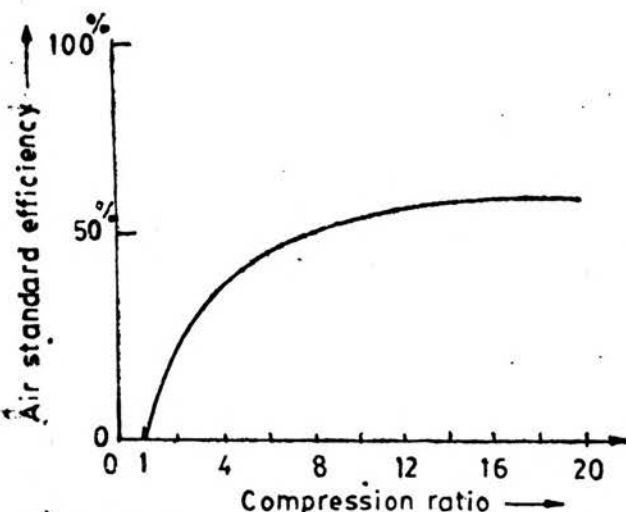


Fig. 6-22. Graph showing variation of air-standard efficiency with compression ratio for Otto cycle engines.

higher compression ratio can be obtained by reducing the clearance space, that is, the combustion space. The smaller the volume of the combustion space the less amount of exhaust gases it will retain, which will result in less dilution of the fresh mixture. This means more uniform burning of the charge, and more power produced. Higher compression ratio produces higher temperature and pressure, which increases the rate of combustion. Hence, higher speed is possible and weaker mixtures can also be burnt.

6.15.1 Limiting compression ratio :

From the above considerations it would appear that a higher compression ratio is available. However, a limit has to be set

to the higher compression ratio for engines in whose cylinder, mixture of air and fuel gas or fuel vapour is compressed together. If the compression pressure is too high, in such a case the resulting temperature during compression stroke is also high enough to ignite the charge before the end of the compression stroke.

In a petrol engine, mixture of petrol vapour and air is compressed. Higher compression ratio in a petrol engine will therefore, cause *pre-ignition* of the charge resulting in the loss of power and possible mechanical damage to the engine. Such a consequence has to be avoided by limiting the compression ratio according to the nature of fuel used. The safe compression ratio can be higher with the lower percentage of hydrogen in the fuel.

The safe compression ratio for Otto cycle engine can be somewhat raised, by spraying a small quantity of water directly into the cylinder during the suction stroke or into the vapouriser resulting in lower compression temperature.

As far as pre-ignition is concerned, there can be no limit to the compression ratio in Diesel engines, where air alone is compressed. But even with this kind of engine, too high value of compression ratio will require very small clearance space. Thus, the value of limiting compression ratio in Diesel engines will depend upon minimum mechanical clearance necessary between piston and cylinder head for safety consideration.

6.15.2 Pre-ignition : In an engine running on Otto cycle, the combustion during the normal working is initiated by an electric spark. The spark is timed to occur at a definite point just before the end of the compression stroke. The ignition of the charge should not occur before this spark is introduced in the cylinder. If the ignition starts, due to any other reason, when the piston is still doing its compression stroke, it is known as *pre-ignition*. Pre-ignition will develop excessive pressure before the end of compression stroke, tending to push the piston in the direction opposite to which it is moving. This will result in loss of power and violent thumping and may stop the engine or do mechanical damage to the engine. The pre-ignition may occur on account of higher compression ratio, over-heated sparking plug points, or incandescent (glowing with heat) carbon deposited on the surface to the cylinder or spark plugs. It may also be due to faulty timing of the spark production.

6.15.3 Detonation : Detonation, pinking or knocking is the name given to violent waves produced within the cylinder of a spark ignition engine. The noise produced is like that produced by a sharp ringing blow upon the metal of the cylinder.

The region in which the detonation occurs is far away from the spark plug, and is known as the *detonation zone*. After the spark is produced, there is a rise of temperature and pressure due to the combustion of the ignited fuel. This rise of temperature and pressure both combine to increase the velocity of flame, compressing the unburnt portion of the charge of the detonation zone. Finally, the temperature in the detonation zone reaches such a high value that chemical reaction occurs at a far greater rate than the advancing flame. Before the flame completes its course across the combustion chamber, the whole mass of remaining unburnt charge ignites instantaneously without external assistance (auto-ignition). This spontaneous ignition of a portion of the charge, sets rapidly moving high pressure waves that hit cylinder walls with such violence that the cylinder wall gives out a loud pulsating noise, called *knocking or pinking*. It is this noise, that expresses or indicates detonation.

If the detonation wave is violent, it may even break the piston. Its other effect is to overheat the spark plug points so as to prepare way for pre-ignition. Detonation and pre-ignition are two distinct phenomena. Pre-ignition occurs before the spark takes place while detonation occurs just after the spark.

6.15.4 Volumetric efficiency : The power developed by an I.C. engine depends on

the mass of mixture of fuel and air or air only which is present in the cylinder at the end of suction stroke. The mass of mixture of air present depends upon the breathing efficiency of the engine. The breathing (inhaling) efficiency is measured by the volumetric efficiency of the engine.

The volumetric efficiency of an I.C. engine is the ratio of the charge taken in (inhaled) during the suction stroke at normal temperature and pressure to the volume swept by the piston, or

$$\text{Volumetric efficiency} = \frac{\text{Volume of fresh charge aspirated per stroke at N.T.P.}}{\text{Volume swept by the piston}}$$

In case of petrol and oil engines, the charge aspirated or taken in per stroke should be replaced by air aspirated per stroke.

In case of gas engines, the charge aspirated per stroke should be replaced by mixture of gas and air aspirated per stroke.

This ratio enables comparison of the respiratory performance of an actual engine with the ideal engine. An ideal engine is assumed to aspirate and fill completely the swept volume with the charge at normal temperature and pressure. The difference between the actual charge drawn into the cylinder per stroke and the swept volume is due to the reasons stated below :

- (i) The suction pressure is less than the atmospheric pressure because of the resistance of the inlet valves and passages. Therefore, the mass of charge drawn in is less than that if atmospheric pressure were maintained.
- (ii) The internal passages and surfaces of the engine being hot, the charge is heated as it enters the cylinder. The increase in temperature of the charge, reduces mass of charge that enters the cylinder.
- (iii) Any gases left in the clearance space of the engine cylinder at the end of the exhaust stroke, are at a pressure above atmospheric and they will expand during the suction stroke to the intake pressure before the new charge begins to enter. The volume of fresh charge taken in during the suction stroke is, therefore, reduced.
- (iv) The mass of the charge drawn in at high altitudes is decreased below that which would be drawn in at sea level, as the pressure of the atmosphere decreases with altitude and consequently the density of the atmospheric air decreases.

The volumetric efficiency on an I.C. engine also decreases with the increase in engine speed. The faster the engine runs, the greater will be throttling of the incoming charge through valves and passages, and lower will be the volumetric efficiency.

The volumetric efficiency of I.C. engine under normal conditions should be of the order of 70 to 80 per cent.

6.15.5 Supercharging : In an ordinary engine, air-fuel mixture or air only is admitted to the cylinder at atmospheric pressure and is known as a naturally aspirated or normally aspirated engine. *Supercharging* is the forcing of the mixture of fuel and air or air only to the cylinder during the suction stroke under pressure with the air pump or compressor, called *supercharger*, in order to increase the mass or density of the mixture or air admitted to the cylinder. When supercharging is used, the engine is known as *supercharged engine*.

In a petrol engine, the supercharger is generally so fitted that it draws air from atmosphere through the carburettor, compresses the resulting mixture (petrol and air), and then delivers it to the cylinder through the induction system (inlet pipe).

Both the spark ignition (S.I.) and compression ignition (C.I.) engines may be supercharged. The amount of supercharging that can be used with S.I. engine is limited by the detonation of the fuel. In the C.I. engine, on the other hand, supercharging is provided to prevent knocking and is limited by the thermal and mechanical stresses and size and power of the supercharger.

The object of supercharging is to increase the power output of an engine, it is, therefore called *boosting*. Supercharging is employed in the following cases for :

- (i) maintaining the power output of an engine working at high altitudes, such as in aero-engines. At high altitude less oxygen is available for combustion of fuel.
- (ii) reducing the bulk of the engine to fit into a limited space, such as in marine engines (ships).
- (iii) reducing the mass of the engine per indicated power developed, such as in aero-engines (aeroplanes).
- (iv) increasing the existing power of an engine when the necessity of increasing its power arises.
- (v) counteracting the drop in volumetric efficiency which may be due to high altitude, as in the case of aero-engines, or due to high speed as in the case of racing car engines.
- (vi) having better air turbulence (bringing air and fuel in contact quickly), and hence more complete combustion, which results in greater power, reduced specific fuel consumption, and smooth running of the engine.

Superchargers : The increased air pressure (supercharging) is obtained by using a compressor which is known as a supercharger.

The compressor may be a reciprocating compressor, a positive displacement rotary compressor (roots blower, vane type blower, etc.) or a non-positive displacement rotary compressor (centrifugal compressor). In practice, generally reciprocating compressor is not preferred, but roots blower, vane type blower or centrifugal compressor is preferred. The supercharger may be driven by the engine through a gear train, belt or chain driven, or direct coupling to the shaft of the engine. This absorbs power from the engine. In such a case, the engine is known as *mechanically supercharged engine*. The supercharger (centrifugal compressor) may also be driven by an exhaust gas turbine. The set of supercharger (compressor and exhaust gas turbine) is known as *turbocharger* and the engine is known as *turbocharged engine*. Advantage of turbocharged engine is that supercharger does not absorb power from the engine itself but some energy of exhaust gases (which is, otherwise, going to be wasted) is converted into mechanical energy in the exhaust gas turbine and is used to drive the supercharger (compressor).

6.16 Thermal Efficiency of I.C. Engines

No engine can convert all the heat energy supplied by fuel to it into work. The fraction which is converted, is thermal efficiency of the engine. The basis upon which the efficiency is calculated may be indicated power or brake power.

Indicated thermal efficiency : This efficiency is designated by η_i and is defined as the ratio,

$$\eta_i = \frac{\text{Heat equivalent of power produced in the cylinders (indicated power) per unit time}}{\text{Heat supplied to the engine in unit time}}$$

The unit of heat and unit of time must be same for the heat equivalent of power produced and heat supplied to the engine. This is very important.

$$\therefore \eta_i = \frac{\text{Indicated power} \times 3,600}{m_f \times \text{C.V.}} \quad \dots(6.1)$$

where, m_f = mass of fuel oil supplied in kg per hour,
 = fuel consumption in litres per hour \times specific gravity of fuel, and
 C.V. = calorific value of fuel oil in kJ/kg.

In case of gas engine,

$$\text{Indicated thermal efficiency, } \eta_i = \frac{\text{Indicated power} \times 3,600}{V_g \times \text{C.V.}} \quad \dots(6.2)$$

where, V_g = volume of gas supplied in m^3 per hour, and
 C.V. = calorific value of gas in kJ/m^3 .

Brake thermal efficiency : This efficiency is designated by η_b and is defined as the ratio,

$$\begin{aligned} \eta_b &= \frac{\text{Heat equivalent of brake power per unit time}}{\text{Heat supplied to the engine in unit time}} \\ &= \frac{\text{Brake power} \times 3,600}{m_f \times \text{C.V.}} \end{aligned} \quad \dots(6.3)$$

where, m_f = mass of fuel oil supplied in kg per hour, and
 C.V. = calorific value of fuel oil in kJ/kg.

Brake thermal efficiency is also termed as *overall efficiency*.

Relative efficiency : This efficiency is designated by η_r and is defined as the ratio,

$$\frac{\eta_i}{\eta_{\text{ideal}}} = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency or ideal thermal efficiency}} \quad \dots(6.4)$$

and shows how close the actual engine comes to the theoretically possible performance.

Relative efficiency (η_r) of an engine operation on the constant volume cycle

$$= \frac{\text{Indicated thermal efficiency}}{1 - \frac{1}{(r)^\gamma - 1}}$$

Relative efficiency on the basis of brake thermal efficiency (η_r) is defined as the ratio,

$$\frac{\eta_b}{\eta_{\text{ideal}}} = \frac{\text{Brake thermal efficiency}}{\text{Air-standard efficiency or ideal thermal efficiency}} \quad \dots(6.5)$$

Problem – 1 : A four-stroke cycle, four-cylinder, petrol engine has 6.25 cm cylinder diameter and 9.5 cm stroke.

On test it develops a torque of 640 N.m when running at 50 r. p. s.. If the clearance volume in each cylinder is 63.5 cm^3 , the brake thermal efficiency ratio based on the air-standard cycle is 0.5 and calorific value of petrol is 44,800 kJ/kg, determine the petrol consumption in litres per hour and the brake mean effective pressure. Take $\gamma = 1.4$ for air and specific gravity of petrol as 0.73.

Clearance volume, $V_c = 63.5 \text{ cm}^3$ (given);

Stroke volume $V_s = \frac{\pi}{4} d^2 \times l = (6.25)^2 \times 9.5 = 292 \text{ cm}^3$

Compression ratio, $r = \frac{V_s + V_c}{V_c} = \frac{292 + 63.5}{63.5} = 5.58$

Using eqn. (5.8),

$$\text{Air-standard efficiency} = 1 - \frac{1}{(r)^\gamma - 1} = 1 - \frac{1}{(5.58)^{0.4}} = 1 - \frac{1}{1.99} = 0.497 \text{ or } 49.7\%$$

Using eqn. (6.5),

$$\left. \begin{array}{l} \text{Relative efficiency on the basis} \\ \text{of brake thermal efficiency} \end{array} \right\} = \frac{\text{Brake thermal efficiency}}{\text{Air-standard efficiency}}$$

$$\text{i.e. } 0.5 = \frac{\text{Brake thermal efficiency}}{0.497}$$

$$\therefore \text{ Brake thermal efficiency, } \eta_b = 0.5 \times 0.497 = 0.2485 \text{ or } 24.85\%$$

$$\text{Now, brake power (engine)} = T \times 2\pi \times N = 640 \times 2\pi \times 50 = 20,106 \text{ watts}$$

$$= 20.106 \text{ kW}$$

$$\text{Brake thermal efficiency, } \eta_b = \frac{\text{Heat equivalent of brake power in kJ /sec.}}{\frac{m_f}{3,600} \times \text{C.V. in kJ/Kg.}}$$

$$\text{i.e. } 0.2485 = \frac{20.106 \times 3,600}{m_f \times 44,800}$$

$$\therefore m_f = \frac{20.106 \times 3,600}{0.2485 \times 44,800} = 6.508 \text{ kg/hr (Petrol consumption in kg/hr.)}$$

$$\therefore \text{ Petrol consumption in litres per hour} = \frac{6.508}{0.73} = 8.915 \text{ litres/hour}$$

$$\text{Brake power per cylinder} = \frac{20.106}{4} = b.m.e.p. \times a \times l \times n$$

(where *b.m.e.p.* = brake mean effective pressure in kPa, and brake power per cylinder in kW.

$$\text{i.e., } \frac{20.106}{4} = b.m.e.p. \times \frac{\pi}{4} \left(\frac{6.25}{100} \right)^2 \times \frac{9.5}{100} \times \frac{50}{2}$$

$$\therefore b.m.e.p. = \frac{20.106 \times 4 \times 2}{4 \times \pi \times (0.0625)^2 \times 0.095 \times 50} = 689.9 \text{ kPa}$$

Problem – 2 : A gas; engine, working on the four-stroke cycle, uses 15 m^3 of gas per hour at a temperature of 28°C and at a pressure of 100 mm of water above the atmospheric pressure of 720 mm Hg. The gas has calorific value of $19,000 \text{ kJ/m}^3$ measured at 760 mm Hg and temperature of 0°C (N.T.P). The indicated power is 17.6 kW and the compression ratio is 6.5 to 1. Find :

- (i) the indicated thermal efficiency, (ii) the ideal thermal efficiency ($\gamma = 1.4$), and (iii) the relative efficiency of the engine

$$(i) p_1 = 720 + \frac{100}{13.6} = 727.35 \text{ mm Hg; } V_1 = 15 \text{ m}^3 \text{ per hour;}$$

$$T_1 = 28 + 273 = 301 \text{ K;}$$

$$p_2 = 760 \text{ mm Hg; } T_2 = 0 + 273 = 273 \text{ K;}$$

Gas consumption (V_2) per hour at N.T.P. (at 760 mm Hg and 0°C) is to be determined.

$$\text{Now, } \frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

$$\therefore V_2 = V_1 \times \frac{p_1}{p_2} \times \frac{T_2}{T_1} = 15 \times \frac{727.35}{760} \times \frac{273}{301} = 13.02 \text{ m}^3/\text{hr.}$$

Heat supplied per hour = $13.02 \times 19,000 = 2,47,380 \text{ kJ/hr.}$

Work done per hour = $17.6 \times 3,600 = 63,360 \text{ kJ/hr.}$

$$\begin{aligned} \text{Indicated thermal efficiency} &= \frac{\text{Work done per hour in kJ}}{\text{Heat supplied per hour in kJ}} \\ &= \frac{63,360}{2,47,380} = 0.2561 \text{ or } 25.61\% \end{aligned}$$

(ii) Using eqn. (5.8), Ideal thermal efficiency (air-standard efficiency)

$$= 1 - \frac{1}{(r)^\gamma - 1} = 1 - \frac{1}{(6.5)^{0.4}} = 1 - 0.473 = 0.527 \text{ or } 52.7\%$$

(iii) Using eqn. (6.4), Relative efficiency,

$$\eta_r = \frac{\text{Indicated thermal efficiency}}{\text{Ideal thermal efficiency}} = \frac{25.61}{52.7} = 0.486 \text{ or } 48.6\%$$

Problem – 3 : A trial carried on a four-stroke, single-cylinder gas engine, governed by hit and miss method of governing, gave the following results :

Cylinder diameter, 30 cm; Piston stroke, 50 cm; Clearance volume, $6,750 \text{ cm}^3$; Explosions per minute, 100; Indicated mean effective pressure, 780 kPa; Net load on the brake, 1,900 newtons; Brake diameter, 1.5 m; Rope diameter, 2.5 cm; Speed, 240 r.p.m.; Gas used, 30 m^3 per hour; Calorific value of gas, $20,500 \text{ kJ/m}^3$.

Determine : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the indicated thermal efficiency, (e) the compression ratio, (f) the air-standard efficiency, (g) the relative efficiency, and (h) the brake mean effective pressure. Take $\gamma = 1.4$ for air

$$(a) \text{ Effective radius of brake, } R = \frac{D + d}{2} = \frac{150 + 2.5}{2 \times 100} = 0.7625 \text{ metre}$$

Brake power = $(W - S) \times 2\pi R \times N$ watts

$$\begin{aligned} &= 1,900 \times 2 \times 3.14 \times 0.7625 \times \frac{240}{60} \\ &= 36,416 \text{ watts} = 36.416 \text{ kW.} \end{aligned}$$

(b) Indicated power = $p_m \times a \times l \times n$ watts

$$\begin{aligned} &= (780 \times 10^3) \times \frac{\pi}{4} \times \left(\frac{30}{100}\right)^2 \times \frac{50}{100} \times \frac{100}{60} \\ &= 45,922 \text{ watts} = 45.922 \text{ kW.} \end{aligned}$$

(c) Mechanical efficiency = $\frac{\text{Brake power}}{\text{Indicated power}} = \frac{36.416}{45.922} = 0.7929$ or 79.29%

(d) Indicated thermal efficiency = $\frac{\text{Heat equivalent of Indicated power in kJ per sec.}}{\frac{V_g}{3,600} \times \text{C.V. in kJ/sec.}}$

$$= \frac{45.922 \times 3,600}{30 \times 20,500} = 0.2688 \text{ or } 26.88\%$$

(e) Compression ratio, $r = \frac{V_s + V_c}{V_c}$

$$V_s = \frac{\pi}{4} \times d^2 \times l = \frac{\pi}{4} \times (30)^2 \times 50 = 35,350 \text{ cm}^3; v_c = 6,750 \text{ cm}^3 \text{ (given).}$$

$$\therefore r = \frac{v_s + v_c}{v_c} = \frac{35,350 + 6,750}{6,750} = 6.237$$

$$(f) \text{ Air standard efficiency} = 1 - \frac{1}{(r)^\gamma - 1} = 1 - \frac{1}{(6.237)^{0.4}} = 1 - \frac{1}{2.08} = 0.52 \text{ or } 52\%$$

$$(g) \text{ Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency}} = \frac{0.2688}{0.52} = 0.517 \text{ or } 51.7\%$$

(h) Brake mean effective pressure (*b.m.e.p.*)

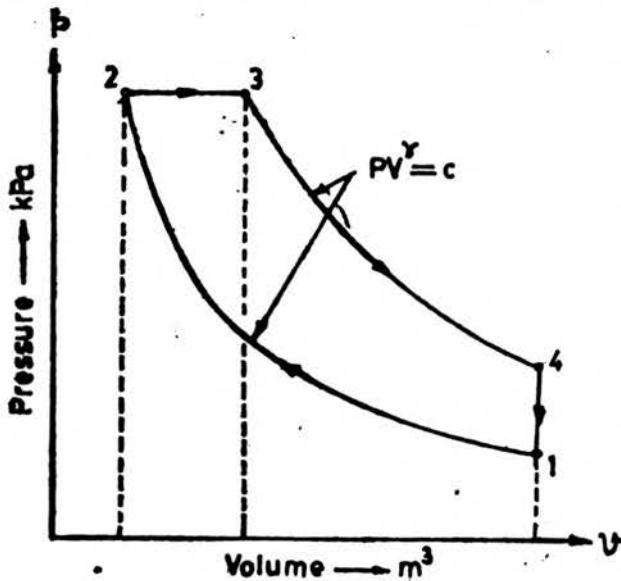


Fig. 6-23.

$$\therefore \text{Cut-off ratio, } \rho = \frac{v_3}{v_2} = \frac{2.04v_2}{v_2} = 2.04$$

$$\text{and compression ratio, } r = \frac{v_1}{v_2} = 14 \text{ (given).}$$

Using eqn. (5.12), Air-standard efficiency

$$= 1 - \frac{1}{(r)^\gamma - 1} \times \left[\frac{(\rho)^\gamma - 1}{\gamma(\rho - 1)} \right]$$

$$\therefore \text{A.S.E.} = 1 - \frac{1}{(14)^{1.4} - 1} \times \frac{(2.04)^{1.4} - 1}{1.4(2.04 - 1)} = 1 - 0.408 = 0.592 \text{ or } 59.2\%$$

$$\begin{aligned} \text{Indicated thermal efficiency} &= \text{Air-standard efficiency} \times \text{Relative efficiency} \\ &= 0.592 \times 0.52 = 0.307 \text{ or } 30.7\% \end{aligned}$$

$$\text{Indicated thermal efficiency} = \frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\frac{M_f}{3,600} \times \text{C.V. in kJ/sec.}}$$

$$\text{i.e., } 0.307 = \frac{1}{\frac{M_f}{3,600} \times \text{C.V.}} \text{ (Taking indicated power as 1 Kw)}$$

$$\therefore M_f = \frac{1 \times 3,600}{0.307 \times 44,000} = 0.2665 \text{ kg per kW-hr.}$$

= Indicated mean effective \times Mechanical efficiency

$$= 780 \times 0.7929 = 618.45 \text{ kPa}$$

Problem - 4 : A Diesel engine has a compression ratio of 14 : 1 and the fuel is cut-off at 0.08 of the stroke. If the relative efficiency is 0.52, estimate the consumption of fuel in litre of calorific value 44,000 kJ/kg which would be required per kW-hour based on indicated power. Take the specific gravity of fuel as 0.82 and $\gamma = 1.4$ for air.

Referring to fig. 6-23,

$$v_3 - v_2 = 0.08 (v_1 - v_2)$$

$$= 0.08 (14 v_2 - v_2) = 1.04 v_2$$

$$\therefore v_3 = 2.04 v_2$$

Fuel consumption in litre per kW-hr based on indicated power

$$= \frac{0.2665}{0.82} = 0.325 \text{ litre/kW-hr.}$$

Problem – 5 : A single-acting, four – stroke Diesel engine develops 37 kW at 210 r.p.m. The mean effective pressure is 740 kPa, compression ratio is 15, fuel is cut-off at 5% of stroke, calorific value of fuel is 43,000 kJ/kg, relative efficiency is 55%. Calculate: (a) the cylinder diameter, if stroke to bore ratio is 1.5, (b) the indicated thermal efficiency, and (c) the fuel consumption in litres/hr. Take $\gamma = 1.4$ for air and specific gravity of fuel as 0.82.

(a) Indicated power = $p_m \times a \times l \times N$ kW (where p_m is in kPa)

$$\text{i.e. } 37 = 740 \times 0.7854 \times \left(\frac{d}{100}\right)^2 \times \frac{1.5d}{100} \times \frac{210}{60 \times 2}$$

$$\therefore d^3 = 24,253.8$$

$$\therefore \text{Cylinder diameter, } d = \sqrt[3]{24,253.8} = 28.95 \text{ cm}$$

(b) Referring to fig. 6-23, compression ratio, $r = \frac{v_1}{v_2} = 15 \therefore v_1 = 15v_2$

Now, stroke volume, $v_s = v_1 - v_2 = 15 v_2 - v_2 = 14 v_2$

$$v_3 = 5\% \text{ of stroke volume} + \text{clearance volume} = \left(\frac{5}{100} \times 14 v_2\right) + v_2 = 1.7 v_2$$

$$\therefore \text{Cut-off ratio, } \rho = \frac{v_3}{v_2} = \frac{1.7 v_2}{v_2} = 1.7$$

$$\text{Using eqn. (5.12), Air-standard efficiency} = 1 - \frac{1}{(r)^\gamma - 1} \times \left[\frac{(\rho)^\gamma - 1}{\gamma(\rho - 1)} \right]$$

$$= 1 - \frac{1}{(15)^{0.4}} \times \frac{(1.7)^{1.4} - 1}{1.4(1.7 - 1)}$$

$$= 1 - 0.3895 = 0.6105 \text{ or } 61.05\%$$

Using eqn. (6.4), Relative efficiency, $\eta_r = \frac{\text{Indicated thermal efficiency}}{\text{Air-standard efficiency}}$

$$\text{i.e., } 0.55 = \frac{\text{indicated thermal efficiency}}{0.6105}$$

$$\therefore \text{Indicated thermal efficiency} = 0.55 \times 0.6105 = 0.3358 \text{ or } 33.58\%$$

(c) Indicated thermal efficiency = $\frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\frac{m_f}{3,600} \times \text{C.V. in kJ/sec.}}$

$$\text{i.e., } 0.3358 = \frac{37 \times 3,600}{m_f \times 43,000}$$

$$\therefore m_f = \frac{37 \times 3,600}{0.3358 \times 43,000} = 9.225 \text{ kg/hr.}$$

$$\therefore \text{Fuel consumption} = \frac{9.225}{0.82} = 11.25 \text{ litres/hour}$$

Problem – 6 : A Diesel engine has a relative efficiency of 0.55 on the brake. If the compression ratio is 13.8, the expansion ratio is 7.4 and the calorific value of fuel oil is

44,000 kJ/kg, calculate the fuel oil consumption in litre per kW-hour based on brake power. Take $\gamma = 1.4$ for air and specific gravity of fuel oil as 0.81.

$$\text{Compression ratio, } r = 13.8; \text{ Cut-off ratio, } \rho = \frac{\text{Compression ratio}}{\text{Expansion ratio}} = \frac{13.8}{7.4} = 1.864$$

$$\begin{aligned} \text{Air-standard efficiency} &= 1 - \frac{1}{(r)^{\gamma-1}} \times \left[\frac{(\rho)^{\gamma} - 1}{\gamma(\rho - 1)} \right] \\ &= 1 - \frac{1}{(13.8)^{1.4-1}} \times \frac{(1.864)^{1.4} - 1}{1.4(1.864 - 1)} \\ &= 1 - \frac{1}{2.86} \times \frac{(2.39 - 1)}{1.4 \times 0.864} = 0.598 \text{ or } 59.8\% \end{aligned}$$

Using eqn. (6.5), Relative efficiency on brake = $\frac{\text{Brake thermal efficiency}}{\text{Air-standard efficiency}}$

$$\text{i.e., } 0.55 = \frac{\text{Brake thermal efficiency}}{0.598}$$

$$\therefore \text{ Brake thermal efficiency, } \eta_b = 0.55 \times 0.598 = 0.33 \text{ or } 33\%$$

$$\text{Brake thermal efficiency, } \eta_b = \frac{\text{Heat equivalent of brake power in kJ/sec.}}{\frac{M_f}{3,600} \times \text{C.V. in kJ/sec.}}$$

$$\text{i.e., } 0.33 = \frac{1}{\frac{M_f}{3,600} \times 44,000} \quad (\text{Taking brake power as } 1 \text{ kW})$$

$$\therefore M_f = \frac{1 \times 3,600}{0.33 \times 44,000} = 0.2479 \text{ kg/kW-hour based on brake power}$$

$$\therefore \text{ Fuel oil consumption} = \frac{0.2479}{0.81} = 0.3061 \text{ litre/kW-hour based on brake power}$$

Problem - 7 : A four-cylinder, four-stroke petrol engine is to be designed to develop indicated power of 40 kW at 50 r.p.s. The bore and stroke are to be equal, the compression ratio is to be 5 and the law of compression and expansion may be taken as $PV^{1.28} = C$. Assuming the suction pressure and temperature to be 100 kPa and 100°C respectively, and that on ignition the cylinder pressure rises instantaneously to 3.5 times the compression pressure, calculate the diameter of cylinder.

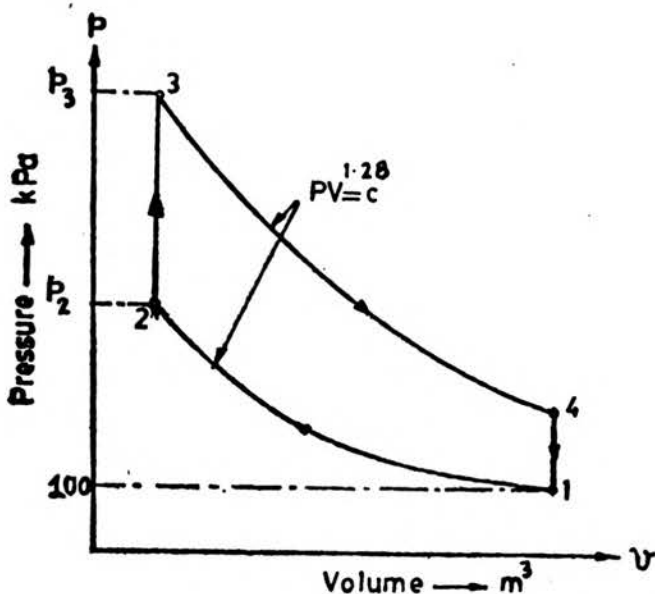


Fig. 6-24.

Referring to fig. 6-24,
 $p_1 = 100 \text{ kPa}$,
 $p_3 = 3.5 p_2$, and $r = 5$.
 Let clearance volume = v_2 ,
 then $v_1 = v_4 = 5 v_2$ and $v_2 = v_3$.

Considering polytropic compression 1-2,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2} \right)^n = (r)^n = (5)^{1.28} = 7.847$$

then $v_1 = v_4 = 5 v_2$ and $v_2 = v_3$.

Considering polytropic expansion 3-4,

$$\frac{p_3}{p_4} = \left(\frac{v_3}{v_4} \right)^n = (r)^n = (5)^{1.28} = 7.847$$

$$\begin{aligned}\therefore p_2 &= p_1 \times 7.847 \\ &= 100 \times 7.847 = 784.7 \text{ kPa.}\end{aligned}$$

$$\text{Now, } \frac{p_3}{p_2} = 3.5 \text{ (given) i.e., } p_3 = p_2 \times 3.5$$

$$\therefore p_3 = 784.7 \times 3.5 = 2,746.45 \text{ kPa}$$

$$\text{Considering polytropic expansion } 3 - 4, \frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right) = (r)^n = (5)^{1.28} = 7.847$$

$$\therefore p_4 = \frac{p_3}{7.847} = \frac{2,746.45}{7.847} = 350 \text{ kPa}$$

Work done per cycle = area 1-2-3-4

= area under (3 - 4) minus area under (2 - 1)

$$= \frac{p_3 v_3 - p_4 v_4}{n - 1} - \frac{p_2 v_2 - p_1 v_1}{n - 1}$$

$$= v_2 \left[\frac{2,746.45 \times 1 - 350 \times 5}{1.28 - 1} - \frac{784.7 \times 1 - 100 \times 5}{1.28 - 1} \right]$$

$$= 2,542 v_2 \text{ kJ}$$

$$\begin{aligned}\text{M.E.P.} &= \frac{\text{Area of the diagram}}{\text{Length of the diagram}} \\ &= \frac{\text{Work done per cycle in kJ}}{v_1 - v_2 \text{ in m}^3} \\ &= \frac{2,542 v_2}{4 v_2} = 635.5 \text{ kPa}\end{aligned}$$

$$\text{Indicated power per cylinder} = \frac{40}{4} = 10 \text{ kW}$$

But indicated power = $p_m \times a \times l \times n$ watts

$$\text{i.e., } (10 \times 10^3) = (635.5 \times 10^3) \times \frac{\pi \left(\frac{d}{100}\right)^2}{4} \times \frac{d}{100} \times \frac{50}{2}$$

$$\therefore d^3 = \frac{10 \times 10^3 \times 4 (10)^4 \times 10^2 \times 2}{10^3 \times 635.5 \times 3.14 \times 50} = 802.1$$

$$\therefore \text{Diameter of cylinder, } d = \sqrt[3]{802.1} = 9.292 \text{ cm.}$$

Problem - 8 : A Diesel engine working on four-stroke cycle has a bore of 30 cm and stroke 40 cm and runs at 5 r.p.s. If the compression ratio is 14 and cut-off takes place at 5% of the stroke, estimate the mean effective pressure of the cycle and indicated power of the engine.

Assume compression index as 1.4 and expansion index as 1.3. The pressure at the beginning of compression is 100 kPa.

Referring to fig. 6-25, let clearance volume = v_2 , then $v_1 = 14 v_2 = v_4$

$$\text{Again, } v_3 = v_2 + \frac{5}{100} (v_1 - v_2)$$

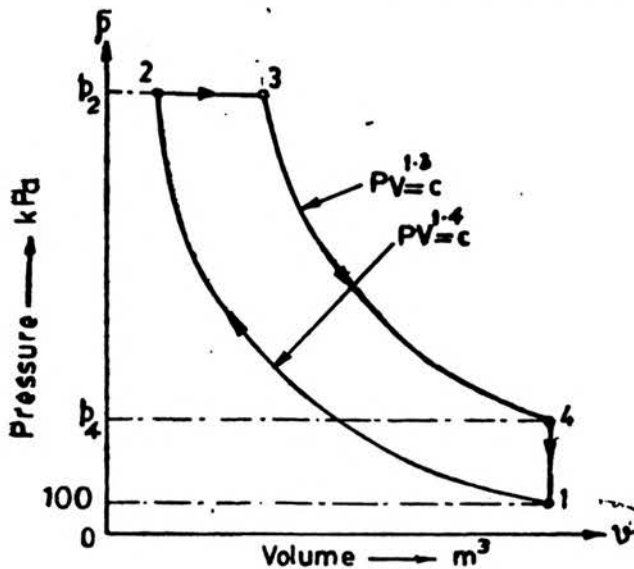


Fig. 6-25.

$$= v_2 + \frac{5}{100} \times 13v_2 = 1.65v_2$$

Considering polytropic compression 1-2,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n = (r)^n$$

$$\therefore p_2 = p_1 \times (r)^n = 100 \times (14)^{1.4} = 4,023 \text{ kPa}$$

$$\therefore p_3 = p_2 = 4,023 \text{ kPa}$$

Considering polytropic expansion 3-4,

$$\frac{p_3}{p_4} = \left(\frac{v_4}{v_3}\right)^n = \left(\frac{14v_2}{1.65v_2}\right)^{1.3} = (8.47)^{1.3} = 16.12$$

$$\therefore p_4 = \frac{p_3}{16.12} = \frac{4,023}{16.12} = 249.6 \text{ kPa}$$

Work done per cycle = area 1-2-3-4

= area under (2-3) + area under (3-4) minus area under (2-1)

$$= p_2(v_3 - v_2) + \frac{p_3 v_3 - p_4 v_4}{n - 1} - \frac{p_2 v_2 - p_1 v_1}{n - 1}$$

$$= v_2 \left[4,023(1.65 - 1) + \frac{4,023 \times 1.65 - 249.6 \times 14}{1.3 - 1} - \frac{4,023 \times 1 - 100 \times 14}{1.4 - 1} \right]$$

$$= v_2 [2,615 + 10,478.7 - 6,557.5] = 6,536.2 v_2 \text{ kJ}$$

$$\text{Using eqn. (5-18) M.E.P.} = \frac{\text{Area of the diagram}}{\text{Length of the diagram}} = \frac{\text{Work per cycle in kJ}}{(v_1 - v_2)} \text{ kPa}$$

(where $v_1 - v_2$ is displacement volume in m^3)

$$= \frac{6,536.2 v_2}{13v_2} = 502.78 \text{ kPa}$$

$$\text{Indicated power} = p_m \times a \times l \times n \text{ kW}$$

$$= 502.78 \times \frac{\pi}{4} (0.3)^2 \times 0.4 \times \frac{5}{2} = 35.54 \text{ kW}$$

Problem - 9 : The following data relate to a twin-cylinder, four-stroke, Diesel engine working on the constant pressure cycle :

Ratio of compression	... 15
Ratio of expansion	... 8
Speed	... 5 r.p.s.
Efficiency ratio or relative efficiency	... 0.6
Consumption of fuel oil per minute	... 0.43 litre
Specific gravity of fuel oil	... 0.81

Calorific value of fuel oil	... 42,000 kJ/kg
Diameter of cylinder	... 30 cm
Piston stroke	... 45 cm
γ for air	... 1.4

Find : (a) the indicated power of the engine, and (b) the indicated mean effective pressure.

$$(a) \text{ Cut-off ratio, } \rho = \frac{\text{Compression ratio}}{\text{Expansion ratio}} = \frac{15}{8} = 1.875$$

$$\begin{aligned} \text{Air-standard efficiency, } &= 1 - \frac{1}{(r)^\gamma - 1} \times \left[\frac{(\rho)^\gamma - 1}{\gamma(\rho - 1)} \right] \\ &= 1 - \frac{1}{(15)^{0.4}} \times \frac{(1.875)^{1.4} - 1}{1.4(1.875 - 1)} \\ &= 1 - \frac{1}{2.96} \times \frac{2.41 - 1}{1.4 \times 0.875} = 0.611 \text{ or } 61.1\% \end{aligned}$$

$$\begin{aligned} \text{Indicated thermal efficiency} &= \text{Air-standard efficiency} \times \text{Relative efficiency} \\ &= 0.611 \times 0.6 = 0.367 \text{ or } 36.7\% \end{aligned}$$

$$\text{Indicated thermal efficiency} = \frac{\text{Indicated power} \times 3,600}{m_f \times C.V.}$$

$$\text{i.e., } 0.367 = \frac{\text{Indicated power} \times 3,600}{(0.43 \times 0.81) \times 60 \times 42,000}$$

$$\therefore \text{ Indicated power of engine} = \frac{0.367 \times (0.43 \times 0.81) \times 60 \times 42,000}{3,600} = 89.23 \text{ kW}$$

$$(b) \text{ Indicated power per cylinder} = \frac{89.23}{2} = 44.615 \text{ kW}$$

$$\text{Indicated power (per cylinder)} = p_m \times a \times l \times n$$

$$\text{i.e., } 44.615 = p_m \times \frac{\pi}{4} \left(\frac{30}{100} \right)^2 \times \frac{45}{100} \times \frac{5}{2}$$

$$\therefore \text{ Indicated mean effective pressure, } p_m = \frac{44.615 \times 2}{0.7854 \times (0.3)^2 \times 0.45 \times 5} = 561.22 \text{ kPa}$$

Problem – 10 : From the following data determine the cylinder diameter and stroke of piston of a four-cylinder, four-stroke Otto cycle engine :

Brake power	... 15 kW
Mechanical efficiency	... 80%
Suction pressure	... 94 kPa
Maximum explosion pressure	... 2,400 kPa
Ratio of compression	... 5
Index of compression curve	... 1.35
Index of expansion curve	... 1.3
Speed	... 1,000 r.p.m.
Stroke : Bore	... 1.4 : 1

Referring to fig. 6-26,

$$v_1 = 5 v_2 ; v_2 = v_3 ; v_4 = 5 v_2 ; p_1 = 94 \text{ kPa} ; p_3 = 2,400 \text{ kPa}.$$

$$\text{Considering polytropic compression (1-2), } \frac{p_2}{p_1} = \left(\frac{v_1}{v_2} \right)^n = (r)^n \quad [\text{where } n = 1.35]$$

$$\therefore p_2 = p_1 \times (r)^n = 94 \times (5)^{1.35} = 826 \text{ kPa}$$

$$\text{Considering polytropic expansion (3-4), } \frac{p_3}{p_4} = \left(\frac{v_4}{v_3} \right)^n = (r)^n \quad [\text{where } n = 1.3]$$

$$\therefore p_4 = \frac{p_3}{(r)^n} = \frac{2,400}{(5)^{1.3}} = 296 \text{ kPa}$$

Work done per cycle = area 1-2-3-4 = area under (3-4) minus area under (2-1)

$$\begin{aligned} &= \left[\frac{p_3 v_3 - p_4 v_4}{n - 1} - \frac{p_2 v_2 - p_1 v_1}{n - 1} \right] \\ &= v_2 \left[\frac{2,400 \times 1 - 296 \times 5}{1.3 - 1} - \frac{826 \times 1 - 94 \times 5}{1.35 - 1} \right] \\ &= v_2 [3,066.7 - 1,017.1] = v_2 2,049.6 \text{ kJ} \end{aligned}$$

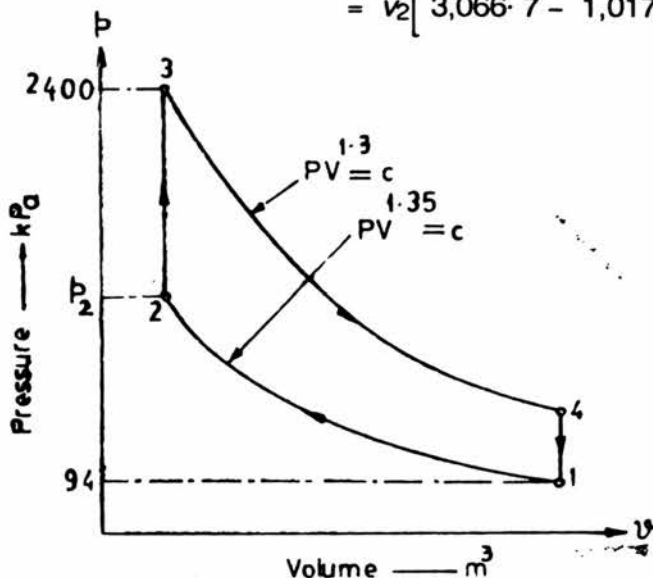


Fig. 6-26.

Using eqn. (5-18),

$$\begin{aligned} \text{M.E.P.} &= \frac{\text{Area of the diagram}}{\text{Length of the diagram}} \\ &= \frac{\text{Work done per cycle in kJ}}{(v_1 - v_2) \text{ m}^3} \text{ kPa} \\ &= \frac{2,049.6 v_2}{4v_2} = 512.4 \text{ kPa} \end{aligned}$$

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}}$$

\therefore Indicated power of engine

$$= \frac{15}{0.8} = 18.75 \text{ kW}$$

$$\text{Indicated power per cylinder} = \frac{18.75}{4} \text{ kW}$$

$$\text{Indicated power/cylinder} = p_m \times a \times l \times n$$

$$\text{i.e. } \frac{18.75}{4} = 512.4 \times 0.7854 \times \left(\frac{d}{100} \right)^2 \times \frac{1.4 d}{100} \times \frac{1,000}{60 \times 2}$$

$$\therefore d^3 = \frac{18.75 \times 60 \times 2 \times 10^4 \times 100}{4 \times 512.4 \times 0.7854 \times 1.4 \times 1,000} = 998.4$$

$$\therefore \text{Diameter of cylinder, } d = \sqrt[3]{998.4} = 10 \text{ cm}$$

$$\therefore \text{Piston stroke, } l = 1.4 d = 10 \times 1.4 = 14 \text{ cm}$$

Problem - 11 : A single-cylinder, four-stroke oil engine working on the dual-combustion cycle has a cylinder diameter of 20 cm and a stroke of 40 cm. The compression ratio is 13.5 and the pressure ratio is 1.42. From the indicator diagram it was found that cut-off

occured at 5.1 per cent of the stroke. If the relative efficiency is 0.6 and the calorific value of fuel oil is 42,000 kJ/kg and the fuel oil consumption is 5 kg/hour, calculate the indicated power of the engine, if it runs at 7 r.p.s. Also calculate the indicated mean effective pressure. Take $\gamma = 1.4$ for air.

$$\text{Stroke volume, } v_s = \frac{\pi}{4} \times d^2 \times l = \frac{\pi}{4} \times (20)^2 \times 40 = 12,560 \text{ cm}^3$$

$$\text{Compression ratio, } r = \frac{v_c + v_s}{v_c} \text{ (where } v_c \text{ is clearance volume)}$$

$$\text{i.e. } 13.5 = \frac{v_c + 12,560}{v_c} \quad \therefore v_c = 1,005 \text{ cm}^3$$

$$\text{Cut-off ratio, } \rho = \frac{v_c + 0.051 v_s}{v_c} = \frac{1,005 + 0.051 \times 12,560}{1,005} = 1.64$$

$$\text{Pressure ratio, } \beta = 1.42 \text{ (given) ; } r = 13.5$$

Using eqn. (5.14), Air-standard efficiency (A.S.E.)

$$\begin{aligned} &= 1 - \frac{1}{(r)^\gamma - 1} \left[\frac{\beta (\rho)^\gamma - 1}{(\beta - 1) + \beta \gamma (\rho - 1)} \right] \\ &= 1 - \frac{1}{(13.5)^{0.4}} \left[\frac{1.42 (1.64)^{1.4} - 1}{(1.42 - 1) + 1.42 \times 1.4 (1.64 - 1)} \right] \\ &= 1 - \frac{1}{2.832} \left[\frac{2.84 - 1}{0.42 + 1.272} \right] \\ &= 1 - 0.3842 = 0.6158 \text{ or } 61.58\% \end{aligned}$$

Indicated thermal efficiency = Air standard efficiency \times Relative efficiency

$$= 0.6158 \times 0.6 = 0.3695 \text{ or } 36.95\%$$

$$\text{Indicated thermal efficiency} = \frac{\text{Indicated power} \times 3,600}{m_f \times \text{C.V.}}$$

$$\text{i.e., } 0.3695 = \frac{\text{Indicated power} \times 3,600}{5 \times 42,000}$$

$$\therefore \text{Indicated power} = \frac{0.3695 \times 5 \times 42,000}{3,600} = 21.56 \text{ kW}$$

$$\text{Now, indicated power} = p_m \times a \times l \times n \text{ kW}$$

$$\text{i.e., } 21.56 = p_m \times 0.7854 \times (0.2)^2 \times 0.4 \times \frac{7}{2}$$

$$\therefore p_m = \frac{21.56 \times 2}{0.7854 \times (0.2)^2 \times 0.4 \times 7} = 490.2 \text{ kPa.}$$

Tutorial - 6

1. Delete the phrase which is not applicable in the following statements :

- (i) A petrol engine works on Otto cycle/Diesel cycle.
- (ii) A two-stroke cycle I.C. engine completes its cycle in one revolution/two revolutions of the crank shaft.
- (iii) A four-stroke cycle I.C. engine completes its cycle in one revolution/two revolutions of the crank shaft.
- (iv) A carburettor is used in petrol/Diesel engine.
- (v) During idling the S.I. (spark ignition) engine needs weak/rich mixture.

- (vi) During starting the S.I. engine needs weak/rich mixture.
- (vii) In a Diesel/petrol engine, the charge during suction stroke consists of air only.
- (viii) Diesel engine employs quantity/quality method of governing.
- (ix) Petrol engine employs quantity/quality method of governing.
- (x) Modern Diesel engines employ air injection/solid injection method of fuel injection.
- (xi) Air cooling is more/less effective than water cooling.
- (xii) In a Diesel engine, the fuel is ignited by spark/heat of compressed air.
- (xiii) The volumetric efficiency of an I.C. engine decreases/increases with the increase in engine speed.

[Delete : (i) Diesel cycle, (ii) two revolutions, (iii) one revolution, (iv) Diesel, (v) weak, (vi) weak, (vii) petrol, (viii) quantity, (ix) quality, (x) air injection, (xi) more, (xii) spark, (xiii) increases]

2. Fill in the blanks to complete the following statements :

- (i) The efficiency of a two-stroke cycle engine is _____ than that of a four-stroke cycle I.C. engine.
- (ii) The compression ratio of a Diesel engine is _____ than that of a petrol engine.
- (iii) Diesel engine are also known as _____ engines.
- (iv) Petrol engines are also known as _____ engines.
- (v) The air-fuel ratio for a chemically correct mixture is about _____.
- (vi) The process of sweeping out exhaust gases from the combustion chamber of the cylinder is known as _____.
- (vii) The process of adding certain chemical to the fuel for suppressing detonation is known as _____ and the chemicals added are called _____.
- (viii) For engines of motor cycles, scooters and mopeds, the cooling system employed is _____.
- (ix) Hit and miss method of governing is used for small size _____ engines.
- (x) In petrol engine if the ignition starts, due to any reason other than spark, when the piston is still doing its compression stroke, the ignition is known as _____.
- (xi) Octane number refers to _____ property of Otto engine fuels.
- (xii) Cetane number refers to _____ of Diesel fuels and is a measure of the Diesel knocking tendency.
- (xiii) Diesel knock is caused by too long a _____ between the initial injection of fuel and the commencement of burning of the fuel.
- (xiv) As compared to petrol engines, Diesel engines are _____ suitable for supercharging.

[(i) less, (ii) more, (iii) compression - ignition, (iv) spark ignition, (v) 15, (vi) scavenging, (vii) doping, dopes, (viii) air cooling, (ix) gas and oil, (x) pre-ignition, (xi) anti-knocking, (xii) ignition quality, (xiii) delay period, (xiv) more]

3. Indicate the correct answer by selecting the proper phrase to complete the following statements :

- (i) The firing order in a four-cylinder petrol engine is
 - (a) 1-2-3-4, (b) 1-3-4-2, (c) 1-2-4-3, (d) 1-4-3-2.
- (ii) In a Diesel engine the fuel is injected
 - (a) during the suction stroke,
 - (b) at the start of compression stroke,
 - (c) at the end of compression stroke,
 - (d) some degrees before the end of compression stroke,
 - (e) at the end of expansion stroke.
- (iii) In a petrol engine the pressure at the end of compression is of the order of
 - (a) 35 to 45 bar (b) 25 to 35 bar, (c) 14 to 25 bar, (d) 7 to 14 bar.
- (iv) The firing order of a six-cylinder I.C. engine is
 - (a) 1-5-3-6-2-4, (b) 6-5-4-1-2-3, (c) 1-2-3-4-5-6, (d) 1-5-2-4-3-6.
- (v) Method of governing employed in a compression-ignition engine is
 - (a) hit and miss, (b) quality, (c) quantity, (d) combination of quantity and quality.
- (vi) In S.I. (spark ignition) engines the secondary winding of the ignition coil generates a voltage of
 - (a) 1,000 to 5,000 volts, (b) 5,000 to 9,000 volts, (c) 9,000 to 12,000 volts, (d) 12,000 to 20,000 volts.

- (vii) The maximum temperature in the I.C. engine cylinder is of the order of
 (a) 2,000 to 2,500°C, (b) 1,500 to 2,000°C, (c) 1,000 to 1,500°C, (d) 500 to 1,000°C.
- (viii) The system of lubrication employed in a crankcase scavenged two-stroke cycle petrol engine is :
 (a) splash lubrication, (b) petrol or mist lubrication, (c) pressure lubrication,
 (d) combined splash and pressure lubrication.
- (ix) Pre-ignition in an Otto cycle engine
 (a) increases efficiency, (b) increases power, (c) causes reduced efficiency and less power,
 (d) increases efficiency and power, (e) decreases efficiency and increases power.
- (x) A reduction in the ignition delay period
 (a) reduces diesel knock, (b) increases diesel knock, (c) results in incomplete combustion,
 (d) causes reduced efficiency and loss of power.
- (xi) The speed of the cam shaft of a four-stroke cycle I.C. engine running at 4,000 r.p.m. is
 (a) 1,000 r.p.m., (b) 2,000 r.p.m., (c) 3,000 r.p.m., (d) 4,000 r.p.m.
- (xii) The Morse test gives
 (a) brake thermal efficiency of a multi-cylinder I.C. engine,
 (b) mechanical efficiency of a multi-cylinder I.C. engine.
 (c) B.S.F.C. (brake specific fuel consumption) of a multi-cylinder I.C. engine,
 (d) air-fuel ratio of a multi-cylinder I.C. engine.
- (xiii) Compression ratio for a petrol engine, without use of dope is usually
 (a) 5, (b) 10, (c) 15, (d) 20.
- (xiv) In a petrol engine, maximum efficiency occurs when air-fuel ratio is
 (a) chemically correct,
 (b) lower than that required for complete combustion,
 (c) higher than that required for complete combustion,
 (d) none of the above.
- (xv) A petrol engine, gives maximum power when air-fuel ratio is
 (a) chemically correct,
 (b) lower than that required for complete combustion.
 (c) higher than that required for complete combustion.
 (d) none of the above.

[(i) b, (ii) d, (iii) d, (iv) a, (v) b, (vi) d, (vii) a, (viii) b,
 (ix) c, (x) a, (xi) b, (xii) b, (xiii) a, (xiv) c, (xv) b].

4. Explain briefly the cycle of operation of (i) a four-stroke cycle petrol engine, and (ii) a four-stroke cycle Diesel engine. Draw indicator and valve timing diagrams for each case.
5. Sketch a typical valve timing diagram of a four-stroke cycle high-speed Diesel engine.
6. Describe, with the help of a neat sketch, two-stroke cycle Diesel engine. Sketch the indicator diagram of such an engine, and discuss the advantages and disadvantages of the two-stroke cycle, compared with the four-stroke cycle operations.
7. Describe, with a neat sketch, the working of a two-stroke cycle petrol engine, giving probable indicator and valve timing diagrams.
8. Describe briefly the principal method adopted for charging and exhausting cylinders of two-stroke cycle engines.
9. What are the most common methods of governing the speed of small size gas engines ? Illustrate your answer by means of sketches.
10. (a) Describe briefly, with the help of sketches, the different methods of governing employed in internal combustion engines.
 (b) Describe briefly, with a neat sketch, any one type of quality governing as applied to oil engines.
 (c) Differentiate clearly between the quality and the quantity governing as applied to I.C. engines.
11. (a) Describe with sketches any one type of fuel pump for a high speed compression-ignition engine, explaining carefully how the amount of oil is adjusted according to the load.
 (b) Sketch and describe a fuel valve working on the solid injection system.
12. Compare from every aspect, solid injection with air injection as a means of supplying the fuel to compression-ignition engines.

- If one method is superior to the other, why is it not in general use ?
13. Enumerate the essential function of a carburettor. Explain briefly, giving suitable sketches, the working of any one type of carburettor. Show clearly the devices incorporated in it for
 - (i) smooth running when idling,
 - (ii) ensuring correct mixture strength at all loads and speeds of the engine, and
 - (iii) supplying large quantity of rich mixture during the accelerating period.
 14. Describe in detail the ignition system used in multi-cylinder petrol engines.
 15. (a) Describe briefly the various methods of ignition used in internal combustion engines.
 (b) Give a wiring diagram for battery ignition system. Explain briefly the function of a condenser in the system.
 16. Write short notes on the following giving neat sketches wherever necessary :
 - (i) Air cooling versus water cooling for I.C. engines.
 - (ii) Scavenging of two-stroke cycle I.C. engines.
 - (iii) Factors affecting volumetric efficiency of I.C. engines.
 - (iv) Methods of ignition used in a multi-cylinder petrol engines.
 - (v) Quantity governing applied to gas engine.
 - (vi) Supercharging of I.C. engines.
 - (vii) Methods of fuel injection employed in Diesel engines and their relative merits and demerits.
 17. Describe briefly with the aid of sketches the following :
 - (i) Any one method of governing employed in I.C. engines.
 - (ii) Any one method of fuel injection for compression-ignition engines.
 - (iii) Cooling system of motor car engines.
 - (iv) Methods of scavenging employed in two-stroke engines.
 - (v) Fuel injection pump of Diesel engine.
 - (vi) Any one system of ignition used in Otto cycle engines.
 18. Write short notes on the following : (i) Carburettor; (ii) Methods of cooling I.C. engines; (iii) Methods of fuel injection in a Diesel engine; (iv) Methods of governing used in I.C. engines; (v) Methods of ignition used in I.C. engines.
 19. Sketch and explain any fuel mixing or fuel injecting device used in I.C. engines.
 20. Compare an engine working on Otto cycle with an engine working on Diesel cycle from the following points of view : (i) Fuel, (ii) Working cycle, (iii) Method of fuel injection, (iv) Method of fuel ignition, (v) Speed, (vi) Method of governing.
 21. What do you understand by the terms "detonation" and "pre-ignition" as applied to internal combustion engines ?
 22. A trial carried out on a four-stroke cycle, single-cylinder gas engine gave the following results :
 Cylinder diameter, 30 cm; piston stroke, 53 cm; clearance volume, $9,200 \text{ cm}^3$, explosions per minute, 110; indicated mean effective pressure, 700 kPa; gas used, 28 m^3 per hour; calorific value of gas, $19,000 \text{ kJ/m}^3$.
 Determine : (a) the compression ratio, (b) the indicated thermal efficiency, (c) the air-standard efficiency (assume $\gamma = 1.4$ for air), and (d) the relative efficiency.
 [(a) $r = 5$; (b) $\eta_i = 31.92\%$; (c) A.S.E. = 47.5% ; (d) $\eta_r = 67.2\%$]
 23. A single-cylinder, four-stroke oil engine working on Otto cycle has bore of 18 cm and stroke of 36 cm. The clearance volume is $1,800 \text{ m}^3$.
 During a test the fuel oil consumption was 4.5 litres per hour; the engine speed 300 r.p.m.; the indicator diagram area 4.25 cm^2 ; length of indicator diagram 6.25 cm; and indicator spring rating 1,000 kPa per cm. If the fuel oil has a calorific value of 43,500 kJ/kg and specific gravity of 0.8, calculate : (i) the indicated thermal efficiency, (ii) the air-standard efficiency, and (iii) the relative efficiency. Take $\gamma = 1.4$ for air.
 [(i) $\eta_i = 35.8\%$; (ii) A.S.E. = 51.46% ; (iii) $\eta_r = 69.57\%$]
 24. A petrol engine working on Otto cycle has clearance volume of 20% of the stroke volume. The engine consumes 8.25 litres of petrol per hour when developing indicated power of 24 kW. The specific gravity of petrol is 0.76 and its calorific value is 44,000 kJ/kg. Determine (i) the indicated thermal efficiency, (ii) the air-standard efficiency, and (iii) the relative efficiency of the engine. Take $\gamma = 1.4$ for air.
 [(i) $\eta_i = 31.32\%$; (ii) A.S.E. = 51.15% ; (iii) $\eta_r = 61.23\%$]
 25. The following particulars refer to a petrol engine working on four-stroke, Otto cycle principle :

Diameter of the cylinder 7.5 cm; stroke 9 cm; clearance volume 81 cm^3 ; indicated power developed 21 kW; specific gravity of petrol 0.76; calorific value of petrol 44,000 kJ/kg. Calculate : (a) the air-standard efficiency, and (b) the petrol consumption in litres/hour, if the relative efficiency of the engine is 65%. Take $\gamma = 1.4$ for air.

[(a) A.S.E. = 50.88%; (b) 6.836 litres/hour]

26. A six cylinder, four-stroke cycle, petrol engine is to be designed to produce brake power of 320 kW at 40 r.p.s. The stroke to bore ratio is to be 1.25 to 1. Assuming a mechanical efficiency of 80% and indicated mean effective pressure of 950 kPa, determine the required cylinder bore and stroke.

If the compression ratio of the engine is to be 6.5, determine the petrol consumption in litres per hour and petrol consumption in litre per kW-hr based on brake power. Take the relative efficiency as 0.55 and calorific value of petrol as 44,000 kJ/kg. Take specific gravity of petrol as 0.76, and $\gamma = 1.4$ for air.

[$d = 15.3 \text{ cm}$; $l = 19.125 \text{ cm}$; 148.5 litres/hour; 0.3711 litre/kW-hr.]

27. A Diesel engine has a relative efficiency of 0.58 on the brake. If the compression ratio is 14 and the expansion ratio is 7 and the calorific value of oil is 44,000 kJ/kg, find : (i) the air-standard efficiency, (ii) the brake thermal efficiency, and (iii) the consumption of oil in litre per kW-hour on brake power basis. Take $\gamma = 1.4$ for air and specific gravity of oil as 0.8.

[(i) 59.3%; (ii) 34.4%; (iii) 0.2973 litre/kW-hr.]

28. A gas engine of 25 cm bore, 45 cm stroke, has a compression ratio of 4.5. At the beginning of compression the charge in the cylinder is at 100 kPa. The law of compression is $pv^{1.35} = \text{constant}$, and the law of expansion is $pv^{1.3} = \text{constant}$. If the pressure is trebled during constant volume explosion, find the mean effective pressure on the piston and the indicated power developed, if the engine makes 85 explosions per minute.

[535.84 kPa; 17.755 kW]

29. A Diesel engine working on the four-stroke cycle has a bore of 25 cm and stroke of 35 cm, runs at 4 r.p.s. If the compression ratio is 14 and the cut-off ratio is 2.2, estimate the indicated mean effective pressure in kPa and the indicated power of the engine. Assume the law of compression to be $pv^{1.32} = \text{constant}$ and the law of expansion is $pv^{1.35} = \text{constant}$. The pressure and temperature at the beginning of compression are 100 kPa and 100°C respectively. Calculate also the temperatures at the salient (key) points of the cycle.

[i.m.e.p. = 604.92 kPa; indicated power = 20.79 kW; 595°C , $1,637^\circ\text{C}$, 727°C]

30. A single-acting, four-stroke cycle Diesel engine develops indicated power of 30 kW at 200 r.p.m. The mean effective pressure is 700 kPa, compression ratio is 14, fuel is cut-off at 6% of the stroke, $\gamma = 1.4$ for air, calorific value of fuel is 43,000 kJ/kg, relative efficiency is 58%. Calculate (i) the cylinder diameter if stroke to bore ratio is 1.25, (ii) the air-standard efficiency, (iii) the indicated thermal efficiency, (iv) the fuel consumption in litres per hour, and (v) the fuel consumption in litre per kW-hour based on indicated power. Take specific gravity of fuel as 0.8.

[(i) 29.7 cm; (ii) 60.45%; (iii) 35.06%; (iv) 8.95 litres/hour; (v) 0.2983 litre/kW-hr.]

31. A four-stroke oil engine working on dual-combustion cycle has a cylinder diameter of 24 cm and a stroke of 35 cm. The clearance volume is $1,615 \text{ cm}^3$, cut-off takes place at 6 per cent of the stroke and the pressure ratio is 1.5. If the relative efficiency is 0.6 and the calorific value of oil is 42,000 kJ/kg and oil consumption is 4.8 kg/hr., calculate the indicated power of the engine. Also calculate the indicated mean effective pressure in kPa if the engine runs at 5 r.p.s. Take $\gamma = 1.4$ for air.

[Indicated power = 19.647 kW; i.m.e.p. = 496.33 kPa]

TESTING OF INTERNAL COMBUSTION ENGINES

7.1 Objectives of Testing

In general, the purposes of testing an internal combustion engine are :

- (i) to obtain information about the engine which cannot be determined by calculations,
- (ii) to confirm data used in design, the validity of which is in doubt, and
- (iii) to satisfy the customer as to the rated power output with the guaranteed fuel consumption.

The majority of tests on internal combustion engines are carried out for commercial purposes in order to check the following :

- (i) rated power (brake power) with the guaranteed fuel consumption (kg/kW-hr.),
- (ii) the quantity of lubricating oil required on brake power basis per kW-hr.,
- (iii) the quantity of cooling water required on brake power basis in kg per kW-hr.,
- (iv) the steadiness of the engine when loaded at different loads, and
- (v) the overload carrying capacity of the engine.

7.2 Thermodynamic Tests

Complete thermodynamic tests are quite different from the commercial tests. They are carried out for the purpose of comparing actual results with the theoretical or ideal performance. For such tests it is necessary to measure losses in addition to the useful part of the energy, and also to draw up a heat balance account. Such trials have been the direct cause of, and incentive to, the improvement in heat engines throughout the period of their development. This interest created a demand for authentic records of engine performance, which could only be satisfied by exhaustive trials carefully observed and calculated. The measurements necessary to determine the mechanical and thermal efficiencies of the engine and to draw up the heat balance account are :

- (i) Indicated power (if possible);
- (ii) Brake power;
- (iii) Morse test for mechanical efficiency in case of multi-cylinder high speed engines;
- (iv) Rate of fuel consumption and its calorific value;
- (v) Rate of flow of cooling water and its rise of temperature, for calculating the heat carried away by jacket cooling water;
- (vi) Heat carried away by the exhaust gases – this is estimated either directly by exhaust gas calorimeter or by measuring air consumption and temperature of exhaust gases, and engine room temperature.

7.2.1 Measurement of indicated power : It is extremely difficult to determine the indicated power, especially when moderate or high engine speeds are used. The strength of the spring to be used in the indicator must be carefully chosen. The ratio of maximum pressure in the engine cylinder to the mean pressure during the cycle in an I.C. engine is much greater than that of any other heat engine. For gas and petrol engine, the

explosion causes the maximum pressure to be reached practically instantaneously. Thus, to prevent vibrations being set up, the spring used must be stiff but at the same time it should give enough height of the indicator diagram. The production of true volume scale is often hindered by the absence or inaccessibility of any suitable point of attachment for the *indicator cord*, so that it may transmit the piston movement, such as is provided by the cross-head of a steam engine. Any miniature crank or cam device attached to the crank shaft must be phased with considerable accuracy, while slackness, inertia and elasticity in the mechanism may give very serious results.

The piston and pencil element used in steam engine practice is useless except at very low speeds, the rate of pressure rise causing violent oscillations which cannot be damped without introducing errors. The replacement of the piston by a diaphragm and the use of high optical or electrical magnification of its deflection, reduce the oscillation problem but fatigue of the diaphragm metal and change of its calibration by heat are both likely to occur. If the diaphragm is separated from the cylinder by means of a cock except during actual recording, these troubles may be reduced, but likely to be replaced by others due to surges of the gas pressure in the connecting passages, and even when these are short the time taken for a change of pressure in the cylinder to reach and deflect the diaphragm, may introduce a phase lag which is serious at high engine speeds.

For rapid determination of the mean effective pressure, a planimeter may be used, being quite accurate enough for all ordinary practical purposes.

The remaining data required for the calculation of the indicated power are the number of explosions or power strokes per minute and the dimensions of the engine cylinder. The number of explosions per minute is *best* given by means of a counter arranged to be actuated from the gas valve, particularly if the engine is governed by the hit and miss method.

Owing to the difficulties of accurate measurement, particularly at high speeds, there is an increasing tendency to disregard indicated power and rely on brake power as a power measurement.

7.2.2 Measurement of brake power : There is very little difficulty in measuring this quantity accurately if ordinary precautions are taken. This may be obtained by the use of either a mechanical, electrical, hydraulic or air brake, etc. The difference between the indicated power and brake power is known as the mechanical or friction loss, and includes the negative loop of the indicator diagram. The following method is adopted to determine the friction power so that the indicated power may be accurately determined.

Motoring test : An approximate value of friction power may be found immediately following a period of running, by measuring the power required to motor the engine (the engine is driven by an electric motor) at the requisite speed and with the ignition switched off. Such a test should be carried out as near as maximum operating temperature possible, the viscosity of the lubricant rising very considerably with a fall of temperature. Unfortunately the thin film of lubricant on the cylinder wall, the shearing of which is the cause of about half the total engine friction, suffers considerable deterioration by heat and oxidation while the engine is running, and on switching off the ignition, this damaged oil on which the piston normally operates, is rapidly washed from the walls and replaced by oil in good condition. The power required to motor the engine thus falls very rapidly within perhaps two minutes, after which it begins to rise slowly owing to the cooling of walls. A reasonable accurate determination of running friction is, therefore, very difficult, if not impossible with normal test equipment.

7.2.3. "Morse" test for mechanical efficiency : For multi-cylinder high speed engine the *Morse test* is available, and is less open to objection than the simple motoring test.

The method of finding indicated power of one cylinder of a multi-cylinder I.C. engine without the use of a high speed indicator is known as the Morse test. The engine is first run under the required condition of load, speed, temperature, etc., and the brake power is measured accurately. Each cylinder is then cut-out in turn; the brake load being rapidly adjusted in each case to bring the engine speed back to the specified value at the given angle of advance and throttle settling.

The fundamental assumptions are that the friction and pumping power of the cut-out cylinder remains the same after cutting out as they were when the cylinder was fully operative (developing power). This would not be a correct assumption if it were not for the fact that it is possible to carry this test in a very short span of time. It should only take a few seconds to cut out one cylinder and adjust the brake load to keep the speed constant. Over this short period the assumption may be considered reasonable. After cutting out one cylinder, the engine should be allowed to run on all cylinders for a short while, before cutting out the next cylinder.

Suppose we have a four-cylinder petrol engine loaded with a hydraulic brake (dynamometer) to measure its brake power. At any given speed with all the four cylinders firing (developing power), the brake power should be accurately measured. Then,

$$\text{Indicated power 4 cylinders} = \text{Brake power 4 cylinders} + \text{Friction power 4 cylinders.(i)}$$

If one cylinder is cut out (spark plug lead is shorted) so that it develops no power, the engine speed will fall. The brake load should then be reduced so that the engine speed increases again to the original given speed. The engine is now developing power in three cylinders, whereas the friction power of all the four cylinders remains the same as already discussed.

Then, the brake power should be measured with the decreased load, i.e., with three cylinders developing power.

$$\text{Then, I.P. 3 cylinders} = \text{B.P. 3 cylinders} + \text{F.P. 4 cylinders} \quad \dots\text{(ii)}$$

Subtracting (ii) from (i), we get,

$$\text{I.P. 4 cylinders} - \text{I.P. 3 cylinders} = \text{B.P. 4 cylinders} - \text{B.P. 3 cylinders}$$

where, B.P. = Brake power, I.P. = Indicated power and F.P., = Friction power.

But, I.P. 4 cylinder - I.P. 3 cylinders is the I.P. of the cylinder that was cut out and hence may be calculated as the difference in readings of B.P. measured when all cylinders were firing and when one cylinder was cut out (i.e., only three cylinders were firing).

By cutting out each cylinder in turn, the I.P. of each cylinder can be determined and the indicated power of the whole engine is then sum of I.P.'s of the separate cylinders. The friction power is given by : total I.P. - total B.P. and the mechanical efficiency is given by dividing the total B.P. by the total I.P. (see illustrative problem No. 10).

7.2.4 Measurement of rate of fuel consumption and its calorific value : This is very easily measured for small capacity engine by noting the time taken to consume a given volume of fuel, although strictly speaking it is the mass of the oil that is required. A simple device, in which two special glass bulbs, one of about 100 c.c. capacity and the other 200 c.c. capacity, may be connected by three-way cocks to the fuel tank and the engine fuel supply line. Three-way cocks help to fill the one bulb when the other is feeding the engine. To reduce the fuel consumption to a mass basis, the specific gravity of the fuel oil should be determined at the temperature of the oil during the trial.

For bigger size oil engine, the simplest and the most accurate method of obtaining the fuel consumption is to support the fuel tank on a weighing machine and supply fuel to the engine. The rate of fuel consumption is then obtained by subtracting the mass of the fuel and tank at the end of the trial, from that at the beginning, the time taken to

discharge this mass of fuel being noted.

The most reliable method of measuring the gas consumption of a gas engine is to pass the gas through a graduated gas holder from which it is drawn by the engine. This is more accurate than the use of a gas meter. *The temperature and pressure of the gas should be taken, so that the volume used may be reduced to normal or standard temperature and pressure.*

A trial of half an hour or even less should suffice, if the engine has settled down to its working conditions.

The heat engine trials committee has recommended to use the gross or higher calorific value of the fuel for the calculation of thermal efficiency and drawing up the heat balance sheet. The higher calorific value for oil fuel can be determined by using Bomb calorimeter and that for gaseous fuel by using Junkers gas calorimeter.

7.2.5 Measurement of heat carried away by cylinder jacket cooling water : In ordinary internal combustion engines, the circulation of cylinder jacket cooling water is maintained by means of natural gravitational current of water or by forced circulation from a pump. In measuring the heat carried away by the jacket water it is necessary to measure the rate of flow of jacket cooling water and also the inlet and outlet temperatures of water. The rate of water circulated in the cylinder jacket is measured by means of water meter fitted in the inlet pipe or by collecting the outflow water in a measuring vessel in a given time interval. The measuring vessel should be supplied with gauge glass reading either in litres or kilograms, or it may be carried on a weighing machine and the mass of water collected in a given time obtained directly. In order to determine the temperature difference all that is necessary is to have two reasonably accurate mercury thermometers which should be inserted in suitable pockets arranged on the inlet and outlet pipes close to the engine.

Let, m_w = mass of cylinder jacket cooling water supplied in kg per minute,
 t_1 = inlet temperature of jacket cooling water, °C,
 t_2 = outlet temperature of jacket cooling water, °C, and
 K = specific heat of water, kJ/kg K.

Then, heat carried away by cylinder jacket cooling water per minute

= mass of cooling water per min. \times specific heat of water \times rise in temperature of cooling water

$$= m_w \times K \times (t_2 - t_1) \text{ kJ/min.} \quad \dots(7.1)$$

There is no reliable method of measuring directly the heat carried away by the air flowing over an air cooled engine and therefore this quantity should be included in the radiation losses, i.e., in the last item of the heat balance sheet.

7.2.6 Measurement of heat carried away by the exhaust gases : In the actual determination of heat carried away by exhaust gases, we are concerned with three quantities, namely the temperature of exhaust gases and room temperature, the mass of exhaust gases, and the mean specific heat of exhaust gases.

Temperature of exhaust gases : The temperature of exhaust gases (t_g) as they leave the engine cylinder can be measured by a thermometer known as *pyrometer*. It works on the principle that when two dissimilar metals are joined together, heating will cause a flow of electricity. The thermo-couple is encased in a tube which is screwed into the exhaust connection of the cylinder. Two wires lead from the thermo-couple to a milli-voltmeter which indicates the minute e.m.f. created by the flow of electricity. The dial of the pyrometer is marked in terms of temperature degrees instead of milli-volts, having been

calibrated by the makers. The warmer the gases, the greater will be the e.m.f. (milli-volts) and the pyrometer dial will read higher temperature.

Mass of exhaust gases : The mass of exhaust gases may be calculated from the measured air consumption by air-box orifice method or by air flow meter in a given time and the fuel consumption in the same time.

$$\text{Air-fuel ratio} = \frac{\text{Air consumption per minute}}{\text{Fuel consumption per minute}}$$

Mass of exhaust gases per minute = Air consumption per min. + Fuel consumption per min.

It is also possible to estimate the air-fuel ratio and the mass of exhaust gases per kg of fuel from the volumetric analysis of exhaust gases by Orsat apparatus and the ultimate analysis of fuel on mass basis.

$$\text{Mass of air supplied per kg of fuel or air-fuel ratio} = \frac{N \times C}{33 (C_1 \times C_2)} = m \text{ kg (say)}$$

where, N , C_1 and C_2 are percentages of nitrogen, carbon dioxide and carbon monoxide by volume in exhaust gases and C is percentage of carbon in fuel on mass basis.

Mass of exhaust gases per kg of fuel = $(.m + 1)$ kg.

\therefore Mass of exhaust gases per minute (m_g) = $(m + 1) \times$ mass of fuel per min. in kg.

Mean specific heat of exhaust gases : The mean specific heat of exhaust products (gases) can be calculated from the knowledge of the constituent products, by allowing the appropriate proportion of specific heat of each constituent.

The value of mean specific heat of exhaust gases (k_p) can be assumed with sufficient accuracy as 1.005 kJ/kg K.

The engine room temperature (t_r) and pressure of the engine room air are measured with ordinary mercury thermometer and mercury barometer respectively. In order to estimate the amount of moisture present in the air it is necessary to read temperatures of dry bulb and wet bulb thermometers.

$$\begin{aligned} \left[\begin{array}{l} \text{Then, heat carried away by} \\ \text{exhaust gases per min.} \end{array} \right] &= \left[\begin{array}{l} \text{mass of exhaust gases per min.} \\ (m_g) \times \text{specific heat of exhaust} \\ \text{gases } (k_p) \times [\text{exhaust gases} \\ \text{temperature } (t_g) - \text{engine room} \\ \text{temperature } (t_r)] \end{array} \right] \\ &= m_g \times k_p \times (t_g - t_r) \text{ kJ/min.} \quad \dots (7.2) \end{aligned}$$

Exhaust gas calorimeter : A direct measurement of the total heat carried away by the exhaust gases may be made by the use of exhaust gas calorimeter shown in fig. 7-4. This consists of a vessel containing a number of tubes through which water is passing. The exhaust gases pass over these tubes and are thereby cooled. By measuring the temperature of exhaust gases leaving the calorimeter and rise in temperature of water and the quantity of water, the heat carried away by exhaust gases can be calculated (See illustrative problem No.2).

$$\begin{aligned} \left[\begin{array}{l} \text{Total heat carried away} \\ \text{by exhaust gases per} \\ \text{min.} \end{array} \right] &= \left[\begin{array}{l} \text{Heat absorbed by water in the exhaust gases calorimeter} \\ \text{per min.} + \text{Heat carried by exhaust gases leaving the} \\ \text{calorimeter per min. in kJ/min.} \end{array} \right] \quad \dots (7.3) \\ &= \left[\begin{array}{l} \text{Rate of flow of water in exhaust gas calorimeter per min.} \\ \times \text{rise in temperature of water} \times \text{specific heat of water} + \\ \text{mass of exhaust gases per min.} \times \text{specific heat of exhaust} \\ \text{gases} \times [\text{exhaust gas temp. at exit from calorimeter} - \\ \text{room temp.}] \text{ kJ/min.} \end{array} \right] \end{aligned}$$

Air consumption : The usual method for measurement of air consumption is to ensure that all the air supplied to the engine is derived exclusively from an air box or tank (fig. 7-1) which is connected to the induction system of the engine by an air tight pipe of a diameter well in excess of that required theoretically for the predicted air flow. A box itself must be air tight. A sharp edged orifice is fitted to the pipe and the pressure difference across it is measured by means of a water manometer as shown in fig. 7-1.

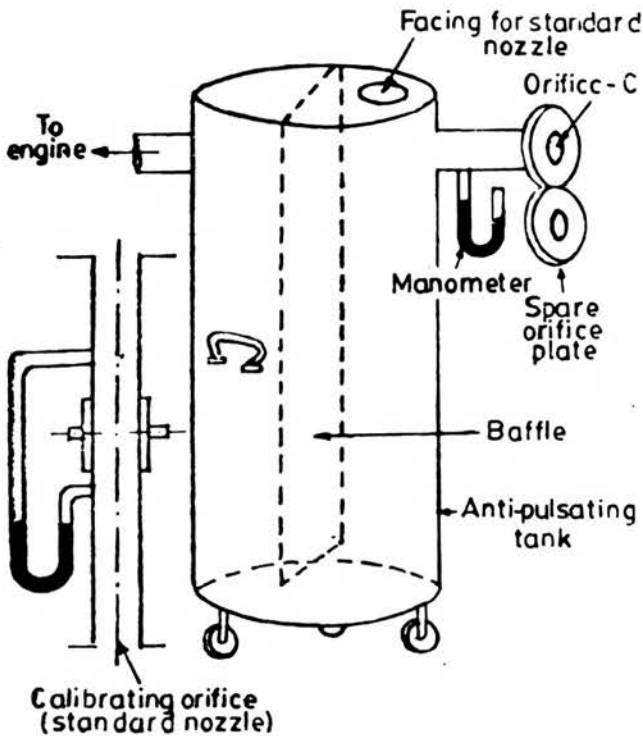


Fig. 7-1. Air box or tank for air flow measurement.

As it is usually desirable to keep the calculations simple it is necessary to keep the water manometer reading down to about 15 cm of water pressure difference, in which case the variation in the density of the air across the orifice is negligible. The air box or tank should have internal baffle so as to avoid any air pulsations, and its volume should be large enough in relation to the total capacity of the engine to be tested (say 200 to 600 times the total capacity), to prevent undue pressure pulsations.

7.3 Heat Balance Account

In a thermodynamic trial of any heat engine, the distribution of the heat supplied per minute or per hour is required. This appears in the heat balance or heat account. In order to complete a heat balance sheet for an internal combustion engine cylinder, the engine should be tested over a period of time under conditions of constant load and speed. All the measurements listed earlier should be taken at regular interval of time. At the completion of the trial the necessary data should be averaged out and a heat account drawn up as follows :

Heat balance sheet in kJ per minute

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel	..	100	(1) Heat equivalent of brake power
			(2) Heat lost to jacket cooling water
			(3) Heat lost to exhaust gases (wet)
			(4) Heat lost to radiation, errors of observation, etc. (by difference)
Total	..	100	Total	..	100

Note : As discussed earlier the heat equivalent of the friction power is not included in the heat balance on the right hand side because most of the heat absorbed in friction will reappear in the jacket cooling water. The heat taken away by the jacket cooling water is already included in the heat balance, and the same amount energy must not be included twice. Some frictional heat will also appear in the heat carried away by exhaust

gases, the remainder being included in the last item of heat balance, i.e., heat lost to radiation, etc. This applies to all types of internal combustion engines.

There are wide variations in the relative proportions of the above losses, depending upon the type, size, and operating conditions of the engine under consideration. For an automobile engine operating on the Otto cycle, the distribution of heat may be : heat converted into work about 25%, heat to the jacket cooling water 25%, heat carried away by exhaust gases 35%, and radiation and other losses 15%. For a Diesel engine, the distribution of heat may be : heat converted into work about 30%, heat to the jacket cooling water 30%, heat carried away by exhaust gases 30%, and radiation and other losses 10%.

The method of estimating the various items in the heat balance sheet is illustrated by solved problems.

7.4 Performance Plotting

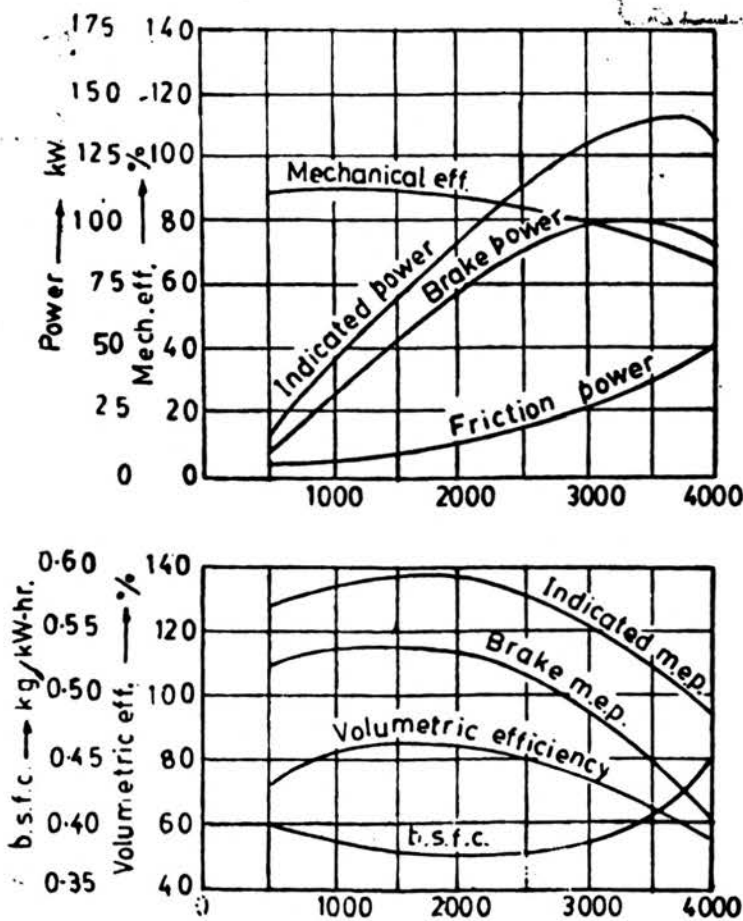


Fig. 7-2. Typical performance curves for an automatic engine at full throttle.

shape of the volumetric efficiency curve depends on the timing of the intake valve. The volumetric efficiency is the highest at medium speeds, say from 1,200 to 2,000 r.p.m., as shown in fig. 7-2. It falls gradually as the speed increases or decreases.

7.4.1 Indicated m.e.p. : Although volumetric efficiency is an important factor in determining the indicated m.e.p. produced by an internal combustion engine, the heating value of the charge and the indicated thermal efficiency are equally important. The heating value of the charge will vary only if the air-fuel ratio it varied. As the engine is not

It is customary to show the performance of a variable speed engine by plotting its characteristics against engine speed in r.p.m. The two chief characteristics are : brake power and brake specific fuel consumption (kg of fuel per kW-hr). It is, however, desirable to study those factors that influence these two characteristics; and auxiliary characteristics, such as volumetric efficiency, indicated mean effective pressure, brake mean effective pressure, torque, indicated power, friction power and mechanical efficiency are often plotted. Typical performance curves are shown in fig. 7-2.

Volumetric efficiency : Volumetric efficiency is a measure of the perfection of the induction process, and may be defined as the ratio of the volume of the induced charge measured under conditions approaching the engine, to the piston displacement. The first characteristic that should be studied is volumetric efficiency, as the power output depends directly on the amount of charge drawn into the cylinder. The

designed for full load operation at very low speed, a poor thermal efficiency results. In general, the i.m.e.p. of an internal combustion engine follows the volumetric efficiency curve rather closely, but at low speeds it falls off a little more than does the volumetric efficiency.

7.4.2. Indicated power : For a given engine, the indicated power is directly proportional to the product of the i.m.e.p and r.p.m. At low and moderate speeds, there is a slight change in i.m.e.p. Hence, over this range, the indicated power curve is almost a straight line, the indicated power being practically proportional to the r.p.m. At higher speeds the decrease in i.m.e.p. causes the indicated power to fall away from a straight line; and at very high speeds, the i.m.e.p. falls off faster than the r.p.m. (speed) and the indicated power decreases.

7.4.3 Friction power : The friction power of a given engine is a function of the product of the frictional resistance and the r.p.m. The major portion of the engine friction in an I.C. engine is the friction between the rings and cylinder walls, and between piston and cylinder walls. With the thick oil film between them, the frictional resistance is directly proportional to the speed. Frictional power should increase faster than the speed. Test results show that this is true.

7.4.4 Brake power : The brake power is the difference between the indicated power and friction power. As the friction power increases faster than the speed, the brake power reaches a maximum value at a speed somewhat lower than that of maximum indicated power. The speed for maximum brake power is known as the peak speed of the engine. This is the speed at which automobile engines are usually rated.

7.4.5 Mechanical efficiency : Mechanical efficiency is the ratio of brake power and indicated power, or ratio of the brake power to the sum of the brake power and friction power. Since the friction power increases faster than the speed and since the brake power fails to increase as fast as speed, the mechanical efficiency must decrease as the speed increases. The decrease is gradual at low speeds, but becomes very rapid at high speeds.

7.4.6 Brake M.E.P. As it is difficult to determine accurately either the indicated m.e.p. or the indicated power for a high speed I.C. engine, the brake m.e.p. is calculated and

used instead. Brake m.e.p. is equal to the product of the indicated m.e.p. and mechanical efficiency. Hence, the curve for brake m.e.p. is quite similar in shape to that for the i.m.e.p., but it falls off faster at high speeds as shown in fig. 7-2.

7.4.7 Torque : Torque is the turning effort produced by an engine. For a given engine, torque is a direct function of brake m.e.p. and as such, the torque curve must have the same shape as the brake m.e.p. curve.

7.4.8 Brake specific fuel consumption : It is the mass of a fuel required per kW-hour on brake power basis. Brake specific fuel consumption

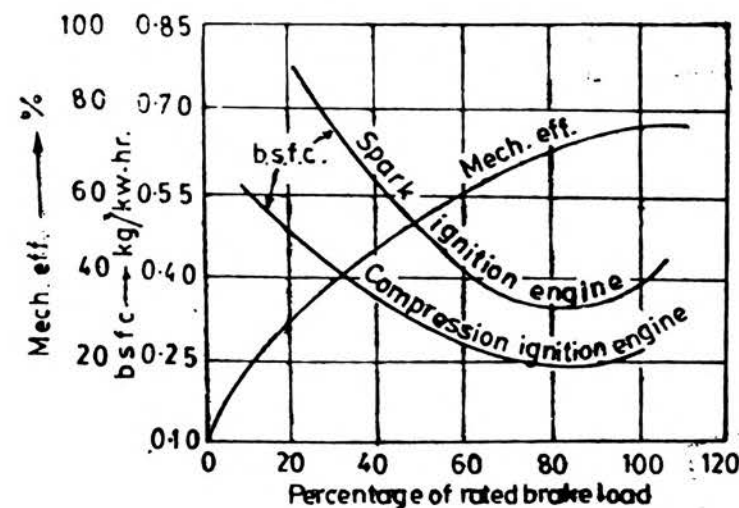


Fig. 7-3. Typical curves of brake specific fuel consumption and mechanical efficiency for constant speed engines.

b.s.f.c.) is inversely proportional to brake thermal efficiency. Since indicated thermal

efficiency falls off at low speeds, the b.s.f.c. becomes relatively high. At high speeds although the indicated thermal efficiency remains high, the excessive frictional losses cause decrease in the brake thermal efficiency and increase in b.s.f.c. Although the curves (fig. 7-2) that have been discussed are those of a variable speed spark-ignition engine, the curves for a Diesel engine (compression-ignition engine) are similar.

For a *constant speed engine*, the curve most commonly plotted is the brake specific fuel consumption versus load, although curves of mechanical and thermal efficiencies may also be plotted as shown in fig. 7-3. For both the Diesel and spark-ignition types of engines, the brake specific fuel consumption increases at heavy loads, primarily because of the large amount of incomplete combustion that accompanies the low air-fuel ratio used, to obtain the heavy loads. At light loads, the brake fuel rates for both types of engines become rather large, primarily because the friction power being substantially constant at a given speed, a large portion of the indicated power output is lost at light loads. Hence, much more fuel must be used per kW-hour on brake power basis at light loads.

Problem - 1 : *The following observations were made during a test on a two-stroke cycle oil engine :*

Cylinder dimensions - 20 cm bore, 25 cm stroke ; speed, 6 r.p.s.; effective brake drum diameter, 1.2 metres; net brke load, 440 newtons; indicated mean effective pressure, 280 kPa; fuel oil consumption, 3.6 kg/hr.; calorific value of fuel oil, 42,500 kJ/kg; mass of jacket cooling water per hour, 468 kg; rise in temperature of jacket cooling water, 28°C; air used per kg of fuel oil, 34 kg; temperature of air in test house, 30°C; temperature of exhaust gases, 400°C; mean specific heat of exhaust gases, 1 kJ/kg K.

Calculate : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the brake mean effective pressure, and (e) brake power fuel consumption in kg per kW-hr. Draw up a heat balance sheet in kJ/min. and as percentages of the heat supplied to the engine. Calculate also the brake thermal efficiency of the engine.

$$\begin{aligned} \text{(a) Brake power} &= (W - S) \times R \times 2\pi \times N \text{ watts} \\ &= 440 \times 0.6 \times 2 \times 3.14 \times 6 = 9,948 \text{ watts or } \mathbf{9.948 \text{ kW}} \end{aligned}$$

$$\begin{aligned} \text{(b) Indicated power} &= p_m \times a \times l \times n \text{ watts} \\ &= (280 \times 10^3) \times \frac{\pi}{4} \left(\frac{20}{100} \right)^2 \times \frac{25}{100} \times 6 \\ &= 13,188 \text{ watts or } \mathbf{13.188 \text{ kW.}} \end{aligned}$$

$$\text{(c) Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{9.948}{13.188} = 0.7541 \text{ or } \mathbf{75.41\%}$$

$$\begin{aligned} \text{(d) Brake m.e.p.} &= \text{Indicated m.e.p.} \times \text{mech. efficiency} \\ &= 280 \times 0.7541 = \mathbf{211.15 \text{ kPa}} \end{aligned}$$

$$\begin{aligned} \text{(e) Fuel consumption in kg per kW-hr. on brake power basis} \\ &= \frac{3.6}{9.948} = \mathbf{0.3616 \text{ kg/kW-hr.}} \end{aligned}$$

Heat supplied per minute :

Heat supplied by combustion of fuel per min.

$$= \frac{3.6}{60} \times 42,500 = 2,550 \text{ kJ/min.}$$

Heat expenditure per minute :

(1) Heat equivalent of brake power per min.

$$= \text{brake power} \times 60 = 9.948 \times 60 = 596.88 \text{ kJ/min.}$$

(2) Using eqn. (7.1), heat lost to jacket cooling water per min.

$$= \text{mass of cooling water per min.} \times \text{specific heat of water} \times \text{rise in temperature of jacket cooling water}$$

$$= m_w \times K \times (t_2 - t_1) = \frac{468}{60} \times 4.187 \times 28 = 914.5 \text{ kJ/min.}$$

(3) 1 kg of fuel combines with 34 kg of air and produces 35 kg of exhaust gases i.e. 1 kg of fuel produces 35 kg of exhaust gases.

Now, as 1 kg of fuel produces 35 kg of exhaust gases,

$$\frac{3.6}{60} \text{ kg of fuel per minute will produce } \frac{3.6}{60} \times 35 = 2.1 \text{ kg of exhaust gases/min.}$$

Using eqn. (7.2),

Heat lost to exhaust gases/min. (wet)

$$= \text{mass of exhaust gases/min.} \times \text{specific heat of exhaust gases} \times (\text{exhaust gas temp.} - \text{room temp.})$$

$$= m_g \times k_p \times (t_g - t_r) = 2.1 \times 1 \times (400 - 30) = 777 \text{ kJ/min.}$$

(4) Heat lost to radiation, errors of observation, etc. per min. (by difference)

$$= 2,550 - (596.88 + 914.5 + 777) = 216.62 \text{ kJ/min.}$$

Heat balance sheet in kJ/minute

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel	2,550	100	(1) Heat equivalent of brake power	596.88	23.41
			(2) Heat lost to jacket cooling water	914.5	35.86
			(3) Heat lost to exhaust gases (wet)	777	30.47
			(4) Heat lost to radiation, errors of observation, etc. (by difference)	261.62	10.26
Total	2,550	100	Total	2,550	100

$$\text{Brake thermal efficiency} = \frac{\text{Heat equivalent of brake power per min.}}{\text{Heat supplied per min.}}$$

$$= \frac{9.948 \times 60}{2,550} = 0.2341 \text{ or } 23.41\%$$

Problem – 2 : In a test of an oil engine running under full load conditions, the following results were obtained :

Brake power, 18.5 kW; Fuel consumption, 5.5 kg/hr; Calorific value of fuel oil, 43,000 kJ per kg; Inlet and outlet temperatures of cylinder circulating water, 15.5°C and 71.2°C respectively; Rate of flow of cylinder circulating water, 4.6 kg/min.; Inlet and outlet temperatures of water to exhaust gas calorimeter, 15.5°C and 54.4°C respectively; Rate of flow of water through calorimeter, 8.1 kg per min.; Temperature of exhaust gases leaving the calorimeter, 82.2°C; Room temperature, 17°C; Air-fuel ratio on mass basis, 20. Take the mean specific heat of exhaust gases including vapour as 1.005 kJ/kg K.

Draw up a heat balance sheet for the test on one minute basis and as percentages

of the heat supplied to the engine.

Heat supplied per minute :

$$\text{Heat supplied by combustion of fuel} = \frac{5.5}{60} \times 43,000 = 3,942 \text{ kJ/min.}$$

Heat expenditure per minute :

(1) Heat equivalent of brake power = $18.5 \times 60 = 1,110 \text{ kJ/min.}$

(2) Heat lost to cylinder jacket circulating water
 $= 4.6 \times 4.187 \times (71.2 - 15.5) = 1,072.8 \text{ kJ/min.}$

(3) (a) Heat absorbed by water in the exhaust gas calorimeter
 $= 8.1 \times 4.187 \times (54.4 - 15.5) = 1,312.5 \text{ kJ/min.}$

(b) Heat remaining in exhaust gases leaving the exhaust gas calorimeter
 $= \left(\frac{5.5}{60} \times 21 \right) \times 1.005 \times (82.2 - 17) = 126.1 \text{ kJ/min.}$

Using eqn. (7.3), total heat carried away by exhaust gases (wet) = (a) + (b)
 $= 1,312.5 + 126.1 = 1,438.6 \text{ kJ/min.}$

(4) Heat lost to radiation, errors of observation, etc. (by difference)
 $= 3,942 - (1,110 + 1,072.8 + 1,438.6) = 320.6 \text{ kJ/min.}$

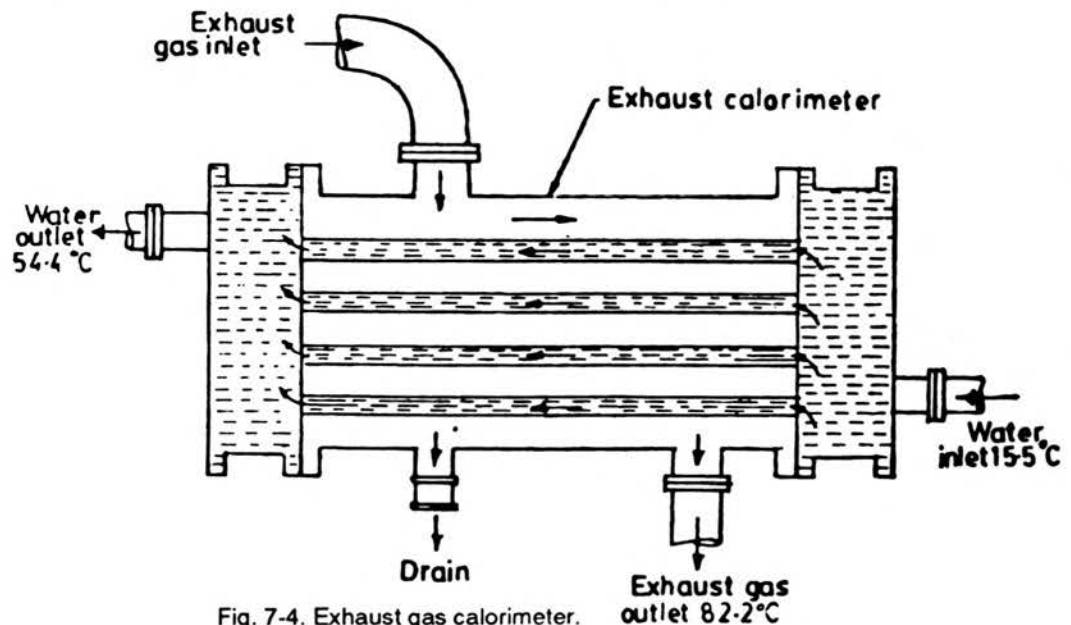


Fig. 7-4. Exhaust gas calorimeter.

Heat balance sheet in kJ/minute

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel oil	3,942	100	(1) Heat equivalent of brake power	1,110.0	28.16
			(2) Heat lost to jacket cooling water	1,072.8	27.22
			(3) Heat lost to exhaust gases (wet)	1,438.6	36.50
			(4) Heat lost to radiation, errors, etc. (by difference)	320.6	8.12
Total	3,942	100	Total	3,942	100.00

Note : Heat to friction = indicated power - brake power, reappears partly in the heat to jacket cooling water and partly in exhaust gases and radiation.

Problem – 3 : The following readings were taken during a test on a single-cylinder, four-stroke cycle oil engine : Cylinder bore, 20 cm; Stroke length, 35 cm; Indicated mean effective pressure, 700 kPa; Engine speed, 4 r.p.s; Fuel oil used per hour, 3.5 kg; Calorific value of oil, 46,000 kJ/kg; Brake torque, 450 N.m; Mass of jacket cooling water per minute, 5 kg; Rise in temperature of jacket cooling water, 40°C; Mass of air supplied per minute, 1.35 kg; Temperature of exhaust gases, 340°C; Room temperature, 15°C; Mean specific heat of dry exhaust gases, 1 kJ/kg K; Hydrogen in fuel, 13.5% on mass basis, k_p of steam in exhaust gases, 2.3 kJ/kg K.

Calculate the mechanical and indicated thermal efficiencies and brake power fuel consumption in kg per kW-hr. Also draw up a heat balance sheet in kJ/min. and as percentages of the heat supplied to the engine.

$$\text{Indicated power} = p_m \times a \times l \times n \text{ watts}$$

$$= (700 \times 10^3) \times \frac{\pi}{4} \left(\frac{20}{100} \right)^2 \times \frac{35}{100} \times \frac{4}{2} = 15,400 \text{ watts or } 15.4 \text{ kW}$$

$$\text{Brake power} = (W - S) R \times 2\pi \times N \text{ watts}$$

$$= T \times 2\pi \times N \text{ watts}$$

$$= 450 \times 2 \times 3.14 \times 4 = 11,304 \text{ watts or } 11.304 \text{ kW}$$

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{11.304}{15.4} = 0.734 \text{ or } 73.4\%$$

$$\text{Indicated thermal efficiency} = \frac{\text{Heat equivalent of indicated power in kJ per min.}}{\text{Heat supplied in kJ per min.}}$$

$$= \frac{15.4 \times 60}{\frac{3.5}{60} \times 46,000} = 0.3443 \text{ or } 34.43\%$$

$$\text{Fuel consumption in kg per kW-hr. on brake power basis} = \frac{3.5}{11.304} = 0.31 \text{ kg/kW-hr}$$

Heat supplied per minute :

$$\text{Heat supplied by combustion of fuel} = \frac{3.5}{60} \times 46,000 = 2,683.3 \text{ kJ/min.}$$

Heat expenditure per minute :

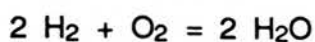
$$(1) \text{ Heat equivalent of brake power} = 11.304 \times 60 = 678.24 \text{ kJ/min.}$$

$$(2) \text{ Heat lost to jacket cooling water} = 5 \times 4.187 \times 40 = 837.4 \text{ kJ/min.}$$

(3) Mass of wet exhaust gases per minute

$$= \text{mass of air per min.} + \text{mass of fuel per min.}$$

$$= 1.35 + \frac{3.5}{60} = 1.4083 \text{ kg per min.}$$



$$1 + 8 = 9 \quad \text{i.e., one kg of H}_2 \text{ produces 9 kg of H}_2\text{O}$$

$$\therefore \text{ Mass of H}_2\text{O produced per kg of fuel burnt} = 9 \times \text{H}_2 = 9 \times 0.135 = 1.215 \text{ kg per kg of fuel.}$$

\therefore Mass of H₂O (steam) produced per minute

$$= 1.215 \times \text{mass of fuel per minute}$$

$$= 1.215 \times \frac{3.5}{60} = 0.0709 \text{ kg per min.}$$

i.e., mass of steam in wet exhaust gases = 0.0709 kg per min.

Mass of *dry* exhaust gases per minute

$$= \text{mass of wet exhaust gases/min.} - \text{mass of steam in wet exhaust gases/min.}$$

$$= 1.4083 - 0.0709 = 1.3374 \text{ kg.}$$

∴ Heat lost to *dry* exhaust gases min. = mass of dry exhaust gases × specific heat of dry exhaust gases × (exhaust gas temp. – room temp.)

$$= 1.3374 \times 1 \times (340 - 15) = 434.66 \text{ kJ/min.}$$

(4) Assuming that the steam in exhaust gases exists as superheated steam at atmospheric pressure (1.01325 bar) and at exhaust gas temperature,

Enthalpy of 1 kg of steam = $H_{sup} - h$

$$= [H_s + kp (t_{sup} - t_s)] - h$$

$$= [2,676.1 + 2.3(340 - 100)] - 15 \times 4.187 = 3,165.3 \text{ kJ/kg}$$

Heat lost to steam in exhaust gases per min.

$$= \text{mass of steam per min.} \times \text{enthalpy of 1 kg of steam}$$

$$= 0.0709 \times 3,165.3 = 224.42 \text{ kJ/min.}$$

(5) Heat lost to radiation, error of observation, etc. per min. (by difference)
 $= 2,683.3 - (678.24 + 837.4 + 434.66 + 224.42) = 508.58 \text{ kJ/min.}$

Heat balance sheet in kJ/minute

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel oil	2,683.3	100	(1) Brake power heat equivalent	678.24	25.29
			(2) Heat lost to jacket cooling water	837.4	31.2
			(3) Heat lost to dry exhaust gases	434.66	16.2
			(4) Heat lost to steam in exhaust gases	224.42	8.36
			(5) Heat lost to radiation, etc. (by difference)	508.58	18.95
Total	2,683.3	100	Total	2,683.3	100

Problem – 4 : A four-stroke, solid injection, Diesel engine coupled to a single-phase A.C. generator gave the following data during a trial of 45 minutes duration :

Fuel oil used	... 2.9 kg
Calorific value of fuel oil	... 46,900 kJ/kg
Analysis of fuel oil on mass basis	... C, 86%; H ₂ , 10%; other matter, 4%
Percentage analysis of dry exhaust gases by volume	... CO ₂ , 7.6; CO, 0.4; O ₂ , 6; N ₂ , 86
Mass of cylinder jacket cooling water	... 260 kg
Room and cylinder jacket water inlet temperature	... 26°C

Temperature of water leaving cylinder jacket	... 73°C
Temperature of exhaust gases	... 290°C
Engine speed	... 250 r.p.m.
Generator voltage and current	... 440 V, 25 Amp.
Efficiency of generator	... 92%

Soon after the test the engine was motored by the dynamo taking current from the mains : Applied voltage - 440 V; Current - 7.8 Amp.; Speed - 250 r.p.m.; Efficiency of the generator as motor - 92%

Draw up a heat balance sheet on percentage basis assuming that the steam in the exhaust gases is at atmospheric pressure (1.01325 bar). Calculate the mechanical efficiency of the engine. Take k_p of dry exhaust gases as 1 kJ/kg K and k_p of steam as 2.1 kJ/kg K.

Heat supplied per 45 minutes :

Heat supplied by combustion of fuel = $2.9 \times 46,900 = 1,36,010$ kJ/ 45 min.

Heat expenditure per 45 minutes :

$$(1) \text{ Brake power} = \frac{440 \times 25}{0.92 \times 1,000} = 11.957 \text{ kW}$$

Heat equivalent of brake power = $11.957 \times 60 \times 45 = 32,284$ kJ/45 min.

(2) Heat lost to jacket cooling water = $260 \times 4.187 (73 - 26) = 51,165$ kJ/45 min.

$$(3) \text{ The mass of air supplied per kg of fuel} = \frac{NC}{33(C_1 + C_2)}$$

where N , C_1 , C_2 are percentages of nitrogen, carbon dioxide and carbon monoxide by volume in exhaust gases, and C is the percentage of carbon in fuel oil on mass basis.

$$\therefore \text{ Mass of air supplied per kg of fuel} = \frac{86 \times 86}{33(7.6 + 0.4)} = 28 \text{ kg.}$$

Out of the above air, oxygen used for combustion of hydrogen
= $0.1 \times 8 = 0.8$ kg.

\therefore Remaining air forming dry exhaust gases = $28 - 0.8 = 27.2$ kg

Hence mass of dry products of combustion (exhaust gases)

= $27.2 + 0.86 = 28.06$ kg/kg of fuel.

\therefore Mass of dry exhaust gases/45 min. = $2.9 \times 28.06 = 81.4$ kg/45 min.

Thus, heat lost to dry exhaust gases = $81.4 \times 1 \times (290 - 26) = 21,490$ kJ/45 min.

(4) Heat lost to steam in exhaust gases per 45 min.

$$\begin{aligned} &= [\text{mass of fuel/45 min} \times 9 H_2] \times [H_s + k_p (t_{sup} - t_s) - h] \\ &= [2.9 \times (9 \times 0.1)] \times [2,676.1 + 2.1 (290 - 100) - 26 \times 4.187] \\ &= 7,741.8 \text{ kJ/45 min.} \end{aligned}$$

(5) Heat lost to radiation, errors of observations, etc. (obtained by difference)

$$= 1,36,010 - (32,284 + 51,165 + 21,490 + 7,741.8) = 23,329.2 \text{ kJ/45 min.}$$

Heat balance sheet in kJ per 45 minutes

Heat supplied per 45 min.	kJ	%	Heat expenditure per 45 min.	kJ	%
Heat supplied by combustion of fuel oil	1,36,010	100	(1) Heat equivalent of Brake power	32,284	23.74
			(2) Heat lost to jacket cooling water	51,165	37.62
			(3) Heat lost to dry exhaust gases	21,490	15.80
			(4) Heat lost to steam in exhaust gases	7,741.8	5.69
			(5) Heat lost to radiation, errors of observation, etc. (by difference)	23,329.2	17.15
Total	1,36,010	100	Total	1,36,010	100

$$\text{Friction power of the engine} = \frac{440 \times 7.8 \times 0.92}{1,000} = 3.157 \text{ kW}$$

$$\begin{aligned} \therefore \text{Mechanical efficiency } \eta_m &= \frac{\text{Brake power}}{\text{Brake power} + \text{Friction power}} \\ &= \frac{11.957}{11.957 + 3.157} = \frac{11.957}{15.114} = 0.7911 \text{ or } 79.11\% \end{aligned}$$

Problem – 5 : A six-cylinder, four-stroke Diesel engine has a bore to stroke ratio of 360 : 500 mm. During the trial, following results were obtained :

Mean area of the indicator diagram, 7.8 cm²; length of the indicator diagram, 7.5 cm; spring number, 700 kPa per cm of compression; brake torque, 14,000 N.m; speed, 8 r.p.s.; fuel consumption, 240 kg/hr; calorific value of fuel oil, 44,000 kJ/kg; jacket cooling water used, 320 kg/minute; rise in temperature of the cooling water, 40°C; piston cooling oil (specific heat, 2.1 kJ/kg K) used, 140 kg/min., with a temperature rise of 28°C. The exhaust gases give up all their heat to 300 kg/minute of water circulating through the exhaust gas calorimeter and raises its temperature through 42°C.

Calculate the brake specific fuel consumption in kg per kW-hour and mechanical efficiency of the engine and draw up a heat balance sheet of the engine on the basis of 1 kg of fuel oil.

$$\text{Indicated mean effective pressure, } = p_m = \frac{7.8}{7.5} \times 700 = 728 \text{ kPa}$$

$$\begin{aligned} \text{Indicated power per cylinder} &= p_m \times a \times l \times n \text{ kW} \\ &= 728 \times [0.7854 \times (0.36)^2] \times 0.5 \times \frac{8}{2} = 148.2 \text{ kW} \end{aligned}$$

$$\text{Total indicated power developed by six cylinders} = 148.2 \times 6 = 889.2 \text{ kW}$$

$$\text{Brake power} = T \times 2\pi \times N = 14,000 \times 2\pi \times 8 = 7,03,720 \text{ watts} = 703.72 \text{ kW}$$

$$\text{Brake specific fuel consumption (B.S.F.C.)} = \frac{240}{703.72} = 0.341 \text{ kg/kW-hr.}$$

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{703.72}{889.2} = 0.7914 \text{ or } 79.14\%$$

Heat supplied per kg of fuel oil :

$$(1) \text{ Heat supplied per kg of fuel oil} = 1 \times 44,000 = 44,000 \text{ kJ/kg of fuel}$$

Heat expenditure per kg of fuel oil :

(1) Heat equivalent of brake power per Kg of fuel oil

$$= 703.72 \times 60 \times \frac{1}{240} = 10,556 \text{ kJ/kg}$$

(2) Mass of jacket cooling water used per kg of fuel oil = $\frac{320 \times 60}{240} = 80 \text{ kg}$.

Heat lost to cylinder jacket cooling water per kg of fuel oil

$$= 80 \times 4.187 \times 40 = 13,398 \text{ kJ/kg}$$

(3) Mass of piston cooling oil used per kg of fuel oil = $\frac{140 \times 60}{240} = 35 \text{ kg}$

∴ Heat lost to piston cooling oil per kg of fuel oil = $35 \times 2.1 \times 28 = 2,058 \text{ kJ/kg}$

(4) Mass of water circulated in exhaust gas calorimeter per kg of fuel oil

$$= \frac{300 \times 60}{240} = 75 \text{ kg}$$

Heat lost to exhaust gases per kg of fuel oil = $75 \times 4.187 \times 42 = 13,189 \text{ kJ/kg}$

(5) Heat lost to radiation, errors of observation, etc. per kg of fuel oil (by difference)

$$= 44,000 - (10,556 + 13,398 + 2,058 + 13,189) = 4,789 \text{ kJ/kg}$$

Heat balance sheet per kg of fuel oil

Heat supplied per kg of fuel oil	kJ	Heat expenditure per kg of fuel oil	kJ
Heat supplied by combustion of fuel oil	44,000	(1) Heat equivalent of brake power	10,566
		(2) Heat lost to jacket cooling water	13,398
		(3) Heat lost to piston cooling oil	2,058
		(4) Heat lost to exhaust gases (wet)	13,189
		(5) Heat lost to radiation, errors of observation, etc. (by difference)	4,789
Total	44,000	Total	44,000

Problem – 6 : A single-cylinder, four-stroke cycle gas engine of 25 cm bore and 36 cm stroke, with hit and miss governing, was tested with the following results :

Duration of trial, one hour; net load on the brake, 1,200 newtons; effective radius of the brake wheel, 0.6 metre; total number of revolutions, 14,400; total number of explosions, 6,600; mean effective pressure from indicator diagram, 700 kPa; gas used, 13.7 m³ at normal temperature (0°C) and pressure (760 mm Hg); calorific value of gas at normal temperature and pressure, 20,000 kJ/m³ ; mass of cooling water passing through the jacket, 600 kg; temperature of jacket cooling water at inlet 15°C and at outlet 50°C; mass of exhaust gases, 210 kg; temperature of exhaust gases, 400°C; room temperature, 15°C; mean specific heat of exhaust gases 1 kJ/kg K.

Calculate the thermal efficiency on indicated power and brake power basis, and draw up a heat balance sheet for the test on one minute basis in kJ and as percentages of the heat supplied to the engine.

$$\text{Brake power} = (W - S) \times R \times 2\pi \times N \text{ watts}$$

$$= 1,200 \times 0.6 \times 2 \times 3.14 \times \frac{14,400}{3,600} = 18,086 \text{ watts or } 18.086 \text{ kW}$$

Indicated power = $p_m \times a \times l \times n$ watts

$$= (700 \times 10^3) \times \left[\frac{\pi}{4} \left(\frac{25}{100} \right)^2 \right] \times \frac{36}{100} \times \frac{6,600}{3,600}$$

$$= 26,666 \text{ watts or } 26.666 \text{ kW}$$

Indicated thermal efficiency = $\frac{\text{Indicated power} \times 3,600}{V_g \times C.V.}$

$$= \frac{26.666 \times 3,600}{13.7 \times 20,000} = 0.2978 \text{ or } 29.78\%$$

Brake thermal efficiency = $\frac{\text{Brake power} \times 3,600}{V_g \times C.V.}$

$$= \frac{18.086 \times 3,600}{13.7 \times 20,000} = 0.2376 \text{ or } 23.76\%$$

Heat supplied per minute :

$$\text{Heat in gas supplied} = \frac{13.7}{60} \times 20,000 = 4,566.67 \text{ kJ/min.}$$

Heat expenditure per minute :

(1) Heat equivalent of brake power = $18.086 \times 60 = 1,085.16 \text{ kJ/min.}$

(2) Heat carried away by jacket cooling water

$$= \frac{600}{60} \times 4.187 (50 - 15) = 1,465.45 \text{ kJ/min.}$$

(3) Heat carried away by exhaust gases (wet)

$$= \frac{210}{60} \times 1 \times (400 - 15) = 1,347.4 \text{ kJ/min.}$$

(4) Heat lost to radiation, errors of observation, etc. (by difference)

$$= 4,566.67 - (1,085.16 + 1,465.45 + 1,347.4) = 668.66 \text{ kJ/min.}$$

Heat balance sheet in kJ per minute

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of gas	4,566.67	100	(1) Heat equivalent of brake power	1,085.16	23.77
			(2) Heat lost to jacket cooling water	1,465.45	32.09
			(3) Heat lost to exhaust gases (wet)	1,347.4	29.51
			(4) Heat lost to radiation, errors of observation, etc. (by difference)	668.66	14.63
Total	4,566.67	100	Total	4,566.67	100.00

Problem - 7 : The following observations were made during a trial of a single-cylinder, four-stroke cycle gas engine having cylinder bore 15 cm and stroke 24 cm :

Duration of trial	... one hour
Engine speed	... 5 r.p.s.
Total number of explosions	... 8,880
Mean effective pressure	... 590 kPa

Net load on brake	... 350 newtons
Effective diameter of brake wheel	... 0.9 metre
Total gas consumption at N.T.P.	... 4.5 m ³
Calorific value of gas at N.T.P.	... 18,000 kJ/m ³
Density of gas at N.T.P.	... 0.97 kg/m ³
Total air consumed	... 47.5 m ³
Density of air at N.T.P.	... 1.293 kg/m ³
Pressure of air	... 725 mm Hg
Temperature of air	... 15°C
Temperature of exhaust gases	... 350°C
Specific heat of exhaust gases	... 1.05 kJ/kg K
Mass of jacket cooling water	... 165 kg
Rise in temperature of jacket cooling water	... 34°C

Calculate the mechanical and the overall efficiency. Also draw up a heat balance sheet on one minute basis.

$$\text{Number of explosions per minute} = \frac{8,880}{60} = 148$$

$$\text{Indicated power} = p_m \times a \times l \times n \text{ kW}$$

$$= 590 \times [0.7854 \times (0.15)^2] \times 0.24 \times \frac{148}{60} = 6.172 \text{ kW}$$

$$\text{Brake power} = (W - S) \times \pi D \times N = 350 \times \pi \times 0.9 \times 5 = 4,948 \text{ watts} = 4.948 \text{ kW}$$

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{4.948}{6.172} = 0.8017 \text{ or } 80.17\%$$

Heat supplied per minute :

$$\text{Gas used per minute} = \frac{4.5}{60} = 0.075 \text{ m}^3$$

$$\text{Heat in gas supplied per minute} = 0.075 \times 18,000 = 1,350 \text{ kJ/min.}$$

Heat expenditure per minute :

$$(1) \text{ Heat equivalent of brake power per min.} = 4.948 \times 60 = 296.9 \text{ kJ/min.}$$

(2) Heat carried away by jacket cooling water per minute

$$= \frac{165}{60} \times 4.187 \times 34 = 391.5 \text{ kJ/min.}$$

(3) Absolute pressure of air, $p_1 = 725 \text{ mm of Hg}$;

$$\text{Absolute temperature of air, } T_1 = 15 + 273 = 288 \text{ K};$$

$$\text{Volume of air consumed per hour, } v_1 = 47.5 \text{ m}^3.$$

Volume of air consumed per hour at 760 mm Hg and 0°C (N.T.P.), v_2 is to be determined.

$$\text{Now, } \frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \therefore \text{Volume of air used per hour at 760 mm Hg and 0°C (N.T.P.),}$$

$$v_2 = v_1 \times \frac{p_1}{p_2} \times \frac{T_2}{T_1} = 47.5 \times \frac{725}{760} \times \frac{273}{288} = 42.9 \text{ m}^3 \text{ per hour.}$$

\therefore Mass of air used per minute,

$$m_1 = \text{density of air at N.T.P.} \times v_2 = 1.293 \times \frac{42.9}{60} = 0.927 \text{ kg.}$$

$$\text{Mass of gas consumed per minute, } m_2 = 0.97 \times 0.075 = 0.0728 \text{ kg.}$$

∴ Mass of exhaust gases produced per minute,

$$m_g = m_1 + m_2 = 0.927 + 0.0728 = 0.9998 \text{ kg.}$$

Heat lost to exhaust gases per minute (wet)

$$= m_g \times \text{sp. heat of exhaust gases} \times (\text{exhaust gas temp.} - \text{room temp.})$$

$$= 0.9998 \times 1.05 \times (350 - 15) = 351.7 \text{ kJ/min.}$$

(4) Heat lost to radiation, errors of observation, etc. (obtained by difference)

$$= 1,350 - (296.9 + 391.5 + 351.7) = 309.9 \text{ kJ/min.}$$

Heat balance sheet in kJ per minute

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of gas	1,350	(1) Heat equivalent of brake power	296.9
		(2) Heat lost to jacket cooling water	391.5
		(3) Heat lost to exhaust gases (wet)	351.7
		(4) Heat lost to radiation, errors of observation, etc. (by difference)	309.9
Total	1,350	Total	1,350

$$\text{Overall efficiency (brake thermal efficiency)} = \frac{\text{Brake power heat equivalent in kJ per hr.}}{\text{Heat supplied in kJ per hr.}}$$

$$= \frac{4.948 \times 3,600}{4.5 \times 18,000} = 0.22 \text{ or } 22\%$$

Problem – 8 : A single-cylinder, 4-stroke cycle gas engine of 20 cm bore and 38 cm stroke, with hit and miss governing, was tested with the following results :

Barometer, 720 mm of Hg; Atmospheric and gas temperatures, 17°C; Gas consumption 0.153 m³/minute at 8.8 mm of water above atmospheric pressure; Calorific value of gas 18,000 kJ/m³ at N.T.P.; Density of gas 0.61 kg/m³ at N.T.P.; Hydrogen content in gas, 13% on mass basis; Air used, 1.45 kg per minute; K_p of dry exhaust gases, 1.05 kJ/kgK; Exhaust gas temperature, 400°C; K_p of steam, 2.1 kJ/kg K :

M.E.P. - Positive loop = 560 kPa at firing;

M.E.P. - Negative loop = 26.5 kPa at firing;

M.E.P. - Negative loop = 36.7 kPa at missing;

Speed, 285 r.p.m., Explosions per minute, 114; Brake-torque, 335 N.m; Cylinder jacket cooling water, 4.5 kg/minute; Rise in temperature of jacket cooling water, 40°C.

Calculate the percentages of the indicated power which are used for pumping and for mechanical friction, and draw up a percentage heat balance sheet.

The p-v diagram (fig. 7-5) consists of two enclosed areas. The negative loop,

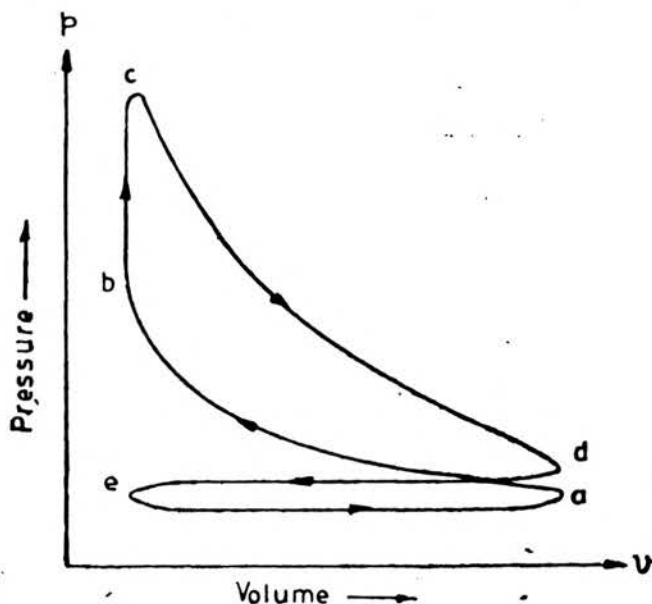


Fig. 7-5.

i.e. smaller enclosed area *dea* gives the pumping loss due to admission of fresh charge and removal of exhaust gases. The larger area *abcd* (positive loop) represents the gross work done by the piston during the cycle (when firing). The negative loop work (indicated power) is to be deducted from the gross work (indicated power) developed to get the net work done (indicated power). The pumping loop (negative loop) is shown much exaggerated in the fig. 7-5.

$$\text{Indicated power} = p_m \times a \times l \times n \text{ kW}$$

(where n = no. of explosions per sec.)

Positive loop indicated power or gross indicated power when firing (hit)

$$= 560 \times \left[0.7854 \times (0.2)^2 \right] \times 0.38 \times \frac{114}{60} = 12.702 \text{ kW}$$

Negative loop indicated power or pumping Indicated power when firing (hit)

$$= 26.5 \times \left[0.7854 \times (0.2)^2 \right] \times 0.38 \times \frac{114}{60} = 0.601 \text{ kW}$$

Indicated power = $p_m \times l \times a \times n_1$ (where n_1 = no. of missed explosions/sec.)

Negative loop indicated power or pumping indicated power when *not firing* (miss)

$$= 36.7 \times \frac{38}{100} \times \frac{\pi}{4} \left(\frac{20}{100} \right)^2 \times \left(\frac{285}{120} - \frac{114}{60} \right) = 0.208 \text{ kW}$$

Hence, total pumping indicated power (when firing and not firing)

$$= 0.601 + 0.208 = 0.809 \text{ kW}$$

Hence, *pumping* indicated power is

$$\frac{0.809}{12.702} \times 100 = 6.37\% \text{ of positive loop indicated power or gross indicated power}$$

Indicated power (net) developed = Positive loop indicated power (or gross indicated power) - total pumping indicated power

$$= 12.702 - 0.809 = 11.893 \text{ kW}$$

$$\text{Brake power} = 2\pi \times N \times T = 2\pi \times \frac{285}{60} \times 335 = 9,998 \text{ watts} = 9.998 \text{ kW}$$

$$\text{Friction power} = \text{Indicated power (net)} - \text{brake power} = 11.893 - 9.998 = 1.895 \text{ kW}$$

Hence, *mechanical friction power* is,

$$\frac{1.895}{12.702} \times 100 = 14.92\% \text{ of positive loop indicated power or gross indicated power}$$

Heat supplied per minute :

$$\text{Gas pressure, } p_1 = \frac{8.8}{13.6} + 720 = 720.646 \text{ mm of Hg; } v_1 = 0.153 \text{ m}^3 ; T_1 = 290 \text{ K;}$$

$$p_2 = 760 \text{ mm of Hg; } T_2 = 273 \text{ K; } v_2 \text{ is to be determined.}$$

$$\text{Now, } \frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \quad \text{or} \quad v_2 = \frac{v_1 \times p_1 \times T_2}{p_2 \times T_1}$$

$$\therefore \text{N.T.P. gas consumption, } v_2 = \frac{0.153 \times 720.646 \times 273}{760 \times 290} = 0.138 \text{ m}^3 \text{ per min.}$$

$$\therefore \text{Heat supplied} = 0.138 \times 18,000 = 2,484 \text{ kJ/min.}$$

Heat expenditure per minute :

$$(1) \text{ Heat equivalent of brake power} = 9.998 \times 60 = 600 \text{ kJ/min.}$$

(2) Heat lost to jacket cooling water = $4.5 \times 4.187 \times 40 = 753.7$ kJ/min.

(3) Mass of fuel gas $0.138 \times 0.61 = 0.0842$ kg/min, mass of air = 1.45 kg/min. (given).

\therefore Mass of exhaust gases (including water vapour) = $1.45 + 0.0842 = 1.5342$ kg/min.

Now, mass of water vapour (steam) of combustion per min.

$$= (9H_2) \times \text{mass of fuel gas per min.} = (9 \times 0.13) \times 0.0842 = 0.0985 \text{ kg/min.}$$

Hence, mass of dry exhaust gases per min.

$$= \text{mass of wet exhaust gases/min.} - \text{mass of water vapour/min.}$$

$$= 1.5342 - 0.0985 = 1.4357 \text{ kg/min.}$$

Heat lost to dry exhaust gases per min. = $1.4357 \times 1.05 \times (400 - 17) = 577.4$ kJ/min.

(4) Assuming the partial pressure of the water vapour as 0.07 bar,

at 0.07 bar, $H_s = 2,572.5$ kJ/kg, $t_s = 39^\circ\text{C}$ (from steam tables).

Enthalpy of 1 kg of water vapour = $H_s + K_p (t_{sup} - t_s) - h$

$$= 2,572.5 + 2.1 (400 - 39) - (17 \times 4.187) = 3,259.4 \text{ kJ per kg}$$

Heat lost to water vapour (steam) per min.

$$= \text{mass of steam formed per min} \times \text{enthalpy of one kg of steam}$$

$$= 0.0985 \times 3,259.4 = 321 \text{ kJ/min.}$$

(5) Heat lost to radiation, errors of observation, etc. (obtained by difference) per min.

$$= 2,484 - (600 + 753.7 + 577.4 + 321) = 231.9 \text{ kJ/min.}$$

Heat balance sheet in kJ per minute

Heat supplied/min.	kJ	%	Heat expenditure/min	kJ	%
Heat supplied by combustion of gas	2,484	100	(1) To brake power	600	24.15
			(2) To jacket cooling water	753.7	30.34
			(3) To dry exhaust gases	577.4	23.25
			(4) To steam	321	12.92
			(5) To radiation, errors of observation, etc. (by difference)	231.9	9.34
Total	2,484	100	Total	2,484	100

Problem – 9 : A single-cylinder, four-stroke, gas engine with explosion in every cycle, used $0.23 \text{ m}^3/\text{min}$. of gas during a test. The pressure and temperature of gas at the meter being 75 mm of water and 17°C respectively. The calorific value of the gas is $18,800 \text{ kJ/m}^3$ at N.T.P. The air consumption was 2.85 kg/min. The barometer reading was 743 mm of Hg. The bore of cylinder is 25 cm and stroke 48 cm. The engine is running at 240 r.p.m.

Estimate the volumetric efficiency of the engine relative to air at N.T.P. (a) taking air and gas mixture into account, and (b) taking air only into account. Assume the volume per kg of air at N.T.P. as 0.7734 m^3 .

$$\text{Absolute pressure of gas at the meter} = 743 + \frac{75}{13.6} = 743 + 5.51 = 748.51 \text{ mm of Hg.}$$

Gas used per min. at 748.51 mm of Hg and 17°C is 0.23 m^3

\therefore Gas used per min. at N.T.P. (760 mm Hg and 0°C)

$$= 0.23 \times \frac{748.51}{760} \times \frac{273}{(273 + 17)} = 0.213 \text{ m}^3 \text{ per min.}$$

$$\therefore \text{Volume of gas used per stroke at N.T.P.} = \frac{0.213}{120} = 0.00178 \text{ m}^3$$

$$\text{Now, volume of air used per stroke at N.T.P.} = \frac{2.85}{120} \times 0.7734 = 0.01836 \text{ m}^3$$

\therefore Volume of air-gas mixture used per stroke at N.T.P.

$$= 0.01836 + 0.00178 = 0.02014 \text{ m}^3$$

Now, stroke volume (swept volume per stroke)

$$= \frac{\pi}{4} d^2 \times l = \frac{\pi}{4} \left(\frac{25}{100} \right)^2 \times \frac{48}{100} = 0.0236 \text{ m}^3$$

(a) Considering that the cylinder is occupied by air-gas mixture,

$$\text{Volumetric efficiency} = \frac{\text{Volume of air-gas mixture per stroke at N.T.P.}}{\text{Swept volume per stroke}}$$

$$= \frac{0.02014}{0.0236} = 0.847 \text{ or } 84.7\%$$

(b) Considering that the cylinder is occupied by air only,

$$\text{Volumetric efficiency} = \frac{\text{Volume of air per stroke at N.T.P.}}{\text{Swept volume per stroke}}$$

$$= \frac{0.01836}{0.0236} = 0.779 \text{ or } 77.9\%$$

Problem – 10 : The following results were obtained during a Morse test on a four-stroke cycle petrol engine :

Brake power developed with all cylinders working	.. 16.2 kW
Brake power developed with cylinder No.1 cut-out	.. 11.5 kW
Brake power developed with cylinder No.2 cut-out	.. 11.6 kW
Brake power developed with cylinder No.3 cut-out	.. 11.68 kW
Brake power developed with cylinder No.4 cut-out	.. 11.57 kW

Calculate the mechanical efficiency of the engine. What is the indicated thermal efficiency of the engine, if the engine uses 7 litres of petrol per hour of calorific value of 42,000 kJ/kg and the specific gravity of petrol is 0.72 ?

When one cylinder is cut-out, the net brake power that we obtain at the shaft is less than the sum of brake power developed by each of the three cylinders, because out of the total brake power developed, some brake power is used in overcoming the friction of the cylinder that is cut-out.

Let B_1 , B_2 , B_3 and B_4 be the brake power of cylinder No. 1, 2, 3 and 4 respectively, and F_1 , F_2 , F_3 , and F_4 be the friction power of cylinder No. 1, 2, 3 and 4 respectively.

Then, total brake power of the engine is

$$B_1 + B_2 + B_3 + B_4 = 16.2 \text{ kW with all cylinders working} \quad \dots (a)$$

$$- F_1 + B_2 + B_3 + B_4 = 11.5 \text{ kW with cylinder No. 1 cut-out} \quad \dots (b)$$

$$B_1 - F_2 + B_3 + B_4 = 11.6 \text{ kW with cylinder No. 2 cut-out} \quad \dots (c)$$

$$B_1 + B_2 - F_3 + B_4 = 11.68 \text{ kW with cylinder No. 3 cut-out} \quad \dots (d)$$

$$B_1 + B_2 + B_3 - F_4 = 11.57 \text{ kW with cylinder No. 4 cut-out} \quad \dots (e)$$

Subtracting by turn (b), (c), (d) and (e) from (a), we get,

$B_1 + F_1 = 16.2 - 11.5 = 4.70 \text{ kW}$	Indicated power ₁
$B_2 + F_2 = 16.2 - 11.6 = 4.60 \text{ kW}$	Indicated power ₂
$B_3 + F_3 = 16.2 - 11.68 = 4.52 \text{ kW}$	Indicated power ₃
$B_4 + F_4 = 16.2 - 11.57 = 4.63 \text{ kW}$	Indicated power ₄

Total indicated power developed = 18.45 kW

Mechanical efficiency, $\eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{16.2}{18.45} = 0.878$ or 87.8%

Indicated thermal efficiency, $\eta_i = \frac{\text{Indicated power} \times 3,600}{m_f \times \text{C.V.}}$
 $= \frac{18.45 \times 3,600}{(7 \times 0.72) \times 42,000} = 0.3138$ or 31.38%

Tutorial - 7

1. Delete the phrase which is not applicable in the following statements :

- (i) Indicated power of an I.C. engine is greater/smaller than brake power.
- (ii) Mechanical efficiency of an engine is $\frac{\text{Brake power}}{\text{Indicated power}} / \frac{\text{Indicated power}}{\text{Brake power}}$
- (iii) Indicated power of an I.C. engine is measured by an indicator / a dynamometer.
- (iv) Brake power of an I.C. engine is measured by an indicator / a dynamometer.
- (v) Morse test enables us to find the indicated power of a single-cylinder/a multi-cylinder I.C. engine without using an indicator.
- (vi) Number of cycles per min. in case of a four-stroke cycle, I.C. engine is equal to $\frac{N}{2} / N$, where N is r.p.m. of the engine.
- (vii) In case of a supercharged I.C. engine, the pressure during the suction stroke is higher/lower than the existing atmospheric pressure.
- (viii) The quantity of burnt gases left in the two-stroke cycle I.C. engine cylinder is more/less than that left in the four-stroke cycle engine cylinder.
- (ix) The warm-up performance of an air-cooled I.C. engine is poor/good as compared to a water-cooled engine.
- (x) For an I.C. engine, friction power increases/decreases with increase in the speed of the engine.

[Delete : (i) smaller, (ii) $\frac{\text{Indicated power}}{\text{Brake power}}$, (iii) a dynamometer, (iv) an indicator, (v) a single-cylinder, (vi) N, (vii) lower, (viii) less, (ix) poor, (x) decreases]

2. Fill in the blanks in the following statements :

- (i) The ratio of brake power to indicated power of an I.C. engine is known as _____ efficiency.
- (ii) Two-stroke cycle I.C. engine gives one working stroke for every _____ revolution of the crankshaft.
- (iii) Complete actual indicator diagram of an I.C. engine consists of _____ loops.
- (iv) In Diesel engines due to higher compression ratio, the temperature at the end of compression is sufficient to _____ the fuel oil which is injected at the end of compression stroke.
- (v) In petrol engine using fuel having fixed octane rating, increase in compression ratio will _____ the knocking tendency.

[(i) mechanical, (ii) one, (iii) two, (iv) ignite, (v) increase]

3. Indicate the correct answer by selecting the proper phrase in the following :

- (i) Morse test is used to determine the mechanical efficiency of
 - (a) single-cylinder S.I. engine,
 - (b) single-cylinder C.I. engine,
 - (c) multi-cylinder I.C. engine.
- (ii) An I.C. engine will develop maximum torque when it :
 - (a) develops maximum power,
 - (b) runs at maximum speed,
 - (c) runs at speed lower than that at which maximum power is developed.
- (iii) For part load operation,

- (a) C.I. engine is economical, (b) S.I. engine is economical, (c) both of the above engines are equally economical, (d) none of the above.
- (iv) The automobile engines generally utilise batteries having voltage of :
(a) 3 V, (b) 6 V, (c) 12 V, (d) 24 V.
- (v) In a Diesel engine, fuel injection pressure required is approximately :
(a) 25 bar, (b) 100 bar, (c) 500 bar, (d) 1,000 bar.
- (vi) For same power and same speed, the flywheel of a four-stroke cycle I.C. engine as compared to two-stroke cycle I.C. engine will be :
(a) smaller, (b) bigger, (c) of the same size.
- (vii) For a four-stroke cycle I.C. engine there is :
(a) one power stroke for every one revolution of the crankshaft,
(b) one power stroke for every two revolutions of the crankshaft,
(c) one power stroke for every four revolutions of the crankshaft,
(d) one power stroke for every half revolution of the crankshaft.
- (viii) Brake specific fuel consumption of a Diesel engine is generally :
(a) less than that of a petrol engine,
(b) more than that of a petrol engine,
(c) equal to that of a petrol engine,
(d) unpredictable.
- (ix) In petrol engines using petrol of fixed octane number, increase in compression ratio will :
(a) increase the knocking tendency,
(b) decrease the knocking tendency,
(c) have no effect on the knocking tendency.
- (x) A petrol engine develops maximum power when it is supplied with air-fuel ratio of :
(a) 17.5 to 18.5, (b) 16 to 17, (c) 12.5 to 13.5, (d) 10.5 to 11.5

[(i) c, (ii) c, (iii) a, (iv) c, (v) b, (vi) b, (vii) b, (viii) a, (ix) a, (x) c]

4. Describe briefly how you would conduct the indicated power test on a small I.C. engine, listing clearly all the observations you would take.

The following observations were recorded during a trial of a four-stroke cycle, single-cylinder oil engine : Duration of trial, 30 min.; Oil consumption, 5.5 litres; Calorific value of oil, 42,000 kJ/kg; Specific gravity of oil, 0.8; Average area of the indicator diagram, 8.4 cm²; Length of indicator diagram, 8.4 cm; Indicator spring scale, 550 kPa/cm; Brake load, 1,700 newtons; Spring balance reading, 200 newtons; Effective brake wheel diameter, 1.5 metres; Speed, 200 r.p.m.; Cylinder diameter, 30 cm; Stroke, 45 cm; Jacket cooling water, 11 kg per minute; Temperature rise of cylinder jacket cooling water, 36°C.

Calculate : (a) the indicated power, (b) the brake power, (c) the mechanical efficiency, (d) the specific fuel consumption in kg/kW-hr. based on brake power, and (e) the indicated thermal efficiency.

Draw up a heat balance sheet for the test on one minute basis in kJ.

[(a) 29.158 kW; (b) 23.562 kW; (c) 80.11% ; (d) 0.372 kg per kW-hr.; (e) 28.4%]

Heat supplied per min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel	6,160	(1) To Brake power	1,413.7
		(2) To Jacket cooling water	1,658.1
		(3) To exhaust, radiation, errors of observation, etc. (by difference)	3,088.2
Total	6,160	Total	6,160.0

5. Describe briefly how you would conduct the brake power test on a small I.C. engine, listing clearly all the observations you would take.

During the trial of a single-cylinder, four-stroke cycle oil engine, the following results were obtained : Cylinder diameter, 20 cm; Stroke, 40 cm; Indicated mean effective pressure, 600 kPa; Brake-torque, 415 N.m; Speed, 250 r.p.m.; Oil consumption, 5.25 litres per hour; Specific gravity of oil, 0.8; Calorific value of the fuel oil, 47,500 kJ/kg; Jacket cooling water, 4.5 kg per minute; Rise in temperature of jacket cooling water, 50°C; Air used per kg of oil, 31 kg; Temperature of exhaust gases, 400°C, Room temperature, 20°C; Mean specific heat of exhaust gases, 1.005 kJ/kg K.

Calculate, the indicated power, the brake power and the brake mean effective pressure and draw up a

heat balance sheet for the test in kJ/min. What are the principal heat losses which are not accounted for in the heat balance sheet ?

[15.708 kW; 10.865 kW; 415 kPa]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel	3,325	(1) To Brake power (2) To jacket cooling water (3) To exhaust gases (wet) (4) To radiation, errors of observation, etc. (by difference)	651.9 942.1 855.5 875.5
Total	3,325	Total	3,325.0

6. A trial carried out on a four-stroke cycle, single-cylinder oil engine working on Otto cycle gave the following results :

Cylinder diameter, 18 cm; stroke, 36 cm; Clearance volume, 1,830 cm³; Speed, 280 r.p.m.; Area of indicator diagram, 4.25 cm²; Length of indicator diagram, 6.25 cm; Spring strength, 1,000 kPa/cm; Net brake load, 600 newtons; Effective brake wheel diameter, 1.2 m; Fuel used per hour, 4.25 litres; Specific gravity of fuel oil, 0.8; Calorific value of fuel oil, 43,000 kJ/kg; Mass of Jacket cooling water, 7 kg/min; Rise in temperature of Jacket cooling water, 27°C; Air used per kg of fuel, 34 kg; Exhaust gas temperature, 410°C; Room temperature, 30°C; Specific heat of exhaust gases, 1.005 kJ/kg K.

Calculate : (a) the mechanical efficiency, (b) the indicated thermal efficiency, (c) the air-standard efficiency, and (d) the relative efficiency. Assume $\gamma = 1.4$ for air.

Draw up a heat balance sheet for the test in kJ/min.

[(a) $\eta_m = 72.62\%$; (b) $\eta_i = 35.79\%$; (c) A.S.E. = 51.15% ; (d) $\eta_r = 69.97\%$]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel	2,436.7	(1) To Brake power (2) To Jacket cooling water (3) To exhaust gases (wet) (4) To radiation, errors of observation, etc. (by difference)	633.4 791.3 757.4 254.6
Total	2,436.7	Total	2,436.7

7. In a test of an oil engine under full load condition the following results were obtained : indicated power, 33 kW; brake power, 26 kW; Fuel used, 10.5 litres per hour; Calorific value of fuel of oil, 43,000 kJ per kg; Specific gravity of fuel oil, 0.8; Inlet and outlet temperatures of cylinder jacket cooling water, 15°C and 70°C; Rate of flow of cylinder jacket cooling water, 7 kg per minute; Inlet and outlet temperatures of water to exhaust gas calorimeter, 15°C and 55°C; Rate of flow of water through exhaust gas calorimeter, 12.5 kg per minute; Final temperature of exhaust gases, 82°C; Room temperature, 17°C; Air-fuel ratio on mass basis, 20; Mean specific heat of exhaust gases including water vapour, 1.005 kJ/kg K. Draw up a heat balance sheet for the test in kJ per minute and estimate the thermal and mechanical efficiencies.

[$\eta_i = 32.89\%$; $\eta_b = 25.91\%$; $\eta_m = 78.79\%$]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel oil	6,020	(1) To Brake power (2) To jacket cooling water (3) To exhaust gases (wet) (4) To radiation, errors of observation, etc. (by difference)	1,560 1,612 2,285.6 562.4
Total	6,020	Total	6,020

8. A six-cylinder, four-stroke cycle, Diesel engine of 34 cm diameter and 38 cm stroke, gave the following results : r.p.m. 350; brake power 175 kW; i.m.e.p. 380 kPa; fuel used per hour 54 litres of calorific value 44,800 kJ/kg; specific gravity of fuel oil 0.815; hydrogen content in fuel 14% on mass basis; air consumption 38 kg/min.; jacket cooling water used 60.2 kg/min. with a temperature rise of 31°C; piston cooling oil of specific heat 2.1 kJ/kg K used, 32 kg/min. with a temperature rise of 20°C; exhaust gas temperature 190°C; room temperature 20°C; specific heat of dry exhaust gases 1.005 kJ/kg K; kp of steam in exhaust gases

2 kJ/kg K; partial pressure of steam in exhaust gases 0.07 bar. Calculate the mechanical efficiency of the engine and draw up a heat balance sheet in kJ per minute indicating the items which may include friction losses.

[$\eta_m = 76.8\%$]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel	32,860	(1) To brake power * (2) To jacket cooling water * (3) To piston cooling oil * (4) To dry exhaust gases * (5) To steam in exhaust gases * (6) To radiation, errors of observation, etc. (by difference)	10,500.0 7,813.8 1,344.0 6,460.4 2,579.3 4,162.5
Total	32,860	Total	32,860

* These items may include friction.]

9. A four-stroke cycle gas engine has a cylinder diameter of 27 cm and piston stroke of 45 cm. The effective diameter of the brake wheel is 1.62 metres. The observations made in a test of the engine were as follows: Duration of test 40 minutes; Total no. of revolutions 8,080; Total no of explosions 3,230; Net load on the brake 920 newtons; Indicated mean effective pressure 575 kPa; Gas used 7.7 m³; Pressure of gas at meter 130 mm of water above atmospheric pressure; Gas temperature 15°C; Height of barometer 750 mm of Hg; Calorific value of gas, 19,500 kJ/m³ at normal temperature (0°C) and pressure (760 mm Hg); Mass of jacket cooling water 183 kg; Rise in temperature of jacket cooling water 50°C.

Calculate the indicated power, brake power and draw up a heat balance sheet for the test in kJ per minute.

[19.938 kW; 15.764 kW]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of gas	3,556.8	(1) To brake power (2) To jacket cooling water (3) To exhaust, radiation, errors of observation, etc. (by difference)	945.8 957.8 1,653.2
Total	3,556.8	Total	3,556.8

10. The following results were obtained in a test on a gas engine : Gas used 0.125 m³ per minute at N.T.P.; Calorific value of gas 16,700 kJ/m³ at N.T.P.; Density of gas 0.64 kg per m³ at N.T.P.; Air used 1.52 kg per minute; Specific heat of exhaust gases 1.005 kJ/kg K; Temperature of exhaust gases 397°C; Room temperature 17°C; Jacket cooling water per minute 6 kg; Rise in temperature of Jacket cooling water 26°C; Indicated power 9.51 kW; Brake power 7.5 kW. Calculate the mechanical efficiency of the engine and draw up a heat balance sheet for the trial on one minute basis in kJ.

[$\eta_m = 78.95\%$]

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of gas	2,087.5	(1) To Brake power (2) To jacket cooling water (3) To exhaust gases (wet) (4) To radiation, errors of observation, etc. (by difference)	450 653.2 611 373.3
Total	2,087.5	Total	2,087.5

11. Describe briefly the method of determining the indicated power of a multi-cylinder petrol engine by cutting out one cylinder at a time. State the assumptions made.

A four-cylinder, four-stroke petrol engine is running on a brake having a radius of 1 metre. When all the four cylinders are firing, the r.p.m. is 1,400. The net brake load is 145 newtons. When spark plug of each cylinder is short circuited in turn, the net loads on the brake are 100, 103, 102 and 99.5 newtons respectively. The speed is maintained constant throughout the test. Estimate the indicated power and mechanical efficiency of the engine when all the cylinders are firing. If the cylinder bore is 9 cm and stroke is 12 cm, what is

brake mean effective pressure ?

[25.73 kW; 82.65% ; 596.3 kPa]

12. During a trial on a single-cylinder oil engine having cylinder diameter of 30 cm., stroke 45 cm, and working on the four-stroke cycle, the following observations were made :

Duration of trial one hour; total fuel oil used 8.1 kg; calorific value of fuel oil 44,800 kJ/kg; total no. of revolutions 12,600; mean effective pressure 690 kPa; net load on the brake 1,550 newtons; diameter of the brake wheel drum 1.78 metres; thickness of the brake belt 2 cm; jacket cooling water circulated 550 litres; inlet temperature of cooling water 16°C; outlet temperature of cooling water 61°C. Estimate the indicated thermal efficiency and brake thermal efficiency of engine. Draw up a percentage heat balance sheet for the trial.

[$\eta_i = 38.1\%$; $\eta_b = 30.34\%$]

Heat supplied per minute	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel oil	6,048	100	(1) To brake power	1,839.7	30.41
			(2) To jacket cooling water	1,727.7	28.56
			(3) To exhaust, radiation, errors of observations, etc. (by difference)	2,480.6	41.03
Total	6,048	100	Total	6,048	100

13. A trial of one hour duration on a petrol engine gave the following results :

Brake power 15 kW; Petrol consumption 6.4 litres; Specific gravity of petrol 0.74; Hydrogen content in petrol 15% on mass basis; Calorific value of petrol 44,400 kJ/kg; Fuel air-ratio 1 : 15; Temperature of exhaust gases 415°C; Room temperature 27°C; Specific heat of dry exhaust gases 1.005 kJ/kg K; Partial pressure of steam in exhaust gases 0.07 bar; Kp of steam 2.1 kJ/kg K; Mass of water passing through the cylinder jackets 270 litres; Rise in temperature of jacket cooling water 50°C.

At the end of the trial the engine was motored and the input power was 4 kW. Calculate the mechanical efficiency of engine and draw up a heat balance sheet for the trial on one minute basis and as percentage of the heat supplied to the engine.

[$\eta_m = 78.95\%$]

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel oil	3,504.6	100	(1) To brake power	900	25.68
			(2) To jacket cooling water	942.1	26.88
			(3) To dry exhaust gases	450.8	12.86
			(4) To steam in exhaust gases	347.6	9.92
			(5) To radiation, errors of observation, etc. (by difference)	864.1	24.66
Total	3,504.6	100	Total	3,504.6	100

14. A single-cylinder, four-stroke cycle gas engine has a bore to stroke ratio of 250/380 mm. During the trial the following results were noted :

Duration of trial 60 minutes; Effective brake load 1,300 newtons; Effective circumference of the brake wheel 3.8 metres; Total no. of revolutions 13,500; Total no. of explosions 6,000; Indicated m.e.p. 700 kPa; Total fuel gas used 16 m³; Temperature of fuel gas 15°C; Pressure of fuel gas above atmospheric pressure 200 mm of water; Barometer reading 742 mm of Hg; Calorific value of fuel gas at N.T.P. (0°C and 760 mm of Hg) 20,500 kJ/m³; Density of fuel gas at N.T.P. 0.8 kg/m³; Hydrogen content in fuel gas on mass basis 14%; Total mass of air used 210 kg; Exhaust gas temperature 400°C; Specific heat of dry exhaust gases 1.005 kJ/kg K; Kp of steam 2.1 kJ/kg K; Total mass of cylinder jacket cooling water 600 kg; Rise in temperature of jacket cooling water 35°C.

Draw up a heat balance sheet on one minute basis and as percentages of the heat supplied to the engine, assuming that the steam in the exhaust gases is at atmospheric pressure. Also calculate the indicated power, brake power and mechanical efficiency of the engine.

[21.762 kW; 18.525 kW; 85.13%]

Heat supplied/min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of gas	5,160	100	(1) To brake power	1,111.5	21.54
			(2) To jacket cooling water	1,465.5	28.41
			(3) To dry exhaust gases	1,334	25.85
			(4) To steam in exhaust gases	822.6	15.94
			(5) To radiation, errors of observation, etc. (by difference)	426.4	8.26
Total	5,160	100	Total	5,160	100

15. The following observations were made during a trial of a single-cylinder, four-stroke cycle gas engine having cylinder diameter 18 cm and stroke 24 cm :

Duration of trial one hour; Total number of revolutions 18,000; Total number of explosions 8,800; i.m.e.p. 590 kPa; Net load on the brake wheel 400 newtons; Effective diameter of brake wheel 1 metre; Total gas used at N.T.P. 4.5 m³; Calorific value of gas at N.T.P. 18,800 kJ/m³; Density of gas at N.T.P. 0.96 kg/m³; Total air used 71.25 m³; Pressure of air 720 mm of Hg; Temperature of air 15°C; Density of air at N.T.P. 1.293 kg/m³; Temperature of exhaust gases 350°C; Room temperature 15°C; Specific heat of exhaust gases 1.005 kJ/kg K; Total mass of cylinder jacket cooling water 160 kg; Rise in temperature of jacket cooling water 35°C.

Calculate the mechanical efficiency and indicated thermal efficiency of the engine. Also draw up a heat balance sheet in kJ on one minute basis.

[$\eta_m = 70.68\%$; $\eta_i = 37.83\%$]

Heat supplied./min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of gas	1,410	(1) To brake power	377.0
		(2) To jacket cooling water	390.8
		(3) To exhaust gases (wet)	488.5
		(4) To radiation, errors of observation, etc. (by difference)	153.7
Total	1,410	Total	1,410.0

16. The following readings were taken during a test on single-cylinder, four-stroke cycle oil engine :

Cylinder diameter	... 280 mm
Stroke length	... 425 mm
Gross i.m.e.p.	... 724 kPa
Pumping i.m.e.p.	... 40 kPa
Engine speed	... 200 r.p.m.
Net load on the brake	... 1,300 newtons
Effective diameter of the brake	... 1.6 meters
Fuel oil used per hour	... 9 kg
Calorific value of fuel oil	... 42,000 kJ/kg
Rate of jacket cooling water per minute	... 11 kg
Temperature rise of jacket cooling water	... 36°C
Mass of air supplied per kg of fuel oil	... 35 kg
Temperature of exhaust gases	... 375°C
Room temperature	... 15°C
Hydrogen content in fuel on mass basis	... 14%
Partial pressure of steam in exhaust gases	... 0.07 bar

Mean specific heat of dry exhaust gases

... 1.005 kJ/kg K

Kp of steam

... 2.1 kJ/kg K

Draw up a heat balance sheet in kJ/minute, indicating which items may include friction losses. Calculate also the indicated power, brake power, mechanical efficiency, indicated thermal efficiency and overall efficiency of the engine.

[29.833 kW; 21.784 kW; $\eta_m = 73\%$; $\eta_i = 28.14\%$; $\eta_b = 20.74\%$

Heat supplied/min.	kJ	Heat expenditure/min.	kJ
Heat supplied by combustion of fuel oil	6,300	(1) To Brake power * (2) To jacket cooling water * (3) To dry exhaust gases * (4) To steam in exhaust gases * (5) To radiation, errors of observation, etc. (by difference)	1,306.9 1,658.1 1,885.3 607.7 842.0
Total	6,300	Total	6,300

* These items may include friction losses]

17. A four-stroke cycle Diesel oil engine gave the following data during a trial of 50 minutes duration :

Brake power, 37 kW; Fuel used, 10 kg; Calorific value of fuel oil, 46,000 kJ/kg; Air used per kg of oil, 35; Temperature of exhaust gases, 380°C, Room temperature, 20°C; Specific heat of exhaust gases, 1.005 kJ/kg K; Mass of jacket cooling water circulated, 750 kg; Temperatures of jacket cooling water at inlet and outlet, 20°C and 70°C respectively. Draw up a heat balance sheet for the test

(i) in kJ per minute, (ii) in MJ per hour, (iii) in MJ per 50 minutes, and (iv) in kJ per kg of fuel oil.

[(i) Heat supplied, 9,200; Heat expenditure : To brake power, 2,220; To cooling water, 3,140; To exhaust, 2,605; To radiation, etc. (by diff.), 1,235

(ii) Heat supplied, 552; Heat expenditure : To B.P., 133.2; To cooling water, 188.4; To exhaust, 156.3; To radiation, etc. (by diff.), 74.1

(iii) Heat supplied, 460; Heat expenditure : To B.P., 111; To cooling water, 157; To exhaust, 130.25; To radiation, etc. (by diff.), 61.75

(iv) Heat supplied, 46,000; Heat expenditure : To B.P., 11,100; To cooling water, 15,700; To exhaust, 13,025; To radiation, etc. (by diff.), 6,175]

18. A two-stroke oil engine gave the following results at full load : Speed, 6 r.p.s.; Net brake load, 600 newtons; Effective brake wheel radius, 0.55 metre; Indicated mean effective pressure, 275 kPa; Fuel oil consumption, 4.25 kg per hour; Jacket cooling water, 480 kg per hour; Temperatures of Jacket cooling water at inlet and outlet, 20°C and 45°C; Temperature of exhaust gases, 370°C.

The following data also apply to the above test : Cylinder diameter, 22 cm; stroke, 28 cm; Calorific value of fuel oil, 42,000 kJ/kg; Hydrogen content in fuel oil, 15% on mass basis; Mean specific heat of dry exhaust gases, 1 kJ/kg K; Kp of steam, 2.1 kJ/kg K. Assume that the steam in exhaust gases exists as super-heated steam at atmospheric pressure and at exhaust gas temperature.

Calculate the indicated power, the brake power and draw up a heat balance sheet for the test in kJ/minute and as percentages of the heat supplied to the engine. Also calculate the indicated thermal efficiency and brake power fuel consumption in kg/kW-hr.

[17.569 kW; 12.434 kW; 35.43%; 0.3418 kg/kW-hr,

Heat supplied per min.	kJ	%	Heat expenditure/min.	kJ	%
Heat supplied by combustion of fuel oil	2,975	100	(1) To brake power (2) To jacket cooling water (3) To dry exhaust gases (4) To steam in exhaust gases (5) To radiation, errors of observation, etc. (by difference)	746.04 837.4 483.95 302.03 605.58	25.08 28.15 16.27 10.15 20.35
Total	2,975	100	Total	2,975	100.00

8

STEAM NOZZLES

8.1 Introduction

In the impulse steam turbine, the overall transformation of heat into mechanical work is accomplished in two distinct steps. The available energy of steam is first changed into kinetic energy, and this kinetic energy is then transformed into mechanical work. The first of these steps, viz., the transformation of available energy into kinetic energy is dealt with in this chapter.

A nozzle is a passage of varying cross-sectional area in which the potential energy of the steam is converted into kinetic energy. The increase of velocity of the steam jet at the exit of the nozzle is obtained due to decrease in enthalpy (total heat content) of the steam. The nozzle is so shaped that it will perform this conversion of energy with minimum loss.

8.2 General Forms of Nozzle Passages

A nozzle is an element whose primary function is to convert enthalpy (total heat) energy into kinetic energy. When the steam flows through a suitably shaped nozzle from zone of high pressure to one at low pressure, its velocity and specific volume both will increase.

The equation of the continuity of mass may be written thus :

$$m = \frac{AV}{v} = A \left(\frac{V}{v} \right) \quad \dots(8.1)$$

where m = mass flow in kg/sec.,

V = velocity of steam in m/sec.,

A = area of cross-section in m^2 , and

v = specific volume of steam in m^3/kg .

In order to allow the expansion to take place properly, the area at any section of the nozzle must be such that it will accommodate the steam whatever volume and velocity may prevail at that point.

As the mass flow (m) is same at all sections of the nozzle, area of cross-section (A) varies as $\frac{V}{v}$. The manner in which both V and v vary depends upon the properties of the substance flowing. Hence, the contour of the passage of nozzle depends upon the nature of the substance flowing.

For example, consider a *liquid*— a substance whose specific volume v remains almost constant with change of pressure. The value of $\frac{V}{v}$ will go on increasing with change of pressure. Thus, from eqn. (8.1), the area of cross-section should decrease with the decrease of pressure. Fig. 8-1(a) illustrates the proper contour of longitudinal section of

a nozzle suitable for liquid. This also can represent convergent nozzle for a fluid whose peculiarity is that while both velocity and specific volume increase, the rate of specific volume increase is less than that of the velocity, thus resulting in increasing value of $\frac{V}{v}$.

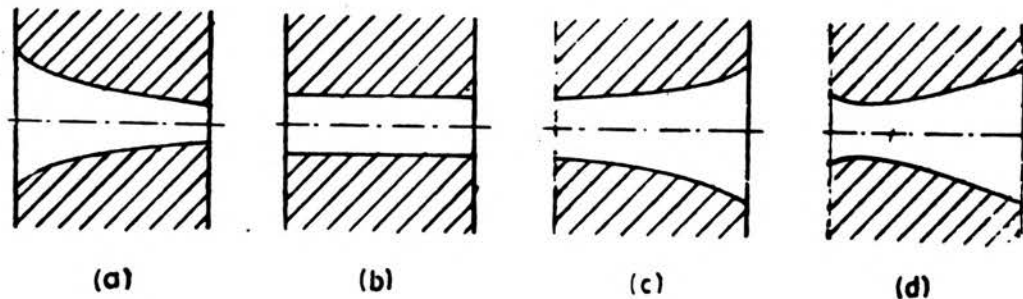


Fig. 8-1. General forms of Nozzles.

Fig. 8-1(b) represents the correct contour for some hypothetical substance for which both velocity and specific volume increase at the same rate, so that their ratio $\frac{V}{v}$ is a constant at all points. The area of cross-section should therefore, be constant at all points, and the nozzle becomes a plain tube.

Fig. 8-1(c) represents a divergent nozzle for a fluid whose peculiarity is that $\frac{V}{v}$ decreases with the drop of pressure, i.e., specific volume increases at a faster rate than velocity with the drop of pressure. The area of cross-section should increase as the pressure decreases.

Table 8-1

Properties of steam at various pressures when expanding dry saturated steam from 14 bar to 0.15 bar through a nozzle, assuming frictionless adiabatic flow.

Pressure p bar	Dryness fraction x	Enthalpy drop $H_1 - H_2$ kJ	Velocity V m/sec.	Specific Volume v_s m^3/kg	Discharge per unit area kg/m^2	Area A m^2	Diameter D metre
14	1.000	-	-	-	-	-	-
12	0.988	38.6	278	0.1633	1,723	0.00058	0.0272
10	0.974	84.1	410	0.1944	2,165	0.00046	0.0242
7	0.950	164.7	574	0.2729	*2,214	0.00045	x0.0239
3.5	0.908	309	786	0.5243	1,651	0.00061	0.0279
1.5	0.872	441.2	939	1.1593	929	0.0011	0.0374
0.70	0.840	555.6	1,054	2.365	531	0.00188	0.049
0.15	0.790	736.7	1,214	10.022	153	0.0065	0.091

* Maximum discharge per unit area

x Smallest diameter

Fig. 8-1(d) shows the general shape of convergent-divergent nozzle suitable for gases and vapours. It can be shown that in practice, while velocity and specific volume both increase from the start, velocity first increases faster than the specific volume, but after

a certain critical point, specific volume increases more rapidly than velocity. Hence the value of $\frac{V}{v}$ first increases to maximum and then decreases, necessitating a nozzle of *convergent-divergent* form. The above statement may be verified by referring to table 8-1, which shows the properties of steam at various pressures when expanding dry saturated steam from 14 bar to 0.15 bar through a nozzle, assuming frictionless adiabatic flow.

8.3 Steam Nozzles

The mass flow per second for wet steam, at a given pressure during expansion is given by

$$m = \frac{AV}{v} = \frac{AV}{xv_s} \text{ kg/sec.} \quad \dots(8.2)$$

where A = Area of cross-section in m^2 ,

V = Velocity of steam in m/sec,

v_s = Specific volume of dry saturated steam, m^3/kg ,

x = Dryness fraction of steam, and

$v = x v_s$ = Specific volume of wet steam, m^3/kg .

As the mass of steam per second (m) passing through any section of the nozzle must be constant, the area of cross-section (A) of nozzle will vary according to the variation of $\frac{V}{xv_s}$ i.e., product of A and $\frac{V}{xv_s}$ is constant. If the factor $\frac{V}{xv_s}$ increases with the drop in pressure, the cross-sectional area should decrease and hence a *convergent* shaped nozzle. The decrease of the factor $\frac{V}{xv_s}$ with pressure drop will require increasing cross-sectional area to maintain mass flow constant and hence the *divergent* shaped nozzle.

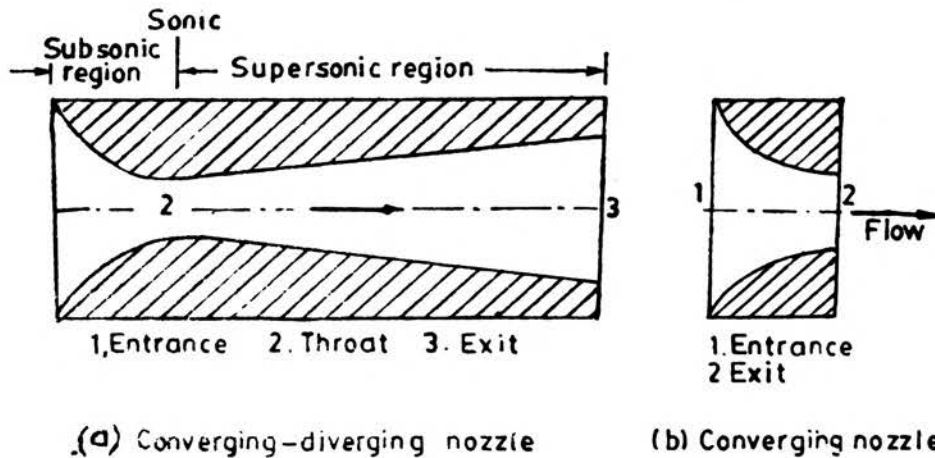


Fig. 8-2. Longitudinal sectional view of steam nozzles.

In practice at first the nozzle cross-section tapers to a smaller section in order to

allow for increasing value of $\frac{V}{xv_s}$; after this smallest diameter is reached, it will diverge to a larger cross-section. The smallest section of the nozzle is known as the *throat*.

A nozzle which first converges to throat and then diverges, as in fig. 8-2(a), is termed as *converging-diverging nozzle*. It is used for higher pressure ratio $\left(\frac{p_2}{p_1}\right)$.

Some form of nozzles finish at the throat and no diverging portion is fitted; this type shown in fig. 8-2(b), is known as *converging nozzle*. In this the greatest area is at the entrance and minimum area is at the exit which is also the throat of the nozzle. This nozzle is used when the pressure ratio, $\frac{P_2}{P_1}$ is less than 0.58 (critical).

8.4 Flow Through Steam Nozzles

From the point of view of thermodynamics, the steam flow through nozzles may be spoken as adiabatic expansion. During the flow of steam through the nozzle, heat is neither supplied nor rejected. Moreover, as the steam expands from high pressure to low pressure, the heat energy is converted into kinetic energy, i.e., work is done in expanding to increase the kinetic energy. Thus the expansion of steam through a nozzle is an adiabatic, and the flow of steam through nozzle is regarded as an adiabatic flow.

It should be noted that the expansion of steam through a nozzle is not a free expansion, and the steam is not throttled, because it has a large velocity at the end of the expansion. Work is done by the expanding steam in producing this kinetic energy.

In practice, some kinetic energy is lost in overcoming the friction between the steam and the side of the nozzle and also internal friction, which will tend to regenerate heat. The heat thus formed tends to dry the steam. About 10% to 15% of the enthalpy drop from inlet to exit is lost in friction. The effect of this friction, in resisting the flow and in drying the steam, must be taken into account in the design of steam nozzles, as it makes an appreciable difference in the results.

Another complication in the design of steam through a nozzle is due to a phenomenon known as *supersaturation*; this is due to a time lag in the condensation of the steam during the expansion. The expansion takes place very rapidly and if the steam is initially dry or superheated, it should become wet as the pressure falls, because the expansion is adiabatic. During expansion the steam does not have time to condense, but remains in an unnatural dry or superheated state, then at a certain instant, it suddenly condenses to its natural state. See illustrative problem no. 14.

Thus, the flow of steam through a nozzle may be regarded as either an ideal adiabatic (isentropic) flow, or adiabatic flow modified by friction and supersaturation.

If friction is negligible, three steps are essential in the process of expansion from pressure p_1 to p_2 :

(i) Driving of steam upto the nozzle inlet from the boiler. The 'flow-work' done on the steam is $p_1 v_1$ and results in similar volume of steam being forced through the exit to make room for fresh charge (steam).

(ii) Expansion of steam through the nozzle while pressure changes from p_1 to p_2 , the work done being $\frac{1}{n-1} (p_1 v_1 - p_2 v_2)$

where n is the index of the isentropic expansion,

v_1 = volume occupied by 1 kg of steam at entrance to nozzle, and

v_2 = volume occupied by 1 kg of steam as it leaves the nozzle.

Alternatively, this work done is equal to the change of internal energy, $\mu_1 - \mu_2$ as during isentropic expansion work is done at the cost of internal energy.

(iii) Displacement of the steam from the low pressure zone by an equal volume discharged from the nozzle. This work amounts to $p_2 v_2$ which is equal to the final flow work spent in forcing the steam out to make room for fresh charge (steam).

Thus, the new work done in increasing kinetic energy of the steam,

$$W = p_1 v_1 + \left[\frac{1}{n-1} (p_1 v_1 - p_2 v_2) \right] - p_2 v_2$$

$$W = \frac{n}{n-1} (p_1 v_1 - p_2 v_2) \quad \dots (8.3)$$

This is same as the work done during Rankine cycle.

Alternatively, $W = p_1 v_1 + (\mu_1 - \mu_2) - p_2 v_2$

$$= (p_1 v_1 + \mu_1) - (\mu_2 + p_2 v_2) = H_1 - H_2 \quad \dots(8.4)$$

where, H_1 and H_2 are the values of initial and final enthalpies allowing for the states of superheating or wetness as the case may be. This is exactly equivalent to the enthalpy drop equivalent to the work done during the Rankine cycle. The value of $H_1 - H_2$ may be found very rapidly from the Mollier chart ($H - \Phi$ chart) or more slowly but with greater accuracy from the steam tables.

In the design of steam nozzles the calculations to be made are :

- (i) the actual velocity attained by the steam at the exit,
- (ii) the minimum cross-sectional area (throat area) required for a given mass flow per second,
- (iii) the exit area, if the nozzle is converging-diverging, and
- (iv) the general shape of the nozzle – axial length.

8.4.1 Velocity of steam leaving nozzle : The gain of kinetic energy is equal to the enthalpy drop of the steam. The initial velocity of the steam entering the nozzle (or velocity of approach) may be neglected as being relatively very small compared with exit velocity.

For isentropic (frictionless adiabatic) flow and considering one kilogram of steam

$$\frac{V^2}{2 \times 1,000} = H_1 - H_2 = H$$

where H is enthalpy drop in kJ/kg and V = velocity of steam leaving the nozzle in m/sec.

$$\therefore V = \sqrt{2 \times 1,000H} = 44.72 \sqrt{H} \text{ m/sec.} \quad \dots (8.5)$$

Let the available enthalpy drop after deducting frictional loss be kH ,

i.e. $(1 - k) H$ is the friction loss,

Then, $V = 44.72 \sqrt{kH} \text{ m/sec.} \quad \dots (8.6)$

If the frictional loss in the nozzle is 15 per cent of the enthalpy drop, then $k = 0.85$.

8.4.2 Mass of steam discharged : The mass flow of steam in kg per second through a cross-sectional area A and at a pressure p_2 can be written as

$$m = \frac{AV_2}{v_2} \quad \text{where } v_2 = \text{specific volume of steam at pressure } p_2.$$

$$\text{But } v_2 = v_1 \left(\frac{p_1}{p_2} \right)^{\frac{1}{n}} = v_1 \left(\frac{p_2}{p_1} \right)^{-\frac{1}{n}} \quad \dots(8.7)$$

where, v_1 = specific volume of steam at pressure p_1 .

Using the value of velocity V from eqns. (8.3.) and (8.5),

$$m = \frac{A}{v_2} \sqrt{\left[2,000 \frac{n}{n-1} (p_1 v_1 - p_2 v_2) \right]} = \frac{A}{v_2} \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left[1 - \frac{p_2 v_2}{p_1 v_1} \right]}$$

Putting the value of v_2 from eqn. (8.7), we get,

$$m = \frac{A}{v_1 \left(\frac{p_2}{p_1} \right)^{-\frac{1}{n}}} \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \right]}$$

$$m = A \sqrt{2,000 \frac{n}{n-1} \times \frac{p_1}{v_1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1} \right)^{\frac{n+1}{n}} \right]} \quad \dots (8.8)$$

8.4.3 Critical pressure ratio : Using eqn. (8.8), the rate of mass flow per unit area is given by

$$\frac{m}{A} = \sqrt{2,000 \frac{n}{n-1} \times \frac{p_1}{v_1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1} \right)^{\frac{n+1}{n}} \right]}$$

The mass flow per unit area has the maximum value at the throat which has minimum area, the value of pressure ratio $\left(\frac{p_2}{p_1} \right)$ at the throat can be evaluated from the above expression corresponding to the maximum value of $\frac{m}{A}$.

All the items of this equation are constant with the exception of the ratio $\left(\frac{p_2}{p_1} \right)$.

Hence, $\frac{m}{A}$ is maximum when $\left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1} \right)^{\frac{n+1}{n}} \right]$ is the maximum.

Differentiating the above expression with respect to $\left(\frac{p_2}{p_1} \right)$ and equating to zero for a maximum discharge per unit area

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

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

$$\text{Hence, } \left[\frac{p_2}{p_1} \right]^{\frac{2-n}{n}} = \frac{n+1}{n} \left[\frac{p_2}{p_1} \right]^{\frac{1}{n}} \text{ or } \left[\frac{p_2}{p_1} \right]^{2-n} = \left[\frac{n+1}{2} \right]^n \left(\frac{p_2}{p_1} \right)$$

$$\text{from which } \left[\frac{p_2}{p_1} \right]^{1-n} = \left[\frac{n+1}{2} \right]^n \text{ or } \frac{p_2}{p_1} = \left[\frac{2}{n+1} \right]^{\frac{n}{n-1}} \quad \dots (8.9)$$

$\frac{p_2}{p_1}$ is known as *critical pressure ratio* and depends upon the value of index n .

The following approximate values of index n and corresponding values of critical pressure ratios may be noted :

Initial condition of steam	Value of index n for isentropic expansion	Nozzle critical pressure ratio $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$
Superheated or supersaturated	1.300	0.546
Dry saturated	1.135	0.578
Wet	1.113	0.582

Dr. Zeuner has suggested a well known equation for value of n in the adiabatic expansion of steam viz. $n = 1.035 + 0.1x_1$, where x_1 is the initial dryness fraction of steam.

The eqn. (8.9) gives the ratio between the throat pressure (p_2) and the inlet pressure (p_1) for a maximum discharge per unit area through the nozzle. The mass flow being constant for all sections of nozzle, maximum discharge per unit area occurs at the section

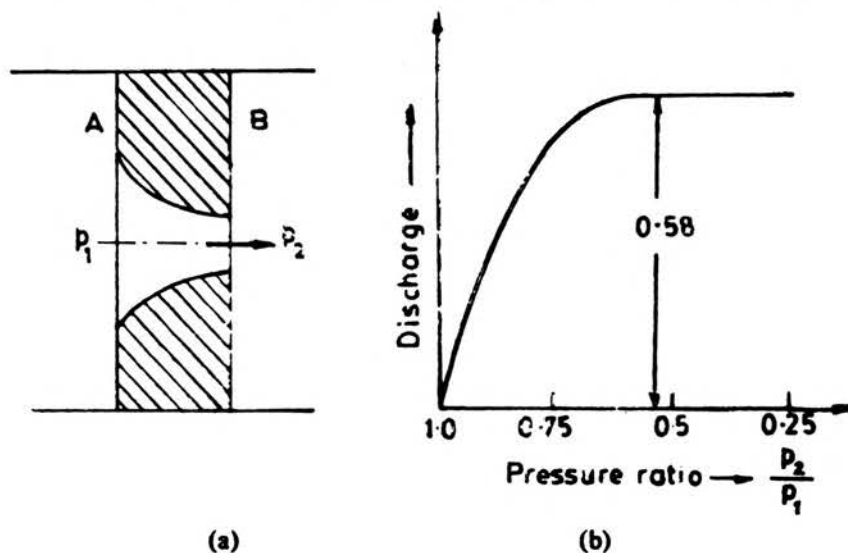


Fig. 8-3.

having minimum area, i.e., at the throat. The area of throat of all steam nozzle should be designed on this ratio. This pressure ratio at the throat is known as *critical pressure ratio*. The pressure at which the area is minimum and discharge per unit area is maximum is termed as the *critical pressure*.

The implication of the existence of a critical pressure in nozzle flow may be expressed in another way. Suppose we have two vessels A and B. A containing steam at a high and steady pressure p_1 . Suppose that the pressure in B may be varied at will. A and B are connected by a diaphragm containing a convergent nozzle, as shown in fig. 8-3(a).

Assume at first that p_2 is equal to p_1 , then there is no flow of steam through the nozzle. Now let p_2 be gradually reduced. The discharge m through the nozzle will increase as shown by the curve of fig. 8-3(b). As the pressure p_2 approaches the critical value, the discharge rate gradually approaches its maximum value, and when p_2 is reduced below the critical value, the discharge rate does not increase but remains at the same value as that at the critical pressure. The extraordinary result that p_2 can be reduced

well below the critical pressure without influencing the mass flow was first discovered by R.D. Napier.

Another explanation can be visualised as follows : the critical pressure will give velocity of steam at the throat equal to the velocity of the sound (*sonic velocity*). The flow of steam in the convergent portion of the nozzle is *sub-sonic*. Thus, to increase the velocity of steam above sonic velocity (*super sonic*) by expanding steam below critical pressure, divergent portion is necessary [fig. 8-2(a)].

8.4.4 Areas of throat and exit for maximum discharge : The first step is to estimate the critical pressure or throat pressure for the given initial condition of steam.

(1) If the nozzle is *convergent*, the nozzle terminates at the throat, hence the throat is the exit end or mouth of the nozzle.

Next, using the Mollier ($H - \Phi$) chart, the enthalpy drop can be calculated by drawing a vertical line to represent the isentropic expansion from p_1 to p_2 (p_2 is throat pressure). Read off from the $H - \Phi$ chart the value of enthalpies H_1 and H_2 or enthalpy drop $H_1 - H_2$ and dryness fraction x_2 as shown in fig. 8-4.

Then, for throat, enthalpy drop from entry to throat, $H_t = H_1 - H_2$ kJ/kg, and velocity at throat, $V_2 = 44.72 \sqrt{H_t}$ m/sec.

$$\text{Then, mass flow, } m = \frac{A_2 V_2}{x_2 v_{s2}} \text{ kg/sec. (if steam is wet at throat)} \quad \dots (8.10)$$

where $A_2 =$ throat area.

The value of v_{s2} (specific volume of dry saturated steam) at pressure p_2 can be obtained from the steam tables.

$$\text{If the steam is superheated at throat, } m = \frac{A_2 V_2}{v_{sup}} \text{ kg/sec.} \quad \dots (8.11)$$

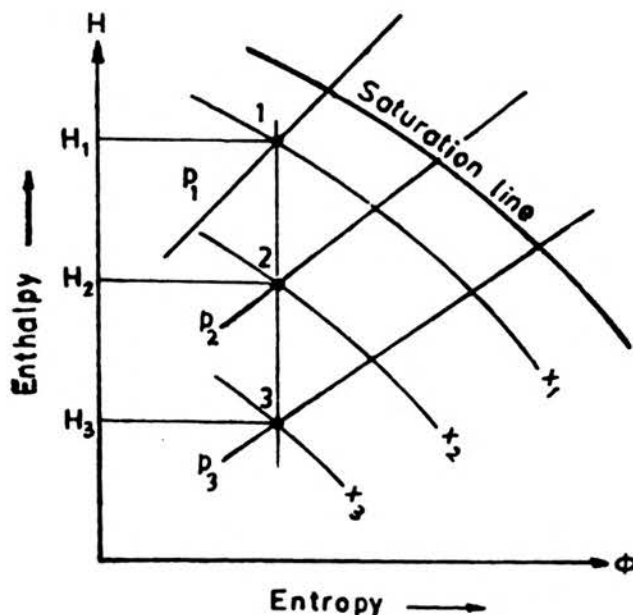


Fig. 8-4. $H - \phi$ diagram.

$$\text{where, } v_{sup} = v_{s2} \left(\frac{T_{sup 2}}{T_{s2}} \right)$$

As the mass of discharge m is known, the area A_2 (throat area) can be calculated.

(2) If the nozzle is *convergent-divergent*, calculation of throat area is the same as for the convergent nozzle in which case the value of p_2 is critical pressure. As the back pressure in this nozzle is lower than critical pressure, the vertical line on the $H - \Phi$ chart is extended up to the given back pressure p_3 at the exit as shown in fig. 8-4.

The value of enthalpy H_3 and the dryness fraction x_3 at exit are read off directly from the $H - \Phi$ chart.

For the exit or mouth of the nozzle, enthalpy drop from entry to exit,

$$H_e = H_1 - H_3 \text{ kJ/kg and velocity at exit,}$$

$$V_3 = 44.72 \sqrt{H_e} \text{ m/sec.}$$

Then, mass flow, $m = \frac{A_3 V_3}{x_3 v_{s3}}$ kg/sec.

... (8.12)

The value of v_{s3} at pressure p_3 can be obtained directly from the steam tables. As the mass of discharge m is known, the exit area A_3 can be calculated by using eqn. (8.12).

Similarly, for any pressure p along the nozzle axis, steam velocity and then the cross-sectional area can be evaluated.

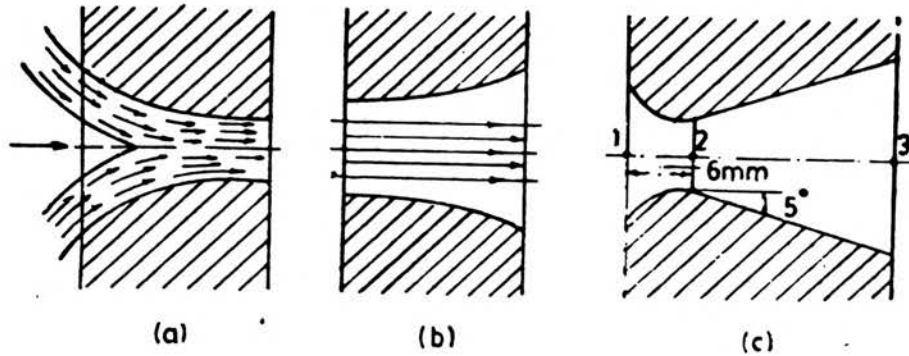


Fig. 8-5.

8.4.5 Length of nozzle : The length of the convergent portion should be short in order to reduce the surface friction, and normally a length of about 6 mm will be found adequate. This rapid change in the area is possible because the convergence of the walls of a passage tends to stabilize the flow as shown in fig. 8-5 (a).

In the divergent portion, high velocity steam has tendency, on account of inertia, to flow along the axis in a form of a circular jet of sectional area equal to throat area. If the divergence is rapid, steam will not occupy the increased area provided. Thus, steam may pass out through the divergent point without drop of pressure as shown in fig. 8.5(b). To avoid this, divergent portion should have sufficient length so that steam has enough time to occupy the full cross-sectional area provided, thus resulting in desired drop of pressure and increase in kinetic energy. This necessitates gradual increase in area. It is found satisfactory in practice to make the length of the nozzle from throat to exit such that the included cone angle is about 10° as shown in fig. 8-5(c).

Problem – 1 : A convergent-divergent nozzle for a steam turbine has to deliver steam under a supply condition of 11 bar with 100°C superheat and a back pressure of 0.15 bar. If the outlet area of the nozzle is 9.7 cm^2 , determine using steam tables, the mass of steam discharged per hour. If the turbine converts 60% of the total enthalpy drop into useful work, determine the power delivered by the turbine. Neglect the effect of friction in the nozzle. Take K_p of superheated steam as 2.3 kJ/kg K .

From Steam Tables

p bar	t_s $^\circ\text{C}$	v_s m^3/kg	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
11	184.09	—	781.34	2,000.4	2,781.7	2.1792	6.5536
0.15	53.97	10.022	225.94	2,373.1	2,599.1	0.7549	8.0085

Let suffixes 1 and 3 represent conditions at entry and exit of the nozzle.

Entropy before expansion = Entropy after expansion

$$\Phi_1 = \Phi_3$$

$$\text{i.e. } \Phi_{s1} + k_p \log_e \frac{T_{sup}}{T_s} = \Phi_{w3} + x_3 (\Phi_{s3} - \Phi_{w3})$$

$$\text{i.e., } 6.5536 + 2.3 \log_e \frac{184.09 + 100 + 273}{184.09 + 273} = 0.7549 + x_3 (8.0085 - 0.7549)$$

$$\therefore 6.5536 + 2.3 \times 0.1976 = 0.7549 + 7.2536 \times x_3$$

$$\therefore x_3 = \frac{6.5536 + 0.4545 - 0.7549}{7.2536} = \frac{6.2432}{7.2536}$$

i.e., $x_3 = 0.862$ (dryness fraction at exit)

$$\text{Enthalpy, } H_1 = H_s + k_p (T_{sup} - T_s) = 2,781.7 + 2.3 (100) = 3,011.7 \text{ kJ/kg}$$

$$\text{Enthalpy, } H_3 = h_3 + x_3 L_3 = 225.94 + 0.862 \times 2,373.1 = 2,271.6 \text{ kJ/kg}$$

Enthalpy drop from inlet to exit,

$$H_e = H_1 - H_3 = 3,011.7 - 2,271.6 = 740.1 \text{ kJ/kg.}$$

$$\text{Using eqn. (8.5), Velocity at exit, } V_3 = 44.72 \sqrt{H_e} = 44.72 \sqrt{740.1} = 1,216.6 \text{ m/sec.}$$

$$\text{For mass continuity, } m = \frac{A_3 V_3}{v_3} \quad [\text{Eqn. (8.2)}]$$

$$= \frac{9.7 \times 1,216.6}{10^4 \times (10.022 \times 0.862)} = 0.1366 \text{ kg/sec.}$$

Mass of steam discharged per hour = $0.1366 \times 3,600 = 491.76 \text{ kg/hour.}$

Useful work done per kg of steam = $0.6 \times 740.1 = 444.06 \text{ kJ/kg.}$

Power delivered = $444.06 \times 0.1366 = 60.66 \text{ kJ/sec. or } 60.66 \text{ kW}$

Note : The enthalpy drop from inlet to exit ($H_1 - H_3$) and the final dryness fraction (x_3) can be found directly from $H - \Phi$ chart by the method as shown in fig. 8-6.

Problem - 2 : A convergent-divergent nozzle is required to discharge 350 kg of steam per hour. The nozzle is supplied with steam at 8.5 bar and 90% dry and discharges against a back pressure of 0.4 bar. Neglecting the effect of friction, find the throat and exit diameters.

Let suffixes, 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle respectively as shown in fig. 8-6.

As the steam is initially wet, critical or throat pressure,

$$p_2 = 0.582 \times p_1 = 0.582 \times 8.5 = 4.95 \text{ bar.}$$

As shown in fig. 8-6, vertical line 1-2-3 is drawn. The values read off from the $H - \Phi$ chart (Mollier chart) are :

$$\text{Enthalpy drop from entry to throat, } H_t = H_1 - H_2 = 102 \text{ kJ/kg,}$$

$$\text{Enthalpy drop from entry to exit, } H_e = H_1 - H_3 = 456 \text{ kJ/kg,}$$

$$\text{Dryness fraction of steam at throat, } x_2 = 0.87 \text{ and}$$

$$\text{Dryness fraction of steam at exit, } x_3 = 0.777$$

$$\text{Velocity at throat, } V_2 = 44.72 \sqrt{H_t} = 44.72 \sqrt{102} = 452 \text{ m/sec.} \quad [\text{eqn. (8.5)}]$$

Specific volume of dry saturated steam at 4.95 bar (by arithmetical interpolation from steam tables), $v_{s2} = 0.3785 \text{ m}^3/\text{kg.}$

\therefore Actual volume of wet steam at throat,

$$v_2 = x_2 \times v_{s2} = 0.87 \times 0.3785 = 0.33 \text{ m}^3/\text{kg.}$$

For mass continuity, $m = \frac{A_2 V_2}{v_2}$ kg/sec.

[eqn. (8.2)]

$$\text{i.e., } \frac{350}{3,600} = \frac{A_2 \times 452}{0.33 \times 10^4}$$

$$\therefore A_2 = \frac{350 \times 0.33 \times 10^4}{3,600 \times 452} = 0.71 \text{ cm}^2$$

$$\therefore \text{Throat diameter, } D_2 = \sqrt{\frac{0.71 \times 4}{\pi}} = 0.951 \text{ cm i.e., } 9.51 \text{ mm}$$

Similarly velocity at exit,

$$V_3 = 44.72 \sqrt{456} = 955 \text{ m/sec.}$$

Specific volume of dry saturated steam at 0.4 bar (from steam tables),

$$v_{s3} = 3.993 \text{ m}^3/\text{kg.}$$

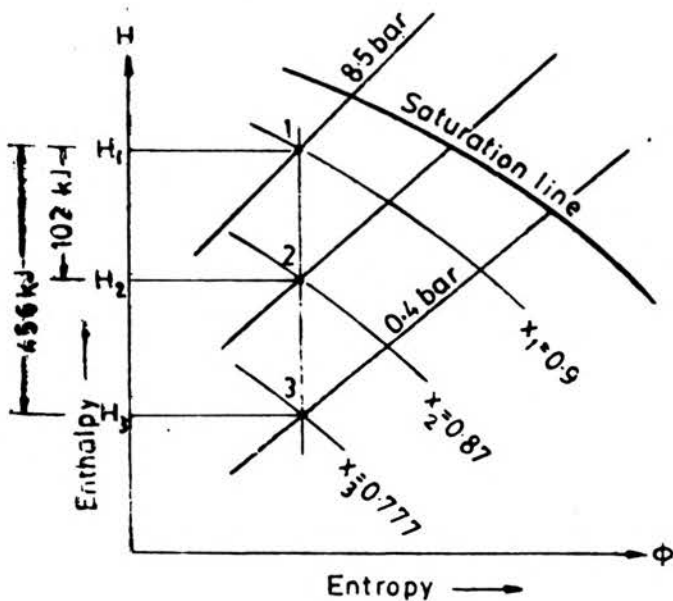


Fig. 8-6. H - φ diagram.

\therefore Actual volume of wet steam at exit, $v_3 = x_3 \times v_{s3}$

$$= 0.777 \times 3.993 = 3.11 \text{ m}^3/\text{kg.}$$

Again for mass continuity,

$$m = \frac{A_3 V_3}{v_3} \quad [\text{eqn. (8.2)}]$$

$$\text{i.e., } \frac{350}{3,600} = \frac{A_3 \times 955}{3.11 \times 10^4}$$

$$\therefore A_3 = \frac{350 \times 3.11 \times 10^4}{955 \times 3,600} = 3.16 \text{ cm}^2$$

\therefore Exit diameter, $D_3 =$

$$\sqrt{\frac{3.16 \times 4}{\pi}} = 2.01 \text{ cm i.e., } 20.1 \text{ mm}$$

Problem - 3 : An impulse turbine which is to develop 175 kW with probable steam consumption of 11 kg per kW-hour is supplied with dry saturated steam at 10 bar. Find the number of nozzles each of about 6 mm diameter at the throat that will be required for the purpose and estimate the exact diameters at the throat and exit of the nozzles. The condenser pressure is 0.15 bar. Neglect the effect of friction in nozzles. Assume index of expansion as 1.135.

(This problem is a continuation of the previous one and its solution is provided in the next page.)

Let suffixes, 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle.

$$\text{From eqn. (8.9), } \frac{p_2}{p_1} = \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}}$$

$$\text{Putting } n = 1.135, \frac{p_2}{p_1} = \left(\frac{2}{2.135} \right)^{\frac{1.135}{0.135}} = (0.936)^{8.4} = 0.578$$

Critical or throat pressure, $p_2 = 0.578 \times p_1 = 0.578 \times 10 = 5.78 \text{ bar.}$

Enthalpy drop from entry to throat, $H_t = H_1 - H_2 = 122 \text{ kJ/kg}$ and

Dryness fraction of steam at throat, $x_2 = 0.957$ (from $H - \Phi$ chart).

Velocity at throat, $V_2 = 44.72 \sqrt{H_t} = 44.72 \sqrt{122} = 494$ m/sec.,

From steam tables at 5.78 bar, $v_{s2} = 0.327$ m³/kg by arithmetical interpolation.

Specific volume at throat, $v_2 = x_2 \times v_{s2} = 0.957 \times 0.327$ m³/kg.

$$\text{For mass continuity, } m = \frac{A_2 V_2}{v_2} = \frac{\frac{\pi}{4} \left(\frac{6}{10}\right)^2 \times 494}{10^4 \times 0.957 \times 0.327} = 0.0446 \text{ kg/sec.}$$

$$\text{Steam consumption per sec.} = \frac{11 \times 175}{3,600} = 0.5347 \text{ kg/sec.}$$

$$\therefore \text{Number of nozzles required} = \frac{0.5347}{0.0446} = 11.99 \text{ say } 12$$

$$\therefore \text{Exact diameter at throat, } D_2 = 6 \sqrt{\frac{11.99}{12}} = 5.997 \text{ mm.}$$

For exit :

Enthalpy drop from entry to exit, $H_e = H_1 - H_3 = 655$ kJ/kg and

Dryness fraction of steam at exit, $x_3 = 0.85$ (from $H - \Phi$ chart).

\therefore Velocity at exit, $V_3 = 44.72 \sqrt{H_e} = 44.72 \sqrt{655} = 1,145$ m/sec.

From steam tables at 0.15 bar, $v_{s3} = 10.022$ m³/kg.

\therefore Taking the number of nozzles as 12,

$$\text{Mass of steam per nozzle} = \frac{0.5347}{12} = 0.0446 \text{ kg/sec.}$$

$$\text{Again for mass continuity, } m = \frac{A_3 V_3}{v_3} \text{ or } A_3 = \frac{m v_3}{V_3} = \frac{m \times v_{s3} \times x_3}{V_3}$$

$$\therefore A_3 = \frac{0.0446 \times (10.022 \times 0.85) \times 10^4}{1,145} = 3.646 \text{ cm}^2.$$

$$\therefore \text{Exact diameter at exit, } D_3 = \sqrt{\frac{3.646 \times 4}{\pi}} = 2.155 \text{ cm i.e., } 21.55 \text{ mm.}$$

Problem - 4 : Steam expands from 17 bar and 80°C superheat to 0.7 bar in a

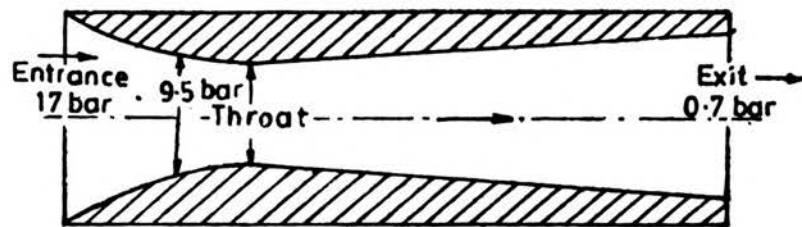


Fig. 8-7

convergent-divergent nozzle. Assuming that the expansion is frictionless adiabatic, and the steam discharged is 0.25 kg/sec., calculate the diameters of the sections of nozzle (i) at a point where the pressure is 9.5 bar, and (ii) at exit.

Take K_p of superheated steam as 2.3 kJ/kg K.

Referring to fig. 8-7,

Let suffixes, 1, 2 and 3 represent conditions at entrance, section of the nozzle where pressure is 9.5 bar and exit respectively.

As the steam is initially superheated,

critical or throat pressure = $0.546 \times p_1 = 0.546 \times 17 = 9.28$ bar.

It means that the nozzle is still converging where the pressure is 9.5 bar.

(1) For section of the nozzle where the pressure is 9.5 bar :

Enthalpy drop from entry to section of nozzle, where the pressure is 9.5 bar,

$H_1 - H_2 = 140$ kJ/kg (from $H - \Phi$ chart);

Temperature of steam, $t_2 = 213^\circ\text{C}$ (from $H - \Phi$ chart).

At 9.5 bar, saturation temperature, $t_s = 177.69^\circ\text{C}$ (from steam tables).

\therefore Steam at section where pressure is 9.5 bar is superheated, i.e., steam is still superheated after expansion.

At 9.5 bar, $v_{s2} = 0.2042$ m³/kg (from steam tables).

Specific volume at 9.5 bar and 213°C ,

$$v_2 = v_{s2} \times \frac{T_{\text{sup}2}}{T_{\text{sat}2}} = 0.2042 \times \frac{(213 + 273)}{(177.69 + 273)} = 0.22 \text{ m}^3/\text{kg}$$

Velocity at section, where pressure is 9.5 bar, $V_2 = 44.72 \sqrt{140} = 529$ m/sec.

For mass continuity, $m = \frac{A_2 V_2}{v_2}$

$$\text{i.e., } 0.25 = \frac{\frac{\pi}{4} (D_2)^2 \times 529}{0.22 \times 10^4}$$

$$\therefore (D_2)^2 = \frac{0.25 \times 0.22 \times 10^4 \times 4}{\pi \times 529} = 1.324$$

\therefore Diameter, $D_2 = \sqrt{1.324} = 1.15$ cm i.e., **11.5 mm**

Diameter of the section of the nozzle at a point where the pressure is 9.5 bar = 11.5 mm.

(ii) For exit :

From $H - \Phi$ chart, Enthalpy drop from inlet to exit, $H_e = H_1 - H_3 = 600$ kJ/kg and dryness fraction, $x_3 = 0.89$.

Velocity at exit, $V_3 = 44.72 \sqrt{H_e} = 44.72 \sqrt{600} = 1,095$ m/sec.

From steam tables, at 0.7 bar, $v_{s3} = 2.365$ m³/kg.

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0.89 \times 2.365$ m³/kg.

For mass continuity, $m = \frac{A_3 V_3}{v_3}$

$$\therefore A_3 = \frac{\pi}{4} (D_3)^2 \times \frac{1}{10^4} = \frac{m \times v_3}{V_3} = \frac{0.25 \times (0.89 \times 2.365)}{1,095}$$

$$\therefore (D_3)^2 = \frac{0.25 \times 0.89 \times 2.365 \times 10^4 \times 4}{\pi \times 1,095} = 6.12$$

\therefore Exit diameter, $D_3 = \sqrt{6.12} = 2.47$ cm, i.e., **24.7 mm**.

Problem – 5 : A convergent-divergent nozzle is supplied with dry saturated steam at 11 bar. If the divergent portion of the nozzle is 6 cm long and the throat diameter is

8 mm, determine the semi-cone angle of the divergent part of the nozzle so that the steam may leave the nozzle at 0.4 bar. Neglect the effect of friction in the nozzle.

For throat :

Let suffixes, 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle.

As the steam supplied is initially dry saturated,

Critical or throat pressure, $p_2 = 0.578 \times p_1 = 0.578 \times 11 = 6.36$ bar.

From $H - \Phi$ chart, Enthalpy drop from entry to throat, $H_t = H_1 - H_2 = 113$ kJ/kg and dryness fraction of steam at throat, $x_2 = 0.96$.

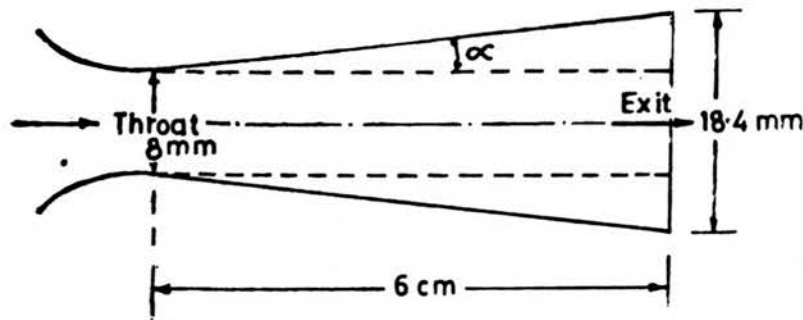


Fig. 8-8.

Velocity at throat,

$$\begin{aligned} V_2 &= 44.72 \sqrt{H_t} \\ &= 44.72 \times 113 \\ &= 475 \text{ m/sec.} \end{aligned}$$

From steam tables at 6.36 bar, $v_{s2} = 0.2995$ m³/kg.

\therefore Specific volume at throat, $v_2 = x_2 \times v_{s2} = 0.96 \times 0.2995$ m³/kg.

For mass continuity,

$$m = \frac{A_2 V_2}{v_2} = \frac{\frac{\pi}{4} \left(\frac{8}{10}\right)^2 \times 475}{0.96 \times 0.2995 \times 10^4} = 0.083 \text{ kg/sec.}$$

For exit :

From $H - \Phi$ chart, Enthalpy drop from entry to exit, $H_e = H_1 - H_3 = 540$ kJ/kg and dryness fraction of steam at exit, $x_3 = 0.834$

Velocity at exit, $V_3 = 44.72 \sqrt{H_e} = 44.72 \sqrt{540} = 1,039$ m/sec.

From steam tables at 0.4 bar, $v_{s3} = 3.993$ m³/kg

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0.834 \times 3.993$ m³/kg.

For mass continuity, $m = \frac{A_3 V_3}{v_3}$

$$\therefore A_3 = \frac{\pi}{4} (D_3)^2 \times \frac{1}{10^4} = \frac{m \times v_3}{V_3} = \frac{0.083 \times (0.834 \times 3.993)}{1,039}$$

$$\therefore (D_3)^2 = \frac{0.083 \times 0.834 \times 3.993 \times 10^4 \times 4}{\pi \times 1,039} = 3.39$$

\therefore Exit diameter, $D_3 = \sqrt{3.39} = 1.84$ cm, i.e., 18.4 mm.

$$\text{Referring to fig. 8-8, } \tan \alpha = \frac{D_3 - D_2}{l} = \frac{18.4 - 8}{6 \times 10} = 0.0866$$

\therefore Semi-cone angle, $\alpha = 4^\circ - 57'$

Problem - 6 : Dry saturated steam at a pressure of 8.5 bar enters a convergent-divergent nozzle, and leaves at a pressure of 1.5 bar.

If the flow is frictionless adiabatic and the corresponding expansion index is 1.135, find using steam tables, the ratio of the cross-sectional area at exit to that at the throat.

Let suffixes 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle respectively.

$$\text{From eqn. (8.9), } \frac{p_2}{p_1} = \left(\frac{2}{n+1} \right)^{\frac{n}{n-1}}$$

$$\text{Putting } n = 1.135, \frac{p_2}{p_1} = \left(\frac{2}{2.135} \right)^{\frac{1.135}{0.135}} = (0.936)^{8.4} = 0.578$$

$$\therefore p_2 = 0.578 \times p_1 = 0.578 \times 8.5 = 4.91 \text{ bar}$$

At 8.5 bar, from steam tables, $\Phi_s = 6.6421 \text{ kJ/kg K}$, $H = 2,771.6 \text{ kJ/kg}$,

At 4.91 bar, from steam tables by arithmetical interpolation, $h = 636 \text{ kJ/kg}$
 $L = 2,110 \text{ kJ/kg}$, $\Phi_w = 1.8550 \text{ kJ/kg K}$, $\Phi_s = 6.860 \text{ kJ/kg K}$, $v_s = 0.38 \text{ m}^3/\text{kg}$;

At 1.5 bar, from steam tables, $h = 467.11 \text{ kJ/kg}$, $L = 2,226.5 \text{ kJ/kg}$,
 $\Phi_w = 1.4336 \text{ kJ/kg K}$, $\Phi_s = 7.2233 \text{ kJ/kg K}$, $v_s = 1.1593 \text{ m}^3/\text{kg}$.

For throat :

Entropy before expansion = Entropy after expansion

$$\Phi_1 = \Phi_{w2} + x_2 (\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e., } 6.6421 = 1.8550 + x_2 (6.860 - 1.8550)$$

$$\therefore x_2 = \frac{4.7871}{5.005} = 0.956$$

$$H_2 = h_2 + x_2 L_2 = 636 + 0.956 \times 2,110 = 2,653.16 \text{ kJ/kg}$$

\therefore Enthalpy drop from entry to throat,

$$H_t = H_1 - H_2 = 2,771.6 - 2,653.16 = 118.44 \text{ kJ/kg.}$$

Velocity at throat, $V_2 = 44.72 \sqrt{118.44} = 487 \text{ m/sec.}$

$$\text{For mass continuity, } m = \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{(x_2 \times v_{s2})}$$

$$\therefore A_2 = \frac{m \times (x_2 \times v_{s2})}{V_2} = \frac{m \times (0.956 \times 0.38)}{487} = 0.000746 \text{ m}$$

For exit :

$$\text{Now, } \Phi_1 = \Phi_{w3} + x_3 (\Phi_{s3} - \Phi_{w3})$$

$$\text{i.e., } 6.6421 = 1.4336 + x_3 (7.2233 - 1.4336)$$

$$\therefore x_3 = \frac{5.2085}{5.7897} = 0.9$$

$$H_3 = h_3 + x_3 L_3 = 467.11 + 0.9 \times 2,226.5 = 2,470.96 \text{ kJ/kg.}$$

\therefore Enthalpy drop from entry to exit,

$$H_e = H_1 - H_3 = 2,771.6 - 2,470.96 = 300.64 \text{ kJ/kg}$$

Velocity at exit, $V_3 = 44.72 \sqrt{300.64} = 775 \text{ m/sec.}$

$$\text{Now, } m = \frac{A_3 V_3}{v_3} = \frac{A_3 V_3}{(x_3 \times v_{s3})}$$

$$\therefore A_3 = \frac{m \times (x_3 \times v_{s3})}{V_3} = \frac{m \times (0.9 \times 1.1593)}{775} = 0.001346 \text{ m}$$

$$\therefore \frac{\text{Area at exit}}{\text{Area at throat}} = \frac{0.001346m}{0.000746m} = 1.8 \text{ (ratio)}$$

Problem - 7 : Assuming frictionless adiabatic flow through a converging-diverging nozzle, show that the maximum discharge per unit area at the throat is given by :

$$\sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \text{ kg per sec. per } m^2}$$

where, n is the index of expansion,

p_1 is the initial pressure of steam in kPa, and

v_1 is the specific volume of steam in m^3/kg at the initial pressure.

The flow or discharge per unit area through the nozzle throat is given by eqn. (8.8),

$$\frac{m}{A} = \sqrt{2,000 \frac{n}{n-1} \cdot \frac{p_1}{v_1} \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}} \right]}$$

This will be maximum when $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

Substituting this value of $\left(\frac{p_2}{p_1}\right)$ in the above equation, we get,

$$\begin{aligned} \frac{m}{A} &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left[\left(\frac{2}{n+1}\right)^{\frac{2}{n} \times \frac{n}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n} \times \frac{n}{n-1}} \right]} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \right]} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left\{ \frac{\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}{\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}} - 1 \right\}} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1} - \frac{n+1}{n-1}} - 1 \right]} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1}\right)^{\frac{2-n-1}{n-1}} - 1 \right]} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left[\left(\frac{2}{n+1}\right)^{-\left(\frac{n-1}{n-1}\right)} - 1 \right]} \end{aligned}$$

$$\begin{aligned} \frac{m}{A} &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left(\frac{n+1}{2} - 1\right)} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n-1}\right) \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \left(\frac{n-1}{2}\right)} \\ &= \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}} \text{ kg per sec. per m}^2. \end{aligned}$$

Problem – 8 : Assuming frictionless adiabatic flow through a converging-diverging nozzle, show that for the maximum discharge, the velocity of steam at throat is given by :

$$V_2 = \sqrt{2,000 \frac{n}{n+1} p_1 v_1} \text{ m/sec.}$$

where n is the index of expansion,

p_1 is the initial pressure of steam in kPa, and

v_1 is the specific volume of steam in m^3/kg at the initial pressure.

For isentropic (frictionless adiabatic) flow, neglecting initial velocity of steam and considering one kilogram of steam, the velocity of steam at throat, V_2 is given by

$$\frac{V_2^2}{2,000} = H_1 - H_2 \text{ or } V_2 = \sqrt{2,000 (H_1 - H_2)}$$

Considering $p v^n = \text{constant}$, a law of isentropic expansion of steam in the nozzle, the above expression of V_2 can be written as

$$\begin{aligned} V_2 &= \sqrt{2,000 \frac{n}{n-1} (p_1 v_1 - p_2 v_2)} \\ &= \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left(1 - \frac{p_2 v_2}{p_1 v_1}\right)} \end{aligned}$$

$$\text{But } \frac{v_2}{v_1} = \left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}} \text{ from } p_1 v_1^n = p_2 v_2^n$$

$$\therefore V_2 = \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}}$$

$$\text{For maximum discharge, } \frac{p_2}{p_1} = \left[\frac{2}{n+1}\right]^{\frac{n}{n-1}}$$

$$\begin{aligned} \therefore V_2 &= \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left\{1 - \left(\frac{2}{n+1}\right)^{\frac{n}{n-1} \times \frac{n-1}{n}}\right\}} \\ &= \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \left\{1 - \frac{2}{n+1}\right\}} \end{aligned}$$

$$= \sqrt{2,000 \frac{n}{n-1} p_1 v_1 \times \frac{n-1}{n+1}}$$

$$V_2 = \sqrt{2,000 \frac{n}{n+1} p_1 v_1} \quad \text{m/sec.}$$

Problem - 9 : Dry saturated steam enters a nozzle at pressure of 10 bar and with an initial velocity of 90 m/sec. The outlet pressure is 6 bar and outlet velocity is 435 m/sec. The heat loss from the nozzle is 6.3 kJ per kg of steam flow. Calculate the dryness fraction and the area at the exit, if the area at the inlet is 12.56 cm².

Steady flow energy equation per kg of steam flow through the nozzle at inlet and outlet can be written as

$$H_1 + \frac{V_1^2}{2,000} = H_2 + \frac{V_2^2}{2,000} + \text{losses}$$

where, H_1 and H_2 are enthalpies at inlet and outlet, kJ/kg, and

V_1 and V_2 are velocities at inlet and outlet, m/sec.

H_1 = enthalpy of dry saturated steam at 10 bar = 2778.1 kJ/kg (from steam tables).

$$\therefore 2,778.1 + \frac{90^2}{2,000} = H_2 + \frac{435^2}{2,000} + 6.3$$

$$\therefore H_2 = 2,778.1 + 4.05 - 94.61 - 6.3 = 2,681.24 \text{ kJ/kg.}$$

As steam is dry saturated at inlet it will be wet at outlet. Let its dryness fraction be x_2 .
 $H_2 = h_2 + x_2 L_2$ [at 6 bar, $h_2 = 670.56$ kJ/kg, $L_2 = 2,086.3$ kJ/kg (from steam tables)].

$$\text{i.e., } 2,681.24 = 670.56 + x_2 \times 2086.3$$

$$\therefore x_2 = \frac{2,681.24 - 670.56}{2,086.3} = 0.964$$

$v_1 = v_{s1} = 0.1944$ m³/kg at 10 bar (from steam tables).

$v_{s2} = 0.3157$ m³/kg at 6 bar (from steam tables).

$v_2 = x_2 \times v_{s2} = 0.964 \times 0.3157 = 0.3043$ m³/kg at 6 bar.

Now, for mass flow continuity,

$$m = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2} \quad \text{kg/sec.}$$

$$\therefore \frac{A_2}{A_1} = \frac{v_2}{v_1} \times \frac{V_1}{V_2}$$

$$\therefore \frac{A_2}{A_1} = \frac{0.3043}{0.1944} \times \frac{90}{435} = 0.3239$$

But $A_1 = 12.56$ cm² (area at inlet)

$$\therefore A_2 = 0.3239 \times 12.56 = 4.068 \text{ cm}^2 \text{ (area at exit)}$$

8.5 Effect of Friction in a Nozzle

As stated earlier, the length of the converging part of the converging-diverging nozzle is very small compared with that of divergent part. Thus, most of friction in the converging-diverging nozzle occurs in the divergent part, i.e., between the throat and the exit. The effect of friction is to reduce the available enthalpy drop for conversion into kinetic energy by about 10 to 15 per cent. The equation for the velocity is then written as

$$V = 44.72 \sqrt{kH} \text{ m per sec.}$$

where k is the coefficient which allows for friction loss,

$$\text{i.e., } k = \frac{\text{Actual or useful enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

It is sometimes termed as *nozzle efficiency*.

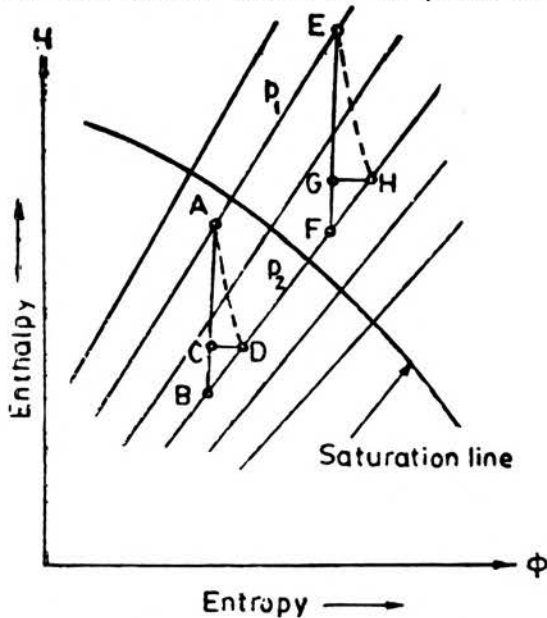
Or, if the initial velocity of the steam entering the nozzle can be neglected compared with the final velocity, the *nozzle efficiency* can also be expressed as

$$\frac{V^2}{2,000 \times (\text{Isentropic enthalpy drop, } H)}$$

The kinetic energy lost in friction is transformed into heat which tends to dry or superheat the steam. Thus, it (friction) will affect the final condition of the steam issuing from the nozzle. Its effect can be represented on the $H - \Phi$ diagram.

Referring to fig. 8-9, let point A represent the initial condition of steam and expansion takes place from pressure p_1 to p_2 . Vertical line AB represents isentropic expansion. The total enthalpy drop AB is reduced by friction to AC such that $\frac{AC}{AB} = k$.

From the known value of k , point C on the diagram can be obtained. But the



expansion must end on the same pressure line p_2 . Hence CD is drawn horizontally to meet p_2 pressure line at D. Then the point D represents the final condition of steam after expansion. It may be noted that dryness fraction at D is greater than that at B. Thus, the effect of friction has been partly to dry the steam.

Actually, drying effect of the friction will occur throughout the whole expansion, so that the actual expansion would be represented by the dotted line AD (fig. 8-9).

Fig. 8-9 The effect of friction in a nozzle.

Similar effect is produced if the initial condition of steam is superheated as represented by point E and expansion takes place wholly or partly in the superheated region. It will be noted that the effect of friction is to further superheat the steam at the end of expansion. The actual expansion is represented by the dotted line EH and the resulting (actual) enthalpy drop is the distance EG.

It thus follows that the effect of friction in the nozzle is to reduce the velocity of the steam and to increase its final dryness fraction or degree of superheat.

Problem – 10 : Steam is supplied at a dryness fraction of 0.97 and 11 bar pressure to a convergent-divergent nozzle and expands down to a back pressure of 0.3 bar. The throat area of the nozzle is 5 cm^2 and 12 per cent of the total enthalpy drop is lost in the divergent part. Determine : (a) steam flow in kg per sec. and (b) the nozzle outlet area.

Let suffixes, 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle.
As the steam is wet initially, critical or throat pressure,

$$p_2 = 0.582 \times p_1 = 0.582 \times 11 = 6.4 \text{ bar.}$$

From steam tables :

p bar	t_s °C	v_s m^3/kg	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
11	-	0.17753	781.34	2,000.4	2,781.7	2.1792	6.5536
6.4	-	0.297	681.6	2,078	2,759.6	1.9566	6.7383
0.3	-	5.229	289.23	2,336.1	2,625.3	0.9439	7.7686

(a) For throat :

$$\Phi_1 = \Phi_2$$

$$\Phi_{w1} + x_1 (\Phi_{s1} - \Phi_{w1}) = \Phi_{w2} + x_2 (\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e., } 2.1792 + 0.97 (6.5536 - 2.1792) = 1.9566 + x_2 (6.7383 - 1.9566)$$

$$\therefore 2.1792 + 4.2432 = 1.9566 + 4.7817 \times x_2$$

$$\therefore x_2 \text{ (dryness fraction at throat) } = 0.934$$

$$\text{Enthalpy, } H_1 = h_1 + x_1 L_1 = 781.34 + 0.97 \times 2,000.4 = 2,722.09 \text{ kJ/kg}$$

$$\text{Enthalpy, } H_2 = h_2 + x_2 L_2 = 681.6 + 0.934 \times 2,078 = 2,622.45 \text{ kJ/kg.}$$

$$\text{Enthalpy drop from inlet to throat, } H_t = H_1 - H_2 = 2,722.09 - 2,622.45 = 99.64 \text{ kJ/kg.}$$

$$\text{Velocity at throat, } V_2 = 44.72 \sqrt{H_t} = 44.72 \sqrt{99.64} = 446.3 \text{ m/sec.}$$

$$\text{Specific volume at throat, } v_2 = x_2 \times v_{s2} = 0.934 \times 0.297 = 0.2774 \text{ m}^3/\text{kg.}$$

$$\text{For mass continuity, } m = \frac{A_2 V_2}{v_2} = \frac{5 \times 446.3}{10^4 \times 0.2774} = 0.8044 \text{ kg/sec.}$$

(b) For exit :

$$\Phi_1 = \Phi_3$$

$$\Phi_{w1} + x_1 (\Phi_{s1} - \Phi_{w1}) = \Phi_{w3} + x_3 (\Phi_{s3} - \Phi_{w3})$$

$$\text{i.e., } 2.1792 + 0.97 (6.5536 - 2.1792) = 0.9439 + x_3 (7.7686 - 0.9439)$$

$$\therefore 2.1792 + 4.2432 = 0.9439 + 6.8247 \times x_3$$

$$\therefore x_3 = \frac{6.4224 - 0.9439}{6.8247} = 0.8028 \text{ (dryness fraction at exit)}$$

$$\text{Enthalpy, } H_1 = 2,722.09 \text{ kJ/kg.}$$

$$\text{Enthalpy, } H_3 = h_3 + x_3 L_3 = 289.23 + 0.8028 \times 2,336.1 = 2,164.23 \text{ kJ/kg}$$

Enthalpy drop from inlet to exit.

$$H_e = H_1 - H_3 = 2,722.09 - 2,164.23 = 557.86 \text{ kJ/kg.}$$

Enthalpy drop after considering friction in the divergent part

$$= 0.88 \times 557.86 = 490.92 \text{ kJ/kg.}$$

$$\text{Velocity at exit, } V_3 = 44.72 \sqrt{490.92} = 990 \text{ m/sec.}$$

$$\text{Specific volume at exit, } v_3 = x_3 \times v_{s3} = 0.8028 \times 5.229 \text{ m}^3/\text{kg}$$

$$\text{For mass continuity, } m = \frac{A_3 V_3}{v_3}$$

$$\therefore A_3 = \frac{m \times v_3}{V_3} = \frac{10^4 \times 0.8044 \times (0.8028 \times 5.229)}{990} = 34.13 \text{ cm}^2 \text{ (outlet area).}$$

Problem – 11 : Steam enters a group of convergent-divergent nozzles at 21 bar and 270°C, the discharge pressure being 0.07 bar. The expansion is in equilibrium throughout and the loss of friction in the converging portion of the nozzle is negligible, but the loss by friction in the divergent section of the nozzle is equivalent to 10 per cent of the enthalpy drop available in that section (i.e., enthalpy drop available in the divergent section).

Calculate the total throat and exit areas in cm², to discharge 14 kg of steam per second. Sketch enthalpy-entropy (H - Φ) chart and show on it the various stages of expansion.

Let suffixes 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle respectively as shown in fig: 8-10.

As the steam is initially superheated, critical or throat pressure,

$$p_2 = 0.546 \times p_1 = 0.546 \times 21 = 11.47 \text{ bar.}$$

A sketch (fig 8-10) of the readings taken from the H - Φ chart is given.

Isentropic enthalpy drop from throat to exit = 770 kJ/kg (from H - Φ chart).

The actual (useful) enthalpy drop from throat to exit is 90% of the isentropic enthalpy drop.

$$\therefore \text{Actual enthalpy drop after allowing for friction in the divergent section} \\ = 0.9 \times 770 = 693 \text{ kJ/kg.}$$

For throat : Enthalpy drop,

$$H_t = H_1 - H_2 = 140 \text{ kJ/kg (from H - Φ chart).}$$

Temperature of steam at throat, $t_2 = 194^\circ\text{C}$ (from H - Φ chart), i.e., steam is superheated at the throat.

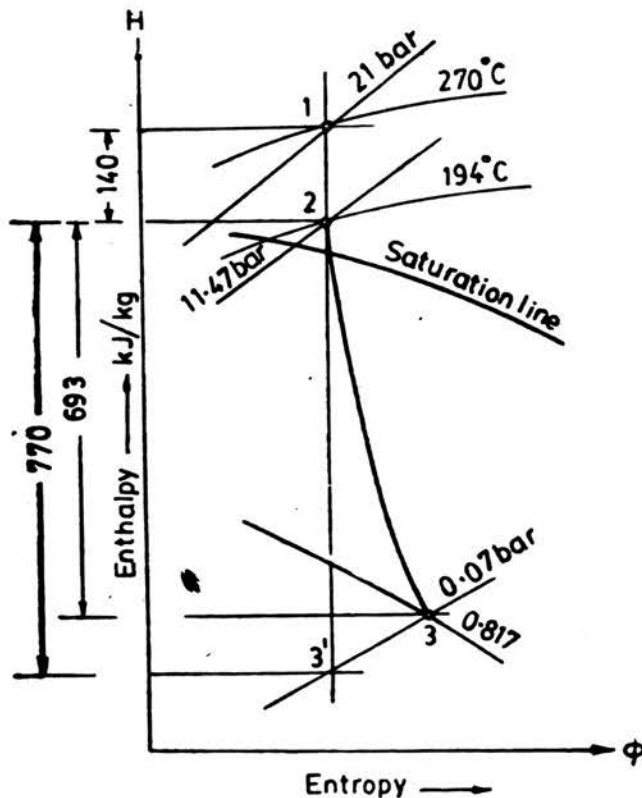


Fig. 8 10. H - φ diagram.

At 11.47 bar (from steam tables by arithmetical interpolation), $t_{s2} = 186^\circ\text{C}$ and $v_{s2} = 0.17 \text{ m}^3/\text{kg}.$

Specific volume at throat,

$$v_2 = v_{s2} \times \frac{T_{sup2}}{T_{s2}} \\ = 0.17 \times \frac{194 + 273}{186 + 273} \\ = 0.173 \text{ m}^3/\text{kg.}$$

Velocity at throat,

$$V_2 = 44.72 \sqrt{H_t} \\ = 44.72 \sqrt{140} = 529 \text{ m/sec.}$$

For mass continuity,

$$m = \frac{A_2 V_2}{v_2} \\ \text{i.e., } 14 = \frac{A_2 \times 529}{0.173 \times 10^4}$$

$$\therefore \text{ Total area of nozzles at throat, } A_2 = \frac{14 \times 0.173 \times 10^4}{529} = 45.78 \text{ cm}^2$$

For exit :

Actual enthalpy drop from entry to exit, $H_e = H_1 - H_3 = 833 \text{ kJ/kg}$, and dryness fraction after reheating, $x_3 = 0.817$ (from $H - \Phi$ chart).

At 0.07 bar, $v_{s3} = 20.53 \text{ m}^3/\text{kg}$ (from steam tables).

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0.817 \times 20.53 \text{ m}^3/\text{kg}$.

Velocity at exit, $V_3 = 44.72 \sqrt{H_e} = 44.72 \sqrt{833} = 1,291 \text{ m/sec}$.

$$\text{For mass continuity, } m = \frac{A_3 V_3}{v_3}$$

$$\text{i.e., } 14 = \frac{A_3 \times 1291}{(0.817 \times 20.53) \times 10^4}$$

$$\therefore \text{ Total area of nozzles at exit, } A_3 = \frac{14 \times 0.817 \times 20.53 \times 10^4}{1,291} = 1819 \text{ cm}^2$$

Problem – 12 : A convergent divergent nozzle is required to pass 1.8 kg of steam per sec. At inlet the steam pressure and actual temperature are 7 bar and 186.2°C respectively and the speed is 75 m/sec. Expansion is stable throughout to the exit pressure of 1.1 bar. There is no loss by friction in the converging section of the nozzle but loss by friction between throat and outlet is equivalent to 70 kJ/kg of steam. Calculate, assuming throat pressure of 4 bar :

(a) the required area of throat in cm^2 ,

(b) the required area of outlet in cm^2 , and

(c) the overall efficiency of the nozzle, based on the heat drop between the actual inlet pressure and temperature and the outlet pressure.

Let suffixes, 1, 2 and 3 represent conditions at inlet, throat and exit (outlet) of the nozzle respectively as shown in fig. 8-11.

(1) Referring to fig. 8-11.

For throat :

Throat pressure, $p_2 = 4 \text{ bar}$ (given) and velocity of steam at entry,

$V_1 = 75 \text{ m/sec}$. (given).

Enthalpy drop from inlet to throat,

$$H_t = H_1 - H_2 = 110 \text{ kJ/kg}$$

and dryness fraction,

$x_2 = 0.985$ (from $H - \Phi$ chart).

$$\text{Now, } \frac{(V_2)^2 - (V_1)^2}{2 \times 1,000} = H$$

where V_1 = velocity of steam at inlet,

V_2 = velocity of steam at throat, and

H = enthalpy drop from entry to throat.

$$\text{i.e., } \frac{(V_2)^2 - (75)^2}{2 \times 1000} = 110. \text{ From which, velocity at throat, } V_2 = 475 \text{ m/sec.}$$

At 4 bar, $v_{s2} = 0.4625 \text{ m}^3/\text{kg}$ (from steam tables).

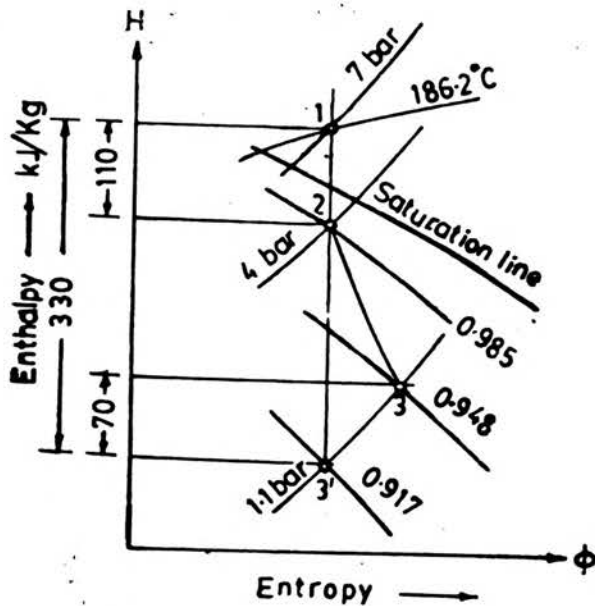


Fig. 8-11. $H - \phi$ diagram.

Specific volume at throat,

$$v_2 = x_2 \times v_{s2} = 0.985 \times 0.4625 \text{ m}^3/\text{kg}.$$

For mass continuity, $m = \frac{A_2 V_2}{v_2}$ i.e.,

$$1.8 = \frac{A_2 \times 475}{(0.985 \times 0.4625) \times 10^4}$$

\therefore Area of throat,

$$A_2 = \frac{1.8 \times 0.985 \times 0.4625 \times 10^4}{475} = 17.3 \text{ cm}^2$$

(b) For exit :

Isentropic enthalpy drop from inlet to exit

$$= H_1 - H_3 = 330 \text{ kJ/kg.}$$

(from $H - \Phi$ chart).

\therefore Actual (useful) enthalpy drop ($H_1 - H_3$) after allowing for friction in the divergent part
 $= 330 - 70 = 260 \text{ kJ/kg.}$

Now, $\frac{(V_3)^2 - (V_1)^2}{2 \times 1,000} = \text{Enthalpy drop, } H$ i.e., $\frac{(V_3)^2 - (75)^2}{2 \times 1,000} = 260$

From which, velocity of steam at exit from the nozzle, $V_3 = 725 \text{ m/sec.}$

Reheated condition at exit or dryness fraction at exit, $x_3 = 0.948$ (from $H - \Phi$ chart).

At 1.1 bar, $v_{s3} = 1.5495 \text{ m}^3/\text{kg}$ (from steam tables).

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0.948 \times 1.5495 \text{ m}^3/\text{kg}.$

For mass continuity, $m = \frac{A_3 V_3}{v_3}$ i.e., $1.8 = \frac{A_3 \times 725}{0.948 \times 1.5495 \times 10^4}$

\therefore Area of outlet (exit), $A_3 = \frac{1.8 \times 0.948 \times 1.5495 \times 10^4}{725} = 36.47 \text{ cm}^2$

(c) Overall efficiency of nozzle = $\frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}} = \frac{260}{330} = 0.788$, i.e., **78.8%**

Problem – 13 : The nozzles in the stage of an impulse turbine receive steam at 17 bar with 80°C superheat and the pressure in the wheel is 9.5 bar. If there are 16 nozzles, find the cross-sectional area at exit of each nozzle for the total discharge of 280 kg per minute. Assume nozzle efficiency of 88 per cent.

If the steam had a velocity of 120 m/sec. at entry to the nozzles, by how much would the discharge be increased ?

Let suffixes 1 and 2 represent conditions at inlet and exit of the nozzle.

As the steam is initially superheated,

critical or throat pressure = $0.546 \times p_1 = 0.546 \times 17 = 9.28 \text{ bar.}$

Since the exit pressure (p_2) is greater than the critical pressure, the nozzles are convergent.

From Mollier chart, isentropic enthalpy drop from inlet to exit,

$$H_e = H_1 - H_2 = 131.0 \text{ kJ/kg.}$$

∴ Actual enthalpy drop after allowing for friction $0.88 \times 131 = 115.3 \text{ kJ/kg.}$

Neglecting the velocity of approach :

$$\text{Velocity of steam at exit, } V_2 = 44.72 \sqrt{115.3} = 480.0 \text{ m/sec.}$$

From Mollier chart, steam at the end of expansion (at 9.5 bar) has a temperature of 222°C (taking friction into account).

At 9.5 bar, $v_{s2} = 0.2042 \text{ m}^3/\text{kg}$, $t_{s2} = 177.69^\circ\text{C}$ (from steam tables).

∴ Specific volume of steam at 9.5 bar and 222°C ,

$$v_2 = 0.2042 \times \left(\frac{222 + 273}{177.69 + 273} \right) = 0.224 \text{ m}^3/\text{kg}$$

$$\text{For mass continuity, } m = \frac{A_2 V_2}{V_2}$$

$$\begin{aligned} \therefore \text{Area at exit for one nozzle, } A_2 &= \frac{m \times v_2}{V_2 \times \text{No. of nozzle}} \\ &= \frac{280 \times 0.224 \times 10^4}{60 \times 480.0 \times 16} = 13.6 \text{ cm}^2 \end{aligned}$$

Considering velocity at inlet or velocity of approach (V_1) of 120 m/sec. :

$$\text{Now, } \frac{(V_2')^2 - (V_1)^2}{2 \times 1,000} = \text{actual enthalpy drop, } H$$

$$\text{i.e., } \frac{(V_2')^2 - (120)^2}{2 \times 1,000} = 115.3 \quad \therefore (V_2')^2 = 2,45,000$$

$$\therefore V_2' = 495 \text{ m/sec. (Velocity of steam at exit)}$$

Since there is no increase in specific volume, the discharge is directly proportional to velocity.

$$\therefore \text{Increase in discharge} = \frac{V_2' - V_2}{V_2} \times 100 = \frac{495 - 480.0}{480.0} \times 100 = 3.13\%.$$

Problem - 14 : *Steam expands through a nozzle under supersaturated adiabatic conditions, from an initial pressure of 8 bar and a temperature of 210°C , to a final pressure of 2 bar, Determine :*

(i) the final condition of the steam, (ii) the exit velocity of the steam, (iii) the degree of undercooling, and (iv) the degree of supersaturation at the end of expansion.

Compare the mass flow through the above nozzle with one in which the expansion takes place under conditions of thermal equilibrium.

For the supersaturated state you may use the following relationship :

$$v = \frac{0.233(H - 1940)}{p}; \quad pv^{1.3} = \text{constant}; \quad \text{and } \frac{p}{(T)^3} = \text{constant.}$$

where, v is the specific volume in m^3/kg , H is the enthalpy in kJ/kg , p is the pressure in kPa and T is the absolute temperature.

Take k_p of superheated steam as 2.3 kJ/kg K .

Expansion under supersaturated conditions (metastable flow) :

From Steam tables :

p bar	t_s °C	v_s m^3/kg	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kgK	Φ_s kJ/kg K
8	170.43	0.2404	721.11	2048	2769.1	2.0411	6.6628
2	120.23	0.8857	504.7	2201.9	2706.7	1.5301	7.1271

(i) Let suffixes 1 and 2 represent the initial and final conditions respectively.

From steam tables, steam enthalpy, $H_1 = 2769.1 + 2.3 (210 - 170.43) = 2,860$ kJ/kg.

$$v_1 = \frac{0.233 (H_1 - 1940)}{p_1} = \frac{0.233 (2,860 - 1940)}{800} = 0.27 \text{ m}^3/\text{kg}.$$

From given equation $p_1 v_1^{1.3} = p_2 v_2^{1.3}$

$$\therefore v_2 = v_1 \left(\frac{p_1}{p_2} \right)^{\frac{1}{1.3}} = 0.27 \left(\frac{8}{2} \right)^{\frac{1}{1.3}} = 0.7843 \text{ m}^3/\text{kg} \text{ (specific volume at exit).}$$

$$\text{But, } v_2 = \frac{0.233 (H_2 - 1,940)}{p_2} \quad \text{i.e., } 0.7843 = \frac{0.233 (H_2 - 1,940)}{200}$$

From which, steam enthalpy, $H_2 = 2,613$ kJ/kg.

At 2 bar, $h = 504.7$ kJ/kg, $L = 2,201.9$ kJ/kg (from steam tables).

$$\text{Dryness fraction at exit, } x_2 = \frac{H_2 - h_2}{L_2} = \frac{2,613 - 504.7}{2,201.9} = 0.9575$$

(ii) Actual enthalpy drop from inlet to exit,

$$H_e = H_1 - H_2 = 2,860 - 2,613 = 247 \text{ kJ/kg}$$

Then, velocity at exit, $V_2 = 44.72 \sqrt{H_1 - H_2} = 44.72 \sqrt{247} = 703$ m/sec.

(iii) Now, $T_1 = 273 + 210 = 483$ K.

$$\text{From given equation } \frac{p_1}{(T_1)^{\frac{13}{3}}} = \frac{p_2}{(T_2)^{\frac{13}{3}}} \quad \text{i.e., } \frac{8}{(483)^{\frac{13}{3}}} = \frac{2}{(T_2)^{\frac{13}{3}}}$$

From which temperature after supersaturation, $T_2 = 350.8$ K,

then, $t_2 = 350.8 - 273 = 77.8^\circ\text{C}$ (actual temperature).

Saturation temperature at 2 bar = 120.23°C (from steam tables)

Degree of undercooling is the difference between the normal saturation temperature corresponding to the pressure and the actual temperature.

$$\therefore \text{Degree of undercooling} = 120.23 - 77.8 = 42.43^\circ\text{C}$$

(iv) Saturation pressure corresponding to $77.8^\circ\text{C} = 0.43$ bar (from steam tables by arithmetical interpolation).

Degree of supersaturation

$$= \frac{\text{Pressure after supersaturation}}{\text{Saturation pressure corresponding to the undercooled temp.}} = \frac{2}{0.43} = 4.65$$

Expansion under supersaturated condition :

$$\text{For mass continuity, } m_2 = \frac{A_2 V_2}{v_2} \text{ (where } A_2 \text{ is the exit area)}$$

$$= \frac{A_2 \times 703}{0.7843} = 896 A_2 \text{ kg/sec.}$$

Expansion under conditions of the thermal equilibrium (stable flow) :

For adiabatic flow,

Entropy before expansion, Φ_1 = Entropy after expansion, Φ_2

$$\Phi_{s1} + k_p \log_e \left(\frac{T_{\text{sup}1}}{T_{\text{sat}1}} \right) = \Phi_{w2} + x_2 (\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e., } 6.6628 + 2.3 \log_e \frac{210 + 273}{170.43 + 273} = 1.5301 + x_2 (7.1271 - 1.5301)$$

$$6.6628 + 2.3 \times 0.0855 = 1.5301 + x_2 \times 5.597$$

$$\therefore x_2 = \frac{6.8595 - 1.5301}{5.597} = 0.952 \quad (\text{dryness fraction at exit})$$

Steam enthalpy, $H_2 = h_2 + x_2 L_2 = 504.7 + 0.952 \times 2,201.9 = 2,603 \text{ kJ/kg.}$

Steam enthalpy, $H_1 = 2,769.1 + 2.3(210 - 170.43) = 2,860 \text{ kJ/kg.}$

Isentropic enthalpy drop from inlet to exit, $H_1 - H_2 = 2,860 - 2,603 = 257 \text{ kJ/kg}$

Velocity at exit, $V_2 = 44.72 \sqrt{H_1 - H_2} = 44.72 \sqrt{257} = 717 \text{ m/sec.}$

From steam tables at 2 bar, $v_{s2} = 0.8857 \text{ m}^3/\text{kg.}$

$$v_2 = x_2 \times v_{s2} = 0.952 \times 0.8857 \text{ m}^3/\text{kg}$$

$$\begin{aligned} \text{For mass continuity, } m_2 &= \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{x_2 \times v_{s2}} \quad (\text{where } A_2 \text{ is the exit area}) \\ &= \frac{A_2 \times 717}{0.952 \times 0.8857} = 850 A_2 \text{ kg/sec.} \end{aligned}$$

Increase of discharge due to supersaturated flow = $896 A_2 - 850 A_2 = 46 A_2 \text{ kg/sec.}$

$$\therefore \text{Increase in discharge due to supersaturated flow} = \frac{46 A_2}{850 A_2} \times 100 = 5.41\%$$

Thus, supersaturated flow increases the discharge by 5.41%.

It should be noted that the effect of supersaturated flow (compared with flow under the conditions of thermal equilibrium) is :

Reduction in the enthalpy drop, which will cause corresponding reduction in velocity, the dryness fraction at exit is higher, and

the rate of discharge is increased.

8.6 Steam Injector

A steam injector utilises the kinetic energy of a steam jet for increasing the pressure and velocity of corresponding quantity of water; they are frequently used for forcing the water into steam boilers under pressure. The action of the injector is illustrated diagrammatically in fig. 8-12.

Steam from the boiler is supplied to the convergent nozzle A, and this steam issuing there from with a high velocity into the mixing cone, is condensed by the cold water flowing from the feed water tank E.

This tank may be either above or below the level of the injector. Owing to conversion of enthalpy of evaporation (latent heat) of steam to kinetic energy, the mixture of water and condensate has a high velocity at B. Thus, mixture issuing from the nozzle B, then flows through the divergent delivery nozzle or diffuser C, in which its kinetic energy is

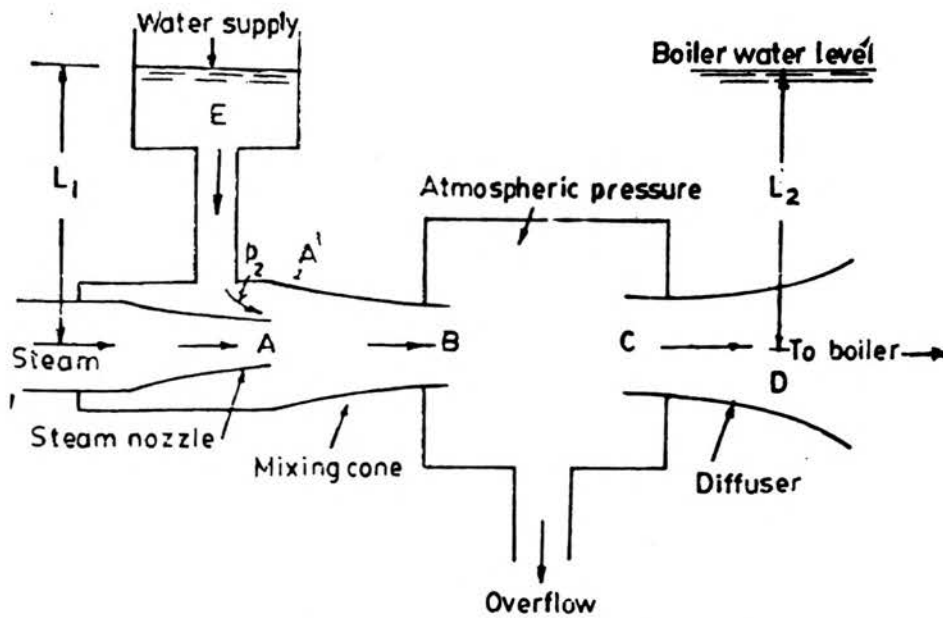


Fig. 8-12. Principle of steam injector.

reduced and converted to pressure energy, until on leaving at D, this pressure energy is sufficient to overcome boiler pressure and to lift the water through the height L_2 , and the water enters the boiler. The pressure of water at D must be about 25 per cent higher than the boiler pressure in order to overcome all resistances. Because of the gap between the nozzles B and C, provided for excess water which may overflow during the starting of the injector, the pressure in the gap is nearly atmospheric.

- Let M_w = Mass of the water per kg of steam entering at A' in kg/sec.,
- V_s = Velocity of steam leaving the nozzle at A in m/sec,
- V_w = Velocity of water entering at A' in m/sec., and
- V_m = Velocity of mixture leaving nozzle at B in m/sec.

Applying the principle of conversion of momentum to the mixing of the steam jet and water supply, per kilogram of steam supplied to the nozzle. Then,

$$\begin{array}{l}
 \text{[Momentum of} \\
 \text{steam entering} \\
 \text{combining nozzle]} \\
 + \\
 \text{[Momentum of} \\
 \text{water entering} \\
 \text{combining nozzle]} \\
 = \\
 \text{[Momentum} \\
 \text{of mixture} \\
 \text{leaving com-} \\
 \text{bining nozzle]}
 \end{array}$$

H.E. 15

$$\text{i.e., } 1 \times V_s + M_w V_w = (1 + M_w) V_m$$

$$\text{or } V_s + M_w V_w = (1 + M_w) V_m$$

If the water level in the tank E is below the level of the injector, then

$$V_s - M_w V_w = (1 + M_w) V_m$$

$$\text{Hence, } M_w = \frac{V_s - V_m}{V_m \pm V_w} \quad \dots (8.13)$$

according to whether the water supply level is below or above the injector level. This formula gives the amount of water injected per kilogram of steam if the velocities are known.

The velocity of steam V_s from the nozzle may be found by assuming that the steam expands isentropically (frictionless adiabatic) from the initial condition to the back pressure p_2 . Using enthalpy-entropy (Mollier) chart or by calculation, the enthalpy drop, H in the

nozzle can be found. Then, $V_s = 44.72 \sqrt{H}$

The velocity of water, V_w entering the annular space A' will be given by the equation,

$$V_w = \sqrt{2g.L_1}$$

To find the velocity of mixture, V_m leaving nozzle B :

Let p_m = pressure at B in kPa, and

w = density of warm water at B in kg/m^3 .

Due to the presence of the gap between nozzles B and C , the pressure p_m of the water in the throat of B may be taken as atmospheric, say 101.33 kPa, and as the water is warm its density may be taken as 995 kg/m^3 .

Then, the total energy, kJ per kilogram of water at $B = \frac{p_m}{w} + \frac{V_m^2}{2,000}$

This energy must be enough to lift the water through a height L_2 metres at the delivery end and inject it into the boiler. The final pressure on leaving at D must therefore be some what greater than this height (L_2) plus the boiler pressure.

At D the pressure energy of 1 kg of water is $\frac{p}{w} + \frac{g}{1,000} L_2$

where p is the absolute pressure of steam in the boiler in kPa.

$$\text{Then, } \frac{p_m}{w} + \frac{V_m^2}{2,000} = \frac{p}{w} + \frac{g}{1,000} L_2 + \frac{V^2}{2,000}$$

The water ultimately comes to rest in the boiler and the kinetic energy $\frac{V^2}{2,000}$ may be taken equal to the pressure energy due to addition of, say, 9.3 metres to the lift L_2 , which approximately corresponds to $V = 13.5 \text{ m/sec.}$ and to an addition of about 90 kPa to the boiler pressure.

$$\text{Hence, } \frac{V_m^2}{2,000} + \frac{101.33}{995} = \frac{p}{995} + \frac{g}{1,000} \times L_2 + \frac{g}{1,000} \times 9.3$$

$$\therefore V_m = \sqrt{2,000 \left[\frac{(p - 101.33)}{995} + \frac{g}{1,000} \times L_2 + \frac{g}{1,000} \times 9.3 \right]} \quad (8.14)$$

If the actual delivery is to be M kilogram of water per sec. and M_s kg of condensed steam per second,

$$\text{Then, } M + M_s = M \left(1 + \frac{1}{M_w} \right)$$

Let a_b = area of throat of B in cm^2 , and d_b = diameter of throat of B in cm,

$$\text{Then, } M \left(1 + \frac{1}{M_w} \right) = \frac{a_b V_m}{10^4 v} \quad \text{where } v = \frac{1}{w} = \frac{1}{995}$$

$$\therefore a_b = \frac{10^4 M \left(1 + \frac{1}{M_w} \right)}{995 V_m} \quad \text{and } d_b = \sqrt{\frac{a_b \times 4}{\pi}} \quad \dots (8.15)$$

Let a_a = area of throat of A in cm^2 ,

d_a = diameter of throat of A in cm, and

v_s = volume of wet steam after expansion in the nozzle A in m^3/kg ,

then, $M_s = \frac{a_a V_s}{10^4 v_s} = \frac{M}{M_w}$

$\therefore a_a = \frac{10^4 M \times v_s}{M_w \times V_s}$ and $d_a = \sqrt{\frac{a_a \times 4}{\pi}}$ (8.16)

The heat balance per kilogram of steam may be determined as follows :

Let H_s = enthalpy per kilogram of steam entering the injector,

h_w = enthalpy per kilogram of water supplied to injector,

h_m = enthalpy per kilogram of water leaving at B.

Then,

$$\begin{aligned} \text{Heat supplied in steam} + \text{Heat supplied in water} &= \text{Kinetic energy of water at A} \\ &= \text{Heat in mixture at B} + \text{Kinetic energy of mixture at B} \end{aligned}$$

i.e., $1 \times H_s + M_w h_w = \frac{M_w V_w^2}{2,000} = (M_w + 1)h_m + \frac{(M_w + 1)}{2,000} V_m^2$... (8.17)

according to whether, the water supply level is below the injector level.

From this equation the temperature of the mixture may be found by taking h_m equal to temperature of mixture multiplied by specific heat of water, i.e., 4.187 kJ/kg K.

Problem – 15 : An injector is required to deliver 100 kg of water per minute from a tank whose constant water level is 1.2 metres below the level of the injector, into a boiler in which the steam pressure is 14 bar. The water level in the boiler is 1.5 metres above the level of the injector. The steam for the injector is to be taken from the same boiler and it is assumed as dry saturated. The temperature of the water in the supply tank is 15°C. Find : (a) the mass of water taken from the supply tank per kg of steam, (b) the diameter of the throat of the mixing nozzle (c) the diameter of the throat of steam nozzle, and (d) the temperature of water leaving the injector. Neglect the radiation losses.

(a) Referring to fig. 8-12, throat pressure, $p_1 = 0.578 \times 14 = 8.08$ bar.

At 14 bar, $H_s = 2,790$ kJ/kg and $\Phi_s = 6.4693$ (from steam tables)

At 8.08 bar, (by arithmetical interpolation),

$\Phi_{w1} = 2.05, \Phi_{s1} = 6.66, h_1 = 723$ kJ/kg.

$L_1 = 2,047$ kJ/kg, $v_{s1} = 0.237$ m³/kg (from steam tables).

$\Phi_s = \Phi_{w1} + x_1 (\Phi_{s1} - \Phi_{w1})$

$6.4693 = 2.05 + x_1 (6.66 - 2.05) \therefore x_1 = 0.96$

$H_1 = h_1 + x_1 L_1 = 723 + 0.96 \times 2,047 = 2,688$ kJ/kg

$\therefore V_s = 44.72 \sqrt{2,790 - 2,688} = 44.72 \sqrt{102} = 452$ m/sec. (velocity of steam)

Now, $V_w = \sqrt{2g L_1}$

$= \sqrt{2 \times 9.81 \times 1.2}$

$= \sqrt{23.55} = 4.85$ m/sec. (velocity of water entering injector)

Using eqn. (8.14), velocity of mixture leaving nozzle,

$V_m = \sqrt{2,000 \left[\frac{(p - 101.33)}{955} + \frac{g}{1,000} \times L_2 + \frac{g}{1,000} \times 9.3 \right]}$

$$V_m = \sqrt{2,000 \left[\frac{(1,400 - 101.33)}{995} + \frac{9.81}{1,000} \times 1.5 + \frac{9.81}{1,000} \times 9.3 \right]}$$

$$= \sqrt{2,822} = 53.1 \text{ m/sec.}$$

Using eqn. (8.13), mass of water,

$$M_w = \frac{V_s - V_m}{V_m + V_w} = \frac{452 - 53.1}{53.1 + 4.85} = 6.88 \text{ kg/kg of steam.}$$

(b) Using eqn. (8.15), area of the throat of mixing nozzle,

$$a_b = 10^4 M \frac{\left(1 + \frac{1}{M_w}\right)}{995 V_m} = \frac{10^4 \times \frac{100}{60} \left(1 + \frac{1}{6.88}\right)}{995 \times 53.1} = 0.361 \text{ cm}^2$$

$$\therefore d_b = \sqrt{\frac{0.361 \times 4}{\pi}} = \sqrt{0.46} = 0.678 \text{ cm (dia. of throat of mixing nozzle)}$$

(c) Specific volume at throat of steam nozzle, $v_s = x_1 \times v_{s1} = 0.96 \times 0.237$
 $= 0.228 \text{ m}^3/\text{kg}.$

Using eqn. (8.16), area of the throat of steam nozzle,

$$a_a = \frac{10^4 M \times v_s}{M_w \times V_s} = \frac{10^4 \times \frac{100}{60} \times 0.228}{6.88 \times 452} = 1.22 \text{ cm}^2$$

$$\therefore d_a = \sqrt{\frac{1.22 \times 4}{\pi}} = \sqrt{1.553} = 1.246 \text{ cm (dia. of throat of steam nozzle).}$$

(d) Using eqn. (8.17),

$$1 \times H_s + M_w h_w - \frac{M_w V_w^2}{2,000} = (M_w + 1) h_m + \frac{(M_w + 1)}{2,000} V_m^2$$

$$1 \times 2790 + 6.88 \times 4.187 \times 15 - \frac{6.88 (4.85)^2}{2,000} = (6.88 + 1) h_m + \frac{(6.88 + 1) (53.1)^2}{2,000}$$

$$2,790 + 432.1 - 0.081 = 7.88 h_m + 11.1$$

$$\therefore h_m = \frac{3,210.9}{7.88} = 407.5 \text{ kJ/kg}$$

Let t_m be the temperature of water leaving the injector. Then,

$$407.5 = 4.187 (t_m - 0)$$

$\therefore t_m = 97.3^\circ\text{C}$ (Temp. of water leaving the injector).

Tutorial – 8

1 Delete the phrase which is not applicable in the following statements :

- (i) When the steam flows through a correctly shaped nozzle, its velocity and specific volume both will decrease/increase.
- (ii) The flow of steam in the convergent portion of the steam nozzle is subsonic/supersonic.
- (iii) Friction is of negligible magnitude between entry and throat/between throat and exit of the nozzle.
- (iv) In a steam nozzle, as the pressure of steam decreases, velocity of steam decreases/increases.
- (v) The length of the converging part of a convergent-divergent steam nozzle is short/long as compared with the length of its diverging part.
- (vi) The effect of friction is to reduce/increase the available enthalpy drop for conversion into kinetic energy.
- (vii) The nozzle critical pressure ratio for initially dry saturated steam is 0.578/0.582

[Delete : (i) decrease, (ii) supersonic, (iii) between throat and exit, (iv) decreases, (v) long, (vi) to increase, (vii) 0.582]

2 Fill in the blanks in the following statements :

- (i) During an adiabatic process with friction, _____ does not remain constant.
- (ii) The smallest section of the convergent-divergent nozzle is known as the _____.
- (iii) Discharge through a nozzle will be _____ when the ratio of pressure at throat to pressure at entry reaches the _____ value.
- (iv) The effect of friction in nozzles is to reduce the _____.
- (v) _____ ratio depends on the value of index, n for isentropic expansion of steam through the nozzle.
- (vi) The value of index n for isentropic expansion of initially superheated steam is _____.
- (vii) The critical pressure ratio, $\frac{P_2}{P_1} =$ _____.
- (viii) The pressure of steam at which the area of the nozzle is minimum and the discharge per unit area is maximum is termed as _____.
- (ix) A nozzle which first converges to throat and then diverges is termed as _____ nozzle.
- (x) The flow of steam in the convergent portion of the nozzle is _____.
- (xi) Friction reduces the enthalpy drop in a steam nozzle by _____ per cent.
- (xii) Friction is of _____ magnitude between entry and throat as most of the friction occurs between the throat and exit of the nozzle.
- (xiii) The supersaturated flow is also called the _____ flow.
- (xiv) A steam injector utilises the kinetic energy of steam jet for _____ the pressure and velocity of corresponding quantity of water.
- (xv) Steam injectors are frequently used for forcing the _____ into steam boilers under pressure.

[(i) entropy, (ii) throat, (iii) maximum, critical, (iv) enthalpy drop, (v) critical pressure, (vi) 1.3, (vii) $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$, (viii) critical pressure, (ix) convergent-divergent nozzle, (x) subsonic, (xi) 4 to 15, (xii) negligible, (xiii) metastable, (xiv) increasing, (xv) feed water]

3 Indicate the correct answer by selecting correct phrases from each of the following :

- (i) The value of index n for isentropic expansion of superheated steam through the nozzle is
(a) 1.4, (b) 1.3, (c) 1.135, (d) 1.113
- (ii) Critical pressure ratio $\frac{P_2}{P_1}$ for steam nozzle in terms of index n for isentropic expansion is given by :
 (a) $\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$ (b) $\left(\frac{n+1}{2}\right)^{\frac{n-1}{n}}$ (c) $\left(\frac{n}{n-1}\right)^{\frac{2}{n+1}}$ (d) $\left(\frac{n-1}{n}\right)^{\frac{n+1}{2}}$
- (iii) For a convergent-divergent nozzle, the mass flow rate remains constant, if the ratio of exit pressure and inlet pressure is
(a) more than critical pressure ratio, (b) less than critical pressure ratio, (c) unity, (d) infinity.
- (iv) For a convergent-divergent nozzle, critical pressure ratio occurs when
(a) nozzle efficiency is maximum,
(b) friction is zero,
(c) decrease in ratio of exit pressure and inlet pressure does not increase steam flow rate.
- (v) The kinetic energy lost in friction is transformed into heat which tends to
(a) dry or superheat the steam,
(b) cool or condense the steam,
(c) increase the pressure of the steam,
(d) decrease the specific volume of steam.
- (vi) The velocity of steam in the divergent portion of a convergent-divergent nozzle is
(a) subsonic, (b) sonic, (c) supersonic.
- (vii) Semi-cone angle of the divergent part of the convergent-divergent steam nozzle is of the order of
(a) 3° to 10°, (b) 13° to 20°, (c) 23° to 30°, (d) 33° to 40°.

[(i) b, (ii) a, (iii) b, (iv) c, (v) a, (vi) c, (vii) a]

- 4 What is the function of a steam nozzle ? Mention the types of nozzles, you know.

Steam flows through a properly designed nozzle and the pressure drops from 12 bar to 0.15 bar. Assuming frictionless adiabatic flow, calculate the dryness fraction and velocity of steam as it leaves the nozzle, when the steam at the higher pressure is (a) dry saturated, and (b) superheated by 60°C.

[(a) 0.7952 dry, 1,159 m/sec; (b) 0.834 dry, 1,197.5 m/sec.]

- 5 What do you understand by the term critical pressure as applied to steam nozzles ?

Dry saturated steam at pressure of 700 kPa is expanded in a convergent-divergent nozzle to 100 kPa.

The mass of steam passing through the nozzle is 270 kg per hour. Assuming the flow to be frictionless adiabatic, determine the throat and exit diameters

[0.966 cm; 1.332 cm]

- 6 Dry saturated steam at a pressure of 12 bar is supplied to a convergent-divergent nozzle and is delivered at a pressure of 0.15 bar. Determine the diameters at the throat and exit of the nozzle if the delivery of steam is 18 kg per minute. Assume frictionless adiabatic flow.

[1.486 cm; 5.125 cm]

- 7 Calculate the diameters at the throat and exit of a nozzle which is to discharge 150 kg of steam per hour. The steam supply to the nozzle is dry saturated at a pressure of 5 bar and the pressure at exit is 0.6 bar. Assume index of expansion for steam as 1.135. Neglect effect of friction

[0.849 cm; 1.24 cm]

- 8 Explain the term "critical pressure" as applied to steam nozzles. Why are the turbine nozzles made divergent after the throat ?

Steam is supplied at a dryness fraction of 0.95 and pressure of 14 bar to a convergent-divergent nozzle and expands down to a back pressure of 0.4 bar. The outlet area of the nozzle is 10 cm². Assuming frictionless adiabatic flow through the nozzle, determine :

- (a) the steam flow in kg per hour, and
(b) the diameter of the nozzle at throat.

[(a) 1,196.3 kg; (b) 1.438 cm]

- 9 A convergent-divergent nozzle is required to discharge 300 kg of steam per hour. The nozzle is supplied with steam at a pressure of 10 bar and 90 per cent dry and discharges against a back pressure of 0.3 bar. Assuming frictionless adiabatic flow, determine the throat and exit diameters.

[0.842 cm; 2.047 cm]

10. Steam is supplied at a dryness fraction of 0.97 and 10 bar to a convergent-divergent nozzle and expands down to a back pressure of 0.3 bar. The throat area is 5 cm². Assuming frictionless adiabatic flow, determine:

- (a) The steam flow in kg. per minute, and
(b) The nozzle outlet area.

[(a) 43.37 kg.; (b) 29.28 cm²]

- 11 A nozzle is required to discharge 8 kg of steam per minute. The nozzle is supplied with steam at 11 bar and 200°C and discharges against a back pressure of 0.7 bar. Assuming frictionless adiabatic flow, determine:

- (a) the throat area, (b) the exit velocity and (c) the exit area. Take K_p of superheated steam as 2.1 kJ/kgK.

[(a) 0.848 cm²; (b) 963.6 m/sec; (c) 2.875 cm²]

- 12 Steam expands from 13 bar and 10°C superheat to 1.4 bar in a convergent-divergent nozzle. The mass of steam passing through the nozzle is 1,800 kg per hour. Assuming the flow to be frictionless adiabatic, determine the condition of steam and the diameters of the nozzle at the throat and exit. Assume that for maximum discharge the throat pressure is 7 bar. Take k_p of superheated steam as 2.1 kJ/kg K.

[(a) 0.964 dry, 1.854 cm; 0.879 dry, 2.802 cm]

- 13 Show that the maximum discharge of steam per unit area, through a nozzle, takes place, when the ratio of the steam pressure at the throat to the inlet steam pressure is

$$\left(\frac{2}{n+1} \right)^{\frac{n}{n-1}} \quad \text{where } n \text{ is the index of adiabatic expansion.}$$

Calculate the discharge in kg/m² at the throat of a nozzle, supplied with dry saturated steam at 700 kPa.

[1020.9 kg/m²]

- 14 From first principles, prove that maximum discharge in a steam nozzle per unit area at the throat is given by

$$M_{\max} = \sqrt{1,000 n \frac{p_1}{v_1} \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}}}$$

where, p_1 = Initial pressure of steam in kPa,

v_1 = Volume of steam in m³/kg at the initial pressure, and

n = Index of expansion.

- 15 A convergent-divergent nozzle is supplied with dry saturated steam at 1,200 kPa. If the divergent portion of the nozzle is 11 cm long and the throat diameter is 1.2 cm, determine the semi-vertical angle of the cone so that steam may leave the nozzle at 15 kPa. Assume frictionless adiabatic flow. [7°-33']
- 16 (a) Draw the "discharge versus ratio of pressures at outlet to inlet" curve for a convergent steam nozzle. Discuss the physical significance of critical pressure ratio.
- (b) Dry saturated steam at a pressure of 8 bar enters a convergent-divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic and the corresponding expansion index is 1.135, find the ratio of cross-sectional areas at exit and throat for maximum discharge. [1.592]
- 17 Dry saturated steam at 1.8 bar is allowed to discharge through a long convergent nozzle into the atmosphere. Taking atmospheric pressure as 1 bar, calculate the mass of steam which should be discharged per second if the exit diameter of the nozzle is 1.2 cm. Neglect friction in the nozzle.
- If the mass of steam actually discharged be 94% of the calculated mass, estimate the percentage of enthalpy drop which is wasted in friction. [0.0309 kg/sec; 11.7%]
- 18 Explain the term "Nozzle efficiency".
- A convergent-divergent nozzle is to pass 4,000 kg of steam per hour. Initially the steam is 0.98 dry at 21 bar and finally it is at 0.7 bar. Assuming that the friction loss in the divergent part is 18 per cent of the total isentropic enthalpy drop, determine the required areas of the throat and outlet. [3.688 cm²; 23.453 cm²]
- 19 A convergent-divergent nozzle is required to pass 360 kg of steam per hour with a pressure drop from 13 bar to 0.15 bar. The steam at the higher pressure is dry saturated. Assuming that the frictional resistance occurs only between throat and exit and is equivalent to 13 per cent of the total isentropic enthalpy drop, determine the diameters at the throat and exit. [0.827 cm; 3.114 cm]
- 20 A convergent-divergent nozzle is to be designed to discharge 0.075 kg of steam per second into a vessel in which the pressure is 1.4 bar, when nozzle is supplied with steam at 7 bar and also superheated to 200°C. Find the throat and exit diameters on the assumption that the friction loss in the divergent part is 10% of the total isentropic enthalpy drop. Take k_p of superheated steam as 2.3 kJ/kg K. [0.971 cm; 1.245 cm]
- 21 Steam at a pressure of 10 bar and a dryness fraction of 0.97 is to be discharged through a convergent-divergent nozzle to a back pressure of 0.15 bar. The mass flow rate through the nozzle is at a rate of 8 kg/kW-hr. If the turbine develops 150 kW, determine : (i) the throat pressure, (ii) the number of nozzle required, the diameter of nozzle at throat being 6.5 mm, and (iii) suitable exit diameter of the nozzle, assuming that 10% of the overall isentropic enthalpy drop reheats the steam in the divergent portion of the nozzle. [(i) 5.82 bar; (ii) 7; (iii) 2.175 cm]
- 22 (a) State what is meant by expansion of steam (i) under stable adiabatic conditions, and (ii) under conditions of supersaturation.
- (b) Discuss the causes of supersaturated flow in nozzles.
- (c) Explain what is meant by the supersaturated expansion of steam and give some idea of the limits within which this condition is possible.
- (d) Steam is expanded in a nozzle from an initial pressure of 10 bar and a temperature 200°C, to a final pressure of 2.5 bar. The expansion is supersaturated.
- Determine : (a) the final condition of steam, (b) the exit velocity of steam, (c) the degree of undercooling, (d) the degree of supersaturation, (e) the actual enthalpy drop, and (f) the isentropic enthalpy drop.
- Compare the mass flow through the above nozzle with one in which expansion takes place under conditions of the thermal equilibrium.
- For supersaturated conditions use the following relationship :
- $$v = \frac{0.233(H - 1,940)}{p}; \quad pv^{1.3} = \text{constant}; \quad \text{and} \quad \frac{p}{(T)^{1.3}} = \text{constant.}$$
- where v is the specific volume in m³/kg, H is the steam enthalpy in kJ per kg, p is the pressure in kPa, and T is the absolute temperature.
- Take k_p of superheated steam as 2.3 kJ/kg K.
- [(a) 0.938 dry; (b) 696.77 m/sec; (c) 56.69°C
(d) 7.764, (e) 242.76 kJ/kg; (f) 254.71 kJ/kg; 9.48%]
- 23 Describe with a neat sketch the working of a steam injector used for a locomotive boiler. Derive the formula for the amount of water injected into the boiler per kilogram of steam.
- 24 An injector is required to deliver 100 kg of water per minute from a tank whose constant water level is

1.2 metres below the level of the injector into a boiler in which the steam pressure is 4 bar. The water level in the boiler is 1.5 metres above the level of the injector. The steam for the injector is to be taken from the same boiler and it is to be assumed as dry saturated. The temperature of the water in the supply tank is 15°C. Find : (a) the mass of water taken from the supply tank per kg of steam, (b) the diameter of the throat of the mixing nozzle, (c) the diameter of the throat of the steam nozzle, and (d) the temperature of the water leaving the injector. Neglect the radiation losses.

[(a) 12.395 kg; (b) 0.9 cm ; (c) 1.708 cm; (d) 62.6°C]

25 Write short notes on the following, illustrating your answers with neat sketches wherever necessary :

- (a) Types of steam nozzles,
- (b) Effect of friction on the flow of steam through convergent-divergent steam nozzles.
- (c) Effect of supersaturated flow in steam nozzles, and
- (d) Steam injector.

STEAM TURBINES

9.1 Introduction

From the early days of the reciprocating steam engines, many attempts were made to develop power from steam without the necessity of the reciprocating mechanism. Modern steam turbine is the result of these efforts. The steam turbine differs from the reciprocating steam engine, both in mechanical construction and in the manner in which power is generated from the steam.

In the reciprocating steam engine a to and fro motion is imparted to the engine piston by the pressure of the steam acting upon it, and this reciprocating motion is converted into rotary motion at the crankshaft through the medium of the crosshead, connecting rod and crank. The expansive property of the steam is not utilized to the fullest, even in the best types of multi-expansion steam engines.

In the steam turbine, rotary motion is imparted directly to the shaft by means of high

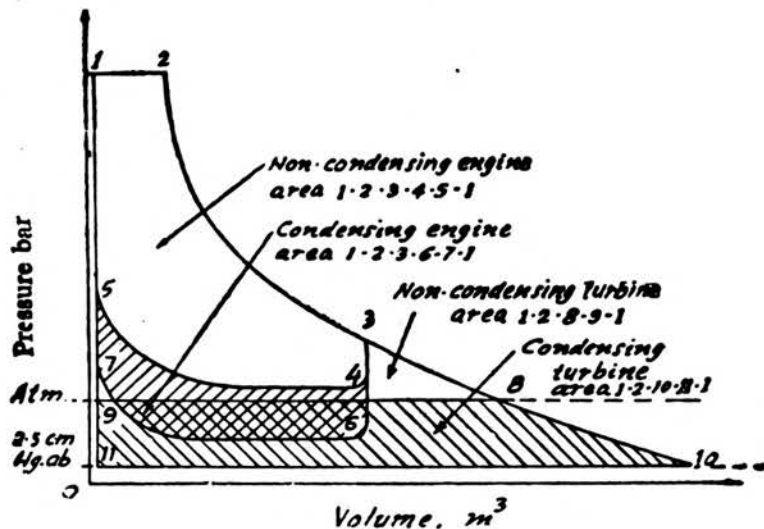


Fig. 9-1. Comparison of gain in work by fitting condenser to a reciprocating steam engine and steam turbine.

velocity steam jets striking the blades fixed on the rim of a wheel which is fastened to the shaft. The turbine is much simpler in mechanical construction, and it utilizes the kinetic or velocity energy of the steam instead of pressure only. The expansive property of the steam is almost utilized in the turbine (fig. 9-1) either in the admission nozzles or in the turbine blading.

Steam turbines are capable of expanding the steam to the lowest exhaust pressure obtainable in the condenser because they are steady flow machines and many

have large exhaust outlets (with no valves) through which the spent (used) steam must be discharged. Steam engines, however, are intermittent (non-continuous) flow machines and must force the expanded steam out through the relatively small exhaust valve. The lowest practical exhaust pressure for most steam engines is therefore 15 to 20 cm of mercury absolute (i.e. 0.2 to 0.3 bar). Steam turbines may expand steam to 2.5 cm of mercury absolute pressure or less.

The main advantages of steam turbine over the reciprocating steam engine are as follows :

- (i) With the turbine much higher speeds may be developed, and a far greater speed range is possible than in the case of the reciprocating steam engine. Because of this, turbine units are much smaller for same power than reciprocating steam

engine units and this in turn means, less floor space is required and may be built to produce very large power.

- (ii) Since the turbine is a rotary machine, perfect balancing is possible. This means foundation of the turbine is lighter and smaller.
- (iii) The ability of turbine to use high pressure and superheated steam and uniflow direction of steam flow through the turbine, combined with its greater range of expansion and ability to utilize a high vacuum to greater advantage, make the steam turbine much more efficient and economical than the reciprocating steam engine for power generation. The mechanical friction losses are very small in case of turbine. The thermal efficiency of the steam turbine therefore is over 30% compared with about 16% efficiency of the best steam engine.
- (iv) The working of the turbine is much smoother than that of the steam engine. The speed of rotation (r.p.m.) is uniform. The torque produced by the turbine is uniform and there is practically no vibration.
- (v) As no internal lubrication is needed, highly superheated steam can be used and exhaust steam contains no lubricating oil.

The steam turbine when properly designed and constructed, is the most durable prime-mover.

The reciprocating steam engine still possesses certain advantages over the steam turbine where frequent stopping, starting, reversing or change of speed may be necessary or where engines are required to operate non-condensing. Mine hoists, locomotives, drilling engines for wells and some types of mill and factory engines are preferably of the reciprocating type for the above reasons. The turbine is a constant high speed machine and really must be operated condensing in order to take full advantage of its greater range of steam expansion.

9.2 Types of Steam Turbines

Steam turbines may be classified into three main types according to the working principles, namely, **impulse turbines**, **reaction turbines** and **combined turbines (impulse-reaction turbines)**.

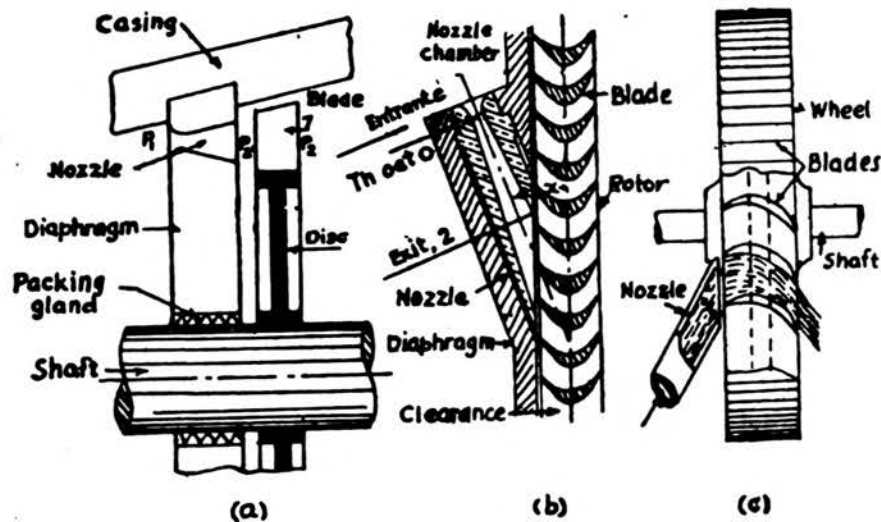


Fig. 9-2.

- (a) Diagrammatic view of a simple impulse turbine.
- (b) Arrangement of blades and nozzle for a simple impulse turbine.
- (c) Simple reaction wheel.

The turbines in which complete process of expansion of steam takes place in stationary nozzle and the velocity energy is converted into mechanical work on the turbine blades, are known as impulse turbines.

An impulse turbine depends almost wholly for its operation on the impulsive force of high velocity steam jet or jets. The high velocity steam jets are obtained by expansion of the steam in the stationary nozzles only, and the steam then passes at high velocity through the moving blades with no drop in pressure but a gradual reduction in velocity. In short, in purely impulse turbines the rotary motion of the shaft is obtained by having high velocity jets of steam directed against the blades attached to the rim of the turbine wheel or rotor. Fig. 9-2(a) illustrates diagrammatic view of a simple impulse turbine. Fig. 9-2(b) shows the arrangement of blades on the rotor and diaphragm carrying convergent-divergent nozzle. The nozzle axis is inclined at a fixed angle to the tangent of the rotor wheel.

A pure reaction turbine [fig. 9-2(c)] is one in which the drop of pressure with expansion and generation of kinetic energy takes place in the moving blades. The steam jets leave the moving blades at greater velocities than those at which they enter these blades. The jets of steam from the moving blades react on the blades and turn them round.

The passages through the moving blades are made convergent so that the steam expands while passing through them, which causes the steam to leave the blades at higher velocity than that at which it entered. The backward motion of the blades is similar to the recoil of a gun when fired. A pure reaction turbine is of little practical importance.

In modern reaction turbines both the impulse and reaction principles work together. The pressure drop is effected partly in the fixed guide blades which are designed to work as nozzles and partly in the moving blades which are also so designed that expansion of the steam takes place in them. The high velocity issuing jet from the fixed guide blades, produces an impulse on the moving blades and the jet coming out at still higher velocity from the moving blades produces a reaction. Therefore, part of the work is due to impulse and the remainder due to the reaction. However, these turbines work more on reaction principle than on impulse. These turbines are generally called reaction turbines but the more correct term should be *impulse-reaction turbines*. A very good example of reaction turbine is a Parsons turbine. In a reaction turbine, the stationary blades and the moving blades are virtually convergent nozzles so that the steam passing through them suffers a fall in pressure.

The circumferential speed of the moving blades is kept the same as the velocity of the steam that enters the blades. This ensures that the steam will flow into the blades without striking them.

Steam turbines may further be classified according to their position of shaft, nature of steam supply, direction of steam flow, construction and arrangement of blades and wheels, and number of stages in expansion. Thus, steam turbines may be further classified (i) according to the position of shaft axis, they are horizontal or vertical, (ii) according to their nature of steam supply and use to which steam is put, they are high-pressure or low-pressure, and bleeder or extraction, (iii) according to the direction of steam flow, they are axial, radial, tangential, single-flow or double-flow, (iv) according to their construction and arrangement of blades and wheels, they are pressure compounded or velocity compounded, and (v) according to number of stages, they are single-stage, two-stage, etc.

As an example of the use of these classifications, we might describe a particular turbine as a horizontal, high-pressure, axial flow, reaction, two-stage, condensing turbine.

9.3 Impulse Steam Turbine

It has been already pointed out that the essential parts of an *impulse turbine* are the *nozzles and blades*. In nozzle the expansive property of the steam is utilized to produce a jet of steam moving with very high velocity. The function of the blades is to change the direction and hence momentum of the jet or jets of steam and so to produce a force which will rotate the blades. It is a matter of prime importance that we should be able to estimate the propelling force that would be applied to a turbine rotor under any given set of conditions. This will also help to estimate the work done and hence the power. Since the force is due to a change of momentum caused mainly by the change in the direction of flow, it becomes essential to draw velocity diagram showing how the velocity of the steam varies during its passage through the blades.

9.3.1 Velocity diagram for moving blades : Fig. 9-3(a) shows the nozzle and blades either of single-stage impulse turbine or of one stage of a multi-stage turbine. Steam enters the nozzle at pressure p_0 and issues from nozzle at pressure p_1 . The velocity of the steam at the nozzle exit is V_1 and it is at an angle α_1 to the tangent of the wheel at the entrance to the moving blades. The tangential component of entering steam V_{w1}

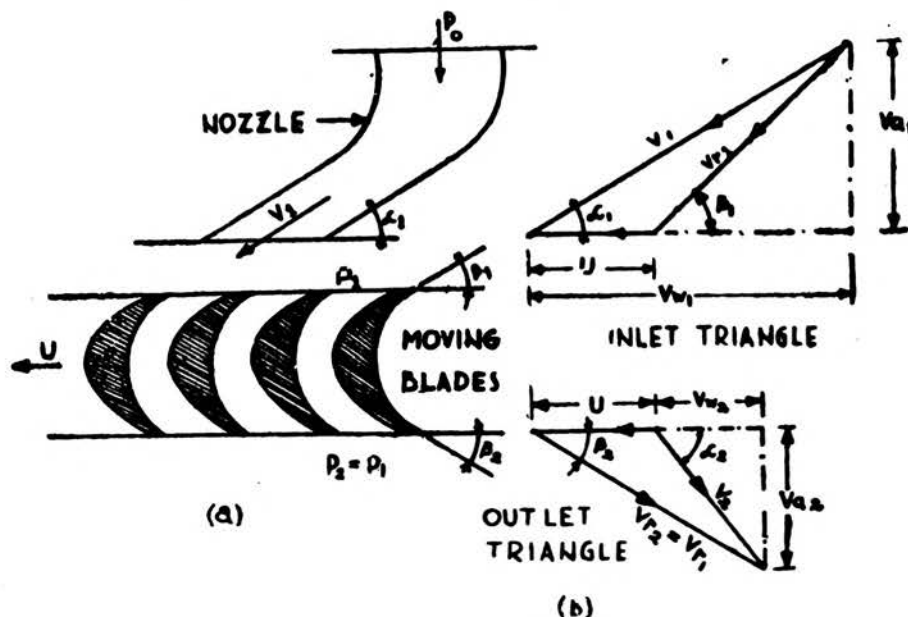


Fig. 9-3. Velocity diagram.

commonly known as *velocity of whirl*, does work on the blades. The axial component V_{a1} of the entering steam jet does not work on the blades because it is perpendicular to the direction of the motion of the blades. This component is also known as the *velocity of the flow or axial velocity*, and it is responsible for the flow of steam through the turbine. Change of velocity in this component causes an axial thrust on the rotor.

As the blade is moving with a tangential velocity u m/sec., the entering jet will have relative velocity of the blades of V_{r1} which makes an angle of β_1 to the wheel tangent. This relative velocity may be obtained by subtracting the vector of blade velocity (u) from velocity of steam (V_1) i.e. $\vec{V}_{r1} = \vec{V}_1 - \vec{u}$. This is shown in fig. 9-3 (b) for velocity triangle at inlet. In order to avoid shock at entry, vector V_{r1} must be tangential to the blade tip at entry, i.e. β_1 must be equal to the angle of blade at entrance.

A similar vector diagram is shown at the outlet tip of the moving blade. The steam glides off the blade with a relative velocity of V_{r2} inclined at an angle β_2 to the tangent; by adding the vector of blade velocity (u) to V_{r2} , the absolute velocity of the leaving

steam (V_2) is obtained. Its inclination is α_2 to the tangent. Having obtained the vector

V_2 its tangential component or velocity of whirl V_{w2} and also its axial component or velocity of flow V_{a2} can be drawn. This completes velocity diagram at exit.

For convenience in solving problems on turbine blades, it is usual to combine the two velocity diagrams of fig. 9-3(b) on a common base representing the blade velocity u . This has been done in fig. 9-4, which shows the complete velocity diagram. This is obtained by turning the inlet diagram through 180° , and by superimposing it on the outlet diagram so that vector u coincides for both diagrams.

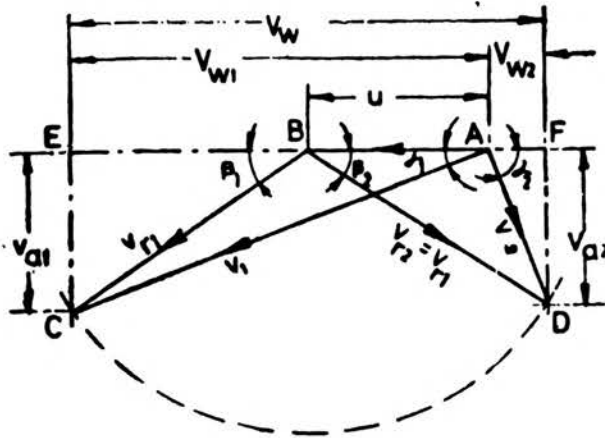


Fig. 9-4. Combined velocity diagram.

In an impulse turbine the relative velocity at inlet V_{r1} has the same magnitude as the relative velocity at outlet V_{r2} if friction is neglected. This is so as there is no fall in steam pressure as it flows over the blades i.e. $V_{r1} = V_{r2}$. The length of the vector V_{r2} may be obtained by drawing a circular arc of radius V_{r1} and centred at B.

It will be noticed that the horizontal distance between the apexes of the inlet and outlet diagrams represented by the distance EF, is the vector difference of V_{w1} and V_{w2} ,

Or change of velocity of whirl

$$= V_{w1} \pm V_{w2} = EF$$

9.3.2 Forces on the blade and work done : Since $V_1(AC)$ is the initial absolute velocity and $V_2 (AD)$ is the final absolute velocity of the steam, the change of velocity which the steam undergoes in passing through the blades is represented by the vector CD (when the apexes of inlet and outlet triangles are joined) = vector $(V_2 - V_1)$ of fig. 9-4.

In general the vector $(V_2 - V_1)$ will not be parallel to u , so that only the tangential component V_w will do useful work; whilst the normal component $(V_{a1} - V_{a2})$ produces an end thrust on the rotor.

Let m = steam flow through blades in kg per sec.

From Newton's second law of motion,

Tangential force on wheel = mass \times acceleration in tangential direction

$$= \text{mass} \times \frac{\text{change of velocity}}{\text{time}}$$

$$= \text{mass per sec} \times \text{change of velocity}$$

$$= m(V_{w1} - V_{w2}) \text{ N} \quad \dots (9.1)$$

It should be noticed from fig. 9-4 that V_{w2} is actually negative as the steam is discharged in the opposite direction to the blade motion. This means that the values of V_{w1} and V_{w2} are added together in eqn. (9.1). Thus writing eqn. (9.1) in a more general way,

$$\text{Tangential force on wheel} = m(V_{w1} \pm V_{w2}) \text{ N} \quad \dots (9.2)$$

+ ve sign is to be used when V_{w2} and u are in opposite direction as shown in fig.

9.4 and - ve sign is to be used when V_{w2} and u are in same direction as shown in fig. 9-9.

$$\begin{aligned} \text{Work done on blade} &= \text{force} \times \text{distance travelled.} \\ &= m(V_{w1} \pm V_{w2}) \times u \text{ N.m/sec. or Joules/sec.} \end{aligned} \quad \dots (9.3)$$

$$\text{Power developed by the wheel} = \frac{m \times (V_{w1} \pm V_{w2})u}{1,000} \text{ kJ/sec. or kW} \quad \dots (9.4)$$

This power is termed as the *rim power* to distinguish it from the actual power transmitted to the shaft.

Blade efficiency : Since available energy of the steam entering the blade is,

$$\frac{m(V_1)^2}{2,000} \text{ kJ/sec.,}$$

the efficiency of the blade alone, $\eta_b = \frac{\text{Work done on the blades}}{\text{Energy supplied to the blades}}$

$$\begin{aligned} &= \frac{\frac{m \times (V_{w1} \pm V_{w2})u}{1,000}}{\frac{m(V_1)^2}{2,000}} = \frac{2u(V_{w1} \pm V_{w2})}{(V_1)^2} \end{aligned} \quad \dots (9.5)$$

The blade efficiency is also called *diagram efficiency* as this is obtained with the help of velocity diagrams.

Stage efficiency : If H_1 and H_2 be the enthalpies before and after expansion through the nozzle, then $(H_1 - H_2)$ is the enthalpy drop (H) in kJ/kg through a stage of fixed blade rings and moving blade rings.

$$\begin{aligned} \text{Stage efficiency, } \eta_s &= \frac{\text{Work done on blade per kg of steam}}{\text{Total energy supplied per stage per kg of steam}} \\ &= \frac{\frac{u(V_{w1} + V_{w2})}{1,000}}{(H_1 - H_2)} = \frac{u(V_{w1} + V_{w2})}{1,000 H} \end{aligned} \quad \dots (9.6)$$

$$\text{Now, nozzle efficiency} = \frac{\frac{(V_1)^2}{2,000}}{(H_1 - H_2)} = \frac{(V_1)^2}{2,000 H} = \frac{(V_1)^2}{2,000 H}$$

$$\begin{aligned} \therefore \text{Blade efficiency} \times \text{nozzle efficiency} &= \frac{2u(V_{w1} + V_{w2})}{(V_1)^2} \times \frac{(V_1)^2}{2,000 H} \\ &= \frac{u(V_{w1} + V_{w2})}{1,000 H} = \text{Stage efficiency} \end{aligned}$$

Axial thrust : The axial thrust on the wheel is due to the difference between the velocities of flow at entrance and outlet.

$$\begin{aligned} \text{Axial force on wheel} &= \text{mass} \times \text{acceleration in axial direction} \\ &= \text{mass per sec.} \times \text{change of axial velocity} \\ &= m(V_{a1} - V_{a2}) \text{ N} \end{aligned} \quad \dots (9.7)$$

The axial force or axial thrust or end thrust on the wheel must be balanced or must be taken by a thrust bearing. It may be noted from eqn. (9.7) that the axial thrust is zero if $V_{a1} = V_{a2}$.

Energy converted to heat by blade friction :

Energy converted to heat by blade friction = Loss of kinetic energy during flow over blade

$$= m \frac{(V_{r1})^2}{2,000} - m \frac{(V_{r2})^2}{2,000} \text{ kJ/sec.} \quad \dots (9.8)$$

where m is steam flow per second.

Problem-1 : Steam issues from the nozzle of a simple impulse turbine with a velocity of 900 m/sec. The nozzle angle is 20° , the mean diameter of the blades is 25 cm and the speed of rotation is 20,000 r.p.m. The mass flow through the turbine nozzles and blading is 0.18 kg of steam per sec. Draw the velocity diagram and derive or calculate the following : (a) Tangential force on blades, (b) Axial force on blades, (c) Power developed by the turbine wheel, (d) Efficiency of the blading, and (e) Inlet angles of

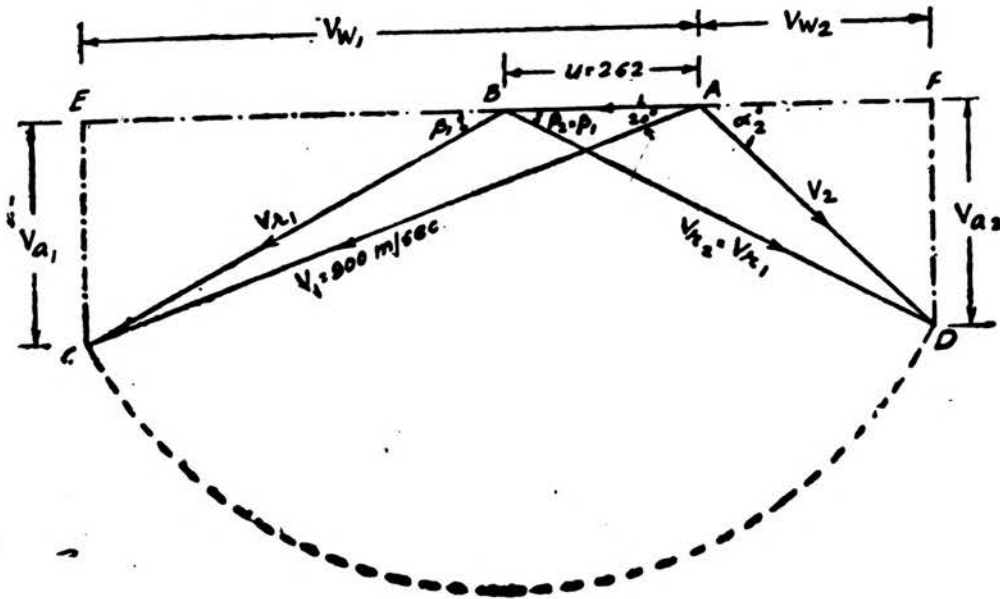


Fig. 9-5. Velocity diagram.

blades for shockless inflow of steam.

Assume that the outlet angle of the blades is equal to the inlet angle and frictional losses are negligible.

Blade speed, $u = \frac{\pi D \times N}{60} = \frac{\pi \times 25}{100} \times \frac{20,000}{60} = 262 \text{ m/sec.}$

The velocity inlet triangle ABC (fig.9-5) may now be constructed to some convenient scale and the following results are obtained graphically (a graphical solution is to be preferred, although calculation is equally possible) :

- Relative velocity at entrance, $V_{r1} = 650 \text{ m/sec.},$
- Axial velocity at inlet, $V_{a1} = 307.8 \text{ m/sec.},$
- Tangential component at inlet, $V_{w1} = 835.7 \text{ m/sec.},$ and
- Inlet blade angle, $\beta_1 = 28.2^\circ.$

Since friction losses are negligible, $V_{r2} = V_{r1} = 650 \text{ m/sec.}$

Also the outlet blade angle, $\beta_2 = \beta_1 = 28.2^\circ.$

The velocity exit triangle ABD may now be constructed. The additional results obtained

are :

$$V_{a2} = V_{a1} = 307.8 \text{ m/sec.}, \text{ and } V_{w2} = 310.8 \text{ m/sec.}$$

(a) Using eqn. (9.2)

$$\begin{aligned} \text{Tangential force on blades} &= m \times (V_{w1} + V_{w2}) \\ &= 0.18 \times (835.7 + 310.8) = 206.37 \text{ N} \end{aligned}$$

(b) As there is no change in axial component of velocity, i.e. $V_{a1} = V_{a2}$, the axial force on blades is **zero**.

(c) From eqn. (9.4),

$$\begin{aligned} \text{Power developed by the wheel} &= \frac{m(V_{w1} + V_{w2})u}{1,000} \\ &= \frac{0.18 \times (835.7 + 310.8) 262}{1,000} = 54.1 \text{ kW} \end{aligned}$$

(d) From eqn. (9.5),

$$\begin{aligned} \text{Efficiency of the blading, } \eta_b &= \frac{2u(V_{w1} + V_{w2})}{(V_1)^2} \\ &= \frac{2 \times 262(835.7 + 310.8)}{(900)^2} = 0.742 \text{ or } 74.2\% \end{aligned}$$

(e) Inlet angles of blades (for shockless inflow of steam), $\beta_1 = 28.2^\circ$ (from velocity diagram).

9.3.3 Effect of blade friction on velocity diagram : In an impulse turbine the relative

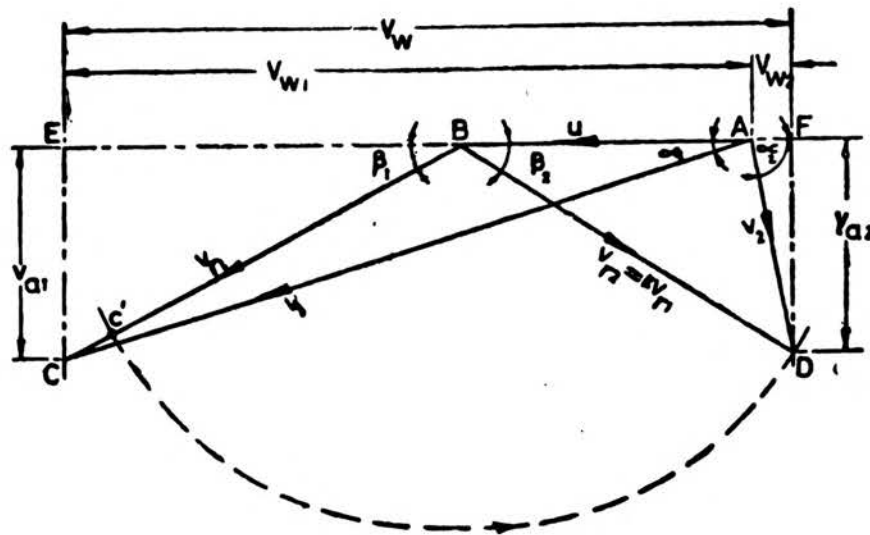


Fig. 9-6. Combined velocity diagram considering friction losses.

velocity will remain unaltered as it passes over the blades, if friction is neglected. In practice the flow of steam over the blades is resisted by friction. The effect of this friction is to reduce the relative velocity of steam as it passes over the blades. In general, there is a loss of 10 to 15 per cent in the relative velocity. Owing to friction in the blades, V_{r2} is less than V_{r1} and we may write,

$V_2 = kV_{r1}$ where, k represents the *blade velocity coefficient or friction factor*.

The velocity diagram of fig. 9-4 can be modified to allow for this blade friction by making $V_2 = k V_{r1}$; this modification is shown in fig. 9-6. In this diagram the inlet diagram is first drawn and the point C' on the BC is marked such that $BC' = kV_{r1}$. With compass centred on B , and arc of radius BC' is drawn to cut line BD at D . Line BD is drawn at given angle β_2 and AD is joined. The line AD then represents absolute velocity V_2 .

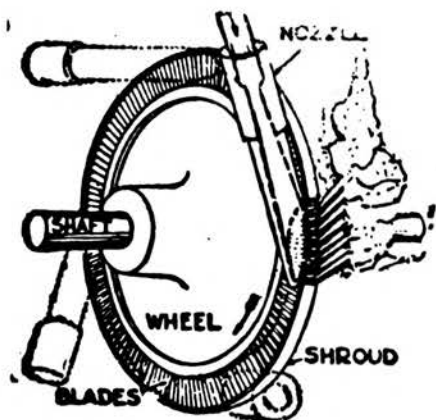


Fig. 9-7. De Laval turbine.

A view of this turbine is illustrated in fig. 9-7. The steam is expanded in nozzles which are inclined to the wheel tangent at an angle of about 20° . The smallest

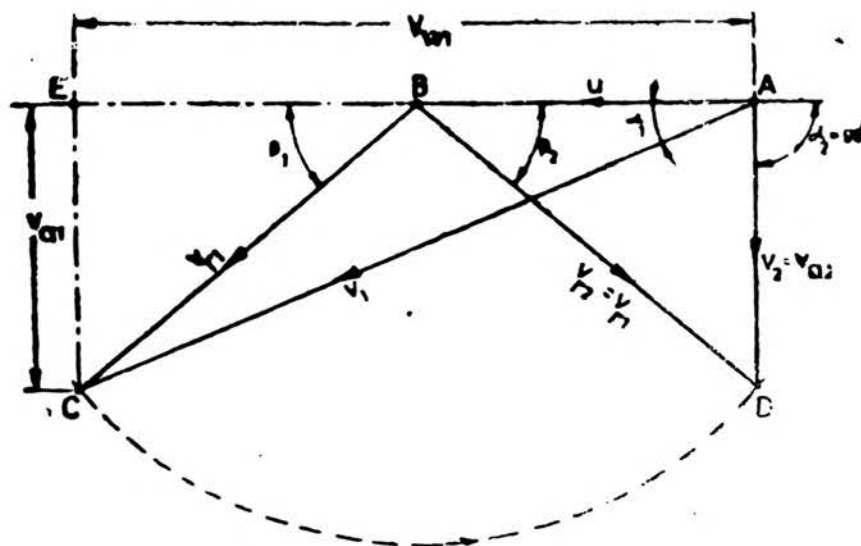


Fig. 9-8. Velocity diagram for De Laval turbine.

De Laval turbine constructed has a wheel diameter of 12.5 cm and a speed of 30,000 r.p.m. It is most suitable for low pressure steam supply. The blades are made symmetrical with angles of about 30° at inlet and outlet. The power developed is about 5 kW and the blade speed is 200 m per sec. It has spherical bearings. Helical gearing is used to reduce the high rotational speed of the wheel to a practical value, without undue noise or friction losses.

The velocity diagram for the De Laval blade is shown in fig. 9-8. Assuming no friction losses for the flow over the blades,

$$\text{Energy supplied per kg of steam} = \frac{(V_1)^2}{2,000} \text{ kJ.}$$

$$\text{Energy rejected per kg of steam} = \frac{(V_2)^2}{2,000} \text{ kJ.}$$

$$\text{Hence, work done per kg of steam} = \frac{(V_1)^2 - (V_2)^2}{2,000} \text{ kJ.}$$

It may be noted that the work done is maximum when V_2 is maximum i.e. when angle α_2 is 90° .

The smallest De Laval turbine constructed has a wheel diameter of 12.5 cm and a speed of 30,000 r.p.m. It is most suitable for low pressure steam supply. The blades are made symmetrical with angles of about 30° at inlet and outlet. The power developed is about 5 kW and the blade speed is 200 m per sec. It has spherical bearings. Helical gearing is used to reduce the high rotational

Considering ΔABD and ΔECB , $V_{r1} = V_{r2}$ (neglecting friction), also $\angle BAD = \angle BEC = 90^\circ$ (for maximum efficiency), and $\beta_1 = \beta_2$ (for the De Laval blade).

Hence, $\Delta ABD = \Delta ECB$.

$$\therefore AB = BE \text{ or } AB = \frac{1}{2} EA, \text{ i.e. } u = \frac{1}{2} V_{w1} = \frac{V_1 \cos \alpha_1}{2} \quad \dots (9.9)$$

$$\text{Also } V_2 = AD = EC = V_1 \sin \alpha_1 \quad \dots (9.10)$$

$$\begin{aligned} \text{Now, blade efficiency, } \eta_b &= \frac{\text{Work done on the blade}}{\text{Kinetic energy supplied to the blade}} \\ &= \frac{\frac{(V_1)^2}{2,000} - \frac{(V_2)^2}{2,000}}{\frac{(V_1)^2}{2,000}} = \frac{(V_1)^2 - (V_2)^2}{(V_1)^2} = \frac{(V_1)^2 - (V_1 \sin \alpha_1)^2}{(V_1)^2} \\ &= 1 - \sin^2 \alpha_1 = \cos^2 \alpha_1 \quad \dots (9.11) \end{aligned}$$

This is the maximum efficiency as α_2 has been assumed to be 90° . Putting α_1 equal to 20° , which is the value adopted in this turbine,

$$\text{Maximum blade efficiency} = \cos^2 20^\circ = 0.883 \text{ or } 88.3\%.$$

This is the theoretical value of the blade efficiency, the actual efficiency is only about 55%.

Although the original machine was great success for mathematicians, it suffered from many defects which made it compare unfavourably with reciprocating engines. *The speed of this wheel is too high to be of practical use.* The chief development of modern turbines has been to devise efficient methods to reduce this high speed; the methods used, such as compounding for velocity and pressure, will be dealt with later in this chapter.

Problem-2 : The rotor of an impulse turbine is 60 cm diameter and runs at 9,600 r.p.m. The nozzles are at 20° to the plane of the wheel, and the steam leaves them at 600 m/sec. The blades outlet angle are 30° and the friction factor is 0.8. Calculate the power developed per kg of steam per second and the diagram efficiency.

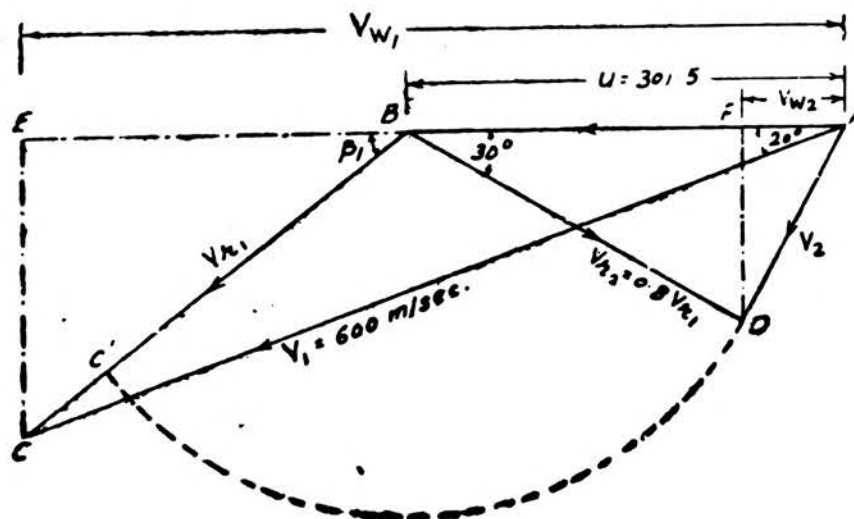


Fig. 9-9. Velocity diagram.

$$\text{Blade velocity, } u = \frac{\pi D N}{60} = \frac{\pi \times 60}{100} \times \frac{9,600}{60} = 301.5 \text{ m/sec.}$$

The velocity diagram may now be constructed to some convenient scale as shown in fig. 9-9. A graphical solution is to be preferred, although calculation is equally possible.

The inlet triangle ABC is readily constructed consisting of $u = 301.5 \text{ m/sec}$, $\alpha_1 = 20^\circ$ and $V_1 = 600 \text{ m/sec}$.

Hence from velocity diagram, $V_{r1} = 332 \text{ m/sec}$.

Since friction factor is 0.8, $V_{r2} = 0.8 \times V_{r1} = 0.8 \times 332 = 265.6 \text{ m/sec}$.

The exit triangle ABD can now be completed by drawing $V_{r2} = 265.6 \text{ m/sec}$ at 30° to u at B.

Hence from velocity diagram, $V_{w1} - V_{w2} = 497.32 \text{ m/sec}$.

From eqn. (9.4),

$$\begin{aligned} \text{Power developed} &= \frac{m(V_{w1} - V_{w2})u}{1,000} \text{ kW} \\ &= \frac{1 \times 497.32 \times 301.5}{1,000} = 150 \text{ kW} \end{aligned}$$

From eqn. (9.5),

$$\begin{aligned} \text{Diagram efficiency, } \eta_b &= \frac{2u(V_{w1} - V_{w2})}{(V_1)^2} \\ &= \frac{2 \times 301.5 \times 497.32}{(600)^2} = 0.836 \text{ or } 83.6\% \end{aligned}$$

Problem-3 : An impulse turbine with a single row wheel is to develop 99.3 kW, the blade speed being 150 m/sec.

A mass of 2 kg of steam per second is to flow from the nozzles at a speed of 350 m/sec. The velocity coefficient of the blades may be assumed to be 0.8 while the steam is to flow axially after passing through the blades ring. Determine the nozzle angle, and the blade angles at inlet and exit assuming no shock. Estimate also the diagram efficiency of the blading.

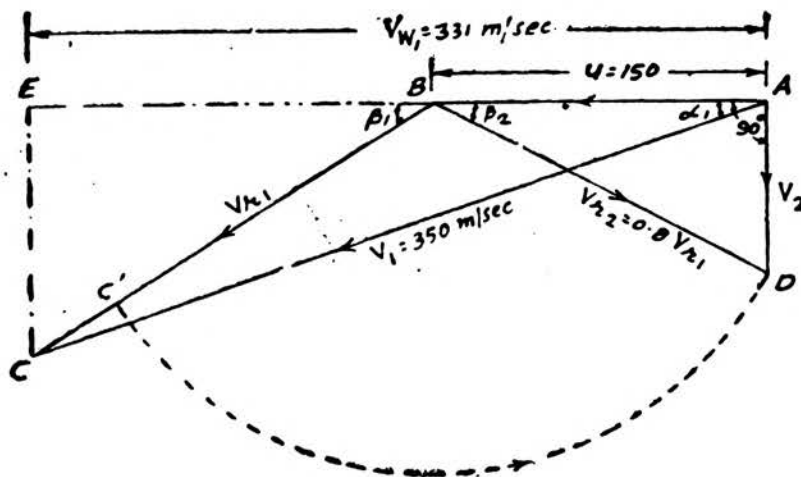


Fig. 9-10. Velocity diagram.

at right angle to the plane of the wheel, then angle BAD is 90° (fig. 9-10). It may be noted that the triangles cannot yet be constructed. The magnitude of $V_{w1} \pm V_{w2}$ can be calculated from eqn. (9.4).

$$\text{From eqn. (9.4), power developed} = \frac{m(V_{w1} \pm V_{w2})u}{1,000} \text{ kW}$$

$$\text{i.e. } 99.3 = \frac{2(V_{w1} \pm V_{w2})150}{1,000}$$

$$\therefore V_{w1} \pm V_{w2} = 331 \text{ m/sec.}$$

As the flow at exit is axial, $V_{w2} = 0 \therefore V_{w1} = 331 \text{ m/sec.}$

The inlet triangle ABC can be constructed to some convenient scale using

$$V_1 = 350 \text{ m/sec., } u = 150 \text{ m/sec, and } V_{w1} = 331 \text{ m/sec.}$$

Hence from the inlet triangle ABC , $V_{r1} = 213 \text{ m/sec.}$, required nozzle angle, $\alpha_1 = 18.7^\circ$, and required inlet blade angle, $\beta_1 = 31.75^\circ$.

$$\therefore V_{r2} = 0.8 V_{r1} = 0.8 \times 213 = 170.4 \text{ m/sec.}$$

Now, exit triangle ABD can be completed.

Hence, from the diagram, required exit blade angle, $\beta_2 = 28.3^\circ$.

Using eqn. (9.5),

$$\begin{aligned} \text{Diagram efficiency, } \eta_b &= \frac{2u(V_{w1} + V_{w2})}{(V_1)^2} \\ &= \frac{2 \times 150 \times 331}{(350)^2} = 0.81 \text{ or } 81\% \end{aligned}$$

Problem-4 : The steam leaves the nozzle of a single-stage impulse wheel turbine at 900 m/sec. The nozzle angle is 20° , the blade angles are 30° at inlet and outlet, and friction factor is 0.8 .

Calculate : (a) the blade velocity, and (b) the steam flow in kg per hour if the power developed by the turbine is 257 kW.

(a) The velocity inlet triangle may be drawn as shown in fig. 9-11 by making u to

any suitable length, say 3 cm , and setting up the given angles. The length of V_1 can then be measured and the scale of the diagram found, since $V_1 = 900 \text{ m/sec.}$ Hence, blade velocity, $u = 312 \text{ m/sec.}$

(b) From the inlet triangle ABC , $V_{r1} = 625 \text{ m/sec.}$

$$\begin{aligned} \therefore V_{r2} &= 0.8 V_{r1} \\ &= 0.8 \times 625 \\ &= 500 \text{ m/sec.} \end{aligned}$$

Now the exit triangle ABD can be completed by drawing $V_{r2} = 500 \text{ m/sec.}$ at 30° to u at B .

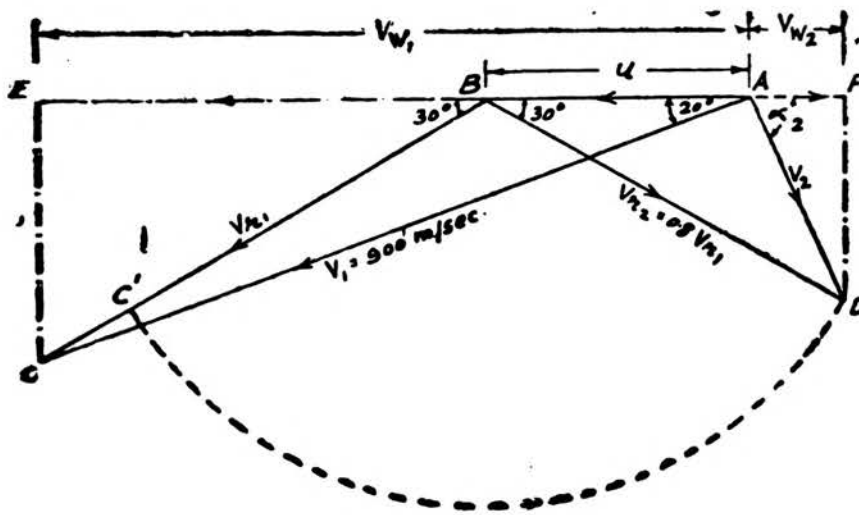


Fig. 9-11. Velocity diagram.

Hence, $V_{w1} + V_{w2} = 966.7 \text{ m/sec.}$ (from velocity diagram).

$$\text{Using eqn. (9.4), power developed in kW} = \frac{m(V_{w1} + V_{w2})u}{1,000}$$

i.e. $257 = \frac{m \times 966.7 \times 312}{1,000} \therefore m = \frac{257 \times 1,000}{966.7 \times 312} = 0.852 \text{ kg/sec.}$

\therefore Steam flow per hour = $0.852 \times 3,600 = 3,067.2 \text{ kg/hr.}$

Problem-5 : The outlet area of the nozzles in a simple impulse turbine is 15.5 cm^2 and the steam leaves them 0.91 dry at 1.4 bar and at 920 m/sec. The blade angles are 30° at inlet and exit, and the blade velocity is 0.25 of the steam velocity at the exit from the nozzle. The friction factor is 0.8 . Find : (a) the nozzle angle, (b) the power developed, (c) the diagram efficiency, and (d) the axial thrust on the blading.

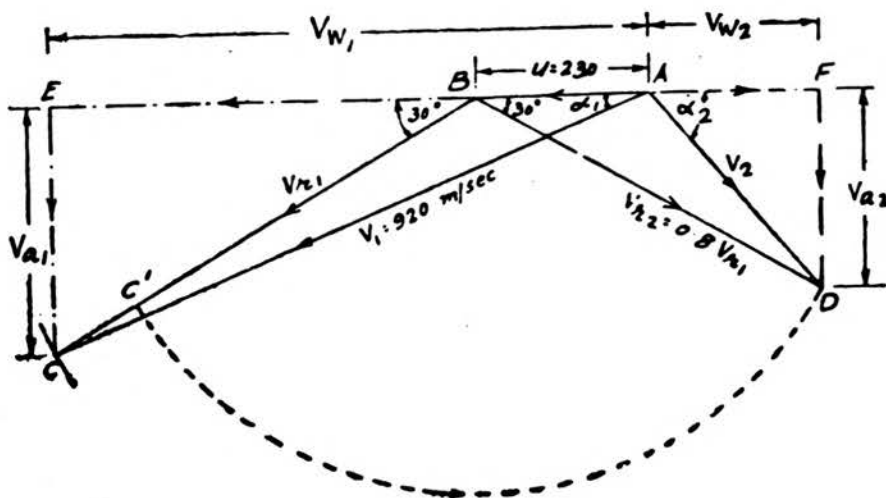


Fig. 9-12. Velocity diagram.

The velocity triangles may be constructed as shown in fig. 9-12 to some convenient scale.

(a) In the velocity diagram (fig. 9-12), $u = 0.25 \times 920 = 230 \text{ m/sec.}$ may be drawn. At B, the inlet blade angle of 30° is drawn. With A as centre and radius equal to 920 m/sec. , an arc is drawn to cut the line (drawn at 30°) at C. The inlet triangle ABC can now be completed. Hence, from the velocity inlet triangle ABC, the required nozzle angle, $\alpha_1 = 23^\circ$ and $V_{r1} = 715 \text{ m/sec.}$

$\therefore V_{r2} = 0.8 V_{r1} = 0.8 \times 715 = 572 \text{ m/sec.}$

The exit triangle ABD can be completed by drawing $V_{r2} = 572 \text{ m/sec.}$ at 30° to u at B.

Hence, from velocity diagram, $V_{w1} + V_{w2} = 1,113 \text{ m./sec.}$, $V_{a1} = 359 \text{ m/sec.}$, and $V_{a2} = 286 \text{ m/sec.}$

At 1.4 bar , from steam tables, $v_s = 1.2366 \text{ m}^3/\text{kg.}$

Steam flow through blades, $m = \frac{AV}{xv_s} = \frac{15.5 \times 920}{10^4 \times 0.91 \times 1.2366} = 1.267 \text{ kg/sec.}$

(b) From eqn. (9.4), Power developed

$= \frac{m(V_{w1} + V_{w2})u}{1,000} \text{ kW} = \frac{1.267 \times 1,113 \times 230}{1,000} = 324 \text{ kW}$

(c) From eqn. (9.5),

Diagram efficiency, $\eta_b = \frac{2u(V_{w1} + V_{w2})}{(V_1)^2}$

$$\eta_b = \frac{2 \times 230 \times 1,113}{(920)^2} = 0.605 \text{ or } 60.5\%$$

(d) From eqn. (9.7),

$$\text{Axial thrust on the blading} = m(V_{a1} - V_{a2}) = 1.267(359 - 286) = 92.5 \text{ N}$$

Problem-6 : A single stage impulse rotor has a blade ring diameter of 57.5 cm and rotates at a speed of 10,000 r.p.m. The nozzles are inclined at 20° to the direction of motion of the blades and the velocity of the issuing steam is 1,050 m/sec. Determine the inlet blade angle in order that the steam shall enter the blades passage without shock. Assume a friction coefficient of the blading equal to 0.85 and that the inlet and outlet angles are equal. Find also : (a) the power developed at the blades for a steam supply of 1,350 kg per hour, (b) the diagram efficiency, and (c) the loss of kinetic energy due to blade friction.

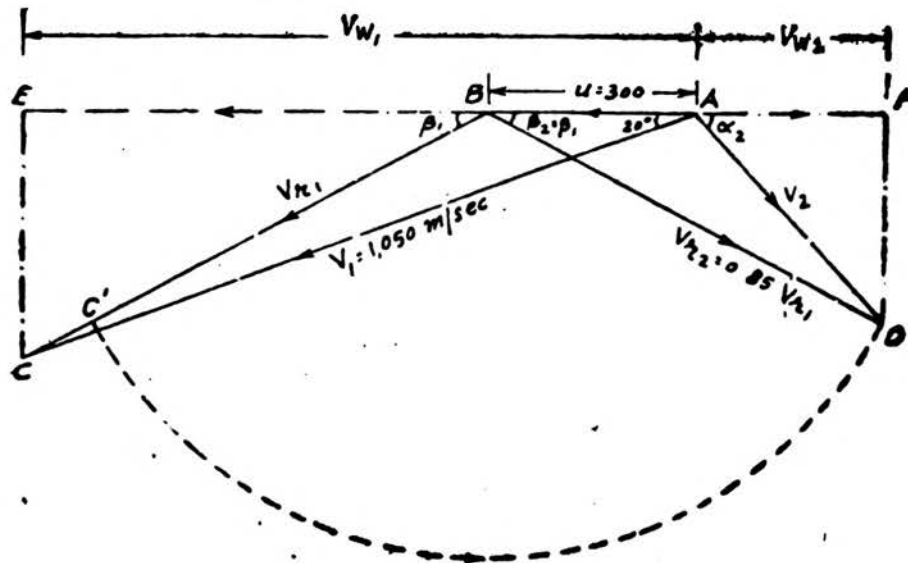


Fig. 9-13. Velocity diagram.

$$\text{Blade speed, } u = \frac{\pi D N}{60} = \frac{\pi \times 57.5}{100} \times \frac{10,000}{60} = 300 \text{ m/sec.}$$

The inlet triangle ABC (fig. 9-13), may now be constructed to some convenient scale and the following results are obtained :

Relative velocity at entrance, $V_{r1} = 775 \text{ m/sec.}$

Tangential component at inlet, $V_{w1} = 986 \text{ m/sec.}$ and inlet blade angle $\beta_1 = 27.6^\circ$

Also the blade outlet angle, $\beta_2 = \beta_1 = 27.6^\circ$

Since $V_{r2} = 0.85 V_{r1} \therefore V_{r2} = 0.85 \times 775 = 658 \text{ m/sec.}$

The exit triangle ABD can now be completed by drawing $V_{r2} = 658 \text{ m/sec.}$ at 27.6° to u at B.

Hence, from velocity diagram, $V_{w2} = 284 \text{ m/sec.}$ and $V_{w1} + V_{w2} = 1,270 \text{ m/sec.}$

(a) From eqn. (9.4),

$$\begin{aligned} \text{Power developed} &= \frac{m(V_{w1} + V_{w2})u}{1,000} \text{ kW} \\ &= \frac{1,350 \times 1,270 \times 300}{3,600 \times 1,000} = 142.8 \text{ kW} \end{aligned}$$

(b) From eqn. (9.5),

$$\begin{aligned} \text{Diagram efficiency, } \eta_b &= \frac{2u(V_{w1} + V_{w2})}{(V_1)^2} \\ &= \frac{2 \times 300 \times 1,270}{(1,050)^2} = 0.692 \text{ or } 69.2\% \end{aligned}$$

(c) From eqn. (9.8),

$$\begin{aligned} \text{Loss of K.E due to blade friction} &= \frac{m}{2 \times 1,000} (V_{r1})^2 - (V_{r2})^2 \text{ kJ/sec.} \\ &= \frac{1,350}{3,600 \times 2 \times 1,000} [(775)^2 - (658)^2] = 31.4 \text{ kJ/sec.} \end{aligned}$$

9.3.5 Methods of reducing rotor speed or compounding of stages : The simple impulse turbine is shown diagrammatically in fig. 9-14. The lower part of the figure shows a longitudinal section through the upper half of the turbine, the middle portion shows a development of the nozzle and blading, while the top portion of the diagram shows approximately how the absolute pressure and absolute velocity of steam vary from point to point during the passage of the steam through the turbine.

If the steam is expanded from the boiler pressure to the condenser pressure, its velocity is extremely high, i.e. about 1,050 m per second. As shown earlier, the velocity of blade for maximum blade efficiency should be about one half of the steam velocity, i.e. about 525 m per second. In practice, the maximum blade velocity reached in this type of *simple single-stage turbine* is about 450 m per sec. As this type of turbine is only employed for relatively small powers, the diameter of rotor is kept fairly small and as a result the rotational speed is very high, reaching 30,000 r.p.m. In practice very few machines are required to be driven at such a high speed, and it is usually necessary to reduce the speed by gearing. Such gearing will be of undue proportions.

One of the chief objects in the development of the steam turbine is to reduce the high speed of the rotor to practical limits. Several methods are used to reduce this high rotor speed. All of these methods consist of a multiple system of rotors in series, keyed on a common shaft and the steam pressure or the steam velocity is absorbed in stages as it flows over the rotor blades. This is known as *compounding*. The following are chief methods for reducing the rotor speed :

9.3.6 Velocity-Compounded impulse turbines : This type of turbine consists of a nozzle or sets of nozzles and a wheel fitted with two or more rows of moving blades. The illustration shown in fig. 9-15 has two-rings of moving blades on the rotor and such a wheel is sometimes referred to as a "two-row wheel". There are also a number of guide or stationary blades arranged between the moving blades and set in the reverse manner as shown in fig. 9-15.

Steam entering the nozzle expands from the initial pressure down to the exhaust pressure, and resulting steam velocity is then utilized by as many sets of rotor blade rings as are necessary. On passing through the first ring of moving blades, the steam gives up only a part of its kinetic energy and issues from this ring of blades with a fairly high velocity. It then enters the guide blades (stationary blades) and is redirected by them into the second ring of moving blades. There is a slight drop in velocity in the fixed guide blades due to friction. In passing through the second ring of moving blades the steam suffers a change of momentum and gives up another portion of its kinetic energy to the rotor. In case of three-row rotor, steam further passes through the next ring of stationary blades and then through the third ring of moving blades and subsequently

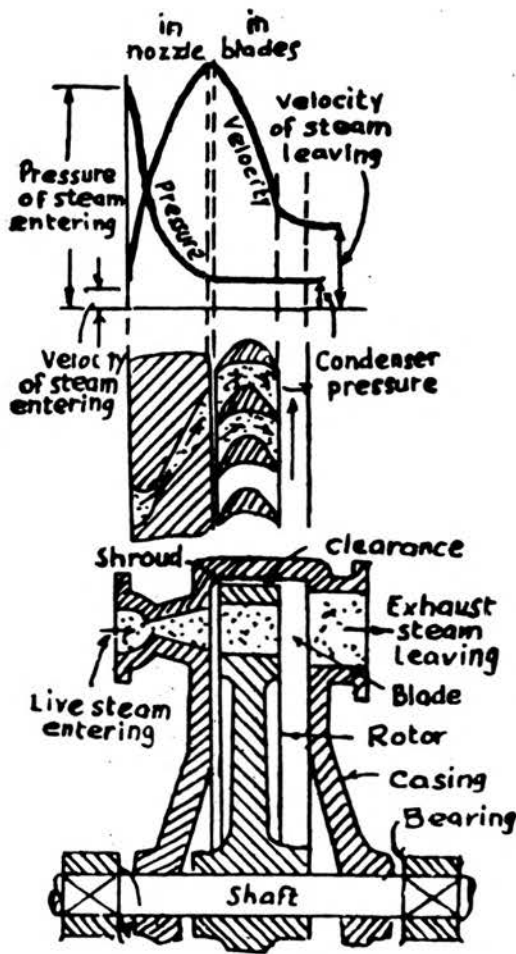


Fig. 9-14. Diagrammatic arrangement of simple impulse turbine showing variation of velocity and pressure of steam as it flows over the blades.

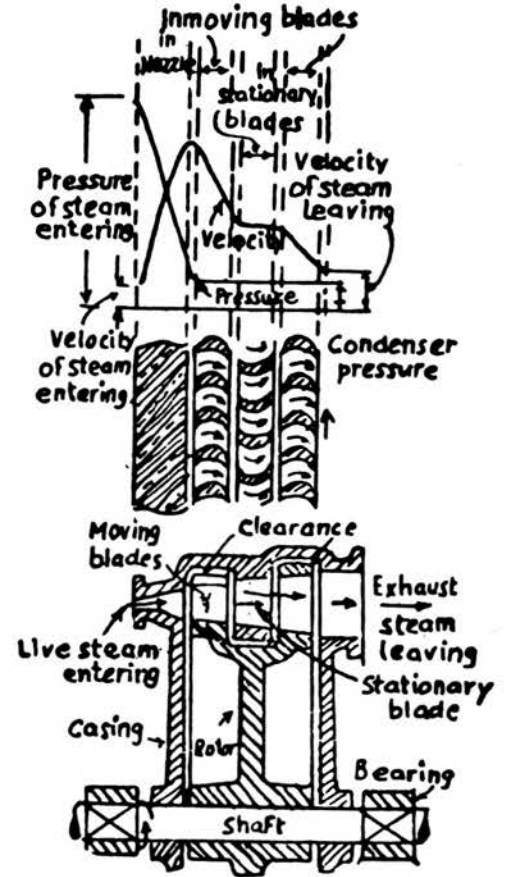


Fig. 9-15. Two - stage velocity-compounded impulse turbine.

leaves the wheel and enters the condenser. It may be noted that a two-row wheel is more efficient than the three-row wheel.

In fig. 9-15 (top portion), the curves of velocity and pressure are shown plotted on a base representing the axis of the turbine.

It will be noticed from the pressure curve that all the pressure drop takes place in the nozzle ring, and the pressure remains constant as the steam flows over the blades.

This method of velocity staging is known as Curtis principle.

9.3.7 Efficiency of a velocity-compounded stage :
The complete velocity diagram (fig. 9-16) for a stage consisting of a two moving blades and one fixed blade ring will consist of two diagrams, one for each set

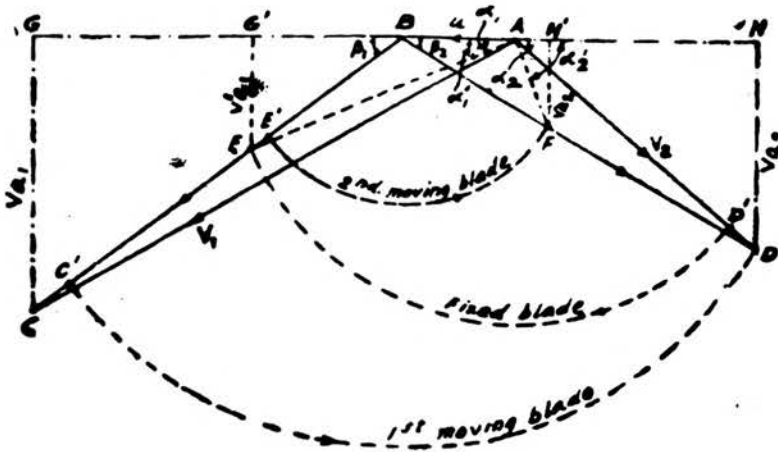


Fig. 9-16. Velocity diagram.

of moving blades. Let us assume that the blading is symmetrical ($\beta_1 = \beta_2$) and steam loses 10% of its velocity when passing over a blade and also blade velocity (u), nozzle angle (α_1), velocity of steam discharged from nozzle (V_1) are known.

Let AC represent V_1 , the velocity of the steam leaving the nozzles and entering the first row of moving blades. The inlet diagram is first drawn and then line BD of an unknown length is drawn at the correct angle β_2 . Mark off on line BC a friction loss of relative velocity CC' . Then BC' equals $0.9 BC = 0.9 V_{r1}$. With compass centred on B , draw an arc of radius BC' to cut BD at D . Then $BD = V_{r2} = 0.9 BC$. By joining A and D the line AD representing V_2 is obtained.

The steam now flows over the fixed blade ring and will lose 10% of its velocity during the passage. Hence mark off DD' to be 0.1 of the absolute velocity V_2 . Thus steam enters the second set of moving blades with absolute velocity AE (shown dotted) at an angle α_1' . The steam now flows over the second moving blades and loses 10% of its relative velocity. Hence, the relative velocity of steam at entry to second set of moving blades is $BE' = 0.9 BE$ at an inlet angle β_1 , i.e. same angle as that for the first set of moving blades. The relative velocity of steam at exit from second set of moving blades is $BF = 0.9 BE = BE'$ at blade exit angle β_2 . The absolute velocity of steam at exit from the second moving blades is AF (shown dotted) at an angle α_2' .

It should be noted that

- α_2 = angle of discharge from first moving blade
- = inlet angle of fixed blade,
- α_1 = outlet angle of fixed blade, and
- α_2' = angle of discharge from second moving blade.

Work done on first set of moving blades per kg of steam

$$= u (GH) = u (V_{w1} \pm V_{w2}) \text{ N.m or Joules}$$

Work done on second set of moving blades per kg of steam

$$= u (G' H') = u (V_{w1}' \pm V_{w2}') \text{ N.m or Joules}$$

\therefore Total work done per stage per kg of steam = $u (GH + G' H')$ N.m or Joules.

$$\text{Power developed per stage} = \frac{m \times (GH + G' H')u}{1,000} \text{ kW}$$

where, m = steam flow through blade in kg per sec.

$$\text{Diagram or blade efficiency} = \frac{u(GH + G' H')}{\frac{V_1^2}{2}} = \frac{2u(GH + G' H')}{V_1^2}$$

$$\text{Stage efficiency} = \frac{u(GH + G' H')}{1,000 H}$$

where, H = enthalpy drop in nozzles in kJ/kg

$$\text{Total axial thrust} = m [(V_{a1} - V_{a2}) + (V_{a1}' - V_{a2}')] \text{ N}$$

Same method may be repeated for velocity diagram, if the stage consists of more than two turbine pairs.

Problem-7 : In a two-stage velocity-compounded impulse turbine, the steam issues from the nozzles at a speed of 800 m/sec. The moving blade angles at entrance and exit are 30° and the blade speed is 180 m/sec. Assuming that the steam enters the blades without loss or shock and the coefficient of friction for the moving and fixed blades is 0.88, find :

- (i) the angle of the nozzle,
- (ii) the angle of the fixed blades discharging tip,
- (iii) the total work done on the blades per kg of steam; and
- (iv) the blade or diagram efficiency for the stage.

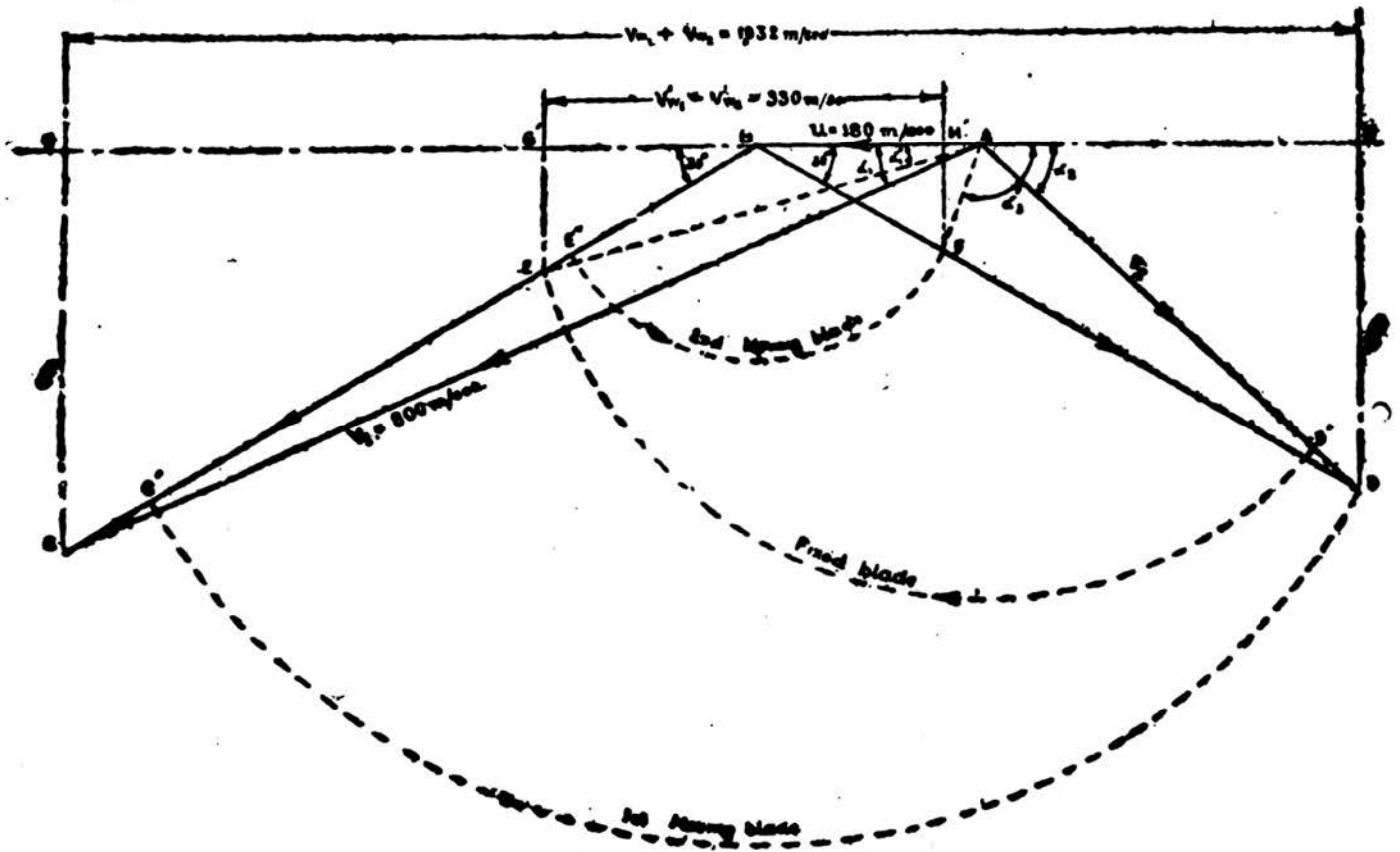


Fig. 9-17. Velocity diagram.

Refer fig. 9-17 for velocity diagram. This can be drawn to some convenient scale from the following data :

$AB(u) = 180$ m/sec., $AC(V_1) = 800$ m/sec., $\beta_1 = 30^\circ$, $\beta_2 = 30^\circ$, and

$BD(V_{r2}) = 0.88 BC(V_{r1})$ for the first set of moving blades.

AE for second moving blade ring = $0.88 AD$ of first moving blade ring.

$BE' = 0.88 BE$ for the second moving blade ring.

The velocity diagram can now be drawn from these values, and the following values can be scaled off the diagram :

(i) $\alpha_1 = 22.5^\circ$ (nozzle angle).

(ii) $\alpha_1' = 16^\circ$ (angle of the fixed blade discharging tip).

(iii) For first moving blade, $GH = V_{w1} + V_{w2} = 1,032$ m/sec.

For second moving blade, $G'H' = V_{w1}' - V_{w2}' = 330$ m/sec.

Total work done per stage per kg of steam = $u(GH + G'H')$

= $180 \times (1,032 + 330) = 2,45,160$ N.m or Joules per kg of steam

$$\begin{aligned}
 \text{(iv) Blade or diagram efficiency} &= \frac{2u(GH + G'H')}{(V_1)^2} \\
 &= \frac{2 \times 180(1,032 + 330)}{(800)^2} = 0.766 \text{ or } 76.6\%
 \end{aligned}$$

Problem-8 : A velocity – compounded impulse turbine has two rows of moving blades with a fixed row of guide blades between them. The steam leaves the nozzles at 900 m/sec. in a direction at 18° to the plane of the rotation, the blade speed is 150 m/sec. and the blade outlet angles are : first moving 24°, fixed 26° and second moving 30°. The friction factor is 0.9 for all rows.

Draw the velocity diagram to as large a scale as possible and from it determine the total change in velocity of whirl and the tangential thrust on the rotor if the steam supply 4,500 kg/hr.

Refer fig. 9-18 for velocity diagram. This can be drawn to some convenient scale from the following data :

$AB(u) = 150$ m/sec., $AC(V_1) = 900$ m/sec., $\beta_2 = 24^\circ$, outlet angle of second moving blade, $\beta_2' = 30^\circ$, $\alpha_1 = 18^\circ$, $\alpha_1' = 26^\circ$, and friction factor of 0.9 for all rows.

The following values can be scaled off the velocity diagram :

For first moving blade, $V_{w1} + V_{w2} = 1,320$ m/sec., and

for second moving blade, $V_{w1}' + V_{w2}' = 556$ m/sec.

∴ Total change in velocity of whirl = $1,320 + 556 = 1,876$ m/sec.

For first moving blade, $V_{a1} - V_{a2} = 0$, and

for second moving blade, $V_{a1}' - V_{a2}' = 53$ m/sec.

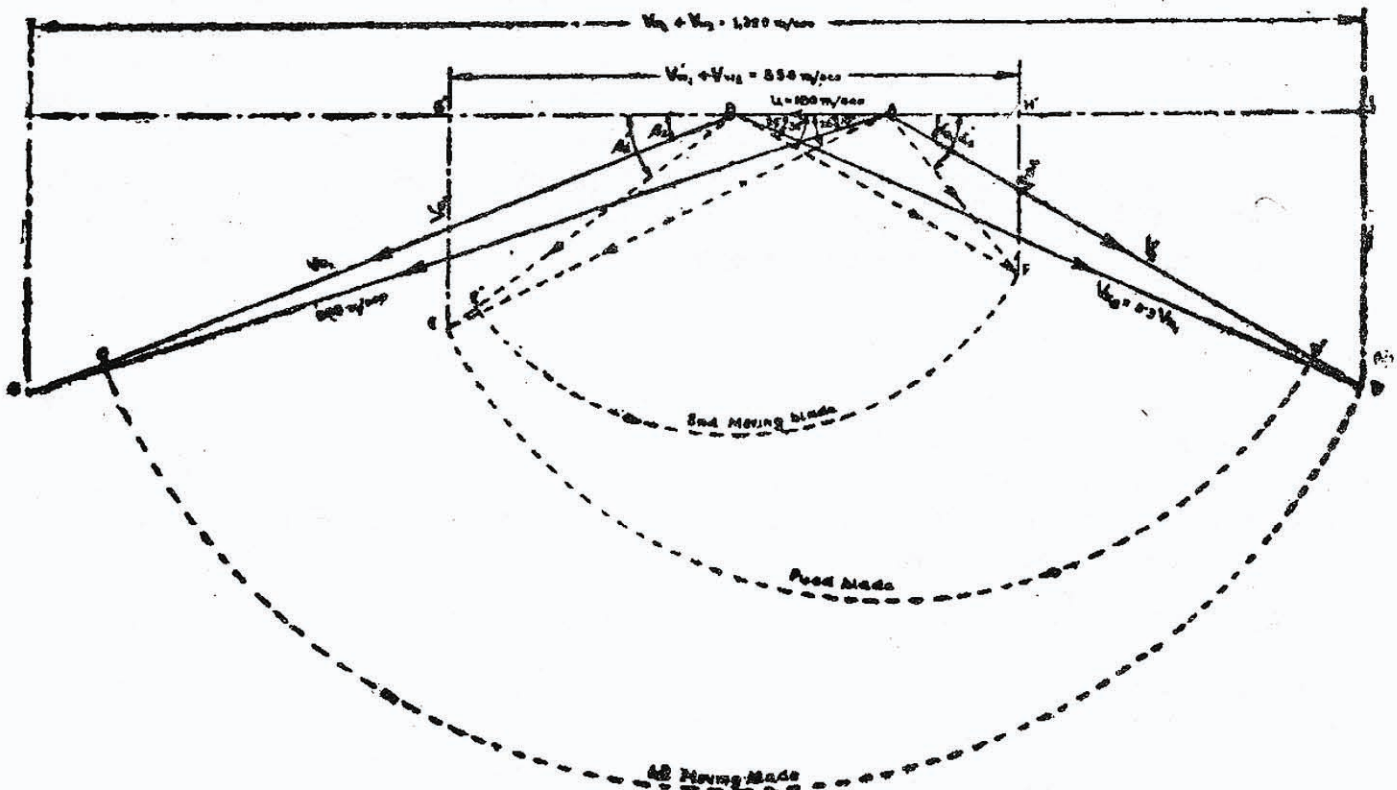


Fig. 9-18. Velocity diagram.

$$\begin{aligned} \therefore \text{Tangential thrust on the rotor} &= m (0 + 53) \\ &= \frac{4,500}{3,600} \times 53 = 66.25 \text{ N} \end{aligned}$$

Velocity diagram for axial discharge : The efficiency of a stage of an impulse turbine is a maximum when the final discharge of the steam is axial, i.e. when the angle of discharge for the second moving blade, $\alpha_2' = 90^\circ$.

In such a case the velocity diagram should be solved in the reverse direction to obtain the blade angles.

Referring to fig. 9-19, draw the blade velocity $AB(u)$ to any convenient length. This gives the blade velocity to an unknown scale. Then triangle ABF' is drawn with angle $BAF' = 90^\circ$ (axial discharge) and angle $ABF' = \beta_2$. This gives outlet diagram ABF' for the second row of moving blades to an unknown scale. Then working in reverse direction we get the inlet diagram ABE' for the second row of moving blades. Then outlet diagram ABD' for the first row of moving blades is completed, and again working in the reverse direction, the inlet velocity diagram ABC , for the first row of moving blades is obtained. Then velocity of steam (V_1) discharged from the nozzle is measured. The construction of velocity diagram is explained in the illustrative problem no.9.

As this velocity of the steam, V_1 is known, the scale of the whole diagram can be obtained. The blade velocity can be obtained by measuring the length AB . Also the blade angles α_1' and α_2 and the nozzle angle α_1 can be obtained from the velocity diagram.

Problem-9 : Steam is supplied to an impulse turbine at a pressure of 12 bar and superheated to 250°C . The pressure in the wheel chamber is 5.5 bar, and in the chamber there are two rings of moving blades separated by fixed blades. The tip of the moving blades are inclined 30° to the plane of the motion. Assuming a 10 per cent friction loss in the nozzle and also reduction of 8 per cent in the velocity of the steam relative to the blade due to frictional resistance in passing through a blade ring, determine the speed of the blade, so that the final velocity of discharge shall be axial. State what should be the inclination of the nozzle to the plane of motion of the blades. Also find out the steam consumption in kg per kW-hour, the diagram efficiency, and the stage efficiency.

Since the pressure drop in the nozzle is from 12 bar and 250°C to 5.5 bar, the total enthalpy drop is 163.5 kJ/kg from $H - \Phi$ chart or Mollier diagram.

The velocity of steam leaving the nozzle,

$$\begin{aligned} V_1 &= 44.72 \sqrt{\text{actual enthalpy drop}} \\ &= 44.72 \sqrt{0.9 \times \text{total enthalpy drop}} = 44.72 \sqrt{0.9 \times 163.5} = 542.5 \text{ m/sec.} \end{aligned}$$

The velocity diagram (fig. 9-19 on the next page) can now be drawn starting from the final velocity of the steam on leaving the second ring of moving blades.

Referring to fig. 9-19, draw blade velocity $u(AB)$ to any convenient length. Then draw triangle ABF' with angle $BAF' = 90^\circ$ (axial discharge) and angle $ABF' = \beta_2 = 30^\circ$. This gives triangle ABF' to an unknown scale.

Make $BE' = \frac{F'B}{0.92}$, BE' being drawn at 30° to AB , and join AE' . Then, as $F'B$ is

8 per cent less than BE' , the figure $ABE'F'$ is the velocity diagram for the second ring of the moving blades drawn to an unknown scale yet to be determined.

Produce BE' and make $AD' = \frac{AE'}{0.92}$. With centre A and radius AD' , cut BF'

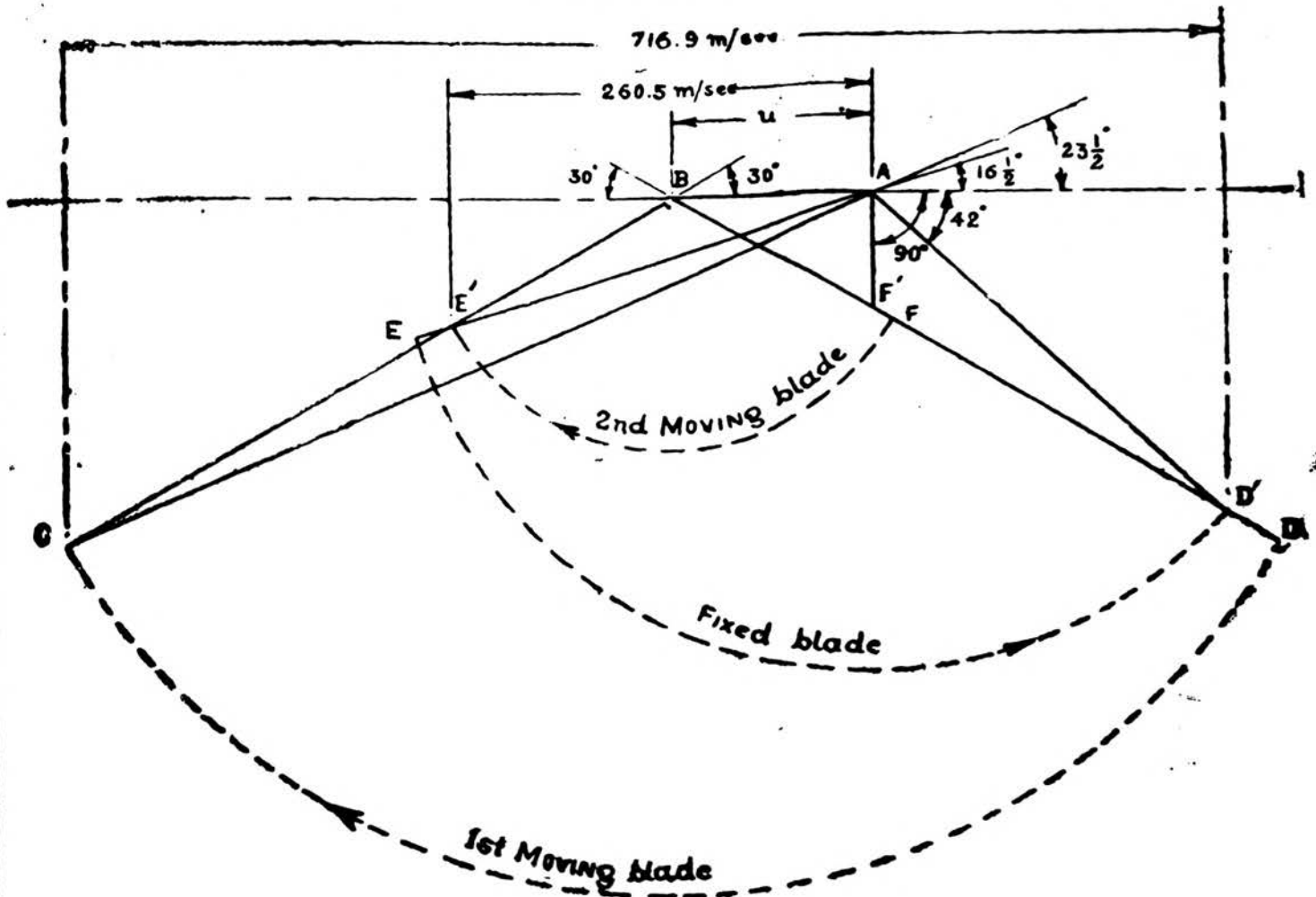


Fig. 9-19. Velocity diagram.

produced at D' and join AD' . Produce BE' to make $BC = \frac{D'B}{0.92}$. With centre B and radius BC , cut BE' produced at C . Join AC . Then the figure $ABCD'$ will be the velocity diagram for the first ring of moving blades, and AC represents the velocity of steam discharged from the nozzle. As the value of velocity of steam, V_1 is calculated at the beginning of the problem, the scale of the whole diagram can now be obtained by measuring length AC which is found to be 17.7 cm.

$$\therefore \text{Scale of the velocity diagram, } 1 \text{ cm} = \frac{V_1}{AC} = \frac{542.5}{17.7} = 30.65 \text{ m/sec.}$$

Hence, blade velocity, $u = AB \times \text{velocity diagram scale} = 4 \times 30.65 = 122.6 \text{ m/sec.}$

From velocity diagram,

fixed blade angles are : $\alpha_1' = 16\frac{1}{2}^\circ$, $\alpha_2 = 42^\circ$,

Nozzle angle, $\alpha_1 = 23\frac{1}{2}^\circ$,

$$V_{w1} + V_{w2} = 716.9 \text{ m/sec.}, \text{ and } V_{w1}' + V_{w2}' = 260.5 \text{ m/sec.}$$

$$\text{Now, power in kW} = \frac{m \times u \times [(V_{w1} + V_{w2}) + (V_{w1}' + V_{w2}')] }{1,000}$$

$$\text{i.e. } 1 = \frac{M \times 122.6 \times (716.9 + 260.5)}{1,000 \times 3,600}$$

(where M is steam consumption in kg per kW-hour)

$$\therefore M = \frac{1,000 \times 3,600}{122.6(716.9 + 260.5)} = 30.04 \text{ kg/kW-hr.}$$

$$\begin{aligned} \text{Diagram efficiency} &= \frac{2 \times u - [(V_{w1} + V_{w2}) + (V_{w1}' + V_{w2}')] }{(V_1)^2} \\ &= \frac{2 \times 122.6 \times (716.9 + 260.5)}{(542.5)^2} = 0.8143 \text{ or } 81.43\% \end{aligned}$$

$$\text{Stage efficiency} = \frac{u[(V_{w1} + V_{w2}) + (V_{w1}' + V_{w2}')] }{1,000 \times H}$$

(where, H is enthalpy drop in the nozzle in kJ/kg)

$$= \frac{122.6(716.9 + 260.5)}{1,000 \times 163.5} = 0.733 \text{ or } 73.3\%$$

9.3.8 Pressure-compounded impulse turbine : It is obvious that by arranging the expansion of the steam in a number of steps, we could arrange a number of simple impulse turbines in series on the same shaft, allowing the exhaust steam from one turbine to enter the nozzles of the succeeding (next) turbine. Each of the simple impulse turbine would then be termed a stage of the turbine, each stage containing a set of nozzles and blades. This is equivalent to splitting up the whole pressure drop into a series of smaller pressure drops; hence the term "Pressure compounding".

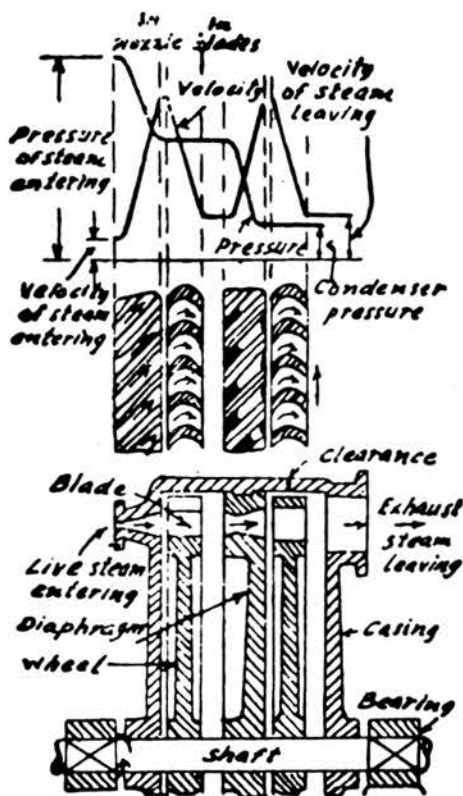


Fig. 9-20 Pressure-compounded impulse

In the fig. 9-20, curves of velocity and pressure are plotted on a base representing the axis of the turbine. It will be noticed that the total pressure drop of the steam does not take place in the first nozzle ring, but is divided equally between the two nozzle rings, and the pressure remains constant during the flow over the moving blades; hence the turbine is an impulse turbine.

Pressure compounding produces the most efficient, although the most expensive

The nozzles are usually fitted into partitions, termed as diaphragms, which separate one wheel chamber from the next. Expansion of steam takes place wholly in the nozzles, the space between any two diaphragms being filled with steam at constant pressure. The pressures on either side of any diaphragm are therefore different. Hence, steam will tend to leak through the space between the bore of the diaphragm and the surface of the shaft. Special devices are fitted to minimise these leakages.

The pressure compounding causes a smaller transformation of heat energy into kinetic energy to take place in each stage as compared to the simple impulse turbine. Hence, steam velocities with pressure compounding are much lower, with the result that blade velocities and rotational speed may be lowered. It is fairly clear that the speed may be reduced at will, simply by increasing the number of stages, but for very low speed the number of stages may become excessive.

In the fig. 9-20, curves of velocity and pressure are plotted on a base representing the axis of the turbine. It will be noticed that the total pressure drop of the steam does not take place in the first nozzle ring, but is divided equally between the two nozzle rings, and the pressure remains constant during the flow over the moving blades; hence the turbine is an impulse turbine.

turbine; so in order to make a compromise between efficiency and first cost, it is customary to combine velocity compounding and pressure compounding.

This type of turbine was developed by the late Professor A. Rateau of Paris and Dr. Zoelly of Zurich.

Pressure – velocity compounded impulse turbine : Another type of impulse turbine is the pressure – velocity compounded turbine. In this turbine both previous two methods are utilized. Total pressure drop of the steam is divided into stages and the velocity in each stage is also compounded.

In this turbine each stage has two or more rows of moving blades and one or more rows of stationary blades, the moving and stationary blades being placed alternately. Each stage is separated from the adjacent stage by a diaphragm containing nozzles. A ring of nozzles is fitted at the commencement of each stage. It is thus compounded both for pressure and velocity.

This method has the advantage of allowing a bigger pressure drop in each stage and consequently less stages are necessary. Hence, a shorter or more compact turbine will be obtained for a given pressure drop.

The pressure-velocity compounded turbine is comparatively simple in construction and is more compact than the multi-stage pressure compounded impulse turbine. Unfortunately its efficiency is not so high.

This method of pressure-velocity compounding is used in the Curtis turbine.

9.4. Reaction Steam Turbine

Though all turbines employ both the impulse and reaction principles to some extent,

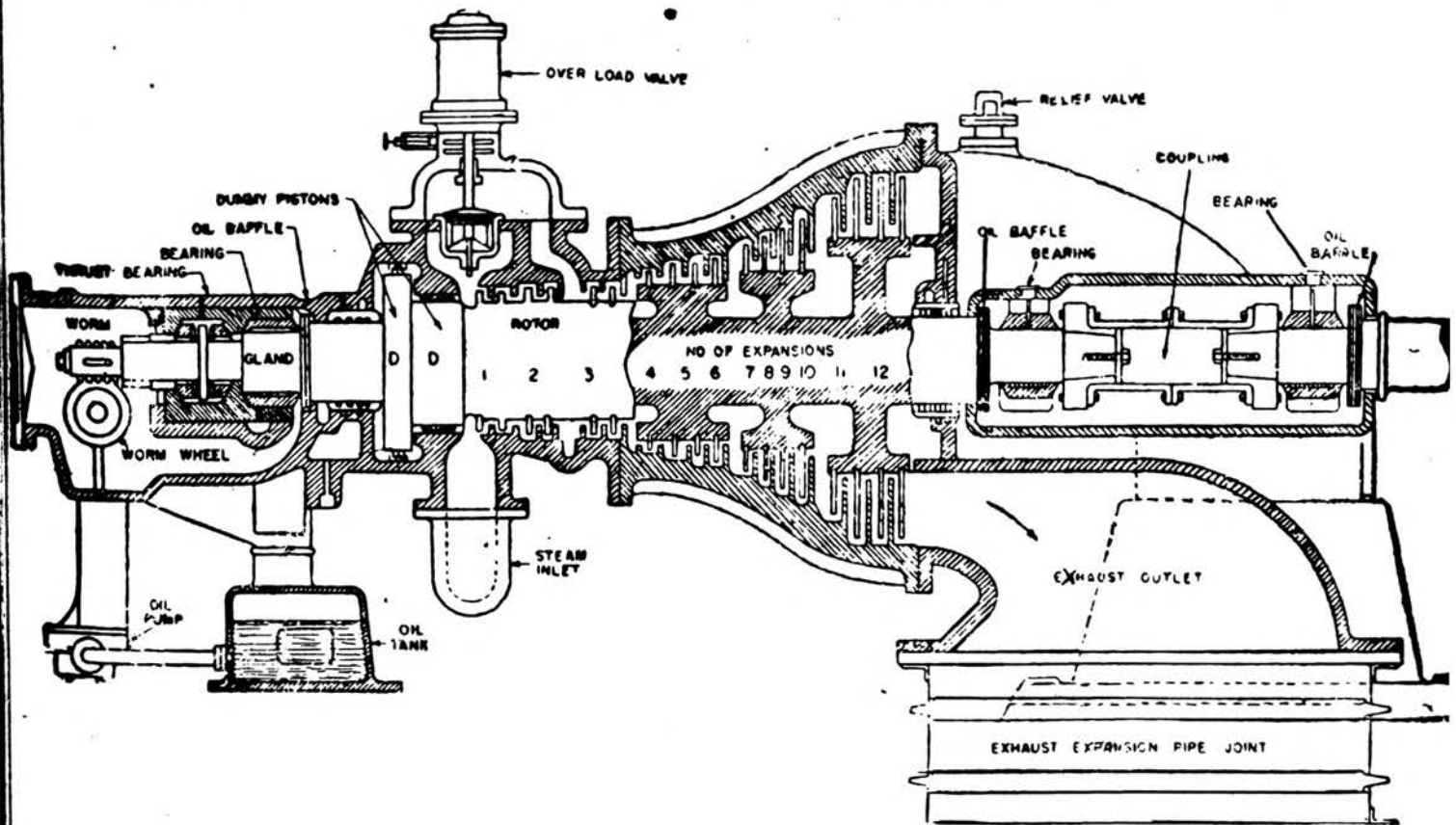


Fig. 9-21. Parson's type reaction turbine.

there is one turbine in which the reaction principle predominates sufficiently to have it commonly described as reaction turbine. The turbine bears the name of its inventor, the late Sir Charles A. Parson. This type, the Parson's reaction turbine, is shown in section in fig. 9-21.

In operation, steam enters the turbine through a double seated throttle valve, which is controlled by a governor driven from a worm gear on the main shaft, and passes in succession through the rings of fixed and moving blades until it reaches the end of the turbine cylinder and passes to the exhaust.

In passing through each ring of blades, the steam drops in pressure and increases in volume. To allow for this increased volume and keep the velocity of steam uniform, the blade ring areas are increased in steps. The blade rings between one step and next form an expansion group, and all the blade rings of particular group have the same external and internal radius. In the turbine shown in fig. 9-21, there are 12 expansion groups.

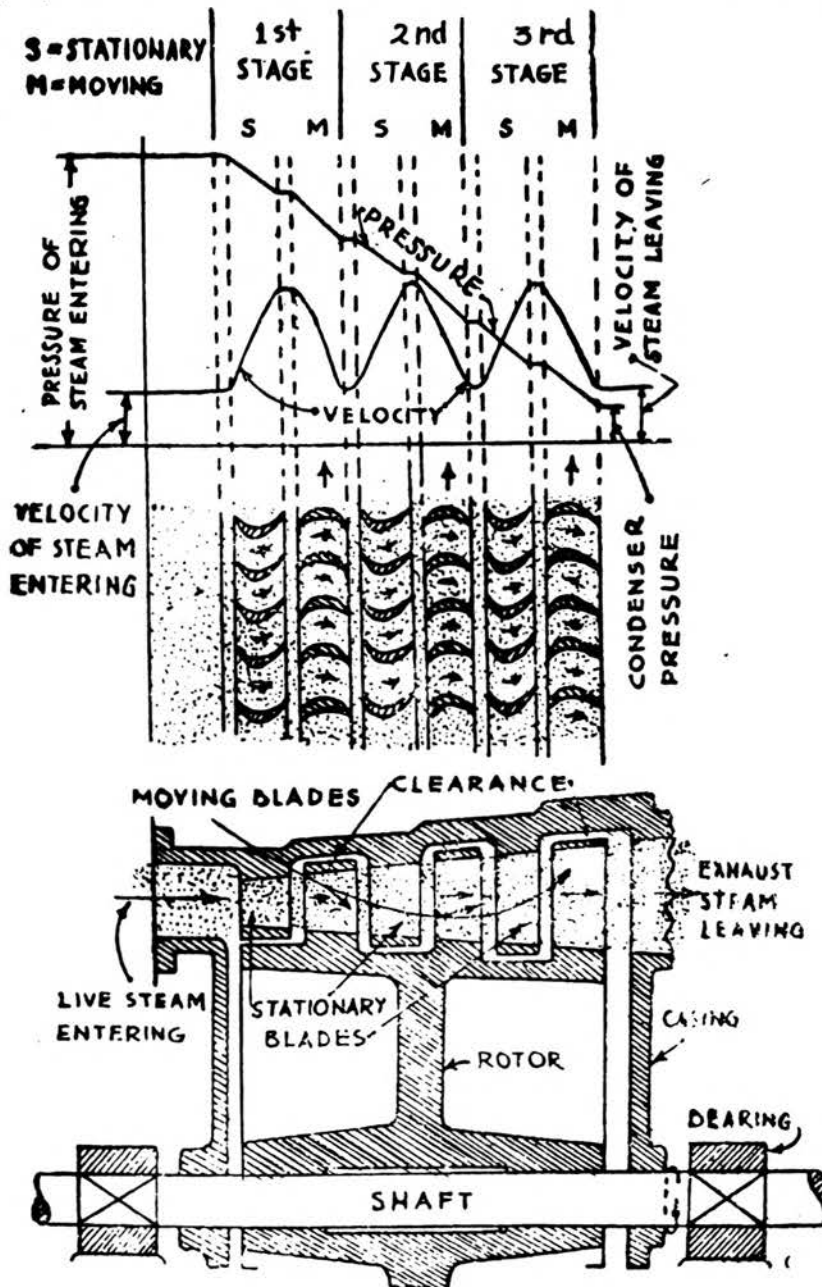


Fig. 9-22. Three-stage axial-flow impulse-reaction turbine.

In impulse turbines, the steam pressure on the back and front of a set of moving blades is the same and any thrust exerted by the steam in the direction of the rotor axis is negligible. In the reaction turbine, this thrust is considerable owing to the fall of pressure within the blades and difference between the blade sizes in the various steps. Dummy pistons and thrust bearings are used to balance this axial thrust. The face of dummy piston D on the right is exposed to entering high pressure steam, while the face of dummy piston D on the left is under steam pressure conveyed by pipe (not shown) from between the third and fourth expansions. The back of the dummy piston on the left is under pressure conveyed by pipe (not shown) from between sixth and seventh expansions. The rotor is a steel forging, and the dummy pistons are solid with it.

Fig. 9-22 shows diagrammatic arrangement of three-stage, axial-flow, impulse - reaction turbine. It also indicates roughly how the blade height increases as the specific volume of the steam increases with reduction in pressure; also how the pressure falls gradually as the steam passes through the groups of blades. It

will be observed from the diagram that there is a pressure drop across each row of blades, fixed and moving. This is of considerable practical importance, specially at the high pressure end of the turbine where the pressure drop is greatest, because this difference of pressure tends to force some steam through the clearance space between the moving blades and casing, and between the fixed blades and the rotor. The available energy possessed by this leaking steam is partly lost.

The steam velocities in this type of turbine are comparatively moderate, the maximum being according to the theory, about equal to the blade velocity. In practice, the steam velocity is commonly arranged to be greater than the blade velocity in order to reduce somewhat the total number of blades rows. The leaving loss for this type of turbine is normally about the same as for the multi-stage impulse turbine having single row wheels. This type of turbine has been, and continues to be, very successful in practice.

9.4.1 Velocity diagram for Parson's reaction turbine : In the reaction turbine, steam expands continuously in the fixed guide blades and moving blades. As its velocity and volume increase, increased area between the blades is required. This is obtained by

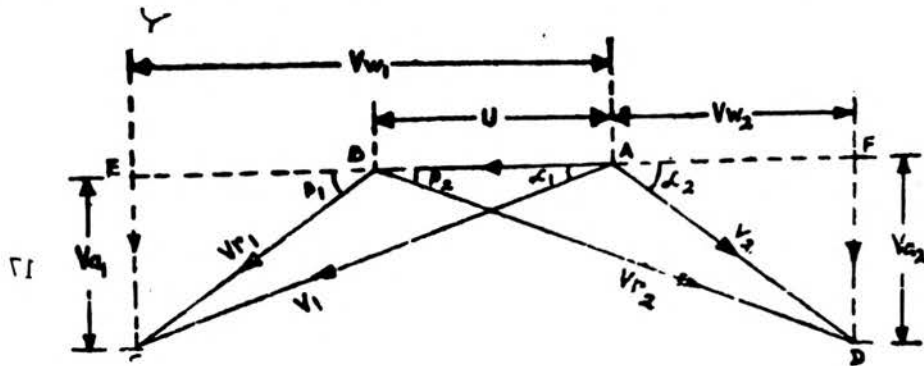


Fig. 9-23. Velocity diagram.

using a tapered rotor with progressively increasing blade heights.

Steam is directed against the moving blades by fixed guide blades, with velocity V_1 , at an angle α_1 . If the blade velocity is u m/sec, the relative velocity is V_{r1} . The steam expands between the moving blades, increasing its velocity and leaving with relative V_{r2} , and exit blade angle β_2 , approximately equal to its entrance velocity V_1 and entrance angle α_1 respectively. Compounding the relative velocity V_{r2} with the blade velocity u , which it has while on the blades, the absolute exit velocity V_2 is obtained.

In a Parsons reaction turbine the fixed and moving blades are made identical, i.e. α_1 and β_2 are equal, also β_1 and α_2 are equal. The velocity diagram from these blades will therefore, be symmetrical about a vertical central line.

In order to draw the velocity diagram (fig. 9-23), AB is drawn to represent the blade speed to a suitable scale. AC and BD are drawn at 20° to AB; and BC and AD at 35° to AB; the points C and D are at the intersection of these lines.

Referring to the fig. 9-23, the heat supplied to the turbine pair is the enthalpy drop of the steam during its passage over the pair; this is obtained from the Mollier diagram ($H - \Phi$ chart), the expansion being assumed isentropic. Then,

$$\text{Work done per pair per kg of steam} = (V_{w1} + V_{w2})u = EF \times AB \text{ N.m or Joules}$$

$$\text{Work done per sec. per pair} = m (EF \times AB) \text{ N.m per sec. or Joules per sec.}$$

where, m = mass of steam flowing over blades in kg per sec.

$$\text{Power developed per pair} = \frac{m(EF \times AB)}{1,000} \text{ kW}$$

$$\text{Efficiency} = \frac{\text{Work done per pair per kg of steam}}{\text{Enthalpy drop per pair}} = \frac{EF \times AB}{1,000 H}$$

where, H is the enthalpy drop per turbine pair in kJ/kg.

Problem-10 : In reaction turbine the fixed and moving blades are of the same shape but are reversed in direction. The angles of the receiving tips are 35° and of the discharging tips 20° . Find the power developed per pair of blades for a steam consumption of 1.1 kg per sec. when the blade speed is 50 m/sec. If the enthalpy drop in the pair is 9.5 kJ per kg, find efficiency of the pair.

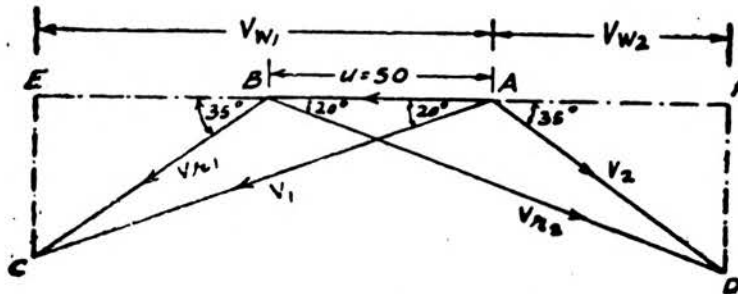


Fig. 9-24. Velocity diagram.

The velocity diagram may be constructed to some convenient scale as shown in fig. 9-24.

From the diagram,

$$(V_{w1} + V_{w2}) = EF = 155 \text{ m/sec.}$$

Work done per pair per kg of steam = $u(V_{w1} + V_{w2})$

$$= 50 \times 155$$

$$= 7,750 \text{ N.m or Joules}$$

$$\begin{aligned} \text{Power developed per pair} &= \frac{m \times u(V_{w1} + V_{w2})}{1,000} \quad [\text{From eqn. (9.4)}] \\ &= \frac{1.1 \times 7,750}{1,000} = 8.525 \text{ kW} \end{aligned}$$

From eqn. (9.6),

$$\begin{aligned} \text{Efficiency} &= \frac{\text{Work done per pair per kg of steam}}{1,000 H} \\ &= \frac{7,750}{1,000 \times 9.5} = 0.816 \text{ or } 81.6\% \text{ (efficiency of the pair)} \end{aligned}$$

9.4.2 Height of blades for reaction turbine : In a reaction turbine the area through which steam flows will always be full of steam. Fig. 9-25 shows the end view of one set of blades.

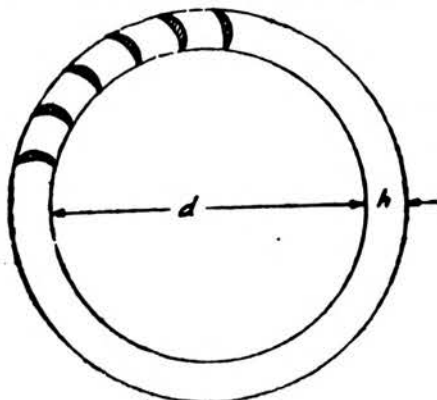


Fig 9-25.

Let d = rotor drum diameter in metres,

h = height of blades in metres = kd

(where, k = design constant)

= 0.08 to 0.1 (usually 0.083)

then, A = area of flow

= $\pi \times$ mean diameter at the mean circumference of blades \times height

$$= \pi (d + h) h$$

$$= \pi (d + kd) kd$$

$$= \pi (1 + k) k (d)^2 \quad (i)$$

$$\text{Now, } m = \text{mass in kg/sec.} = \frac{A \times V_f}{xv_s}$$

where, V_f = velocity of flow in m/sec. and

v_s = specific volume of steam entering the stage in m^3/kg .

$$\therefore m = \frac{\pi(1+k)k(d)^2}{xv_s} \times V_f \quad \dots (9.12)$$

Let n be the speed of wheel in revolutions/minute, then,

$$\text{Blade speed, } u = \frac{\pi(d+h)n}{60} \text{ m/sec.}$$

But, the velocity diagram shown in fig. 9-23 is similar for all blade rings of this turbine, as the blades are similar throughout. Therefore, V_f is proportional to u . Hence, $(V_{w1} + V_{w2})$ is proportional to u .

$$\text{Let } V_f = k_1 u \quad \dots (ii)$$

$$\text{and } (V_{w1} + V_{w2}) = k_2 u \quad \dots (iii)$$

where, k_1 and k_2 are constants that can be obtained from velocity diagram.

Then, substituting these values in eqn. (9.12),

$$m = \frac{\pi(1+k)k(d)^2}{xv_s} k_1 u \quad \dots (9.13)$$

$$\text{Work done per pair} = m(V_{w1} + V_{w2}) u = m k_2 u^2 \text{ Joules/sec.}$$

$$\therefore \text{Power developed per pair} = \frac{m k_2 u^2}{1,000} \text{ kW} \quad \dots (9.14)$$

Problem-11 : A low pressure reaction turbine running at 600 r.p.m. is supplied with 14 kg of steam/sec. Find the drum diameter and the height of the blades at the section of the turbine where pressure is 1.0 bar and dryness is 0.9, if the discharge angle of the blade is 20° and the blade velocity is 0.7 of the relative velocity of the steam at outlet of the blade. Assume that the blade height is to be $\frac{1}{12}$ th of drum diameter.

From steam tables, at 1 bar, $v_s = 1.694 \text{ m}^3/\text{kg}$.

$$\text{Now, } m = \frac{\pi(d+h)h V_f}{xv_s} \quad \dots (i)$$

The velocity diagram will be geometrically similar to fig. 9-23, but the scale is not known. Measuring from the velocity diagram, $V_f = 0.5 u$,

$$\text{also } u = \frac{\pi(d+h)n}{60} = \pi \left(d + \frac{d}{12} \right) \times \frac{600}{60}$$

Substituting the values of m , V_f and u in (i), we have

$$14 = \frac{\pi \left(d + \frac{d}{12} \right) \frac{d}{12} \times 0.5 \left[\pi \left(d + \frac{d}{12} \right) \frac{600}{60} \right]}{0.9 \times 1.694} = 3.17 (d)^3$$

$$\therefore d = \sqrt[3]{\frac{14}{3.17}} = 1.64 \text{ m i.e. } 164 \text{ cm (drum diameter)}$$

$$\therefore h = \frac{d}{12} = \frac{1.64}{12} = 0.137 \text{ m i.e. } 13.7 \text{ cm (blade height)}$$

Problem-12 : A group of reaction blading consists of three fixed and three moving rings all of the same height, and the mean blade speed of the moving rings is 65 m/sec. For the mean moving ring the inlet absolute and relative velocities are 80 and 30 m/sec. respectively and the specific volume is $0.156 \text{ m}^3/\text{kg}$. Determine for a flow of 2.25 kg/sec.: (a) the required area of blade annulus, (b) the power developed by the group, (c) the required enthalpy drop for the group if the steam expands with an efficiency ratio of 0.8.

Assume that both fixed and moving blades are of the same section.

- (a) If both fixed and moving blades are of the same section, then the moving blades exit angle β_2 will be equal to α_1 ; also V_2 must be so inclined as to enter the fixed blades without shock, i.e. $\alpha_2 = \beta_1$. Thus, this is a Parson's turbine.

The inlet triangle ABC is easily drawn, as the length of the three sides is known

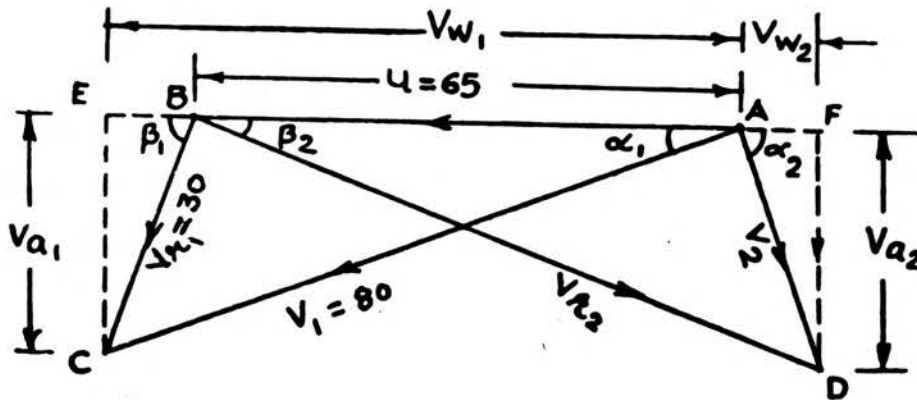


Fig. 9-26. Velocity diagram.

(scale .1 cm = 10 m/sec.), as shown in fig. 9-26.

$$AB = u = 65 \text{ m/sec.} = 6.5 \text{ cm}$$

$$AC = V_1 = 80 \text{ m/sec.} = 8.0 \text{ cm.}$$

$$BC = V_{r1} = 30 \text{ m/sec.} = 3.0 \text{ cm.}$$

Mass of steam flowing over blades, $m = 2.25 \text{ kg/sec.}$ (given).

Specific volume of steam, $v = 0.156 \text{ m}^3/\text{kg}$ (given).

The construction results in constant velocity of flow,

$$V_{a1} = V_{a2} = 28.3 \text{ m/sec.}$$

For continuity of flow, $m = \frac{AV}{v} \text{ kg/sec.}$

(where A is area of blade annulus in m^2)

$$\therefore A = \frac{mv}{V} = \frac{2.25 \times 0.156}{28.3} = 0.0124 \text{ m}^2$$

(b) From velocity diagram, $V_{w1} + V_{w2} = 84.5 \text{ m/sec.}$

Since there are three pairs in the group or expansion and the diagram is that for the mean pair,

$$\begin{aligned} \text{Power developed by the group} &= 3 \times \frac{m \times (V_{w1} + V_{w2}) u}{1,000} \\ &= 3 \times \frac{2.25 \times 84.5 \times 65}{1,000} = 37.1 \text{ kW} \end{aligned}$$

(c) The enthalpy change in the fixed blades will be $\frac{V_1^2 - V_2^2}{2 \times 1,000} \text{ kJ/kg}$

and enthalpy change in the moving blades will be $\frac{(V_{r2})^2 - (V_{r1})^2}{2 \times 1,000} \text{ kJ/kg.}$

It is clear from the construction of the diagram that these enthalpy changes will be equal.

$$\therefore \text{Useful enthalpy change per pair} = 2 \times \frac{(V_1)^2 - (V_2)^2}{2 \times 1,000} = 2 \times \frac{80^2 - 30^2}{2 \times 1,000} \text{ kJ/kg.}$$

$$\therefore \text{Actual enthalpy change per pair} = 2 \times \frac{80^2 - 30^2}{2 \times 1,000 \times 0.8} \text{ and}$$

$$\begin{aligned} \text{Actual enthalpy change for the group} &= 3 \times 2 \times \frac{80^2 - 30^2}{2 \times 1,000 \times 0.8} \\ &= 20.625 \text{ kJ/kg} \end{aligned}$$

9.5 Re-heat Factor

In pressure compounding, the pressure of the steam is made to fall progressively (step by step) in number of stages of the turbine from initial pressure p_1 to exhaust pressure p_b . If the friction in the blading is neglected, the expansion of steam can be considered isentropic; however there is always considerable friction resisting the flow of steam and hence the isentropic enthalpy drop in any stage is not fully utilized in raising the kinetic energy, i.e. owing to friction there is a loss of kinetic energy. The kinetic energy thus lost is converted into heat with the result that steam becomes dry or superheated. This process of friction heating always causes an increase in entropy and, consequently, slight increase in final enthalpy drop.

The behaviour of the steam as it passes through the successive stages is best studied

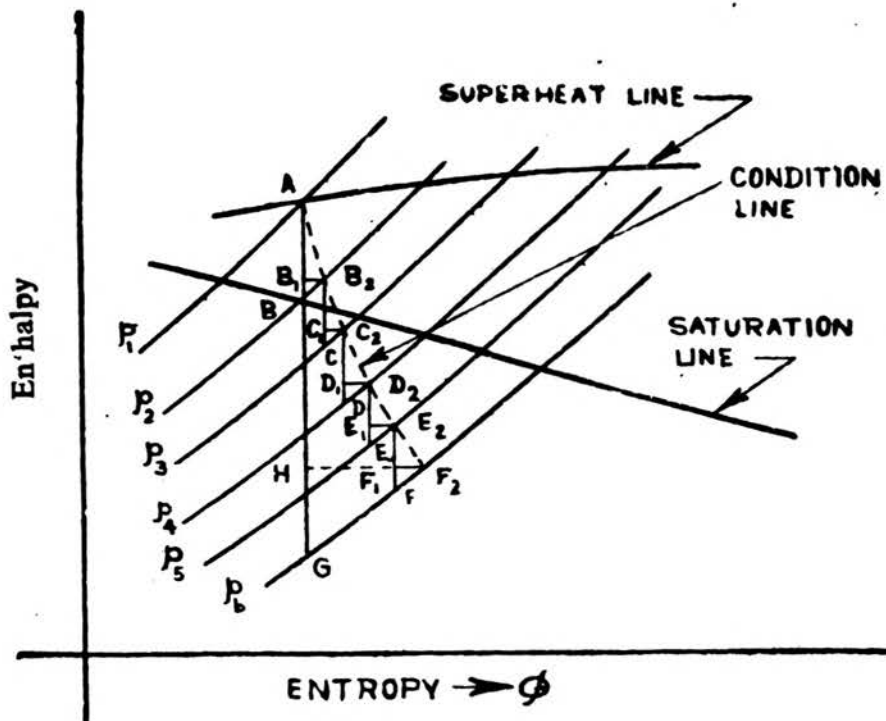


Fig. 9-27. The condition curve on H - Φ diagram.

by reference to the Mollier diagram ($H - \Phi$ diagram). A portion of diagram is shown in fig. 9-27.

Let us consider a steam turbine with five stages between pressure range of p_1 and p_b (fig. 9-27). The initial condition of the steam as it enters the turbine is represented

by the point A. In the first stage the steam expands isentropically from pressure p_1 to p_2 . The expansion is represented by the vertical line AB, a line of constant entropy. Mark off BB_1 on vertical line AB to represent friction loss of energy in the first stage due to blade friction. From point B_1 draw a horizontal line to meet the first stage back pressure line p_2 at B_2 . Then the point B_2 represents the final condition of the steam when discharged from the first stage. In the first stage, AB is the isentropic enthalpy drop neglecting friction and AB_1 is the actual or adiabatic enthalpy drop for that stage. The friction loss in the first stage, measured in heat units, is represented by BB_1 and the total enthalpy of steam as it enters the second stage is shown by level of point B_1 .

The same process is repeated for the remaining stages, that is, second, third, fourth and fifth stage, and condition of the steam at the end of each stage is obtained. The final condition of the steam at the end of each stage is represented by points C_2 , D_2 , E_2 and F_2 . The isentropic enthalpy drop in the second, third, fourth and fifth stage is represented by the lines B_2C , C_2D , D_2E and E_2F and actual or adiabatic enthalpy drop is represented by the lines B_2C_1 , C_2D_1 , D_2E_1 and E_2F_1 .

If the friction be neglected, the isentropic expansion of the steam through all stages is represented by the vertical line AG. It will be seen from the Mollier diagram (fig. 9-27) that the constant pressure line diverges from left to right and the effect of the friction is to move isentropic expansion line for each stage towards the right of the diagram. This means that the isentropic enthalpy drop, as represented by the lines AB, B_2C , C_2D , D_2E , and E_2F , has slightly increased. The ratio of the sum of the isentropic enthalpy drops in all stages to the isentropic enthalpy drop when expansion is carried out in a single stage, is known as *re-heat factor* for the turbine. The re-heat factor will be denoted by R.F.,

$$\text{i.e. Re-heat factor, R.F.} = \frac{AB + B_2C + C_2D + D_2E + E_2F}{AG}$$

The value of the re-heat factor varies with the type and efficiency of the turbine; an average value is 1.05.

The effect of the re-heat factor is to increase the final enthalpy drop; so the efficiency of the turbine is increased by the same ratio. This increase in efficiency due to friction is very small compared with net loss in friction.

$$\begin{aligned} \therefore \text{Turbine efficiency, } \eta &= \text{Stage efficiency} \times \text{Re-heat factor} \\ &= \eta_{\text{stage}} \times R.F. \end{aligned}$$

The *isentropic efficiency* of the turbine or efficiency of all the stages combined is the ratio of actual enthalpy drop to isentropic enthalpy drop of the steam. Actual or adiabatic enthalpy drop is represented by the vertical line AH and isentropic enthalpy drop (without friction) is represented by the line AG (fig. 9.27).

$$\text{Isentropic efficiency} = \frac{\text{Vertical projection of } AF_2 \text{ i.e., } AH}{AG}$$

The curve joining the points A, B_2 , C_2 , D_2 , E_2 , and F_2 will represent the condition of the steam at any instant. This curve is shown dotted and is called the *condition curve* or *line of condition* for the turbine.

Problem - 13 : Steam at 13 bar and 200°C is expanded in a turbine through six stages of equal isentropic enthalpy drop to a pressure of 0.1 bar. There is a 20% loss of enthalpy drop due to friction throughout the expansion. Calculate the re-heat factor.

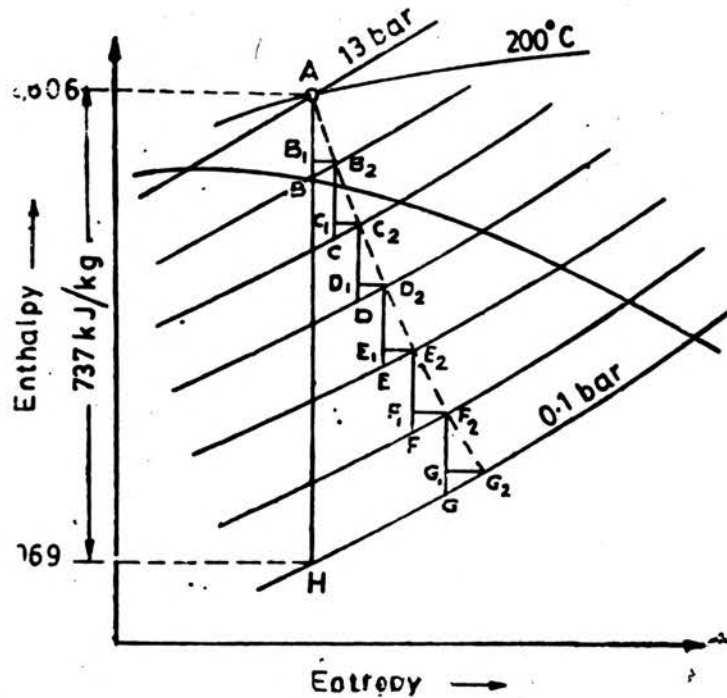


Fig. 9-28. H - φ diagram.

Overall isentropic enthalpy drop, $AH = 737 \text{ kJ/kg}$ (obtained from $H = \phi$ chart).

$$\therefore \text{Isentropic enthalpy drop during each stage} = \frac{737}{6} = 122.83 \text{ kJ/kg}$$

Dividing AH equally into six equal parts, the pressure lines for each stage were found to be 7, 3.5, 1.6, 0.67 and 0.27 bar.

Isentropic enthalpy drop for the remaining five stages is then found out from $H - \phi$ chart and shown in the table below

Turbine stage	Isentropic enthalpy drop (without friction) in each stage in kJ/kg	Frictional loss 20% in kJ/kg	Adiabatic enthalpy drop in each stage in kJ/kg
1	$AB = 122.8$	$BB_1 = 24.56$	$AB_1 = 98.24$
2	$B_2C = 125.6$	$CC_1 = 25.1$	$B_2C_1 = 100.50$
3	$C_2D = 129.8$	$DD_1 = 26.0$	$C_2D_1 = 103.80$
4	$D_2F = 131.9$	$EE_1 = 26.4$	$D_2E_1 = 105.50$
5	$E_2F = 131.9$	$FF_1 = 26.4$	$E_2F_1 = 105.50$
6	$F_2G = 134.0$	$GG_1 = 26.8$	$F_2G_1 = 107.20$

$$\therefore \text{Re-heat factor, R.F.} = \frac{\sum \text{Isentropic enthalpy drop}}{AH}$$

$$= \frac{122.8 + 125.6 + 129.8 + 131.9 + 131.9 + 134.0}{737} = \frac{776}{737} = 1.0529$$

9.6 Re-heating of Steam

The steam becomes wet as it expands. The wet steam has in it suspended water particles. The water particles which are heavier than steam particles, cause erosion on the turbine blades. In order to increase the life of the turbine blades, it is necessary to

keep steam dry during expansion. This is done by taking out steam from the turbine at the section where it becomes just dry saturated and is re-heated at constant pressure by the flue gases until it is again superheated to the same temperature as on entry to

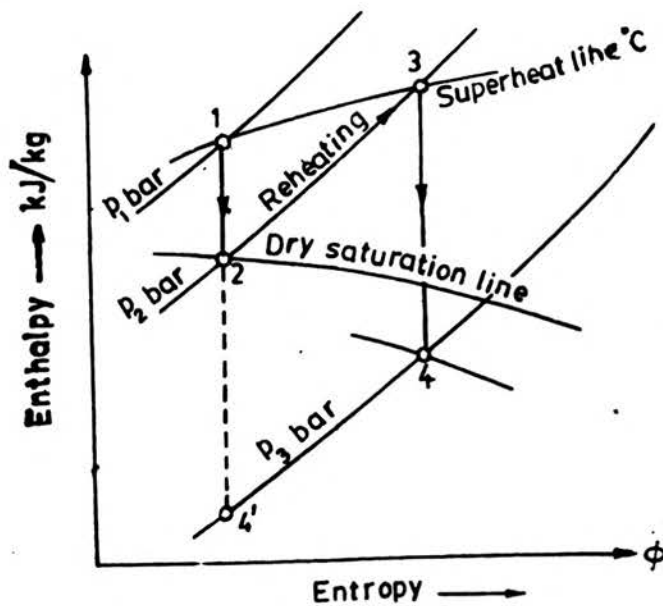


Fig. 9-29. Re-heating of steam.

the turbine. It is then taken back into the next stage of the turbine where further expansion takes place. This process is known as "re-heating". Within certain limits this process will cause increase in work done. It may be noted that increase in work done is at the cost of additional heat supplied in re-heating the steam and therefore there will be no appreciable change in the efficiency.

This process is indicated on the Mollier diagram (fig. 9-29). The initial condition of the steam entering the turbine is represented by the point 1. The steam then expands isentropically through the turbine along the line 1-2. At a certain point 2, at which the steam has become just dry saturated, it is re-heated

back to its initial temperature at constant pressure to point 3; at this point the steam is again in a superheated state and is at pressure p_2 . It then continues its isentropic expansion through next stage of the turbine until the condenser pressure p_3 is reached at point 4. Neglecting the friction, the total enthalpy drop is $[(H_1 - H_2) + (H_3 - H_4)]$ and the total heat supplied is the enthalpy at point 1, i.e., H_1 , plus the heat supplied during the re-heating process between points 2 and 3, i.e., $H_3 - H_2$.

Work done per kg of steam = total enthalpy drop = $[(H_1 - H_2) + (H_3 - H_4)]$ kJ/kg.

Total heat (net) supplied per kg of steam = $[H_1 + (H_3 - H_2) - h_4]$ kJ/kg
where, h_4 is the enthalpy of water at point 4.

∴ Efficiency with re-heating

$$= \frac{\text{Work done}}{\text{Heat supplied}} = \frac{[(H_1 - H_2) + (H_3 - H_4)]}{[H_1 + (H_3 - H_2) - h_4]} \quad \dots (i)$$

If steam had not been re-heated, then the expansion through the turbine would be represented by the vertical line 1 - 4'.

Then, work done per kg of steam = total enthalpy drop = $H_1 - H_4'$ kJ/kg

Heat supplied per kg of steam = $H_1 - h_4'$ kJ/kg

$$\therefore \text{Efficiency without re-heating} = \frac{H_1 - H_4'}{H_1 - h_4'} \quad \dots (ii)$$

where H_4' and h_4' are the enthalpies of steam and water respectively at point 4',

Actual working of a specific problem with the help of eqn. (i) and (ii), it will be found that the effect of re-heating may not cause appreciable change in efficiency, but will cause increase in the work done per kilogram of steam used. Refer illustrative problem No. 14.

This process of re-heating may be repeated if required during the expansion of the steam through the turbine in more than two stages.

The following advantages may be claimed by re-heating of steam :

(i) The quality of steam at exit from the turbine is improved; this reduces the erosion (wearing out) trouble on the turbine blades.

(ii) Work per kilogram of steam increases and hence specific steam consumption of steam turbine decreases. This reduces the amount of water required in condenser of the turbine.

Problem - 14 : Steam at a pressure of 28 bar and 50°C superheat, is expanded through a turbine to a pressure, where the steam is just dry saturated. It is then re-heated at constant pressure to its original temperature, after which it completes its expansion through the turbine to an exhaust pressure of 0.2 bar. Calculate the ideal efficiency of the plant and the work done, (a) taking the re-heating into account, and (b) if the steam was expanded direct to exhaust pressure without any re-heating.

(a) With re-heating :

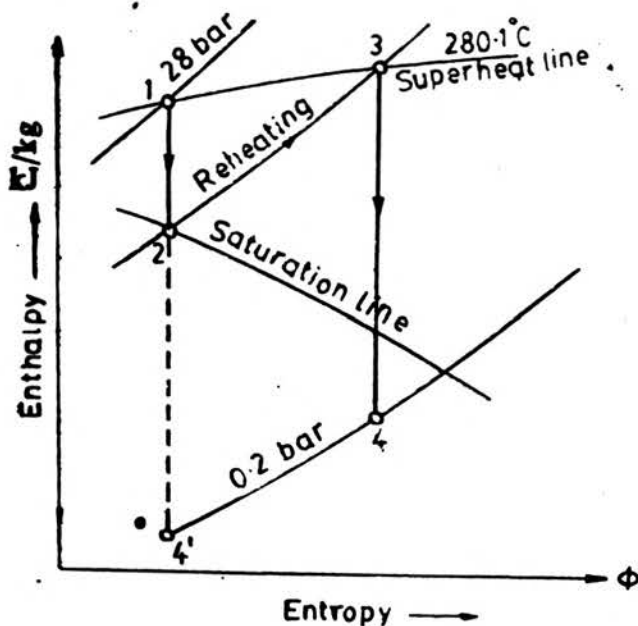


Fig. 9-30. $H - \Phi$ diagram.

From $H - \Phi$ chart (fig. 9-30), enthalpies, $H_1 = 2,920$ kJ/kg, $H_2 = 2,793$ kJ/kg (corresponding to dry saturated steam at pressure of 15.5 bar), $H_3 = 2,960$ kJ/kg (corresponding to pressure of 15.5 bar and temperature of 280.1°C), $H_4 = 2,230$ kJ/kg (corresponding to exhaust pressure of 0.2 bar); and $h_4 = 251.4$ kJ/kg (enthalpy of water at 0.2 bar obtained from steam tables).

Ideal efficiency with re-heating

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

$$= \frac{(H_1 - H_2) + (H_3 - H_4)}{H_1 + (H_3 - H_2) - h_4}$$

$$= \frac{(2,920 - 2,793) + (2,960 - 2,230)}{2,920 + (2,960 - 2,793) - 251.4} = \frac{857}{2,835.6} = 0.3022 \text{ or } 30.22\%$$

$$\text{Work done} = (H_1 - H_2) + (H_3 - H_4) = (2,920 - 2,793) + (2,960 - 2,230) = 857 \text{ kJ/kg}$$

(b) Without re-heating ;

From $H - \Phi$ chart, $H_1 = 2,920$ kJ/kg, $H_4' = 2,120$ kJ/kg (at 0.2 bar), and $h_4' = 251.4$ kJ/kg (at 0.2 bar) from steam tables.

Ideal efficiency without re-heating.

$$= \frac{\text{Work done}}{\text{heat supplied}} = \frac{H_1 - H_4'}{H_1 - h_4'} = \frac{2,920 - 2,120}{2,920 - 251.4} = 0.30 \text{ or } 30\%$$

$$\text{Work done} = H_1 - H_4' = 2,920 - 2,120 = 800 \text{ kJ/kg.}$$

It can be seen that by re-heating there is no appreciable change in the efficiency, but it has increased the work done; thus an increase in the power is obtained from a

given size of turbine.

9.7 Regenerative Feed Heating or Bleeding

The process of draining steam from the turbine, at certain points during its expansion, and using this steam for heating the feed water supplied to the boiler, is known as *bleeding* and this process of feed heating is known as regenerative feed heating.

At certain sections of the turbine, a small quantity of wet steam is drained out from the turbine, as shown in fig. 9-31. This bled steam is then circulated around the feed water pipe leading from the hot-well to the boiler. The relative cold feed water causes this bled steam to condense, the heat thus lost by steam being transferred to the feed water. The condensed steam then drains into the hot-well. The result of this process is to supply the boiler with hotter feed water whilst a small amount of work is lost by the turbine. This definitely increases efficiency of plant, but there is also a decrease in the work done per kilogram of steam; this process is shown in illustrative problem No. 15.

In the absence of precise information as to the actual temperature of the feed water entering and leaving the heaters and the condensate temperature, the following assumptions are made :

(i) The bled steam just condenses, i.e., gives up its superheat (if any) and all its enthalpy of evaporation only. The condensed steam therefore leaves the heater at the saturation temperature corresponding to the bleeding pressure.

(ii) The feed water is heated to the saturation temperature at the pressure of the bled steam.

Feed heating systems : Different systems of feed water heating are shown in fig. 9-31 and fig. 9-32. In each case 2 heaters are used. In actual practice, heaters may vary from 2 to 6.

(i) *When the bled steam does not mix with feed water or Cascade system* :

Fig. 9-31 shows two surface heaters in which the feed water condenses the bled steam, i.e., the bled steam does not mix with the feed water. This system is also known as *Cascade system*.

Consider bleeding at point 1 (fig. 9-31). Let ' w_1 ' be the mass of bled steam per

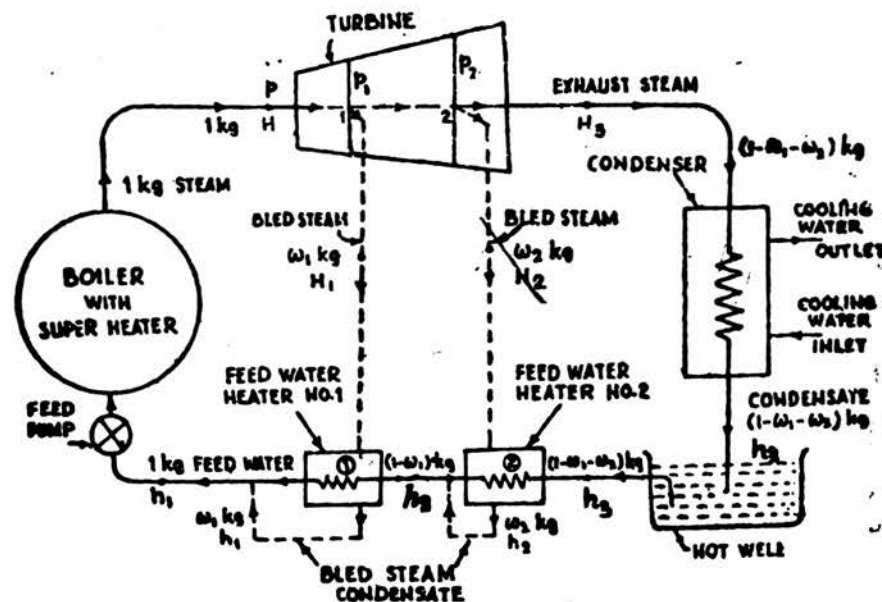


Fig. 9-31 Cascade system.

kilogram of feed water heated. Then,

Heat lost by steam = Heat gained by feed water;

$$\text{i.e., } w_1 (H_1 - h_2) = (h_1 - h_2)$$

where, h_2 is enthalpy of the feed water coming from heater No.2 and entering heater No.1.

$$\therefore w_1 = \frac{(h_1 - h_2)}{(H_1 - h_2)} \text{ kg}$$

Now consider bleeding at point 2. Let ' w_2 ' be the mass of bled steam at point 2 per kilogram of feed water heated. Then,

$$w_2 (H_2 - h_3) + w_1 (h_2 - h_3) = h_2 - h_3$$

$$\therefore w_2 = \frac{(1 - w_1)(h_2 - h_3)}{(H_2 - h_3)} \text{ kg}$$

where, h_3 is the enthalpy of feed water entering heater No.2,

Mass of steam in turbine per kilogram of feed water between points 1 and 2 } = $1 - w_1$

Mass of steam between point 2 and exhaust } = $1 - w_1 - w_2$

Work done in turbine per kilogram of feed

water between entrance and point 1 } = $(H - H_1)$

Work done between point 1 and point 2 = $(1 - w_1)(H_1 - H_2)$

Work done between point 2 and exhaust = $(1 - w_1 - w_2)(H_2 - H_3)$

Total heat supplied per kilogram of feed water = $(H - h_1)$

$$\therefore \text{Efficiency of the plant (including the effect of bleeding)} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{(H - H_1) + (1 - w_1)(H_1 - H_2) + (1 - w_1 - w_2)(H_2 - H_3)}{H - h_1}$$

(ii) When bled steam is mixed with feed water or Drain pump system :

It is a common practice in bleeding installation to mix the bled steam with the feed water. The mixture of bled steam and feed water is then supplied direct to the boiler. This system is also known as *drain pump system*. A diagrammatic arrangement of such an installation with two feed water heaters is shown in fig. 9-32.

At a point in the turbine installation at which the steam pressure is p_1 , w_1 kilogram of steam is abstracted (removed) and mixed with the feed water, which has been raised to a temperature of t_2 , by the previous bled steam. Then, on the basis of one kilogram of feed water, heat lost by bled steam = heat gained by feed water,

$$\text{i.e., } w_1 (H_1 - h_1) = (1 - w_1)(h_1 - h_2)$$

$$\therefore w_1 = \frac{h_1 - h_2}{H_1 - h_2}$$

Similarly, for the bled steam at pressure p_2 , $w_2 (H_2 - h_2) = (1 - w_1 - w_2)(h_2 - h_3)$

$$\therefore w_2 = \frac{(1 - w_1)(h_2 - h_3)}{H_2 - h_3}$$

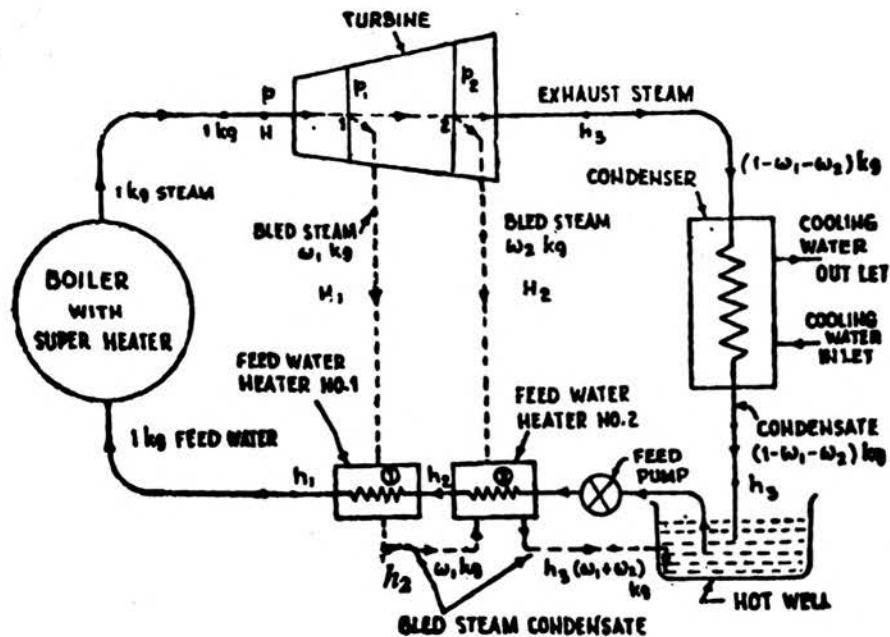


Fig. 9-32. Drain pump system.

Problem - 15 : Two stages of feed heating are employed in a steam turbine installation, steam being bled for these at pressures of 3.4 bar and 0.6 bar respectively. The temperature of the feed water is raised to that of the bled steam, and the condensate from each heater may be taken as being at the same temperature as the feed water entering the heater.

The steam is supplied to the turbine at 17 bar with 4.5°C superheat, and condenser pressure is 0.06 bar. The stage efficiency between pressures 17 bar and 3.4 bar is 0.7, and in the other two stages is 0.65. Estimate :

(i) the mass of steam bled to each heater,

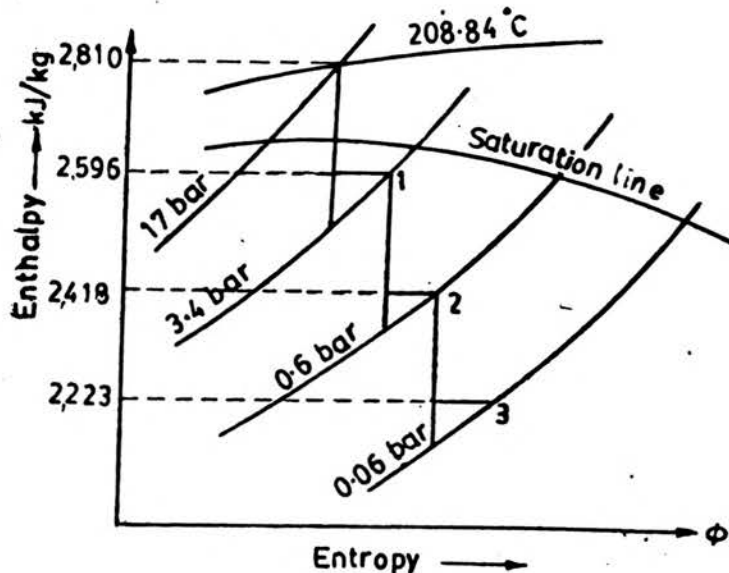
(ii) the total work done per kilogram of steam supplied to the turbine, and

(iii) the overall thermal efficiency of the cycle.

Refer to fig. 9-31 and fig. 9-33.

For stage 1, stage efficiency = 0.7, and for stages 2 and 3, stage efficiency = 0.65. The required enthalpy values for different stages may be read from the $H - \Phi$ chart after considering stage efficiency as shown in fig. 9-33. From $H - \Phi$ chart, $H = 2,810$ kJ/kg, $H_1 = 2,596$ kJ/kg, $H_2 = 2,418$ kJ/kg, $H_3 = 2,223$ kJ/kg.

From steam tables, at 3.4 bar, $h_1 = 579.97$ kJ/kg,

Fig. 9-33. $H - \Phi$ diagram.

at 0.6 bar, $h_2 = 359.86$ kJ/kg and

at 0.06 bar, $h_3 = 151.53$ kJ/kg.

(i) For heater No. 1

Heat lost by bled steam = Heat gained by feed water, i.e., $w_1 (H_1 - h_2) = h_1 - h_2$

$$\therefore w_1 = \frac{h_1 - h_2}{H_1 - h_2} = \frac{579.97 - 359.86}{2,596 - 359.86} = 0.0984 \text{ kg}$$

For heater no.2

$$w_2 (H_2 - h_3) + w_1 (h_2 - h_3) = h_2 - h_3$$

$$\therefore w_2 = \frac{(h_2 - h_3)(1 - w_1)}{H_2 - h_3} = \frac{(359.86 - 151.53)(1 - 0.0984)}{2,418 - 151.53} = 0.0829 \text{ kg}$$

(ii) Total work done per kg of steam supplied to turbine

$$\begin{aligned} &= (H - H_1) + (1 - w_1)(H_1 - H_2) + (1 - w_1 - w_2)(H_2 - H_3) \\ &= (2,810 - 2,596) + (1 - 0.0984)(2,596 - 2,418) + (1 - 0.0984 - 0.0829) \\ &\quad \times (2,418 - 2,223) \\ &= 214 + (0.9016 \times 178) + (0.8187 \times 195) \\ &= 214 + 160.45 + 159.55 = 534.0 \text{ kJ/kg} \end{aligned}$$

(iii) Total heat supplied per kg of steam = $H - h_1 = 2,810 - 579.97 = 2,230.03$ kJ/kg

$$\therefore \text{Overall thermal efficiency} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{534}{2,230.03} = 0.2394 \text{ or } 23.94\%$$

Problem - 16 : The steam supply to a turbine is at 40 bar with 38°C superheat. Steam is bled for feed heating at 13 bar and at 3.6 bar. The condenser pressure is 0.1 bar.

Calculate the optimum mass of bled steam at each stage and the cycle efficiency. Assume an efficiency ratio of 0.8 for each portion of the expansion and that the feed water leaving each heater is raised to the temperature of the steam entering the heater, the bled steam being pumped into the feed line after each heater.

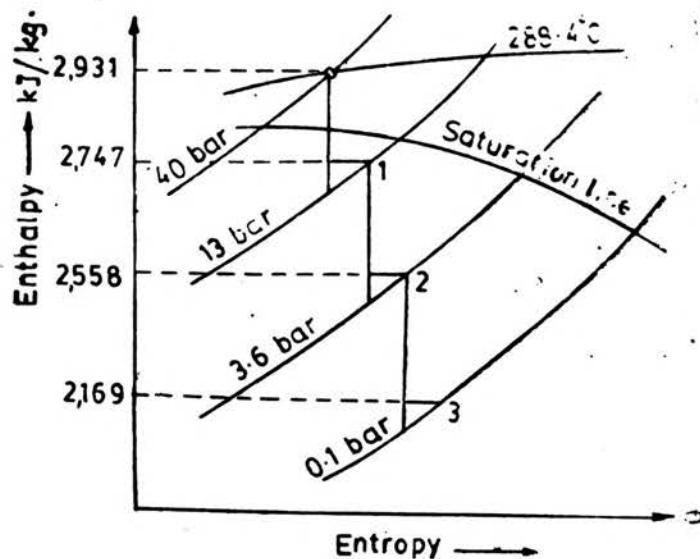


Fig. 9-34. $H - \Phi$ diagram.

Refer to figs. 9-32 and 9-34. For each stage, efficiency ratio of 0.3 is assumed.

From steam tables, at 13 bar $h_1 = 814.93$ kJ/kg, at 3.6 bar, $h_2 = 588.59$ kJ/kg, at 0.1 bar, $h_3 = 191.83$ kJ/kg.

The required enthalpy values may be read from the $H - \Phi$ chart as shown in fig. 9-34. From $H - \Phi$ chart, $H = 2,931$ kJ/kg, $H_1 = 2,747$ kJ/kg, $H_2 = 2,558$ kJ/kg, $H_3 = 2,169$ kJ/kg.

For heater No.1

Heat lost by bled steam = Heat gained by feed water

$$\text{i.e., } w_1 (H_1 - h_2) = (1 - w_1) (h_1 - h_2)$$

$$\therefore w_1 = \frac{h_1 - h_2}{H_1 - h_2} = \frac{814.93 - 588.59}{2,747 - 588.59} = \frac{226.35}{2,158.41} = 0.1049 \text{ kg}$$

For heat No. 2

$$w_2 (H_2 - h_3) = (1 - w_1 - w_2) (h_2 - h_3)$$

$$w_2 = \frac{(1 - w_1) (h_2 - h_3)}{(H_2 - h_3)} = \frac{(1 - 0.1049) (588.59 - 191.83)}{2,558 - 191.83}$$

$$= \frac{0.8951 \times 396.76}{2,366.17} = 0.15 \text{ kg}$$

Work done per kg of steam supplied to turbine

$$= (H - H_1) + (H_1 - H_2) (1 - w_1) + (H_2 - H_3) (1 - w_1 - w_2)$$

$$= (2,931 - 2,747) + (2,747 - 2,558) (1 - 0.1049) + (2,558 - 2,169) \times (1 - 0.1049 - 0.15)$$

$$= 184 + 169.16 + 289.84 = 643 \text{ kJ/kg}$$

Heat supplied per kg of steam = $H - h_1 = 2,931 - 814.93 = 2,116.07 \text{ kJ/kg}$

$$\therefore \text{ Cycle efficiency} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{643}{2,116.07} = 0.304 \text{ or } 30.4\%$$

9.8 Steam Turbine Governing

In a normal condensing steam turbine driving an alternator or a D.C. generator, the energy output will vary in accordance with the load on the alternator, and the function of the governor is to regulate the supply of steam to the turbine so that the speed of rotation shall remain almost constant at all loads. The principal methods of governing steam turbines are :

- (i) Throttle governing,
- (ii) Nozzle control governing,
- (iii) By-pass governing,
- (iv) Combined throttle and nozzle control governing, and
- (v) Combined throttle and by-pass governing.

9.8.1 Throttle governing : In throttle governing, the pressure of steam is reduced before reaching the turbine at part loads. Throttle governing is most widely used, particularly on small turbines, because its initial cost is less and mechanism is simple. The flow of steam entering the turbine is restricted by a balanced throttle valve which is controlled by the centrifugal governor. In turbines of small power in which the valves are light and the forces on them due to steam flow are negligible, the governor may be arranged to actuate (move) the throttle valve directly. For large machines, the frictional forces would require a large and powerful governor to actuate directly on the valve. The difficulty is easily overcome by the use of a relay, a device in which the relatively small force produced by the governor for a small change of speed is caused to produce a large force (if such is necessary) to actuate (move) the throttle valve.

A simple differential relay is shown diagrammatically in fig. 9-35. This throttle valve is actuated by the relay piston sliding in the cylinder. A floating or differential lever is attached at one end to the governor sleeve and the other end to throttle valve spindle, and at some intermediate point to a pilot or piston valve which consists of two small

piston valves covering ports without any lap, i.e., the length of the valve is just equal to the length of the ports. The operating medium is usually lubricating oil supplied by a pump at a pressure of 3 to 4 bar. The pipes G are open to the oil drain tank.

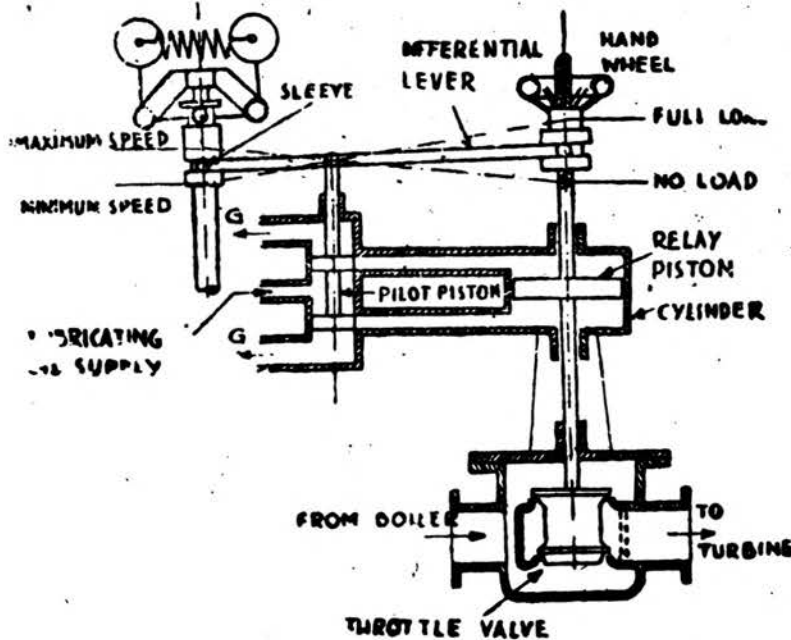


Fig. 9-35. Diagrammatic arrangement of throttle governing with oil relay.

accelerate the rotor. The turbine and governor speed will now rise and thus cause a lift of the governor sleeve. For the time being, the throttled valve spindle is stationary and the pilot (piston) valve is, therefore, raised. The upper port is opened to the oil pressure and lower port to drain. The relay piston is thus forced downwards and throttled valve partially closed. The downward movement of the throttle valve lowers the piston valve and so closes the port. As soon as the oil ports are covered, the relay piston is locked in position. This will occur only when the opening of the throttled valve is correct for the load on the turbine.

Since, for equilibrium of the governor mechanism, the piston valve must always be in its central position and covering both oil ports, the position of the governor sleeve will vary according to the position of the throttle valve. The position of the floating lever is indicated by chain dotted lines in fig. 9-35 for no-load and full load.

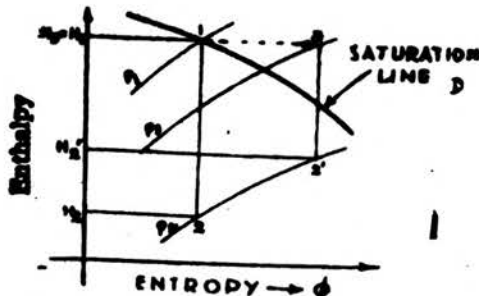


Fig. 9-36. Illustration of decrease in available energy due to throttling.

Although in throttling no energy is lost, the available energy (enthalpy drop) is decreased as illustrated in fig. 9-36. This figure shows dry saturated steam which may be expanded isentropically from point 1 (pressure p_1) to point 2 (pressure p_2) with isentropic total enthalpy drop ($H_1 - H_2$). If the governor first decreases pressure from p_1 to p_3 by throttling (point 1 to point 3), the isentropic total enthalpy drop is $H_3 - H_2'$. This is far less than available isentropic enthalpy drop ($H_1 - H_2$) without throttling.

This reduces the efficiency of the turbine at part load. This relationship between load and steam consumption for a turbine governed by throttling, is given by the well known Willain's straight line.

9.8.2 Nozzle control governing : Nozzle control is accomplished as shown in fig. 9-37. Poppet-type valves uncover as many steam passages as necessary to meet the load, each passage serving a group of nozzles. The control governor has the advantage of using steam at full boiler pressure.

In automatic governed land turbines, various arrangements of valves and groups of nozzle are employed. The arrangements are shown diagrammatically in fig. 9-37. An arrangement, often adopted with large steam turbines and with turbines using high-pressure steam, is shown in fig. 9-37 (A). The nozzle are divided into group N_1 , N_2 and N_3 and the control valves V_1 , V_2 and V_3 respectively. The number of nozzle groups may vary from three to five or more.

In fig. 9-37(A), three sets of nozzle N_1 , N_2 and N_3 having 10, 4 and 3 nozzles respectively, are shown. Thus, there are 17 nozzles in all and for the sake of illustration we shall assume that total power of the turbine is 340 kw. In order to simplify the explanation, let it be assumed that the steam delivered by each nozzle under the full pressure drop is sufficient to develop 20 kw. Actually this assumption is not justified. Then, at full load all the 17 nozzles will be delivering steam at full pressure and the turbine will operate at maximum efficiency. Similarly at 200 kw only the valve V_1 controlling the set of 10 nozzles would be open and at 280 kw valves V_1 and V_2 controlling the set of 14 nozzles would be open.

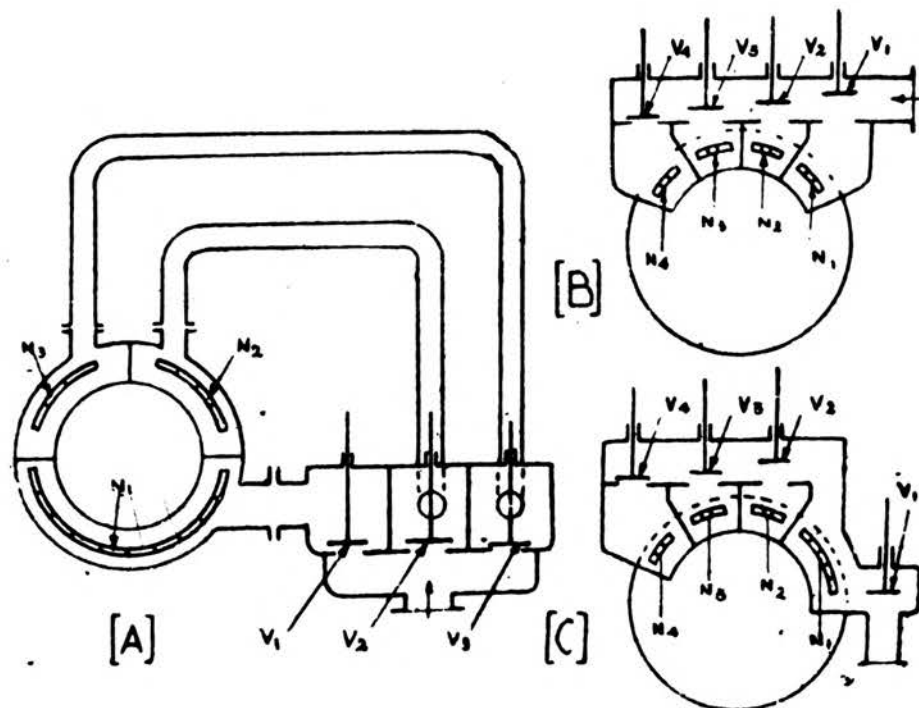


Fig. 9-37. Diagrammatic arrangement of nozzle control governing.

In fig. 9.37(B) arrangement is similar to fig. 9-37(A) except that all the nozzle control valves are arranged in a casting forming part of the cylinder or bolted thereto and containing passages leading to the individual nozzle groups. Although this arrangement is compact, the nozzles are contained in the upper half of the cylinder and the arc of admission is usually limited to 180° or less. The number of nozzle groups varies from four to twelve.

Fig. 9-37(C) shows an arrangement sometimes employed. The group of nozzle N_1 is under the control of the valve V_1 , through which all the steam entering the turbine passes. Further admission of steam is through the valves V_2 , V_3 in turn. In some instances, the nozzle group N_1 has been arranged in the lower half of the turbine and supplied with steam through a throttle valve V_1 up to, say, half load. For loads greater than half load, a further supply of steam is admitted through the valves V_2 , V_3 , etc.,

Whatever method of construction is adopted, the nozzle control is necessarily restricted to the first stage of the turbine, the nozzle areas in the other stages remaining constant. It follows that, provided the condition of the steam at inlet to the second stage is not materially affected by the changed condition in the first stage, the absolute pressure of the steam in front of the second stage nozzles will be directly proportional to the rate of steam flow through the turbine.

It is observed that there is a greater enthalpy drop available when nozzle control is employed but this greater enthalpy drop is not efficiently utilised at part load. Comparative tests shows that when there is a fairly large enthalpy drop in the first stage, nozzle control reduces the steam consumption.

9.8.3. By-pass governing : In modern impulse turbines, and specially those operating at very high pressure, the H.P. turbine comprises a number of stages of comparatively small mean diameter. All such turbines are usually designed for a definite load termed economical load, at which efficiency is the maximum. The economical load is made about 80% of the maximum continuous load.

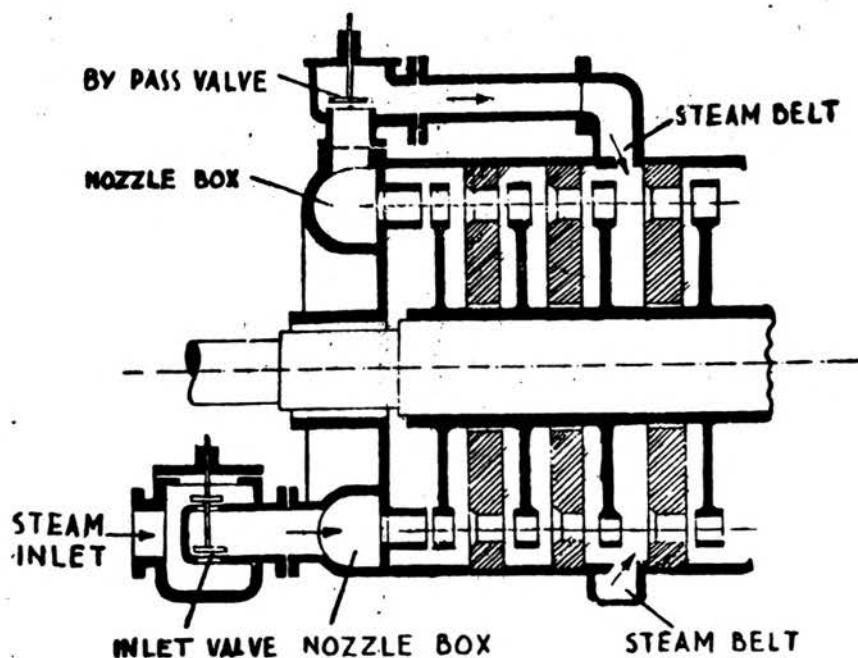


Fig. 9-38. Diagrammatic arrangement of by-pass governing.

Owing to the very small enthalpy drop in the first stage, it is not possible to employ nozzle control governing efficiently. Further-more, it is desirable to have full admission in the H.P. stage at the economic load so as to reduce losses.

These difficulties of regulation are overcome by the employment of by-pass governing as shown in fig. 9-38. All the steam entering the turbine passes through the inlet valve (which is under the control of the speed governor) and enters the nozzle box or steam chest. In certain cases, for example, this would suffice for all loads upto the economical

load, the governing being effected by throttling. For loads greater than the economical loads, a by-pass valve is opened allowing steam to pass from the first stage nozzle box into the steam belt and so into the nozzle of the fourth stage. The by-pass valve is not opened until the lift of the valve exceeds a certain amount; also as the load is diminishing the by-pass valve closes first. The by-pass valve is under the control of the speed governor for all loads within its range.

9.9. Special Forms of Steam Turbines

There are several industries such as paper making, textile, chemical, dyeing, sugar refining, carpet making, etc., where combined use of power and heating for process work is required. It is wasteful to generate steam for power and process purposes separately, because about 70 per cent of the heat supplied for power purposes will normally be carried away by the cooling water. But if the engine or turbine is operated with a normal exhaust pressure, then the temperature of the exhaust steam is too low to be of any use for heating purposes. By suitable modification of the initial steam pressure and of the exhaust pressure, it would be possible to generate the required power and still have available for process work a large quantity of heat in the exhaust steam. It follows, therefore, that from the practical stand-point, the thermal efficiency of a combined power and heating plant may approach unity.

There are two types of turbines employed in combined power and process plants, namely, the *back-pressure turbines* and the *steam extraction or pass-out turbines*.

9.9.1 Back-pressure turbine : The back-pressure turbine takes steam at boiler pressure and exhausts into a pipe which leads neither to a condenser nor to atmosphere, but to a process plant or other turbine. This may be employed in cases where the power generated by expanding steam from an economical initial pressure down to the heating pressure is equal to or greater than, the power requirements. Usually the exhaust steam from the turbine is superheated and in most cases it is not suitable for process work, partly because it is impossible to control its temperature and partly because of the fact that rate of the heat transfer from superheated steam to the heating surface is lower than that of saturated steam. For these reasons, a de-superheater is often used.

It is unlikely that the steam required for power generation will always be equal to that required for process work, and some means of controlling the exhaust steam pressure, must be employed if variations in the pressure and therefore of the steam saturation temperature are to be avoided.

In order to increase the power capacity of a existing installation, a high pressure boiler and a back-pressure turbine are added to it. This added high pressure boiler supplies steam to back-pressure turbine which exhausts into the old low pressure turbine.

9.9.2 Pass-out or extraction turbine : In many cases the power available from a back-pressure turbine through which the whole of the heating steam flows, is appreciably less than that required in the factory. This may be due to the small heating or process requirements, to a relatively high exhaust pressure, or a combination of both. In such a case it would be possible to install a back-pressure turbine to provide the heating steam and a condensing turbine to generate the extra power; but it is possible and usual, to combine functions of both machines in a single turbine. Such a machine is called pass-out or extraction turbine. In this, at some point intermediate between inlet and exhaust, some steam is extracted or passed out for process or heating purposes.

Since the power and speed of the turbine, as well as the quantity of process steam, are controlled by external conditions, while in the turbine the two are more or less related, it is obvious that some special form of governing is required. This usually takes the form of a sensitive governor which controls admission of steam to the high-pressure section,

so as to maintain constant speed – regardless of the power or process requirements.

9.9.3 Exhaust or low-pressure turbine : If a continuous supply of low pressure steam is available - for example from reciprocating steam engines exhaust - the efficiency of the whole plant may be improved by fitting an exhaust or low-pressure turbine. The exhaust turbine is chiefly used where there are number of reciprocating steam engines which work intermittently (not continuously); and, of necessity, are non-condensing, such as rolling mills and colliery engines. The exhaust steam from these engines, which would otherwise pass into the atmosphere and be wasted, is expanded in an exhaust turbine and then condensed.

In such a turbine some form of *heat accumulator* is required to collect the more or less irregular supply of low pressure steam from the non-condensing steam engines and deliver it to the turbine at the rate required. In some cases when the supply of low pressure steam falls below the demand, live steam from the boiler, with its pressure and temperature reduced, is used to make up the deficiency.

The pressure drop may be obtained by means of a reducing valve, or for large flows, more economically by expansion through another turbine. Sometimes the high-pressure and low-pressure turbines are combined on a common spindle. This combined unit is known as a *mixed pressure turbine* because of two supply pressures.

9.10 Material of Construction in Steam turbines

The different parts of steam turbine work under varying service conditions. For long operating life and low cost, appropriate material selection for each part is the essential requirement for an economic design. The most important part is that which meets with the entering steam having high temperature and pressure. The most commonly used materials for different components are as under :

Casing and steam and nozzle chests are usually prepared from steel castings. For steam temperature upto 450°C, steel with 0.3% C, 1.0% Mn, 0.6% Si, 0.06% S, 0.05% P, is used. Further for operating steam temperatures between 565° and 600°C, austenite steel is used. Its composition is 0.08 C, 16.0% Cr, 13.0% Ni, 2.0% Mo, and 0.8% Cb.

Rotors are assemblies of shafts, discs or drums and blades or buckets. Each of these have a wide choice of materials. Shaft material for low temperature may be hot rolled heat treated carbon steel bar stock or alloy steel forgings. For temperature upto 570°C, shaft is made from forging with composition 0.37% C, 1.0% Mn, 0.35% Si, 0.035% S, 0.035% P, 1.25% Cr, 1.5% Mo, 0.5% Ni, and 0.3% V. Above 570°C temperature, the shaft is of ferritic alloy forging with composition 0.3% C, 0.5% Ni, 1.0% Cr, and 1.25% Mo. For wheels for 345°C temperature, composition is 0.45% C, 0.9% Mn, 0.15% Si, 0.035% S, and 0.035% P.

Blades are made of cold rolled drawn steel. Usually stainless steel having 0.06% C, 0.25% Mn, 0.5 Si, 0.03% S, 0.03% P, 11.5% Cr, 0.4% Mo, 0.5% Ni is used.

For low temperature service, *nozzles, rings and diaphragms* are often of cast iron or malleable iron. As temperature goes higher, materials range through steel plate, cast steel, steel and stainless steel forging. Some diaphragms are rolled, some are cast from aluminum chromium steel.

Seals and gland packings are made from carbon to stainless steel, leaded bronze, leaded nickel brass, non-hardened stainless iron and corrosion resistant chrome-molybdenum materials, as per the requirements. *Springs* for holding packing are of inconel, monel or stainless steel.

Bearings are usually cast on bronze, steel or cast iron backs with inner lining of high tin-babbit. Journals and collars are usually integral part of the shaft and are of the same

material as the shaft. Some times they are built up of sprayed metal to make a hard surface.

The *bolts* of high pressure casing raises special problems due to high pressures because of *creep*. Gradual elongation under stress relaxes bolts hold on the casing joint. They are usually of 13-chrome-tungsten-molybdenum-vanadium alloy steel for higher temperature (above 450°C). This material resists temper embrittlement and oxidation and has higher notch-bar rupture strength.

Piping range from carbon steel for temperature below 450°C and medium pressures, to stainless steel of temperature upto 600°C with heavy thick walls.

Inlet pipe seals for turbine in 540°C – 565°C range are of stellite which is an alloy of chromium, cobalt, molybdenum and tungsten. These sealing rings allow pipe connections between separated steam chest and nozzle chests to move axially and transversely during start ups and shut downs. Piping oxidation at joints must be prevented. It freezes the sealing rings, resulting rigid connection, transmits piping expansion and contraction forces to the turbine casing, causing serious misalignment.

Governing valves are usually provided at the front end of the turbine and are made of carbon-chrome alloy steel. Steam must resist oxidation to prevent freezing in packings.

9.11 Steam Turbines for Power Generation

The continuous increase in the use of electrical energy has made necessary the construction of several additional generating stations at various parts of the country. Reliability, economy in first cost, and operating costs are achieved by installing the largest units practicable. Brief particulars of the turbines used for power generation are as under:

Power in MW	Steam pressure bar	Steam temperature °C	Reheat temperature °C	Approx. final feed temp. °C
30	40	455	-	171
60	60	482	-	196
100	100	566	-	204
120	100	538	538	224
200	160	566	538	236

Most of the turbines for power generation operate at 3,000 r.p.m. The final feed temperature lies between 0.7 and 0.73 times the initial steam saturation temperature and it has proved to be economical. The fairly general features of power generation turbines are as under :

- (i) Steam chests are usually placed alongside the high pressure turbine.
- (ii) Velocity compounding is done in first stage of H.P. turbine in order to reduce the pressure and temperature of steam to which H.P. turbine cylinder is exposed.
- (iii) Some form of turning gear at the coupling between L.P. turbine and generator, is provided for slow turning of turbine during warming up process and in cooling down process before coming to rest. This is required to prevent bending of rotor shaft.
- (iv) In last two or three stages of L.P. turbine, draining arrangement of water flung off the blades by centrifugal action, is made.
- (v) Cylinders are supplied in such a way that freedom of expansion and contraction due to temperature changes is adequate and simultaneously it does not disturb the vertical

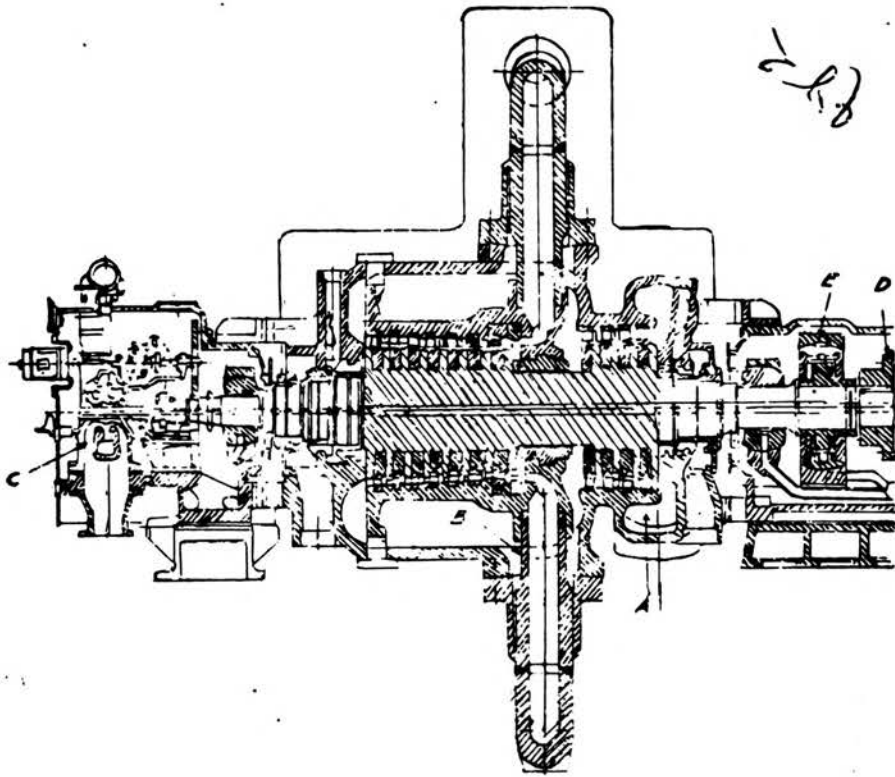


Fig. 9-39

alignment, i.e., cylinder and rotor remain concentric. In double shell construction, the inner-shell is so supported that they remain co-axial and hence concentric with the rotor.

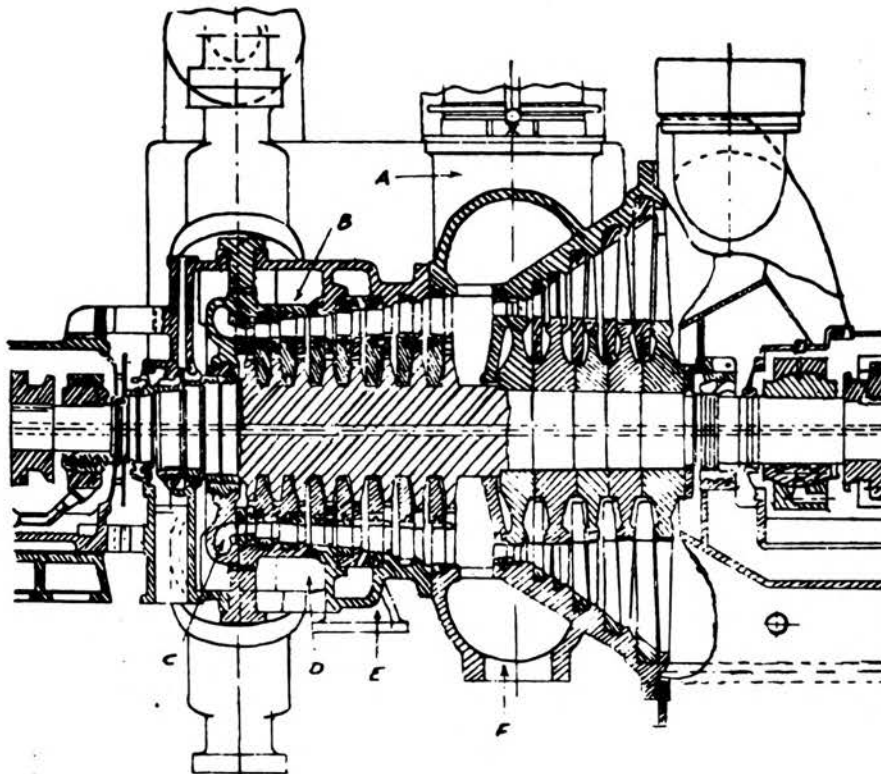


Fig. 9-40.

(vi) When wheels are shrunk on the rotor-spindle, the running speed of the rotor may be above the first transverse critical speed.

Typical construction of 200 MW steam turbine is shown in figs. 9-39, 9-40 and 9.41. Some of the details of three cylinders tandem turbine operating at 3000 r.p.m. are :

Inlet steam pressure	... 160 bar
Inlet steam temperature	... 565°C
Reheat temperature	... 538°C
No. of stages of reheating	... 6
Final temperature of feed heating	... 238°C
Vacuum	... 724 mm of Hg.

Fig. 9-40 shows H.P. turbine, part of which is of double-shell construction. Steam enters the nozzle box through four radial pipes *B*. After partial expansion in eight stages of impulse blading, the steam flows in reverse direction in the space between inner and outer space to enter the last four impulse stages for further expansion. Then it goes for reheating. Steam from reheater enters I.P. turbine nozzle box *C* of a short inner cylinder *B* (fig. 9-40) by way of four radial steam pipes. This cylinder *B* contains three impulse stages and is located by pads and keys so that while being free to expand and contract due to temperature changes, it remains concentric with the outer cylinder and with rotor. They are followed by five more impulse stages in which steam further expands. At this point steam flow divides. About one-third steam passes through single flow L.P. turbine arranged in the same casing as the I.P. stages, while about two-third of the steam passes through two connecting pipes *A* in to the centre of the double flow L.P. turbine as shown in fig. 9-41. All three I.P. expansions exhaust into common exhaust chamber and single shell condenser. Steam is bled from double flow L.P. cylinder for feed heating but not from the corresponding stages in I.P. turbine casing.

The H.P. and I.P. rotors are solid forging. The L.P. turbine have disc shrunk on and keyed to the shaft. The first stage of each L.P. turbine is impulse and the remaining stages have reaction blading. The active length of the blades in last stage is about 70 cm.

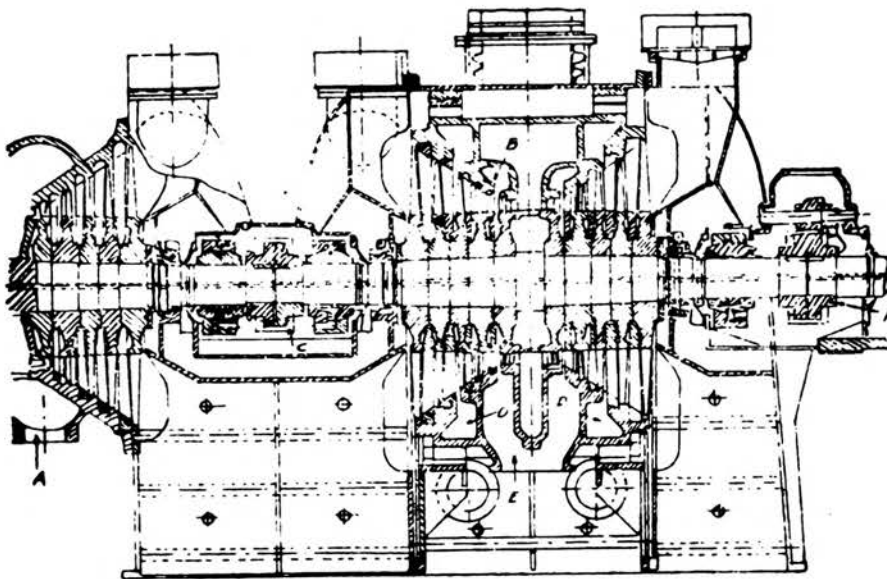


Fig. 9-41.

The three rotors are coupled together by "solid" coupling *E* as shown. One thrust block *F* is also provided between H.P. and I.P. cylinders to minimize the differential expansion between rotating and stationary parts. The overall length of the turbine is about 17 metres.

Full admission is done to all stages of H.P. turbine at all times. This is done because vibrations may occur due to partial ad-

mission. The outer casing of the double flow L.P. turbine is fabricated, due to their large size and difficulty in transportation. The inner casing carrying diaphragms and fixed blades is of steel castings. The main oil pump is double inlet C.F. pump G driven directly by turbine.

9.12 Other General Purpose Steam Turbines

9.12.1 Single casing condensing type : Fig. 9-42 shows this type of turbine. First stage consists of velocity compounding and impulse blading, while remaining are reaction stages. Dummy piston at the end of the first stage helps the thrust bearing counter balance the un-

balanced force of reaction stage. Cylinder is made up of forged sections welded together. After heat treatment, cylinder is slotted to receive reaction blading. Steam is removed for feed heating at four points. This type is used for power reaction generation.

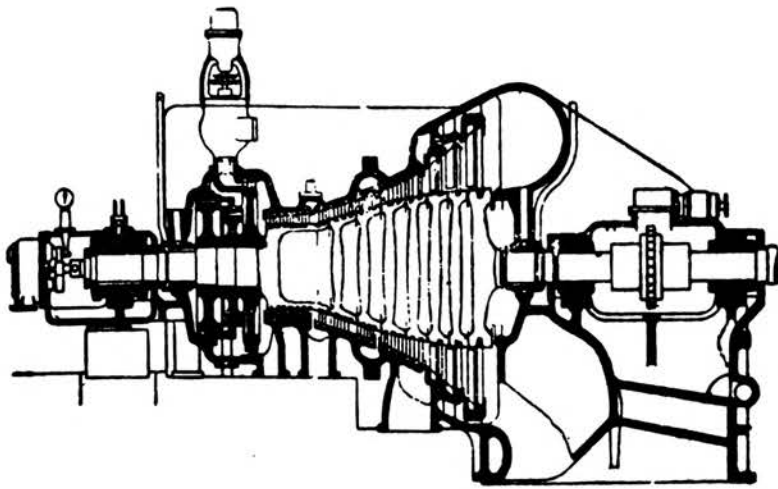


Fig. 9-42

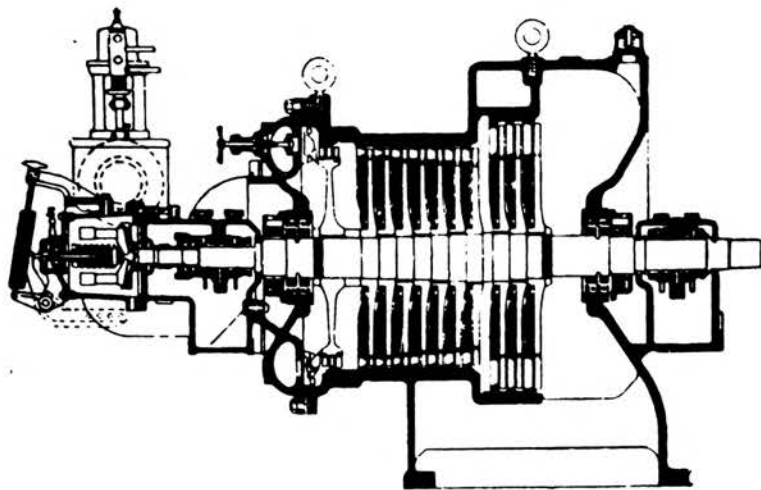


Fig. 9-43

9.12.2 Single stage multi-stage condensing type : Fig. 9-43 shows this type of turbine. First stage is velocity compounded and is followed by ten impulse stages. Ball thrust bearing keeps shaft aligned axially. On left, a centrifugal governor is provided to control steam flow. This unit is fitted with non-automatic extraction openings to bleed steam for feed water heating. Carbon ring seals are used at diaphragm and casing glands.

9.12.3 Radial flow double rotation turbine : Fig. 9-44 shows this type of turbine. This unit drives two A.C. generators, one on each shaft. Generators are coupled together electrically, to keep the oppositely rotating shafts in synchronism for best blade speed to steam speed ratio of the reaction stages. Multi-disc turbine is so arranged that the high pressure steam enters from below. It first flows into the annular steam chest, then through holes in the overhung blade disc to the centre area at the shaft. Steam then flows radially outward through first concentric set of blades. Then it turns 180° to flow radially inward

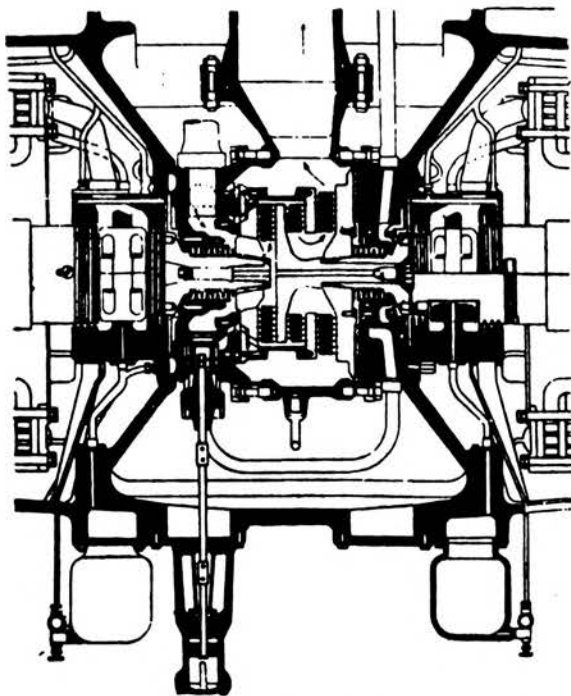


Fig. 9-44

seals at the two shafts, control steam flow through these clearances. This unit is used for only power generation.

through a second set of concentric blades. It again makes a 180° turn to flow radially outward through the third set of blades. From here steam flows into annular space leading to exhaust pipe at bottom of the turbine. By-pass valve to the right of disc in annular steam chest lets high pressure steam to skip first set of blades to enter the second set, providing overload operation at reduced efficiency. Maximum power developed by this type of turbine is 7,500 kW. It can be designed for automatic or non-automatic extraction of partly expanded steam. Strip type Labyrinth seals on the moving blade rings reduce the steam leakage past the blades, while concentric Labyrinth seals between over hung discs and inner casings cut down leakage short circuiting the blading. Labyrinth gland

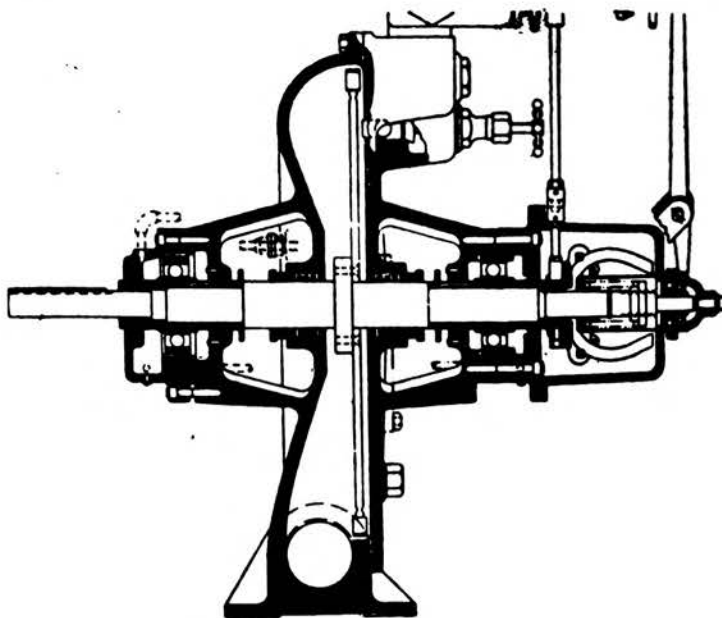


Fig. 9-45

9.13 Some Examples of Mechanical Drive Turbines

Mechanical drive turbines are usually single-stage velocity compounded. Fig. 9-45 shows a single-stage impulse turbine. Maximum rating is 75 kw. It is similar to De-laval turbine.

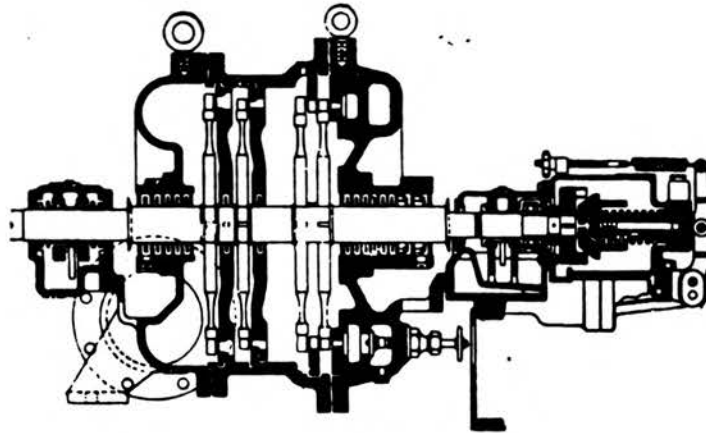


Fig. 9-46

Fig. 9-46 shows multi-stage turbine. First stage is velocity compounded and it uses two separate wheels. This type may be condensing type and runs at 10,000 r.p.m. It has carbon ring seals ring oiled journal bearings and a double thrust ball bearing to control position of the shaft.

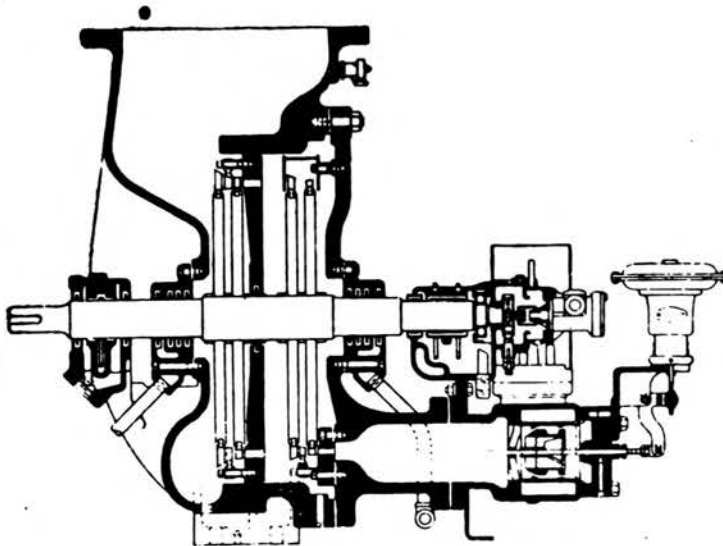


Fig. 9-47

Fig. 9-47 shows variable speed turbine. Usually this type is used to drive the compressor with range of speed 3,500 to 6,000 r.p.m. They are usually condensing type. In this type, as shown in fig. 9-47, two velocity compounded stages are provided.

Tutorial - 9

1. Delete the phrase which is not applicable in the following statements :

- (i) The thermal efficiency of a steam turbine is higher/lower than that of a steam engine.
- (ii) Steam turbine is an internal/external combustion thermal prime mover.
- (iii) Balancing is perfect in case of steam turbine/steam engine.
- (iv) Steam turbines work on modified Rankine/Rankine cycle.
- (v) A steam turbine develops power at a uniform/changing rate and hence does not need any flywheel.
- (vi) In an impulse turbine, steam expands in nozzles/blades.
- (vii) The speed of simple impulse wheel is too high/low for practical purposes.
- (viii) The steam turbines are mostly of axial/radial flow type.
- (ix) In case of an impulse turbine, the relative velocity at outlet is greater/less than that at inlet, due to friction.
- (x) If friction is neglected in case of an impulse turbine, relative velocity at inlet and relative velocity at outlet are equal/different in magnitude.

[Delete : (i) lower, (ii) internal, (iii) steam engines, (iv) modified Rankine, (v) changing, (vi) blades, (vii) low, (viii) radial (ix) greater, (x) different].

2. Fill in the blanks to complete the following statements :

- (i) The two main types of steam turbines are _____ and _____.
- (ii) Speed obtained in case of steam turbines may be as high as _____ r.p.m.
- (iii) The method of abstracting steam at certain section of turbine is known as _____.
- (iv) In an impulse turbine the expansion of steam takes place in the _____ only, where the pressure decreases and velocity increases.
- (v) In an impulse turbine, the pressure of steam remains constant while it passes over the _____ of the turbine.
- (vi) In case of reaction steam turbine, the steam expands as it flows over the _____.
- (vii) An actual reaction steam turbine is a combined _____ and _____ steam turbine.
- (viii) In an actual reaction turbine, steam expands partly in stationary blades and partly as it flows over the _____.
- (ix) Degree of reaction is defined as the ratio of isentropic enthalpy drop in the moving blades to isentropic enthalpy drop in the _____ of the reaction turbine.
- (x) In case of reaction turbines, since the steam expands continuously in both the fixed and moving blades, its relative velocity does not remain constant but _____ due to the expansion of steam.
- (xi) The velocity of a simple impulse steam turbine is too _____ for practical purposes and as such the speed has to be _____ by some suitable means.
- (xii) The _____ turbine was the first impulse steam turbine successfully built in 1889.

[(i) impulse, reaction; (ii) 30,000, (iii) bleeding, (iv) nozzles, (v) blades, (vi) moving blades, (vii) impulse, reaction, (viii) moving blades, (ix) entire stage, (x) increases, (xi) high, reduced, (xii) De Laval]

3. Select the correct phrase out of the phrases given below for each statement :

- (i) In a Parsons reaction turbine, the relative velocity at outlet is
 - (a) less than that at the inlet,
 - (b) greater than that at the inlet,
 - (c) equal to that at the inlet,
 - (d) equal to blade speed.
- (ii) In the impulse turbine the steam is expanded
 - (a) in nozzles,
 - (b) in blades,
 - (c) partly in nozzles and partly in blades,
 - (d) neither in nozzles nor in blades.
- (iii) In a condensing steam turbine the steam is exhausted
 - (a) at atmospheric pressure,

- (b) below atmospheric pressure,
- (c) above atmospheric pressure,
- (d) at any pressure.
- (iv) De Laval turbine is a
 - (a) simple reaction turbine,
 - (b) simple impulse turbine,
 - (c) velocity compounded impulse turbine,
 - (d) pressure compounded impulse turbine.
- (v) Steam turbine works on
 - (a) Rankine cycle,
 - (b) modified Rankine cycle,
 - (c) Bell-Coleman cycle,
 - (d) Carnot cycle.
- (vi) Parsons reaction turbine is basically
 - (a) an impulse-reaction turbine,
 - (b) a pressure compounded impulse turbine,
 - (c) a velocity compounded impulse turbine,
 - (d) a pure reaction turbine.
- (vii) Curtis turbine is basically
 - (a) a velocity compounded impulse turbine,
 - (b) a pressure compounded impulse turbine,
 - (c) a simple impulse turbine,
 - (d) an impulse-reaction turbine,
 - (e) a pure reaction turbine.
- (viii) Rateau turbine is basically
 - (a) a velocity compounded impulse turbine,
 - (b) a pressure compounded impulse turbine,
 - (c) an impulse-reaction turbine,
 - (d) a pure reaction turbine.
- (ix) The main advantage of reaction turbine as compared to impulse turbine* is
 - (a) high blade speed,
 - (b) low blade speed,
 - (c) high efficiency,
 - (d) high output.
- (x) Most widely used method of governing steam turbine is
 - (a) throttle governing,
 - (b) nozzle control governing,
 - (c) by-pass governing.

[(i) b, (ii) a, (iii) b, (iv) b, (v) a, (vi) a, (vii) a, (viii) b, (ix) b, (x) a]

4. Steam issues from the nozzles of a single impulse turbine at 850 m/sec. on to blades moving at 350 m/sec. The blades tip angles at inlet and exit are each 36° . The steam is to enter the blades without shock and the flow over the blades is frictionless. Determine : (a) the angle at which the nozzles are inclined to the direction of motion of the blades, and (b) the diagram efficiency. [(a) 22° ; (b) 84.6%]
5. Steam leaves the nozzle of a single impulse wheel turbine at 900 m/sec. The nozzle angle is 20° and the blade angles are 30° at inlet and outlet. What is the blade velocity and the work done per kilogram of steam ? Assume the flow over the blades as frictionless. [312 m/sec; 333 kJ]
6. A stage in an impulse turbine consists of converging nozzles and one ring of moving blades. The nozzle angles are 22° and the moving blades have both tip angles of 35° . If the velocity of steam at the exit from the nozzles is 450 m/sec., find the blade speed so that the steam shall pass on to the blades without shock, and the stage efficiency, neglecting frictional losses, if the blades run at this speed.
If the relative velocity of the steam is reduced by 15% in passing through the blade ring, find the actual efficiency and the thrust on the shaft, when the blade ring develops 36.8 kW.

[176 m/sec; 83.6%; 77.6%, 11.7 N]

7. An impulse turbine with a single row wheel is to develop 99.3 kW, the blade speed being 150 m/sec. A mass of 2 kg of steam per second is to flow from the nozzles at a speed of 350 m/sec. The velocity coefficient of the blades may be assumed to be 0.8 while the steam is to flow axially after passing through the blade ring. Determine the nozzle angle, the blade angles at inlet and exit assuming no shock. Estimate also the diagram efficiency of the blading.

[nozzles angle = 18.7° ; Inlet blade angle = 31.75° ; exit blade angle = 28.3° ; Diagram efficiency = 81%]

8. Compare steam turbine with the reciprocating steam engine on the basis of the mechanical construction. What are the advantages of steam turbine plant over the reciprocating steam engine plant?

A De Laval steam turbine has a wheel 30 cm mean diameter and runs at 12,000 r.p.m. The nozzles are inclined at 20° to the plane of the wheel and escape velocity of steam from nozzles is 850 m/sec. There is a 10% loss of velocity in the blades and the inlet and outlet angles of the blades are equal. Determine :

- (a) the blade angles,
 (b) the absolute velocity of the steam at the exit from the blades, and
 (c) the wheel or diagram efficiency.

[(a) 25.5° ; (b) 446 m/sec; (c) 60.3%]

9. The steam from the nozzles of a single-stage impulse turbine has a velocity of 800 m/sec. and are inclined at 20° to the direction of motion of the blades. Determine the necessary inlet angle of the blades so that no shock occurs for a blade speed of 300 m/sec.

Assuming that friction reduces the relative velocity of the steam by 10% as it passes over the blades and the blade angles are equal, find the work done per kg of steam supplied.

[31.2° ; 257.5 kJ]

10. The nozzle of a turbine stage delivers 4 kg of steam per second at an angle of 18° and a speed of 425 m/sec. If the blading outlet angle is 22° and the blade velocity coefficient is 0.76, determine the blade power developed and the blade inlet angle. Take the peripheral speed of the wheel as 170 m/sec.

[288 kW; 29.3°]

11. At one stage in impulse turbine the steam is expanded from 8.5 bar and 95% dry, to 3 bar. If the flow through the nozzle is frictionless adiabatic, find the velocity of the steam as it leaves the nozzle. If the nozzle is inclined at 20° to the direction of the blades and the blade angle at exit is 30° to the same direction, the blade speed is 0.4 of the steam velocity at exit from the nozzle, and the velocity of steam relative to the blades suffers a 10 per cent drop in passing over the blades, find the power developed when the steam flow is 4.5 kg/sec.

[604 m/sec; 681 kW]

12. The outlet area of the nozzles in a simple impulse turbine is 22.5 cm^2 and steam leaves them 0.9 dry at 3 bar and at 750 m/sec. The nozzles are inclined at 20° to the plane of the wheel, the blade speed is 300 m/sec., the blade outlet angles are 30° and the blade velocity coefficient is 0.82. Calculate : (a) the power developed in the blades, (b) the steam used per kW-hour, (c) the diagram efficiency, (d) the axial thrust on the shaft, and (e) loss of kinetic energy due to blade friction.

[(a) 680 kW; (b) 16 kg/kW-hour; (c) 79.6%; (d) 181 N; (e) 114 kJ]

13. In a De Laval steam turbine the blade angles are 30° at inlet and exit. The steam leaves the nozzle at 380 m/sec. and the blade speed is 75 m/sec. If the relative velocity of the steam is reduced by 15 per cent during its passage through the blades, find : (a) the nozzle angle, and (b) the blade efficiency.

[(a) 24.4° ; (b) 52.3%]

14. Steam leaves the nozzle of a simple impulse turbine at 900 m/sec. The nozzle angle is 22° , and the blade angles are 30° at inlet and outlet, and the blade velocity coefficient is 80 per cent. Calculate :

- (a) the blade velocity, and
 (b) the steam flow in kg per hour if the power developed by turbine is 235 kW.

[(a) 250 m/sec; (b) 3,225 kg per hr.]

15. The steam supplied to a single-row impulse wheel turbine expands in the nozzle over such a range that the adiabatic enthalpy drop is 88 kJ/kg. The nozzle efficiency is 93% and nozzle angle is 15° . If the blading speed is 175 m/sec., the outlet blade angle is 18° and the velocity coefficient for the blading is 0.82, determine : (a) suitable inlet angle for the moving blade, (b) the speed of the steam after discharge from the blading, (c) the diagram efficiency, and (d) the power developed by the turbine if 2,750 kg of steam per hour is supplied to the turbine.

[(a) 25.8° ; (b) 62 m/sec; (c) 86.2%; (d) 54 kW]

16. In a stage of an impulse turbine provided with a single-row wheel, the mean diameter of the blade ring is 80 cm and the speed of rotation is 3,000 r.p.m. The steam issues from the nozzle with a velocity of 275 m per sec and the nozzle angle is 20° . The inlet and outlet angles of the blades are equal, and due

to friction in the blade channels the relative velocity of the steam at outlet from the blade is 0.86 times the relative velocity of steam entering the blades. What is the power developed in the blading when the axial thrust on the blades is 120 N ?

[285 kW]

17. The mean diameter of the blades of impulse turbine with a single-row wheel is one metre and the speed of rotation is 3,000 r.p.m. The nozzle angle is 18° , the ratio of blade speed to steam speed is 0.42, the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of blade is to be 3° less than the inlet angle. The steam flow is 7 kg per second.

Determine : (a) the tangential force on the blades, (b) the power developed in the blades, (c) the blading efficiency, and (d) the axial thrust on the blades.

[(a) 2,600 N; (b) 407 kW; (c) 83.2%; (d) 190 N]

18. Steam issues from nozzle of a De Laval turbine with a velocity of 1,000 m/sec. The nozzle angle is 20° , the mean blade velocity is 365 m/sec. and the inlet and outlet angles of the blades are equal. The steam flow through the turbine is 800 kg per hour. The ratio of relative velocity at outlet from the blades to that at inlet is 0.8.

Calculate :

- the blade angles,
- the relative velocity of the steam entering the blades,
- the tangential force on the blades,
- the power developed, and
- the blade efficiency

[(a) 30.8° ; (b) 669 m/sec; (c) 230.5 N; (d) 84 kW; (e) 75.8%]

19. Steam issues from the nozzles of a De Laval turbine with a velocity of 920 m per sec. The nozzle angle is 20° , the mean diameter of the blades is 25 cm and the speed of rotation is 20,000 r.p.m. The steam flow through the turbine is 0.18 kg per sec. If the ratio of relative velocity at outlet from the blades to that at inlet is 0.82, calculate :

- The tangential force on blades,
- The work done on blades per sec.,
- The power of the wheel,
- The efficiency of blading,
- The axial force on blades, and
- The inlet angle of blades for shockless inflow of steam.

Assume that the outlet angle of blades is equal to the inlet angle.

[(a) 197 N; (b) 51.8 kJ; (c) 51.8 kW; (d) 68%; (e) 10.1 N; (f) 27.6°]

20. Enumerate the types of steam turbines. Explain why impulse turbines are compounded and explain with diagrams the methods of compounding.
21. Explain with the aid of neat sketches the various methods adopted to reduce the rotor speed of the impulse steam turbines.

Enumerate the advantages and disadvantages of velocity compounded impulse turbines.

In a velocity compounded impulse turbine, the initial speed of the steam is 700 m per sec and turbine uses 4.5 kg of steam per second. The nozzle discharge angle is 16° and the outlet angles for the blades are :

First moving blades 20° , fixed blades 25° , and second moving blades 28° .

The blade speed is 150 m/sec and the ratio between the relative velocities at the outlet and inlet edges of the blades is 0.9. Draw the velocity diagrams to a scale of 1 cm = 25 m/sec. and determine : (a) the power developed, (b) the diagram or blade efficiency, and (c) the axial thrust on moving blades.

[(a) 864 kW; (b) 78.4%; (c) 292.3 N]

22. The outlet angle of the blade of Parsons turbine (reaction turbine) is 20° and the axial velocity of flow of steam is 0.5 times the mean velocity of the blade. Draw the velocity diagram for a stage consisting of one fixed and one moving row of the blades, given that the mean diameter is 70 cm and that speed of rotation is 3,000 r.p.m. Find the inlet angles of the blades if steam enters without shock.

If the mean steam pressure is 5.5 bar and the blade height is 6.25 cm, and the steam is dry saturated, find the power developed per pair of blades.

[$53^\circ 54'$; 457 kW]

23. A reaction turbine runs at 300 r.p.m. and its steam consumption is 15,500 kg/hour. The pressure of steam at a certain pair is 1.8 bar, and its dryness is 0.92. The power developed by the pair is 3.31 kW and the

discharge blade tip angle is 20° for both fixed and moving blades, and the axial velocity of flow is 0.72 of the blade speed. Find the drum diameter and the blade height. Neglect blade thickness.

[92.55 cm; 10.45 cm]

24. What is the object of compounding in steam turbines ? Distinguish between velocity compounding and pressure compounding. With the help of suitable curves show the variations of pressure and velocity in the above methods of compounding.

25. Write briefly on the following, giving sketches wherever necessary :

(a) The reason for velocity compounding and pressure compounding of steam turbines.

(b) Principle of working of reaction steam turbines, and

(c) Blade friction and its effects on velocity diagrams of impulse steam turbines.

26. An impulse stage of a turbine has two rows of moving blades separated by fixed blades. The steam leaves the nozzles at an angle of 20° with the direction of motion of the blades. The exit angles are : 1st moving 30° ; fixed, 22° ; 2nd moving 30° .

If the adiabatic enthalpy drop for the nozzle is 188 kJ/kg and the nozzle efficiency is 90%, find the blade speed necessary if the final velocity of the steam is to be axial. Assume a loss of 15% in the relative velocity for all blade passages. Find also blade efficiency and the stage efficiency.

[116.4 m/sec; 70.04%; 63.6%]

27. Define the term "re-heat factor" used in connection with steam turbines.

In a four-stage pressure compounded turbine the steam is supplied at pressure of 24 bar and superheated to a temperature of 350°C . The exhaust pressure is 0.07 bar, and the overall turbine efficiency is 0.72. Assuming that the work is shared equally between the stages, and that the condition line is straight, estimate: (a) the stage pressures, (b) the efficiency of each stage, and (c) the re-heat factor.

[(a) 7 bar, 1.84 bar, 0.4 bar; (b) 61%, 65.5%, 68.8%, 73.5%; (c) 1.07]

28. Steam at 21 bar with 60°C of superheat expands in a turbine to 3.5 bar. It is then re-heated at this pressure to its original temperature and finally expanded in a second turbine to 0.15 bar, the efficiency being 0.8 for each expansion. Sketch the enthalpy - entropy diagram for the whole process and mark on it the heat content of the steam at the beginning and end of each expansion. Determine the final condition of the steam and the work done per kilogram of steam.

[0.977; 774 kJ/kg]

29. Explain the process of feed heating by "bleeding." Show that in general, bleeding improves the efficiency of steam plant.

Find the theoretical thermal efficiency of a steam plant working between the pressures 10 bar, steam being dry saturated, and 0.06 bar, (a) without bleeding, (b) when the correct mass of steam is bled at 1.5 bar.

[(a) 28.7%; (b) 30.0%]

30. What are advantage of feed heating by bled steam ?

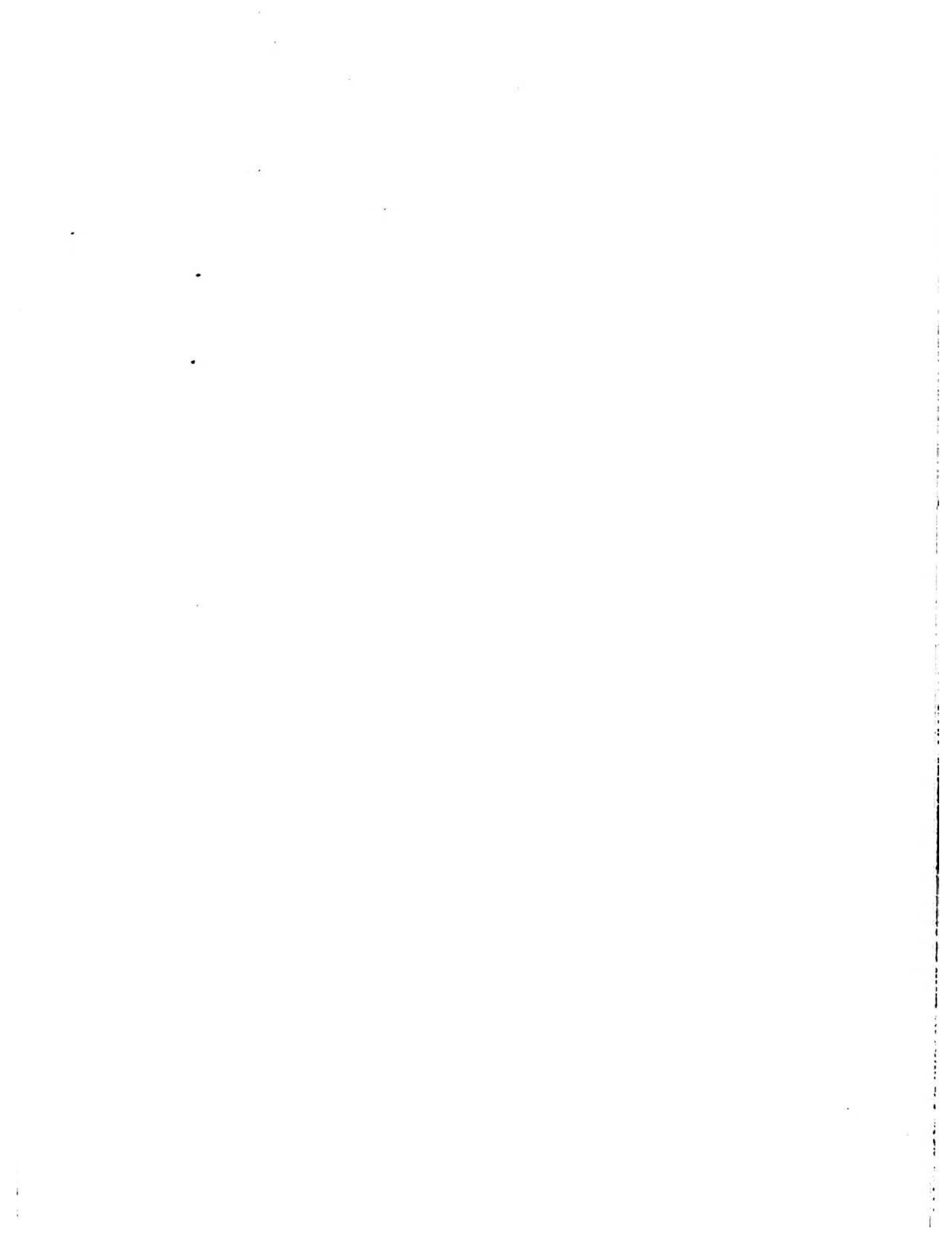
A steam turbine is fitted with a regenerative feed water heating system in which the heating is performed by steam extracted from the turbine at two different pressures. The heating steam, condensed to water in the high-pressure heater, is drained into the steam space of the low-pressure heater and, together with the water condensed in the low-pressure heater, is then drained to the condenser. The following table gives particulars of the process :

	Total enthalpy in kJ/kg
Steam entering turbine	.. 3,232
Steam entering high-pressure heater	.. 2,830
Steam entering low-pressure heater	.. 2,604
Steam entering condenser	.. 2,324
	Temperature
Feed water leaving hot-well and entering low-pressure heater	.. 28°C
Feed water entering high-pressure heater	.. 75°C
Feed water leaving high-pressure heater	.. 123°C
Drain water leaving low-pressure heater and entering condenser	.. 78°C
Drain water leaving high-pressure heater	.. 127°C

Assuming that the mass of feed water passing through the heaters is equal to the mass of steam entering the turbine, each being 13,500 kg per hour, find the mass of bled steam passing per hour into each heater, the power developed by steam in the turbine, and the thermal efficiency of the process.

[heater No.1 - 1,180 kg/hr; Heater No. 2 - 1,070 kg/hr; Power = 3,154 kW; Thermal eff. = 30.97%]

31. Explain what do you understand by *bleeding* as applied to steam turbine practice.
32. Write short notes on the following, giving sketches wherever necessary :
 - (i) Governing of steam turbines, (ii) Choice of materials for turbine blades, (iii) Back-pressure steam turbines, (iv) Steam extraction turbines, (v) Balancing end thrust of reaction steam turbines, (vi) Exhaust steam turbines, and (vii) Turbines for power generation.
33. Write detailed note on the governing of steam turbines.
34. What is the material of construction in the steam turbines components ? Suggest the material for low cost and long life of critical parts of steam turbines.
35. Illustrate some examples of mechanical drive steam turbines.



STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m^3/kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
.006113	0.1	206.136	.01	2501.3	2501.4	.0000	9.1562
.007	1.89	181.255	7.91	2496.9	2504.4	.0288	9.1064
.008	3.77	159.675	15.81	2492.5	2508.3	.0575	9.0578
.009	5.45	142.789	22.89	2488.5	2511.4	.0829	9.0142
.010	6.98	129.208	29.30	2484.9	2514.2	.1059	8.9756
.011	8.37	118.042	35.17	2481.6	2516.8	.1268	8.9408
.012	9.66	108.696	40.58	2478.6	2519.1	.1460	8.9091
.013	10.86	100.755	45.60	2475.7	2521.3	.1637	8.8792
.014	11.98	93.922	50.31	2473.1	2523.4	.1802	8.8529
.015	13.03	87.980	54.71	2470.6	2525.3	.1957	8.8279
.016	14.02	82.763	58.87	2468.3	2527.1	.2102	8.8044
.017	14.95	78.146	62.80	2466.0	2528.8	.2238	8.7825
.018	15.84	74.030	66.54	2463.9	2530.5	.2368	8.7618
.019	16.69	70.337	70.10	2461.9	2532.0	.2401	8.7422
.020	17.50	67.004	73.48	2460.0	2533.5	.2607	8.7237
.021	18.28	63.981	76.74	2458.2	2534.9	.2719	8.7060
.022	19.02	61.226	79.85	2456.4	2536.3	.2826	8.6892
.023	19.73	58.705	82.83	2454.8	2537.6	.2928	8.6732
.024	20.42	56.389	85.72	2453.1	2538.8	.3026	8.6579
.025	21.08	54.254	88.49	2451.6	2540.0	.3120	8.6432
.026	21.72	52.279	91.17	2450.1	2541.2	.3211	8.6290
.027	22.34	50.446	93.75	2448.6	2542.3	.3299	8.6155
.028	22.94	48.742	96.27	2447.2	2543.4	.3384	8.6024
.029	23.52	47.152	98.7	2445.8	2544.5	.3466	8.5898
.030	24.08	45.665	101.5	2444.5	2545.5	.3545	8.5776
.032	25.16	42.964	105.57	2441.9	2547.5	.3697	8.5545
.034	26.19	40.572	109.84	2439.5	2549.3	.3840	8.5327
.036	27.16	38.440	113.90	2437.2	2551.1	.3975	8.5123
.038	28.08	36.527	117.77	2435.0	2552.8	.4104	8.4930
.040	28.96	34.800	121.46	2432.9	2554.4	.4226	8.4746

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.042	29.81	33.234	125.00	2430.9	2555.9	.4343	8.4572
.044	30.62	31.806	128.39	2429.0	2557.4	.4455	8.4406
.046	31.40	30.500	131.65	2427.2	2558.8	.4562	8.4248
.048	32.15	29.299	134.79	2425.4	2560.2	.4665	8.4096
.050	32.88	28.192	137.82	2423.7	2561.5	.4764	8.3951
.055	35.58	25.769	144.95	2419.6	2564.5	.4997	8.3613
.060	36.16	23.739	151.53	2415.9	2567.4	.5210	8.3304
.065	37.63	22.014	157.67	2412.4	2570.0	.5408	8.3020
.070	39.00	20.530	163.40	2409.1	2572.5	.5592	8.2758
.075	40.29	19.238	168.79	2406.0	2574.8	.5764	8.2515
.080	41.51	18.103	173.88	2403.1	2577.0	.5926	8.2287
.085	42.67	17.099	178.70	2400.3	2579.0	.6079	8.2073
.090	43.76	16.203	183.29	2397.7	2581.0	.6224	8.1872
.095	44.81	15.399	187.65	2395.2	2582.9	.6362	8.1632
0.10	45.81	14.674	191.83	2392.8	2584.7	.6493	8.1502
.11	47.69	13.415	199.67	2388.3	2588.0	.6738	8.1168
.12	49.42	12.361	206.92	2384.1	2591.1	.6963	8.0863
.13	51.04	11.465	213.67	2380.2	2593.9	.7172	8.0584
.14	52.55	10.693	219.99	2376.6	2596.6	.7366	8.0325
.15	53.97	10.022	225.94	2373.1	2599.1	.7549	8.0085
.16	55.32	9.433	231.56	2369.9	2601.4	.7720	7.9860
.17	56.59	8.910	236.89	2366.8	2603.7	.7882	7.9649
.18	57.80	8.445	241.95	2363.8	2605.8	.8035	7.9451
.19	58.96	8.027	246.78	2361.0	2607.8	.8181	7.9263
.20	60.06	7.649	251.40	2358.3	2609.7	.8320	7.9085
.21	61.12	7.307	255.83	2355.7	2611.6	.8452	7.8916
.22	62.14	6.995	260.08	2353.2	2613.3	.8579	7.8756
.23	63.12	6.709	264.18	2350.8	2615.0	.8701	7.8602
.24	64.06	6.446	268.12	2348.5	2616.6	.8818	7.8455
.25	64.97	6.204	271.93	2346.3	2618.2	.8931	7.8314

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			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
.26	65.85	5.980	275.61	2344.1	2619.7	.9040	7.8179
.27	66.70	5.772	279.17	2342.0	2621.2	.9145	7.8049
.28	67.53	5.579	282.62	2340.0	2622.6	.9246	7.7924
.29	68.33	5.398	285.98	2338.0	2624.0	.9344	7.7803
.30	69.10	5.229	289.23	2336.1	2625.3	.9439	7.7686
.32	70.60	4.922	295.48	2332.4	2627.8	.9622	7.7465
.34	72.01	4.650	301.40	2328.8	2630.2	.9723	7.7257
.36	73.36	4.408	307.05	2325.5	2632.5	.9956	7.7061
.38	74.64	4.190	312.43	2322.3	2634.7	1.0159	7.6876
.40	75.87	3.993	317.58	2319.2	2636.8	1.0259	7.6700
.42	77.05	3.815	322.51	2316.2	2638.7	1.0400	7.6534
.44	78.18	3.652	327.26	2313.4	2640.6	1.0536	7.6375
.46	79.27	3.503	331.83	2310.6	2642.5	1.0666	7.6223
.48	80.32	3.367	336.23	2308.0	2644.2	1.0790	7.6078
.50	81.33	3.240	340.49	2305.4	2645.9	1.0910	7.5939
.55	83.72	2.964	350.54	2299.3	2649.8	1.1193	7.5615
.60	85.94	2.732	359.86	2293.6	2653.5	1.1453	7.5320
.65	88.01	2.535	368.54	2288.3	2656.3	1.1694	7.5048
.70	89.95	2.365	376.70	2283.3	2660.0	1.1919	7.4766
.75	91.78	2.217	384.39	2278.6	2663.0	1.2124	7.4564
.80	93.50	2.087	391.66	2274.1	2665.8	1.2329	7.4346
.85	95.14	1.972	398.57	2269.8	2668.4	1.2517	7.4141
.90	96.71	1.869	405.15	2265.7	2670.9	1.2695	7.3949
.95	98.20	1.777	411.83	2261.8	2673.2	1.2864	7.3766
1.00	99.63	1.694	417.46	2258.0	2675.5	1.3026	7.3594
1.01325	100.00	1.6729	419.04	2257.0	2676.1	1.3069	7.3549
1.05	101.00	1.6184	423.24	2254.4	2677.6	1.3184	7.3430
1.10	102.31	1.5495	428.79	2250.9	2679.7	1.3329	7.3273
1.15	103.58	1.4864	434.15	2247.5	2681.7	1.3471	7.3124
1.20	104.80	1.4284	439.32	2244.2	2683.5	1.3608	7.2981
1.25	105.99	1.3749	444.32	2241.0	2685.4	1.3740	7.2844

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			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
1.30	107.13	1.3254	449.15	2238.0	2687.1	1.3867	7.2712
1.35	108.24	1.2794	453.13	2235.0	2688.8	1.3990	7.2586
1.40	109.31	1.2366	458.39	2232.1	2690.4	1.4109	7.2464
1.45	110.36	1.1967	462.80	2229.1	2692.0	1.4224	7.2347
1.50	111.37	1.1593	467.11	2226.5	2693.6	1.4336	7.2233
1.55	112.36	1.1243	471.28	2223.8	2695.0	1.4444	7.2123
1.60	113.32	1.0914	475.36	2221.1	2696.5	1.4550	7.2017
1.65	114.26	1.0604	479.33	2218.5	2697.9	1.4652	7.1914
1.70	115.17	1.0312	483.20	2216.0	2699.2	1.4752	7.1814
1.75	116.06	1.0036	486.99	2213.6	2700.6	1.4849	7.1717
1.80	116.93	.9775	490.68	2211.2	2701.8	1.4944	7.1623
1.85	117.79	.9528	494.30	2208.8	2703.1	1.5036	7.1532
1.90	118.62	.9293	497.84	2206.5	2704.3	1.5127	7.1443
1.95	119.43	.9070	501.31	2204.2	2705.5	1.5225	7.1356
2.00	120.23	.8857	504.70	2201.9	2706.7	1.5301	7.1271
2.05	121.02	.8655	508.03	2199.8	2707.8	1.5386	7.1189
2.10	121.74	.8462	511.29	2197.6	2708.9	1.5468	7.1109
2.15	122.53	.8277	514.48	2195.5	2710.0	1.5549	7.1030
2.20	123.27	.8101	517.63	2193.4	2711.0	1.5628	7.0953
2.25	124.00	.7933	520.72	2191.3	2712.1	1.5706	7.0878
2.30	124.71	.7771	523.74	2189.3	2713.1	1.5782	7.0805
2.35	125.41	.7616	526.72	2187.3	2714.1	1.5856	7.0733
2.40	126.13	.7467	529.65	2185.4	2715.0	1.5930	7.0663
2.45	126.77	.7324	532.53	2183.5	2716.0	1.6002	7.0594
2.50	127.44	.7187	535.37	2181.5	2716.9	1.6072	7.0527
2.55	128.09	.7055	538.15	2179.7	2717.8	1.6142	7.0461
2.60	128.73	.6918	540.90	2177.8	2718.7	1.6210	7.0396
2.65	129.37	.6805	543.60	2176.0	2719.6	1.6277	7.0333
2.70	129.99	.6687	546.27	2174.2	2720.5	1.6343	7.0270
2.75	130.60	.6573	548.89	2172.4	2721.3	1.6408	7.0209

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m ³ /kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
2.80	131.21	.6463	551.48	2170.7	2722.1	1.6472	7.0149
2.85	131.81	.6357	554.02	2168.9	2723.0	1.6535	7.0090
2.90	132.39	.6254	556.54	2167.2	2723.8	1.6597	7.0032
2.95	132.97	.6154	559.02	2165.5	2724.5	1.6658	6.9975
3.00	133.55	.6058	561.47	2163.8	2725.3	1.6718	6.9919
3.05	134.11	.5965	563.88	2162.2	2726.1	1.6777	6.9864
3.10	134.67	.5875	566.27	2160.6	2726.8	1.6835	6.9810
3.15	135.22	.5787	568.62	2158.9	2727.6	1.6893	6.9756
3.20	135.76	.5702	570.95	2157.3	2728.3	1.6950	6.9704
3.25	136.30	.5620	573.25	2155.8	2729.0	1.7006	6.9652
3.30	136.83	.5540	575.52	2154.2	2729.7	1.7061	6.9601
3.35	137.35	.5463	577.76	2152.6	2730.4	1.7116	6.9551
3.40	137.87	.5387	579.97	2151.1	2731.1	1.7169	6.9502
3.45	138.38	.5314	582.16	2149.6	2731.8	1.7222	6.9453
3.50	138.88	.5243	584.33	2148.1	2732.4	1.7275	6.9405
3.55	139.38	.5173	586.47	2146.6	2733.1	1.7327	6.9358
3.60	139.87	.5106	588.59	2145.1	2733.7	1.7378	6.9311
3.65	140.36	.5040	590.68	2143.7	2734.4	1.7428	6.9265
3.70	140.84	.4926	592.75	2142.2	2735.0	1.7478	6.9219
3.75	141.32	.4914	594.81	2140.8	2735.6	1.7528	6.9175
3.80	141.79	.4853	596.83	2139.4	2736.2	1.7577	6.9130
3.85	142.26	.4794	598.84	2138.0	2736.8	1.7625	6.9087
3.90	142.72	.4736	600.83	2136.6	2737.4	1.7673	6.9044
3.95	143.18	.4680	602.80	2135.2	2738.0	1.7720	6.9001
4.00	143.63	.4625	604.64	2133.8	2738.6	1.7766	6.8959
4.1	144.53	.4519	608.59	2131.1	2739.7	1.7858	6.8877
4.2	145.40	.4417	612.36	2128.4	2740.8	1.7948	6.8796
4.3	146.26	.4321	616.06	2125.8	2741.9	1.8036	6.8717
4.4	147.10	.4228	619.68	2123.2	2742.9	1.8122	6.8641
4.5	147.93	.4140	623.25	2120.7	2743.9	1.8207	6.8565

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m^3/kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
4.6	148.74	.4055	626.76	2118.2	2744.9	1.8290	6.8492
4.7	149.54	.3974	630.21	2115.7	2745.9	1.8371	6.8420
4.8	150.32	.3896	633.60	2113.2	2746.8	1.8451	6.8349
4.9	151.10	.3821	636.94	2110.8	2747.8	1.8530	6.8280
5.0	151.86	.3749	640.23	2108.5	2748.7	1.8607	6.8213
5.1	152.60	.3689	643.46	2106.1	2749.6	1.8682	6.8146
5.2	153.34	.3613	646.65	2104.8	2750.5	1.8757	6.8081
5.3	154.06	.3549	649.78	2101.5	2751.3	1.8830	6.8017
5.4	154.78	.3487	652.87	2099.3	2752.1	1.8902	6.7955
5.5	155.48	.3427	655.93	2097.0	2753.0	1.8973	6.7893
5.6	156.17	.3369	658.93	2094.8	2753.8	1.9043	6.7832
5.7	156.86	.3313	661.90	2092.7	2754.5	1.9112	6.7773
5.8	157.53	.3259	664.83	2090.5	2755.3	1.9180	6.7714
5.9	158.20	.3207	667.71	2088.4	2756.1	1.9246	6.7657
6.0	158.85	.3157	670.56	2086.3	2756.8	1.9312	6.7600
6.1	159.50	.3108	673.37	2084.2	2757.5	1.9377	6.7545
6.2	160.14	.3060	676.15	2082.1	2758.3	1.9441	6.7490
6.3	160.77	.3015	678.89	2080.1	2759.0	1.9504	6.7436
6.4	161.39	.2970	681.60	2078.0	2759.6	1.9566	6.7383
6.5	162.01	.2927	684.28	2076.0	2760.3	1.9627	6.7331
6.6	162.61	.2885	686.93	2074.0	2761.0	1.9688	6.7279
6.7	163.21	.2844	689.55	2072.1	2761.6	1.9748	6.7228
6.8	163.81	.2805	692.13	2070.1	2762.3	1.9807	6.7178
6.9	164.39	.2766	694.69	2068.2	2762.9	1.9865	6.7129
7.0	164.97	.2729	697.22	2066.3	2763.5	1.9922	6.7080
7.1	165.55	.2692	699.72	2064.4	2764.1	1.9979	6.7032
7.2	166.11	.2657	702.20	2062.5	2764.7	2.0035	6.6985
7.3	166.67	.2622	704.64	2060.7	2765.3	2.0091	6.6938
7.4	167.23	.2589	707.07	2058.8	2765.9	2.0146	6.6892
7.5	167.78	.2556	709.47	2057.0	2766.4	2.0200	6.6847

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m ³ /kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
7.6	168.32	.2524	711.85	2055.2	2767.0	2.0253	6.6802
7.7	168.86	.2493	714.20	2053.4	2767.5	2.0306	6.6758
7.8	169.39	.2463	716.52	2051.6	2768.1	2.0359	6.6714
7.9	169.91	.2433	718.83	2049.8	2768.6	2.0411	6.6670
8.0	170.43	.2404	721.11	2048.0	2769.1	2.0462	6.6628
8.1	170.95	.2376	723.38	2046.3	2769.7	2.0513	6.6585
8.2	171.46	.2349	725.62	2044.5	2770.2	2.0563	6.6544
8.3	171.97	.2322	727.83	2042.8	2770.7	2.0612	6.6502
8.4	172.47	.2295	730.04	2041.1	2771.2	2.0662	6.6462
8.5	172.96	.2270	732.22	2039.4	2771.6	2.0710	6.6421
8.6	173.46	.2245	734.38	2037.7	2772.1	2.0758	6.6381
8.7	173.94	.2220	736.52	2036.1	2772.6	2.0806	6.6342
8.8	174.43	.2196	738.64	2034.4	2773.0	2.0853	6.6303
8.9	174.90	.2173	740.75	2032.8	2773.6	2.0900	6.6264
9.0	175.38	.2150	742.82	2031.1	2773.9	2.0946	6.6226
9.1	175.85	.2127	744.91	2029.5	2774.4	2.0992	6.6188
9.2	176.31	.2105	746.96	2027.9	2774.8	2.1038	6.6151
9.3	176.78	.2084	749.00	2026.3	2775.3	2.1083	6.6114
9.4	177.24	.2063	751.02	2024.7	2775.7	2.1127	6.6077
9.5	177.69	.2042	753.02	2023.1	2776.1	2.1172	6.6041
9.6	178.14	.2022	755.01	2021.5	2776.5	2.1215	6.6005
9.7	178.59	.2002	756.98	2019.9	2776.9	2.1259	6.5969
9.8	179.03	.1982	758.94	2018.4	2777.3	2.1302	6.5934
9.9	179.47	.1963	760.88	2016.8	2777.7	2.1345	6.5899
10.0	179.91	.19444	762.81	2015.3	2778.1	2.1387	6.5865
10.2	180.71	.19080	766.63	2012.2	2778.9	2.1471	6.5796
10.4	181.62	.18730	770.38	2009.2	2779.6	2.1553	6.5729
10.6	182.46	.18392	774.08	2006.2	2780.3	2.1634	6.5664
10.8	183.28	.18067	777.74	2003.3	2781.0	2.1713	6.5599
11.0	184.09	.17753	781.34	2000.4	2781.7	2.1792	6.5536

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m ³ /kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
11.2	184.89	.17449	784.89	1997.5	2782.4	2.1869	6.5473
11.4	185.68	.17156	788.40	1994.6	2783.0	2.1945	6.5412
11.6	186.46	.16873	791.86	1991.8	2783.6	2.2020	6.5351
11.8	187.23	.16599	795.28	1989.0	2784.2	2.2093	6.5292
12.0	187.99	.16333	798.65	1986.2	2784.2	2.2166	6.5233
12.2	188.74	.16077	801.98	1983.4	2785.4	2.2238	6.5176
12.4	189.48	.15828	805.28	1980.7	2786.0	2.2309	6.5119
12.6	190.20	.15586	808.53	1978.0	2786.5	2.2378	6.5063
12.8	190.93	.15353	811.75	1975.3	2787.1	2.2447	6.5008
13.0	191.64	.15125	814.93	1972.7	2787.6	2.2515	6.4953
13.2	192.34	.14905	818.07	1970.0	2788.1	2.2582	6.4900
13.4	193.04	.14691	821.18	1967.4	2788.6	2.2648	6.4847
13.6	193.72	.14483	824.85	1964.8	2789.1	2.2714	6.4795
13.8	194.40	.14281	827.29	1962.3	2789.6	2.2778	6.4743
14.0	195.07	.14084	830.30	1959.7	2790.0	2.2842	6.4693
14.2	195.74	.13893	833.28	1957.2	2790.5	2.2905	6.4643
14.4	196.39	.13707	836.28	1954.7	2790.9	2.2968	6.4593
14.6	197.04	.13525	839.14	1952.2	2791.3	2.3029	6.4544
14.8	197.69	.13349	842.03	1949.7	2791.7	2.3090	6.4496
15.0	198.32	.13177	844.89	1947.3	2792.2	2.3150	6.4448
15.5	199.88	.12766	851.92	1941.2	2793.1	2.3298	6.4331
16.0	201.41	.12380	858.79	1935.2	2794.0	2.3442	6.4218
16.5	202.89	.12006	865.50	1929.4	2794.9	2.3582	6.4108
17.0	204.34	.11673	872.06	1923.6	2795.7	2.3718	6.4000
17.5	205.76	.11349	878.50	1917.9	2796.4	2.3851	6.3896
18.0	207.15	.11042	884.79	1912.4	2797.1	2.3981	6.3794
18.5	208.51	.10751	890.96	1906.8	2797.8	2.4109	6.3694
19.0	209.84	.10475	897.02	1901.4	2798.4	2.4233	6.3597
19.5	211.14	.10212	902.96	1896.0	2799.0	2.4354	6.3502
20.0	212.42	.09963	908.79	1890.7	2799.5	2.4474	6.3409

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m^3/kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
20.5	213.67	.09725	914.52	1885.5	2800.0	2.4590	6.3318
21.0	214.90	.09498	920.15	1880.3	2800.5	2.4704	6.3229
21.5	216.10	.09281	925.69	1875.2	2800.9	2.4817	6.3141
22.0	217.29	.09073	931.14	1870.2	2801.3	2.4927	6.3056
22.5	218.45	.08875	936.49	1865.2	2801.7	2.5035	6.2972
23.0	219.60	.08685	941.77	1860.2	2802.0	2.5141	6.2890
23.5	220.72	.08502	946.97	1855.4	2802.3	2.5245	6.2809
24.0	221.83	.08327	952.09	1850.5	2802.6	2.5347	6.2729
24.5	222.92	.08159	957.13	1845.7	2802.9	2.5448	6.2651
25.0	223.99	.07998	962.11	1841.0	2803.1	2.5547	6.2575
26	226.09	.07692	971.85	1831.6	2803.6	2.5740	6.2425
27	228.12	.07409	981.34	1822.4	2803.8	2.5927	6.2280
28	230.10	.07145	990.59	1813.4	2804.0	2.6109	6.2139
29	232.02	.06899	999.61	1804.5	2804.1	2.6285	6.2002
30	233.90	.06668	1008.42	1795.7	2804.2	2.6457	6.1869
31	235.72	.06452	1017.04	1787.1	2804.1	2.6624	6.1740
32	237.51	.06249	1025.47	1778.6	2804.1	2.6787	6.1614
33	239.54	.06057	1033.72	1770.2	2803.9	2.6946	6.1491
34	240.94	.05877	1041.82	1761.9	2803.7	2.7101	6.1370
35	242.60	.05707	1049.75	1753.7	2803.4	2.7253	6.1253
36	244.23	.05545	1057.53	1745.6	2803.1	2.7401	6.1138
37	245.82	.05392	1065.17	1737.6	2802.8	2.7546	6.1025
38	247.38	.05247	1072.68	1729.7	2802.4	2.7688	6.0915
39	248.91	.05109	1080.05	1721.8	2801.9	2.7828	6.0807
40	250.40	.04978	1087.31	1714.1	2801.4	2.7964	6.0701
42	253.31	.04733	1101.48	1698.8	2800.3	2.8229	6.0495
44	256.12	.04510	1115.22	1683.8	2799.0	2.8485	6.0296
46	258.83	.04306	1128.58	1669.0	2797.6	2.8732	6.0103
48	261.45	.04118	1141.57	1654.5	2796.0	2.8970	5.9916
50	263.99	.03944	1154.23	1640.1	2794.3	2.9202	5.9734

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m ³ /kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
52	266.45	.03783	1166.58	1626.0	2792.6	2.9427	5.9557
54	268.84	.03634	1178.66	1612.0	2790.7	2.9645	5.9385
56	271.17	.03495	1190.46	1598.2	2788.6	2.9858	5.9217
58	273.43	.03365	1202.02	1584.5	2786.5	3.0065	5.9052
60	275.64	.03244	1213.35	1571.0	2784.3	3.0267	5.8892
62	277.78	.03130	1224.46	1557.6	2782.1	3.0464	5.8734
64	279.88	.03023	1235.37	1544.3	2779.7	3.0657	5.8580
66	281.93	.02922	1246.09	1531.2	2777.2	3.0845	5.8428
68	283.93	.02827	1256.63	1518.1	2774.7	3.1030	5.8279
70	285.88	.02737	1267.00	1505.1	2772.1	3.1211	5.8133
72	287.79	.02652	1277.21	1492.2	2769.4	3.1389	5.7989
74	289.67	.02571	1287.28	1479.4	2766.7	3.1563	5.7847
76	291.50	.02494	1297.19	1466.6	2763.8	3.1734	5.7707
78	293.30	.02421	1306.98	1453.9	2760.9	3.1902	5.7569
80	295.06	.02352	1316.64	1441.3	2758.0	3.2068	5.7432
82	296.79	.02286	1326.18	1428.7	2754.9	3.2230	5.7297
84	298.49	.02222	1335.61	1416.2	2751.8	3.2391	5.7164
86	300.16	.02162	1344.93	1403.7	2748.7	3.2549	5.7032
88	301.80	.02104	1354.14	1391.3	2745.4	3.2704	5.6902
90	303.40	.02048	1363.26	1378.9	2742.1	3.2858	5.6772
92	304.99	.01995	1372.29	1366.5	2738.8	3.3009	5.6644
94	306.54	.01944	1381.23	1354.1	2735.4	3.3159	5.6517
96	308.07	.01895	1390.08	1341.8	2731.9	3.3306	5.6391
98	309.58	.01848	1398.86	1329.5	2728.3	3.3452	5.6265
100	311.06	.018026	1407.56	1317.1	2724.7	3.3596	5.6141
102	312.46	.017588	1416.19	1304.8	2721.0	3.3739	5.6017
104	313.96	.017167	1424.76	1292.5	2717.3	3.3880	5.5894
106	315.38	.016760	1433.26	1280.2	2713.5	3.4020	5.5771
108	316.77	.016367	1441.70	1267.9	2709.6	3.4158	5.5649
110	318.15	.015987	1450.10	1255.5	2705.6	3.4295	5.5527

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m^3/kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	'Sat. steam Φ_s
112	319.50	.015620	1458.4	1243.2	2701.6	3.4430	5.5406
114	320.84	.015264	1466.7	1230.8	2697.5	3.4565	5.5285
116	322.16	.014920	1474.9	1218.4	2693.4	3.4698	5.5165
118	323.46	.014587	1483.1	1206.0	2689.2	3.4831	5.5044
120	324.75	.014263	1491.3	1193.6	2684.9	3.4962	5.4924
122	326.02	.013949	1499.4	1181.1	2680.5	3.5092	5.4804
124	327.27	.013644	1507.5	1168.6	2676.0	3.5222	5.4684
126	328.51	.013348	1515.5	1156.0	2671.5	3.5351	5.4564
128	329.73	.013060	1523.5	1143.4	2666.9	3.5478	5.4444
130	330.93	.012780	1531.5	1130.7	2662.2	3.5606	5.4323
132	332.12	.012508	1539.5	1118.0	2657.5	3.5732	5.4203
134	333.30	.012242	1547.4	1105.2	2652.6	3.5858	5.4082
136	334.46	.011983	1555.3	1092.4	2647.7	3.5983	5.3961
138	335.61	.011731	1563.2	1079.5	2642.7	3.6108	5.3839
140	336.75	.011485	1571.1	1066.5	2637.6	3.6232	5.3717
142	337.87	.011245	1579.0	1053.4	2632.4	3.6356	5.3595
144	338.98	.011010	1586.9	1040.2	2627.1	3.6479	5.3471
146	340.08	.010781	1594.8	1026.9	2621.7	3.6603	5.3348
148	341.16	.010557	1602.6	1013.5	2616.1	3.6726	5.3223
150	342.24	.010337	1610.5	1000.0	2610.5	3.6848	5.3098
152	343.30	.010123	1618.4	986.4	2604.8	3.6971	5.2971
154	344.35	.009912	1626.3	972.7	2598.9	3.7093	5.2844
156	345.39	.009706	1634.2	958.8	2593.0	3.7216	5.2716
158	346.42	.009504	1642.1	944.8	2586.9	3.7338	5.2586
160	347.44	.009306	1650.1	930.6	2580.6	3.7461	5.2455
162	348.44	.009111	1658.0	916.2	2574.3	3.7584	5.2323
164	349.44	.008920	1666.0	901.7	2567.7	3.7707	5.2189
166	350.43	.008732	1674.1	886.9	2561.0	3.7830	5.2053
168	351.40	.008547	1682.2	872.0	2554.2	3.7954	5.1916
170	352.37	.008364	1690.3	856.9	2547.2	3.8079	5.1777

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m^3/kg v_s	Enthalpy kJ/kg			Entropy kJ/kg K	
			Sat. water h	Evap. L	Sat. steam H	Sat. water Φ_w	Sat. steam Φ_s
172	353.33	.008185	1698.5	841.5	2540.0	3.8204	5.1635
174	354.28	.008008	1706.8	825.8	2532.6	3.8330	5.1491
176	355.21	.007833	1715.1	809.9	2525.0	3.8457	5.1345
178	356.14	.007660	1723.5	793.6	2517.2	3.8586	5.1196
180	357.06	.007489	1732.0	777.1	2509.1	3.8715	5.1044
182	357.98	.007320	1740.7	760.1	2500.8	3.8846	5.0889
184	358.88	.007153	1749.4	742.8	2492.2	3.8978	5.0730
186	359.77	.006987	1758.3	725.1	2483.3	3.9112	5.0567
188	360.66	.006822	1767.3	706.1	2474.1	3.9249	5.0400
190	361.54	.006657	1776.5	688.0	2464.5	3.9388	5.0228
192	362.41	.006493	1785.9	668.7	2454.6	3.9530	5.0050
194	363.27	.006329	1795.6	648.6	2444.2	3.9676	4.9866
196	364.12	.006165	1805.5	627.8	2433.3	3.9825	4.9676
198	364.97	.006000	1815.7	606.1	2421.8	3.9979	4.9477
200	365.81	.005834	1826.3	583.4	2409.7	4.0139	4.9269
202	366.64	.005665	1837.3	559.6	2396.9	4.0305	4.9050
204	367.46	.005495	1848.9	534.3	2383.2	4.0479	4.8819
206	368.28	.005320	1861.1	507.3	2368.4	4.0663	4.8571
208	369.09	.005139	1874.2	478.1	2352.3	4.0861	4.8304
210	369.89	.004952	1888.4	446.2	2334.6	4.1075	4.8013
212	370.69	.004754	1904.2	410.6	2314.8	4.1313	4.7690
214	371.47	.004538	1922.2	369.6	2291.8	4.1585	4.7318
216	372.25	.004298	1943.8	320.3	2264.1	4.1914	4.6876
218	373.03	.004007	1972.6	255.1	2227.6	4.2351	4.6298
220	373.80	.003568	2025.0	140.6	2165.6	4.3154	4.5327
220.9	374.14	.003155	2099.3	0	2099.3	4.4298	4.4298

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