ELEMENTS OF HEAT ENGINES

VOLUME I

(IN SI UNITS)

By

Late Prof. R. C. Patel
B.E. (Mech.) Hons., B.E. (Elect.) Hons.,
A.R.C.S.T. (Mech.). Hons., (Glasgow),
M. Sc. (Bermingham),
F.I.E. (India), F.I.A.E. (India), M.I.M (India), M.I.S.T.E.,
Former Vice Chancellor
M.S. University of Baroda, BARODA.

AND

Late C. J. Karamchandani
D.M.E.E. (Karachi),
Senior Lecturer (Retd.),
Mechanical Engineering Department,
Polytechnic,
M.S. University of Baroda, BARODA.

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Preface
to the
Eighteenth Edition

It is a matter of great satisfaction to the author of this book that he is required to bring out the Eighteenth 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 throughout the country.

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, Baroda, and Shri Surendra J. Shah of Parijat Printery, Ahmedabad, for getting the book printed so nicely and in time.

Baroda
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C. J. Karamchandani

Preface
to the
Fifteenth Edition

The overwhelming support by the readers to the Fourteenth Edition in the SI Units has encouraged the author to bring out the Fifteenth Edition in such a short span of time. The author feels delighted in presenting this edition to the readers.

Suggestions received from the readers are duly acknowledged and considered. It is hoped that the publication will continue to be valuable text as before.

The author expresses his profound sorrow at the sudden demise of his co-author, Prof. R. C. Patel, former Vice-Chancellor of M.S. University of Baroda, BARODA.

The author sincerely thanks Prof. A. A. Patel, Prof. of Mechanical Engineering, Faculty of Technology and Engineering, Shri G. C. Karamchandani, B. Arch., A.I.I.A. and Shri R. R. Patel, D.M.E., P.D.D. (R & A.C.), for checking the calculations and reading the proofs.

The author also thanks Shri J. C. Shah and brothers of M/s. Acharya Book Depot, Baroda and Shri Surendra J. Shah of M/s. Parijat Printery, Ahmedabad, for getting the volume printed so nicely and in time.

Baroda
10th June, 1990

C. J. Karamchandani

Preface to the Second Edition

It is, indeed, a matter of gratification to the authors of this book that they are required to bring out the second 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 the Engineering Students throughout the country.

This edition has been rewritten in M.K.S. units to comply with the decision of Government of India for change over in the system of units. The major part of this book is overhauled to enhance the usefulness of the book by making it more lucid in expression, more simplified in its subject matter and by suitable modification in the general arrangement and addition of subject matter at some places.

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 and useful 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.) Hons., M.M. Patel, 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 due to Shri A.K. Karkhanis for redrawing the sketches. Our sincere thanks are also due to the publishers.

Baroda
4th October, 1965

R. C. Patel
C. J. Karamchandani

Preface to the First Edition

This volume is intended for the use of students preparing for the second year diploma examinations in Mechanical and Electrical Engineering of the Universities and State Technical Education 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 problems 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 volumes 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 examinations in Electrical Engineering and Volumes II and III for the final year diploma examinations in Mechanical Engineering.*

Utmost care has been taken in making 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 Sarvashri M.C. Karamchandani, B.E. (Mech.) Hons., D.S. Tamhne, D.M.E. and A.V. Dongre, D.M.E. for solving the problems. Our thanks are due to Shri Parmar for preparing sketches.

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

Baroda, R. C. Patel

18th December, 1962 C. J. Karamchandani

CONTENTS

1

BASIC CONCEPTS OF THERMODYNAMICS

1.1 Introduction

Thermodynamics is a branch of science that deals with energy in all its forms and the laws governing the transformation of energy from one form to another. Since, there are many forms of energy such as mechanical, thermal or heat, chemical, electrical, etc., this science covers a very broad field of application and is a base to many branches of natural science and engineering and technology. That part of the subject which applies to engineering, is generally referred to as *Engineering Thermodynamics or Applied Thermodynamics*.

Thermodynamics deals with the behaviour of gases and vapours (working substance) when subjected to variations of temperature and pressure and the relationship between heat energy and mechanical energy, commonly referred to as work. When a substance undergoes a change from one condition to another in a process, energy transformation may occur. Common processes are those of heating or cooling and expansion or compression in the cylinder or passages with or without production or supply of mechanical work. Chemical reaction and/or change of phase may occur in some processes involving liberation of heat.

Engineers concerned with power generating machinery should have a working knowledge of all matters dealing with the conversion of heat energy into work or power. The laws based on experimental results obtained from the study of gases and vapours are useful in the design of boilers, steam engines, steam turbines, internal combustion engines, gas turbines, refrigerating machines and air compressors. In the present days of industrialisation, demand for energy is increasing rapidly. It is, therefore, necessary to design and operate thermal plants and machines at their highest level of performance for efficient utilization of fuels and natural resources available.

1.2 Working Substance or Medium

Any thermodynamic process or change involves the use of *working substance or thermodynamic medium,* which has the ability to receive, store and give out (or reject) energy as required by the particular process. The medium may be in any one of the four physical states or phases, namely—solid, liquid, vapour and gaseous. The vapours and gaseous substances are sometimes termed together as gases.

The power generating machines (plants) which operate according to thermodynamic laws, require the use of some working medium. For example, steam power generating plants use water vapour and refrigerator or ice plant uses ammonia or freon as the working substance. Steam is a very suitable medium for steam power plants because it readily absorbs heat, flows easily, exerts pressure on the piston or blade while it moves and allows considerable expansion of its volume. Ammonia or Freon is a suitable medium for an ice plant because it boils at a temperature below 0°C and at a moderate pressure and absorbs heat from water, making it to freeze into ice. A mixture of air and fuel forms a working medium in the initial processes and product of combustion in remaining processes of internal combustion engines and gas turbines.

1.3 System

A thermodynamic *system* is a specific portion of matter, with definite boundary, on which our attention is focussed. The system *boundary* may be real or imaginary, fixed or deformable. Everything outside the system which has direct bearing on its behaviour is known as *surroundings* as shown in fig. 1-1. Thermodynamic problems are concerned with the transfer of energy and mass across the boundry of a system. There are three types of systems :

- Isolated systems,
- Closed system, and
- Open system.

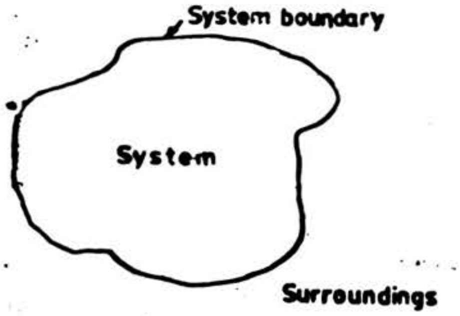
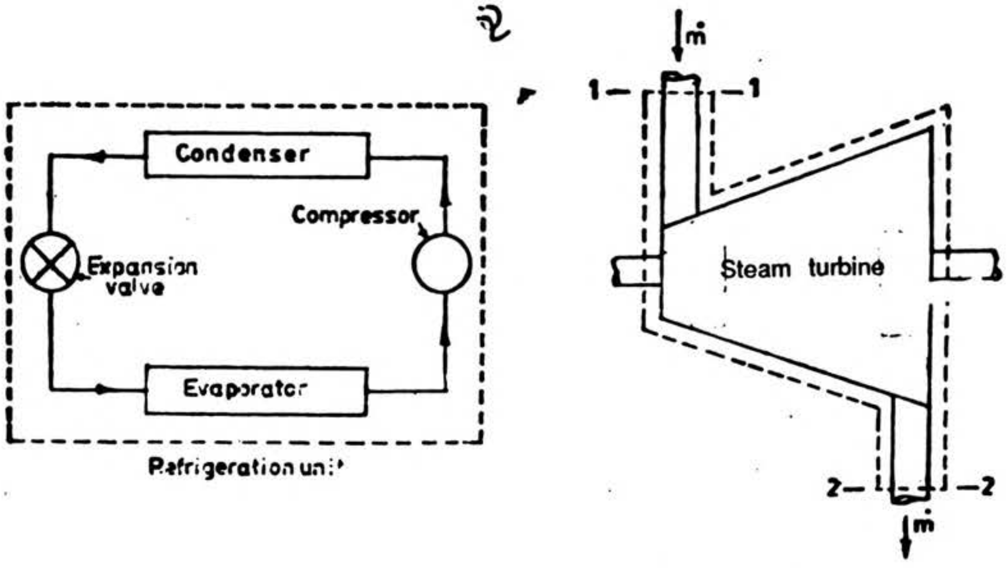


Fig. 1-1. System and surroundings.

An *isolated system* cannot exchange both energy and mass with its surroundings. System and surroundings together form a Universe. Universe is, thus, considered as an isolated system.



(a) Closed system **(b) Open system**

Fig. 1-2. Closed and open systems.

Across the boundry of *closed system*, transfer of energy (work and or heat) takes place but transfer of mass does not take place. Compression of a gas in a piston cyclinder assembly, refrigerator, heating of water in a closed vessel, are examples of closed system. Figure 1-2(a) shows a refrigeration unit as an example of closed system.

In an *open system*, mass and energy both may be transferred between the system and surroundings. Gas turbine, axial flow and centrifugal air compressors, boiler delivering steam, etc. are some examples of open system. Steam turbine is also an example of open system which is shown in fig. 1-2(b).

1.4 State and Properties of a Substance

The exact condition of a substance is called its *state* and variables which determine the state are spoken of as its *properties* or *parameters*. The principal properties are pressure, volume, temperature, internal energy, enthalpy and entropy. The fundamental properties are pressure, volume and temperature. The other properties listed above are dependent in some manner on one or more of these fundamental properties. Any two of the properties such as pressure, volume, temperature, internal energy, enthalpy, entropy, much be known to determine the *'thermodynamic state'* of the working medium. Thus, if the thermodynamic state is fixed, all these properties are fixed with it.

1.5 Process and Cycle

A change of state occurs when one or more of the properties of a system changes. When a system undergoes changes on its state, it is said to have undergone a *process*. Thus, a process is the path joining succession of states passed through by a system. Process is named according to its specification, i.e., constant pressure process, constant volume process, etc.

A *cycle* is a process or a combination of processes so conducted that the initial and final states of the system are the same. A thermodynamic cycle is also known as a *cyclic operation of processes.*

1.6 Basic Properties

Pressure, volume and temperature are named as basic properties or parameters as they may be determined by direct observations or simple measurements.

1.6.1 Pressure : Pressure is a force applied over a unit area.

In SI (international system) units, the unit for pressure is the force of one newton (N) acting on a square metre area, which is called the *pascal* (Pa)., i.e. 1 Pa = 1 N/m^2.

As this unit is rather small for practical use, engineers measure pressure in kilopascals and megapascals. We note that,

1 kilopascal = 1,000 pascals = 1 kPa

1 megapascal = 10,00,000 pascals = 1 MPa

Other units for pressure, not included in the SI units, commonly used are :

bar and standard atmosphere (atm).

1 bar = 10^5 Pa = 100 kPa (kilopascal) = 0·1 MPa (megapascal), and

1 standard atmosphere (atm) = 1,01,325 Pa, which is somewhat larger than a bar.

The atmosphere surrounding the earth, exerts a pressure on its surface equivalent to the weight of air over a unit area of the earth's surface. The pressure of the atmosphere is recorded by a barometer. The height of the mercury column also varies with the altitude above or below sea level. At sea level, 760 mm of mercury column at 0°C is taken as the standard barometric pressure. This is known as a *physical atmosphere* or *barometric atmosphere (atm).*

To express this physical atmosphere in kilogram per square centimetre (technical or metric atmosphere), we must first determine the weight of a column of mercury with a cross-section of one square centimetre and a height of 760 mm. The density of mercury is 13, 595 kilograms per cubic metre at 0°C; hence the standard height of 760 mm at 0°C of a mercury column corresponds to a pressure of

$$\rho g h = 13{,}595 \times 9{\cdot}80665 \times \frac{760}{1{,}000} = 1{,}01{,}325 \text{ N/m}^2 \text{ or } = 1{,}01{,}325 \text{ Pa}$$

Thus, one physical or standard atmosphere (1 atm) = 760 mm of Hg

= 1,01,325 Pa = 101·325 kPa = 0·101325 MPa = 1·01325 bar.

From this, it can be shown that 1 mm of Hg = 0·13332 kPa, = 0.0013332 bar, and

750 mm of Hg = 100 kPa = 1 bar.

Vacuum may be defined as the absence of pressure. Quite often the pressure of a confined fluid is less than that of the surrounding atmosphere. The confined fluid is then said to be under a partial vacuum. In such a case, the instrument used to measure the pressure is vacuum gauge. It records the difference between the surrounding atmospheric pressure and that within the vessel in millimetres of mercury. A vessel having no pressure within it is said to have a perfect vacuum.

In practice, pressure of a fluid is measured by means of instruments known as gauges. Gauges which measure pressure greater than atmospheric pressure are called *pressure gauges*. A pressure gauge attached to a vessel indicates the difference between the pressure inside the vessel and the pressure of the atmosphere.

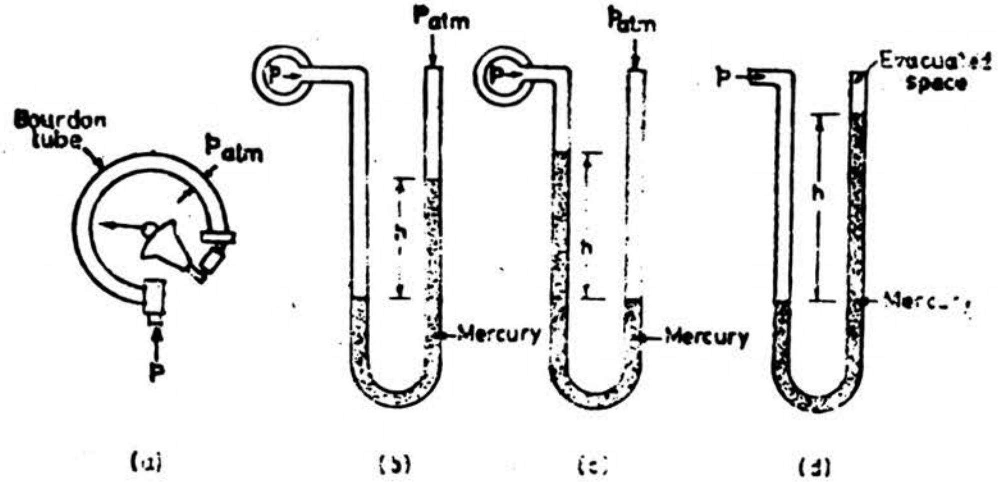


Fig. 1-3. Measurements of pressure.

Figure 1-3(a) illustrates a *Bourdon pressure gauge*. This pressure gauge is commonly used to measure the pressure of steam in a boiler. The reading of the pressure gauge is called *gauge pressure*. The absolute pressure of steam within the boiler is, therefore, equal to the observed gauge pressure plus the atmospheric pressure.

Gauges measuring pressure less than atmospheric pressure are called *vacuum gauges*. The principle of working of pressure gauges is the same. The pressure in a steam condenser is usually less than atmospheric pressure. A vacuum gauge attached to the condenser, measures the difference between the atmospheric pressure and the pressure inside the condenser. This difference is called the *vacuum* and is expressed in millimetres of mercury. The absolute pressure within the condenser is, therefore, equal to atmospheric pressure minus the vacuum.

A manometer is employed to measure slight variation in the pressure above or below atmospheric pressure. Manometers read the pressure in millimetres of mercury or water, or in terms of height of the column of any fluid that may be used in the manometer tube.

Figure 1-3(b) shows the principle of working of a U-tube manometer, measuring pressure above atmospheric pressure. One limb of the U-tube is open to atmosphere and the other end is connected to a vessel having pressure more than that of atmosphere. Suppose the manometer indicates a reading of *h* millimeters as difference between the mercury levels in the two limbs of the U-tube. Then, the pressure of the gas *p* is being balanced by the atmospheric pressure, P_{atm} plus the pressure due to the column of mercury, h. Hence,

$$p = p_{atm} + h \qquad \text{...(1.1)}$$

i.e absolute pressure of gas (p) in the vessel or pipe line is the sum of atmospheric pressure and the manometer reading.

Figure 1-3(c) shows a U-tube manometer measuring pressure less than atmospheric pressure (vacuum). One limb of the U-tube is open to atmosphere and the other end is connected to a vessel having pressure less than that of the atmosphere. Here,

$$p = p_{atm} - h \qquad \text{...(1.2)}$$

i.e. absolute pressure in the vessel is the difference of atmospheric pressure and the manometer reading.

Figure 1-3(d) shows a closed U-tube indicating absolute pressure. If p is the atmospheric pressure, then this gauge is known as a *barometer.*

Practically in all thermodynamic analysis and calculations, absolute pressures are used.

Problem - 1 : *Convert the following readings of pressure to kilopascal (kPa) : (i) 8 bar, (ii) 10 atm, (iii) 2 MPa, (iv) 1,12,000 N/m^2, (v) 200 kN/m^2, (vi) 0.05 N/mm^2, and (vii) 1,500 mm of Hg.*

(i) 8 bar = 8×10^5 Pa = 8,00,000 Pa or **800 kPa.**

(ii) 10 atm = $10 \times$ 1,01,325 Pa = 10,13,250 Pa or **1,013·25 kPa.**

(iii) 2MPa = 2×10^6 Pa = 20,00,000 Pa or **2,000 kPa.**

(iv) 1,12,000 N/m^2 = 1,12,000 Pa = **112 kPa.**

(v) 200 kN/m^2 = 2,00,000 N/m^2 = 2,00,000 Pa = **200 kPa.**

(vi) 0·05 N/mm^2 = $0{\cdot}05 \times 10^6$ N/m^2 = 50,000 N/m^2 = 50,000 Pa = **50 kPa.**

(vii) 1500 mm of Hg = 1,500 × 0·13332 kPa = **200 kPa.**

Problem - 2 : *A pressure gauge reads 2.3 MPa, and the barometer reads 98 kPa. Calculate the absolute pressure in MPa.*

Absolute pressure = atmospheric pressure + gauge pressure

$$\Rightarrow 0{\cdot}098 \text{ MPa} + 2{\cdot}3 \text{ MPa} = \mathbf{2{\cdot}398 \text{ MPa}}$$

Note : In this text, the pressure stated in Pascals will indicate the absolute pressure. Gauge pressure will be indicated as gauge pressure or pressure gauge reading.

Problem - 3 : *A vacuum gauge on the condenser reads 620 mm of mercury and at the same time barometer reads 740 mm of mercury. What is the absolute pressure in the condenser in kPa ?*

Using eqn. (1.2), Absolute pressure in the condenser

= atmospheric pressure in mm of Hg - vacuum gauge reading in mm of Hg

= 740 − 620 = 120 mm of Hg.

$\therefore$ Absolute pressure in the condenser = 120 × 0·13332 ≈ **16 kPa.**

Problem - 4 : *The pressure of the gas supplied to an engine is measured as 76·2 mm of water gauge. If the barometer reads 730 mm of mercury, what is the absolute pressure of gas in mm of mercury and in bar ?*

Very low pressures are generally measured by the height of water column; this is convenient because the height of water column is 13·6 times greater than that mercury column for any given pressure, i.e. 1 mm of Hg = 13·6 mm of H_2O.

Gauge pressure of gas = 76·2 mm of water

$$\text{or } = \frac{76{\cdot}2}{13{\cdot}6} = 5{\cdot}6 \text{ mm of mercury.}$$

Absolute presure of gas = gauge pressure + barometric pressure

= 5·6 mm of Hg + 730 mm of Hg = **735·6 mm Hg**

Absolute pressure of gas in bar = 735·6 × 0·13332 = 98·07 kPa = **0·9807 bar.**

1.6.2 Volume : Volume of gas is defined as the space which the gas occupies and it is measured in cubic metres. A widely used unit of volume is the litre which is 1,000th part of a cubic metre,

i.e. 1 litre = 10^{-3} m^3

The *specific volume* of a substance is its volume per unit mass. It is generally in cubic metres per kilogram. One kilogram of air at 0°C and under a pressure of 101·325 kPa (760 mm of Hg) has a volume of 0·7734 cubic metre. Therefore, the specific volume of air under these conditions is 0·7734 m^3/kg.

The *density* of a substance is its mass per unit volume is generally stated in kilogram per cubic metre.

From definitions of density and specific volume, it is clear that density of any substance is the reciprocal of its specific volume and vice-versa. Thus, if specific volume of air is 0·7734 m^3/kg, its density is 1/0·7734 = 1·293 kg/m^3 under the same conditions of pressure and temperature. Density is denoted by the symbol ρ (roh).

Problem - 5 : *Three kilograms of dry hydrogen occupy 33 cubic metres. What is the specific weight (density) of hydrogen ?*

$$\text{Specific volume} = \frac{33}{3} = 11\,m^3/\text{kg. Density} = \frac{1}{\text{Specific volume}} = \frac{1}{11} = \mathbf{0{\cdot}09\ kg/m^3}.$$

1.6.3 Temperature : The temperature of a substance may be defined as measure of hotness or degree of coldness of a body. A body is said to be hot when it has a relatively high temperature; it is said to be cold when it has a relatively low temperature. It is known fact that heat energy has a tendency to flow from a hot body to one that is cooler. Thus, temperature determines which way (direction) the heat flow will take place. It is not a measure of quantity of energy possessed by the body but it indicates level of internal energy possessed by the body as explained in art 1·7·5.

Ordinarily temperatures are *measured* by instruments called *thermometers.* Very high temperatures are measured by instruments known as *pyrometers.* Small and precise changes of temperature can be measured by resistant thermometers and thermocouples.

Engineers measure temperature by Centigrade thermometer. In the Centigrade thermometer the freezing point of water is marked as zero and the boiling point as 100. The freezing point and the boiling point of water at atmospheric pressure (101·325 kPa or 760 mm of Hg) are called fixed points of the thermometer; other temperatures are measured by reference to these points. The Centrigrade scale of temperature is constructed by dividing the thermometer stem between the fixed points in 100 equal parts or degrees.

Centigrade scale may be extended above the boiling point and below the freezing point. Temperatures lower than zero are denoted by negative sign. Thus, – 15°C means 15 Centigrade degrees below the freezing point.

The centigrade scale in SI units is now named after Mr. Celsius, the Swedish astronomer, who invented this scale in 1953. Therefore, this scale is called *Celsius scale.*

The *absolute scale of temperature* is based on so called absolute zero of the temperature.

Absolute zero is the temperature at which all vibratory, translatory and rotational motions of molecules of a substance is supposed to have ceased, i.e., internal energy becomes zero. A gas on cooling will contract in volume as the temperature falls. Charles found that with perfect gases, the decrease in volume per degree centigrade decrease in temperature is 1/273rd of its initial volume at 0°C, pressure remaining constant. Thus, the volume of gas will be zero at temperature – 273°C. This temperature 273°C below 0°C (or – 273°C) is called the *absolute zero of temperature.*

The absolute temperature is the temperature measured above the point of absolute zero. Absolute temperatures are expressed by the capital letter T. Absolute temperature is expressed in degrees of the Kelvin scale or K.

Temperature K = Temperature°C + 273

$$\text{i.e. } T = t + 273 \text{ K} \qquad \text{...(1.3)}$$

$$\text{or } t = T - 273^{\circ}\text{C}$$

The exact relationship between Kelvin scale and Celsius scale is

Temperature K = Temperature°C + 273.15

Problem - 6 : *(i) The temperture of steam in a boiler on absolute scale (Kelvin scale) is 343. What will be its temperature on Celsius scale ?*

(ii) The temperature of steam in a boiler is 200°C. What will be its temperature on the absolute or Kelvin scale ?

(i) Temperature K = Temperature°C + 273

i.e. 343 = Temperature°C + 273

∴ Temperature°C = 343 – 273 = **70°C**

(ii) Temperature K = Temperature°C + 273 = 200 + 273 = **473 K**

1.7 Energy

Energy may be defined as the capacity, a body possesses for doing work. Energy appears in many forms, i.e. mechanical (potential and kinetic), internal, electrical, chemical, nuclear, work, heat, etc. Energy cannot be seen and does not have mass.

All forms of energy mainly can be classified as *stored energy* and *energy in transition.*

Energy residing in a system is called ***stored energy.*** Energy is found stored in many forms, such as mechanical, internal (or thermal), electrical, chemical and nuclear energy. *Mechanical energy* is concerned with the system as a whole. It may be in the form of kinetic and or potential energy of a body as a whole. *Internal energy* is associated with the molecules of a system (kinetic and potential energy of molecules of a system). The system has the ability to receive, to store and to emit energy and undergo a *thermodynamic process* that results as the change in properties of the system. The change in stored energy of a system must derive completely from energy which has transferred across its boundary (in the form of work and heat), since energy can be neither created nor destroyed.

Energy in transition is the energy that is transferred to or from a system. Transient form of energy has significance only while energy is being transferred. Work and heat are forms of energy in transition which can only cross the boundaries of a system. Work and heat are not stored by system hence there is nothing like work of a body or heat of a body. *Work and heat are not properties or state parameter.* They are not state functions. They are only path functions. They exist only in transition or transfer.

1.7.1 Work : Work is a transient form of energy. When a force acts upon a body, causing the body to move and to overcome continually a resistance, work is said to be done. This work is equal to the force multiplied by the distance through which it acts. The time element is of no consideration. The unit of work done is *newton-metre (N–m)*, which is the product of a unit force (one newton) and a unit distance (one metre) moved in the direction the force. This unit of work is also known as *joule (J),*

that is 1 joule = 1 newton-metre (N·m)

1 kilojoule (kJ) = 1,000 joules

When work is done by a system, it is arbitrarily taken to be positive. and when work is done on a system it is taken to be negative. Work of compression or expansion of a gas, shaft work and flow work are different forms of work.

We have seen that unit of work done is independent of time. Suppose one machine does the same amount of work as another, but in one-fourth the time, it is evident that the first machine can do four times the work of the second in the same time and hence to compare or to rate the machine, time must be considered. The rate at which work is done by or upon the system is known as *power.*

The unit for power is a rate of one joule per second (J/s) which is a watt (W),
i.e. 1 watt = 1 joule per second.

A common unit for power is kilowatt (kW).

1 kW = 1,000 W or 1,000 J/s or 1,000 N.m/s.

kWh (kilowatt hour) is work done by a source of 1 kW in one hour. Electrical energy consumption is measured in terms of kWh.

1 kWh = 1,000 x 3,600 joules or 3,600 kJ

The unit for time in SI units is the second (s). Other units for time often used are the hour (hr.) and the day, although neither one is a basic SI unit.

1.7.2 Heat : Heat is that form of energy which is transferred from one body to another on account of temperature difference. Heat energy, therefore, is not a stored form of energy but occurs only in transition and when present, energy of some other form is being transferred from one body to another. Heat energy may be transferred in three ways, by conduction, convection and radiation. In all the three modes of heat transfer there must be temperature difference and the direction of heat transfer is in the direction of decreasing temperature. It may be noted that conduction and convection require some definite medium while radiation can occur in vacuum also.

If one end of metal rod, insulated against heat transfer to the air, is heated to a higher temperature, heat energy will flow from the hot end to the cold end and the temperature of the cold end will rise. The heat is transferred from one cross-section to the next by molecular activity. The molecular activity at the hot end causes the molecules in an adjacent section to increase their molecular activity. Thus, energy is carried along the entire length of the bar, and the temperature of cold end rises. If the two ends of the bar are at same temperature, the molecules of the two ends have the same level of molecular activity and there is no tendency of energy to be transferred from one cross-section to the next, i.e., there is no heat transfer. This mode of heat transfer is called *conduction* and is predominent in solid form of matter. Over and above molecular communication, free electrons present in the material also help to conduct heat from hot end to cold end. For details refer chapter-I of vol. III.

In *convection,* heat is transmitted from one place to another by fluid currents. As gases or liquids are heated by conduction through the wall of containing vessel, they tend to expand and rise, and their place is taken by the upper colder layers which being heavier than the heated liquid or gas tend to flow downwards. In this way convection currents are set up and the whole volume of gas or liquid is gradually heated to uniform temperature. Water in a steam boiler is heated uniformly throughout by the convective currents set up by upward flow of the lighter heated water in contact with the heated surface and by the downward flow of heavier colder water from above.

Radiant heat is in the form of temperature excited electromagnetic waves which passes from one body to another without much raising the temperature of the medium through which it passes. Substances which are transparent to light usually allow radiant heat to pass freely through them even when they are poor conductors of heat. This is one reason why radiant heat is so important in boilers, since it will pass through a thin film of gas clinging to the tube or plate which offers considerable resistance to the passage of heat by conduction.

In a steam boiler, the heat from the surface of the burning coal in a furnace is transferred to the crown and sides of furnace by radiation; it passes through the furnace plate by conduction, and the water is heated by convection.

As discussed earlier, heat like work is a form of energy which is transferred to and from a system. Therefore, the unit for heat, and for any other form of energy, is the same as the unit for work in the International System of Units viz. joule (J).

1.7.3 Potential Energy : Potential energy is the energy of a body due to its position or elevation relative to some datum plane. Therefore, the term potential energy is exclusively used for gravitational energy. The potential energy of a substance is equal to the work that can be done by allowing a substance to fall from the given position to the surface of the earth or other datum. The maximum possible work done is the product of gravitational force (or weight) of a falling body and the distance through which this body falls. The unit of potential energy is newton-metre (N.m) and the symbol used to represent this energy is *P.E.* If a body of m kg mass is allowed to fall from an elevation L_2 to an elevation L_1, the change in potential energy,

$$\begin{aligned}\Delta PE &= PE_2 - PE_1\\ &= mg\ (L_2 - L_1) \qquad \text{...(1.4)}\end{aligned}$$

1.7.4 Kinetic Energy : The kinetic energy (KE) possessed by a body is due to its motion. The unit of (KE) is also N.m. Water held behind a dam has potential energy and if released, the water in its flow has its potential energy changed into kinetic energy. The energy, therefore, could be utilized or work could be done equal to the energy possessed by the water, by permitting it to flow through blades of a water wheel or a water turbine.

A body of mass *m* kg moving with a velocity *V* possesses a certain amount of kinetic energy (*KE*) with reference to earth. Kinetic energy,

$$KE = m \cdot \frac{V^2}{2} \quad ...(1.5)$$

Potential energy and kinetic energy are extensive properties, since they depend upon the mass of the system.

1.7.5 Internal Energy : Matter is composed of an aggregation or collection of molecules which are moving continuously. The movement of molecules is more pronounced in gases than in liquids. When the gas is stored in closed vessel it is stagnant, that is, not moving as a whole. However, it posseses a considerable amount of *internal kinetic energy* due to the motion of its molecules within the limits of its containing vessel. In addition to the internal kinetic energy, substance have *internal potential energy* due to the relative position of their molecules with respect to one another.

A change in mechanical potential energy of a body occurs when the elevation of the body relative to the earth as a datum is changed. The force acting in this case is the force of gravity. Now, since there is a cohesive force between the molecules of a substance there will be a change in the internal potential energy, if anything happens which increases or decreases the average distance between the molecules. It may be noted that distance between the molecules can be more widely increased or decreased in a gas than in a liquid, say water. During the formation of steam, the molecules of water are separated against their attractive force which require a large amount of energy. The energy used to overcome the attractive force is stored in the steam as internal potential energy.

The *internal energy, u* of a substance may be defined as the algebraic sum of internal kinetic energy and the internal potential energy of its molecules. Internal energy is an extensive property, since it depends upon the mass of the system.

Thus, the internal energy of a substance is the energy stored within the substance and it is due to the motion and configuration (relative position) of its molecules. If the temperature of a gas is increased, the molecular activity increases. Therefore, the internal energy is a function of temperature and its value can be increased or decreased by adding or subtracting heat to or from the substance. The absolute value of internal energy of a body cannot be determined, but, it is possible to estimate changes of internal energy from changes of state of the body. The internal energy value of a substance in any event is measured above an arbitrary datum condition, at which the substance or medium is said to posses zero internal energy. Value of internal energy is expressed in joules. The change of internal energy when the substance passes from state 1 to state 2 can be expressed in a general way,

$$\Delta u = u_2 - u_1 \quad ...(1.6)$$

1.7.6 Enthalpy : The other terms in use for enthalpy are *total heat* and *heat content*. It is said that the term enthalpy is more convenient than the term heat content and total heat. Enthalpy is an energy term and is defined as follows :

$$H = u + pv$$

where *u* is the internal energy, *p* is the absolute pressure, and *v* is the volume.

Since, we cannot measure the absolute quantity of internal energy, it is impossible to measure the absolute enthalpy of a substance. What we wish to know in actual practice is the change in internal energy and the change in enthalpy. Wherever the change of internal energy can be calculated and whenever the product pv is known for any two states of any fluid, the change in enthalpy, ΔH can be evaluated as

$$\Delta H = H_2 - H_1 = (u_2 - u_1) + p_2v_2 - p_1v_1 \qquad ...(1.7)$$

To find the simpler expression for change in enthalpy of a perfect gas, we know that

$$u_2 - u_1 = mk_v\,(T_2 - T_1),\; p_2v_2 = mRT_2 \text{ and } p_1v_1 = mRT_1.$$

Substituting these values in eqn. (1.7), we have

$$\Delta H = H_2 - H_1 = mk_v\,(T_2 - T_1) + mRT_2 - mRT_1$$

$$= mk_v\,(T_2 - T_1) + mR\,(T_2 - T_1)$$

But from eqn. (1.27), $k_p - k_v = R$ or $k_v = k_p - R$

$$\therefore \text{ Change of enthalpy, } \Delta H = H_2 - H_1 = mk_p\,(T_2 - T_1) \qquad ...\ (1.8)$$

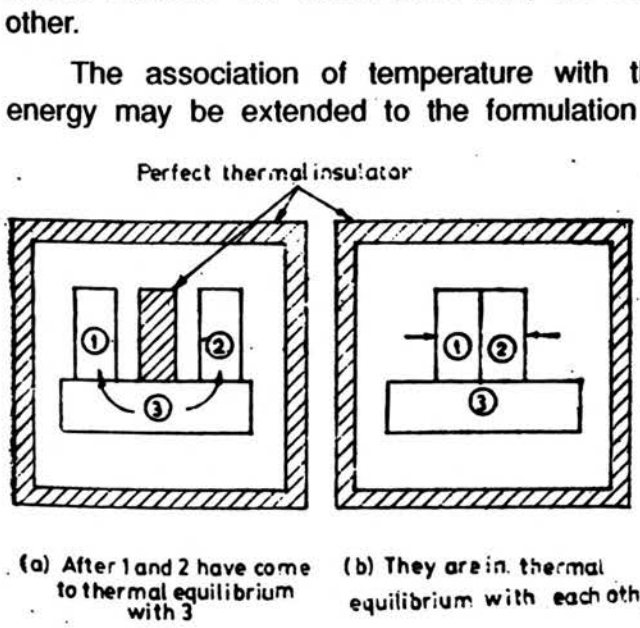
1.8 Laws of Thermodynamics

Laws of thermodynamics lie at the centre of the classical thermodynamics. These laws (zeroth, first, second and third) are natural or fundamental laws formulated on the basis of natural observations. They are not derived from any mathematics.

1.8.1 Zeroth Law of Thermodynamics : If we take two bodies, one hotter than the other and bring them into contact, we shall find after some time that both the bodies are equally hot. When this state is attained we say that two bodies are in thermal equilibrium. The bodies in *thermal equilibrium* will have some property in common and this property is called *temperature.* In other words, bodies in thermal equilibrium will be at the same temperature. As stated above, temperature indicates the intensity of molecular activities (kinetic fraction of internal energy). The higher the temperature, the greater is the level of activities. When two bodies whose temperature are different, are brought in contact, an increase will take place in the molecular activities of one body (which is at lower temperature) and a decrease in the other body (which is at higher temperature). These changes in molecular activities will continue till the time the temperature of the bodies become the same. After this, the bodies will be in thermal equilibrium with each other.

The association of temperature with the quantitative evaluation of internal kinetic energy may be extended to the formulation of the *Zeroth law of thermodynamics.*

As a basis for describing when bodies will be in thermal equilibrium, we consider systems 1 and 2 insulated from each other but in good thermal contact with a third or a common system. As indicated in fig. 1-4(a), systems 1 and 2 will come to thermal equilibrium with system 3. If the insulator is now removed and systems 1 and 2 are brought into contact as shown in fig. 1-4(b), we find that there is no further change. This means that the combined system has come to thermal equilibrium; consequently systems 1 and 2 are in thermal equilibrium. This experimental observations lead to the conclusion that *two systems in thermal equi-*



(a) After 1 and 2 have come to thermal equilibrium with 3

(b) They are in thermal equilibrium with each other

Fig. 1-4. Schematic illustration of the Zeroth law.

librium with a third are in equilibrium with each other. This statements is known as the Zeroth law.

The above law suggests the existance of a system property. The fact that two systems can be said to be in thermal equilibrium with each other before they are placed in contact implies that there must be some characteristic of the systems indicative of this. We recognize this characteristic as temperature, which can be defined as that property of a system which determines whether or not it is in thermal equilibrium with other systems. In the logical development of the science of thermodynamics, the Zeroth law preceeds the First and Second laws but chronologically, the First and Second laws were established prior to the presentation of this statement and hence it is designated as the Zeroth law.

If the properties of a system are uniform throughout, then, so long as the external conditions are unaltered, the system is said to be in *thermodynamic equilibrium.* A system in thermodynamic equilibrium satisfies all the three mechanical, thermal and chemical equilibriums, i.e., a system is in thermodynamic equilibrium if it in mechanical, thermal and chemical equilibrium.

Mechanical equilibrium exists if there are no net forces between the system and surroundings and if there are none within the system.

Thermal equilibrium exists if the temperature of the system is uniform throughout and equal to that of the surroundings with which it is in thermal contact.

Chemical equilibrium exists if the composition of each compound present is constant and if diffusion and solution do not occur.

1.8.2 First Law of Thermodynamics : This law is the same as the law of the conservation of energy, which states that *energy can neither be created nor destroyed if mass is conserved. The sum total of the energy in the universe is constant. Energy, however, can be converted from one form into another form.* This is the thermodynamic aspect of first law. A machine cannot create work from nothing nor it can deliver more work than it receives. In a steam generating plant, the chemical energy of the fuel is converted into heat energy in the boiler, which in turn is converted into mechanical work in the steam engine or steam turbine. If the turbine is coupled to an electric dynamo, the mechanical energy is converted into electrical energy. If the dynamo is supplying the electrical energy produced by it to drive an electric motor, the electrical energy is again converted into mechanical energy.

It was established by Joule that heat and mechanical energies are mutually convertible. Heat requires for its production, a definite number of units work for each unit of heat produced. Similarly, heat produces by its disappearance, a definite number of units of work for each unit of heat converted. This is known as the first law of thermodynamics.

Joules experiments showed that for a closed system during a cyclic process, the sum of the work transferred is equal to the sum of the heat transferred. Mathematically, it is written as

$$\oint dW = \oint dQ \qquad \text{... (1.9)}$$

The circle on integral sign represents a cyclic process.

This means during any cycle, a closed system executes, the cyclic integral of work is equal to cyclic integral of heat. Work and heat both being measured in joules (J) or kilojoules (kJ).

For a non-cyclic process, a closed system (in absence of *KE* and *PE*) executes, the work transferred and heat transferred may not be equal, and the difference between the two is accounted for by a change in internal energy, *u* of the system. This can be stated mathematically as

$$Q - W = \Delta u$$

$$\text{or } Q = \Delta u + W$$

$$\text{or } Q_{1-2} = u_2 - u_1 + W_{1-2} \qquad \text{... (1.10)}$$

In differential form this can be written as

$$dQ = du + dW \qquad \text{... (1.11)}$$

Equation (1.11) is the mathematical form of the first law of thermodynamics. It may be noted that heat and work are not state functions, but internal energy is a state function or property.

1.8.3 Second Law of Thermodynamics : This law states that *"It is impossible for self-acting machine, unaided by any external agency, to convey heat from a body at low temperature to a body at higher temperature", i.e., heat cannot itself pass from a cold body to a warmer body.* This statement, known as second law of thermodynamics, was given by Clausius.

The second law states that heat will not pass automatically from a colder to a hotter body. Heat can be forced to pass to a higher temperature, as in the action of a refrigerating machine, but only by applying an 'external agency' to drive the machine, i.e., by doing work on the system.

Number of other statements have been put forth to formulate the second law of thermodynamics; all these can be shown logically equivalent and any one statement can be derived from the other. The second law like the first law is the statement of the net result of common experience. The following are the statements of the second law formulated by different authors at different times. However, they all are more or less indicative of one and the same meaning.

.. No apparatus can operate in such a way that its only effect (in system and surroundings) is to convert the heat taken in completely into work.

.. It is impossible to convert the heat taken in completely into work in a cyclic process.

.. For heat to be converted into work there must be, in addition to the source of heat, a cooling agent possessing a lower temperature, i.e., there must be a drop in temperature.

.. The heat of the cooler body in the given system cannot serve as a source of work.

.. It is impossible to construct an engine that operating in a cycle, will produce an effect other than the extraction of heat from a single reservoir and the performance of an equivalent amount of work, – Kelvin-Planck.

.. It is impossible to take heat from a reservoir and convert it into work by cyclic process, without transferring it to a colder reservoir.

.. All natural or spontaneous processes take place in one direction only and cannot be reversed.

.. All natural or spontaneous processes can be made to do work but the maximum work can only result from a reversible process.

.. Any process which consists solely in the transfer of heat from one at a lower temperature to another at a higher temperature is impossible.

.. Heat passes spontaneously only from a system at higher temperature to another at lower temperature. The passage of heat in the reverse direction requires the expenditure of work from an external source.

.. It is impossible to construct a device that, operating in a cycle, will produce no effect other than the transfer of heat from one body to a second body at higher temperature-Clausius.

.. All natural or spontaneous processes are in transition from a less probable to a more probable state.

A little thought would reveal that all these statements essentially convey the same meaning but each one has definite aspect of thermodynamics to cover.

First statement does not imply that heat cannot be converted into work, but does mean that changes other than those resulting directly the conversion of heat into work must occur in either the system or surroundings. Consider the case of an ideal gas in a vertical cylinder-piston assembly, expanding reversibly at constant temperature. Work is produced in the surroundings (consider the gas as the system) equal to the integral of the pressure times the change in volume. Since the gas is ideal, $\Delta u = 0$. Then, according to the First law the heat absorbed by the gas from the surroundings is equal to the work produced in the surroundings because of the rerversible expansion of the gas. At first this might seem to be a contradiction of first statement since in the surroundings the only result has been the complete conversion of heat into work. However, the second law statement requires that there also be no change in the system, a requirement which has not been met in this example. Since the pressure of the gas has decreased, this process could not be continued indefinitely. The pressure of the gas would soon reach that of the surroundings, and further expansion would be impossible. Therefore, a method of continuously producing work from heat by this method fails. If the original state of the system were restored in order to meet the requirements of first statement, it would be necessary to take energy from the surroundings in the form of work in order to compress the gas back to its original pressure. At the same time energy as heat would be transferred to the surroundings in order to maintain constant temperature. This reverse process would require just the amount of work gained from the expansion; hence the net work produced would be zero.

In the second statement viz it is impossible to convert the heat taken in completely into work in a cyclic process which is an alternative way of expressing the Second law, the term cyclic requires that the system be restored periodically to its original state. In the previous example, the expansion and compression back to the original state constituted a complete cycle. If the process is repeated, it becomes a cyclic process. The restriction of cyclic process in statement 2 amounts to the same limitation as that introduced by the words *only effect* in first statement. The second law does not prohibit the production of work from heat, but it does place a limitation upon the efficiency of any cyclic process. Thus, partial conversion of heat into work forms the basis for mechanisms, called heat engines, without which the conversion is impossible.

A process of a system in which reverse or back movement of system restoring the system as well as surroundings along the same path is possible, is called a *reversible process*. In a reversible process all means of energy dissipation (due to friction, viscosity, electric resistance, magnetic hysterisis, plastic deformation, etc.) are absent. Reversible processes are hypothetical and are useful for comparison purposes. In a reversible process, a system must be in thermodynamic equilibrium at all states.

Any process that is not reversible is known as an *irreversible process.* All naturally occurring (i.e. spontaneous) processes are irreversible i.e. heat transfer, frictional losses and mass transfer.

Entropy is an important derived thermodynamic property introduced by Clausius. The property entropy is consequence of the Second law of thermodynamics. Entropy is defined as the ratio of heat supplied or rejected during a reversible process and the absolute temperature at which the heat is supplied or rejected.

It is the change of entropy for the system which is of interest, since absolute values cannot be determined. The change of entropy for the same thermodynamic states would have the same value regardless of the process between the states.

Entropy is discussed in detail in chapter-4.

Problem - 7 : *500 kg of coal per hour are burned in the furnace of a boiler. Supposing each kg of coal produces 33·5 MJ of heat, find how much work in MJ could be done per hour, if 20 per cent of heat developed were converted into work.*

1 kg of coal gives out 33·5 MJ of heat.

∴ 500 kg of coal per hour will give out 33·5 x 500 = 16,750 MJ per hour.

Heat converted into work is 20 per cent of the heat produced.

∴ Work done = 16,750 x 0·2 = **3,350 MJ per hour.**

Problem - 8 : *(i) One kg of steam at 700 kPa contains 2,760 kJ of heat. Change this total heat to equal amount of mechanical energy in newton-metres (N.m).*

(i) Find the heat equivalent of work done in kJ, when a weight of 500 kg is raised through a height of 60 metres.

(i) Now, 1 kJ = 1kN.m

∴ 2,760 kJ = 2,760 kN.m = 2,760 × 1,000 = **27,60,000 N.m**

(ii) Work done = force × distance through which it acts

= mass x gravitational acceleration × distance through which it acts

= (500 x 9·81) x 60 = 2,94,300 N.m, or 2,94,300 J, or **294·3 kJ.**

Problem - 9 : *Find the mechanical energy in N.m given out when one kilowatt is maintained for one hour. Find the heat equivalent of this energy in kJ.*

One kW = 1,000 N.m per second

One kW-hour = 1,000 x 3,600 = **36,00,000 N.m-hr.**

or = 36,00,000 J per hour or **3,600 kJ per hour.**

1.9 Specific Heat

The specific heat of a substance may be defined as the amount of heat that must be supplied to the substance to raise the temperature of unit mass of the substance through one degree.

In differential terms, specific heat *k* of a substance is defined as

$$k = \frac{dq}{dT} \qquad \text{...(1.12)}$$

where dq is heat supplied per unit mass and dT is change in its temperature.

When a body is heated, the heat energy is used to speed up the molecules and also to provide the work necessary to expand the body. In a solid or a liquid, the amount of expansion is very small and the work of expansion is similarly small. When heating a gas, however, the expansion may be considerable, and value of specific heat will depend on nature of heating process. Gas has two important values of specific heat, namely, *specific heat* at *constant volume* and *specific heat at constant pressure.*

Consider 1 kg of gas being heated in a closed vessel (fig. 1-5), so that no expansion of the gas is allowed. The number of heat units required to raise the temperature of 1 kg of gas through 1°C under these conditions is called the specific heat at constant volume (k_v). In differential form it is expressed as

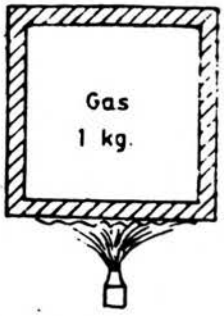


Fig. 1 5 Heating at constant volume.

$$k_v = \left(\frac{\partial q}{\partial T}\right)_v \qquad ...(1.13)$$

In this case there is no work of expansion, because the gas cannot move and all the heat supplied is used to increase the internal energy, i.e., kinetic energy and potential energy of molecules. The value of k_v for air is 0·7165 kJ/kg K.

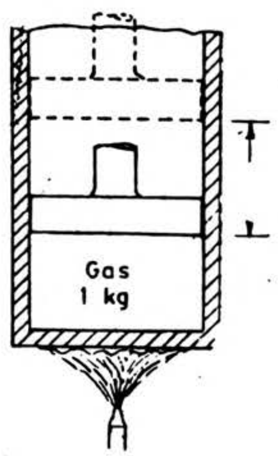


Fig. 1-6. Heating at constant pressure.

Consider now 1 kg of gas being heated in a cylinder (fig.1-6) fitted with a movable piston which exerts a constant pressure on the gas. When the gas is heated it will expand and move the piston through some distance *h*. In this case, in addition to the heat required to increase the kinetic energy of the molecules, further heat must be added to perform the work of moving the piston through the distance *h*.

The number of heat units required to raise the temperature of 1 kg of gas through 1°C under these conditions, is called the *specific heat at constant pressure* (k_p). In differential form it is expressed as

$$k_p = \left(\frac{\partial q}{\partial T}\right)_p \qquad ...(1.14)$$

The value of the specific heat of gas at constant pressure will always be greater than that at constant volume by the amount of expansive work done. The value of k_p for air is 1·0035 kJ/kg K.

The *ratio of two specific heats,* k_p and k_v of any given gas is assumed to be constant. This is expressed by the symbol γ (gamma). Thus,

$$\gamma = \frac{k_p}{k_v} \qquad ...(1.15)$$

For atmospheric air the value of k_p and k_v are 1·0035 kJ/kg K and 0·7165 kJ/kg K respectively.

Then, for air

$$\gamma = \frac{k_p}{k_v} = \frac{1{\cdot}0035}{0{\cdot}7165} = 1{\cdot}4$$

It should be noted that no such term like γ is in use for vapours since there is no definite value for the specific heat of vapour.

Values of k_p and k_v at temperatures between 15°C and 30°C for some common gases are given in table 1-1.

Table 1 - 1 Values of specific heats for gasses

Gas	Chemical formula	k_p kJ/kg K	k_v kJ/kg K	$\frac{k_p}{k_v}$	$R = k_p - k_v$ kJ/kg K
Air	—	1·0035	0·7165	1·40	0·2870
Carbon dioxide	CO_2	0·8419	0·653	1·289	0·1889
Oxygen	O_2	0·9216	0·6618	1·393	0·2598
Nitrogen	N_2	1·0416	0·7448	1·40	0·2968
Ammonia	NH_3	2·177	1·69	1·288	0·4870
Carbon monoxide	CO	1·0413	0·7445	1·40	0·2968
Hydrogen	H_2	14·209	10·085	1·409	4·1240
Steam	H_2O	1·8723	1·4108	1·33	0·4615
Argon	Ar	0·52	0·312	1·667	0·2080
Gasoline (Octane)	C_8H_{18}	1·711	1·6385	1·044	0·0725
Helium	He	5·1926	3·1156	1·667	2·0770
Methane	CH_4	2·2537	1·7354	1·299	0·5183

Problem - 10 : *A copper vessel weighs 2·5 kg. Calculate the quantity of heat required to raise its temperature from 20°C to 70°C. Take specific heat of copper as 0·377 kJ/kg K.*

Quantity of heat required,

Q = mass of copper, m × specific heat of copper, k × rise in temperature, $(t_2 - t_1)$

$= 2{\cdot}5 \times 0{\cdot}377 \times 50 =$ **47·125 kJ**

Problem - 11 : *The specific heat of a gas at constant pressure (k_p) is 0·984 kJ/kg K and the value of γ for the gas is 1·351. If 0·5 kg of this gas is heated at constant volume from 25°C to 375°C, how many kJ need be supplied.*

$$\gamma = \frac{k_p}{k_v},\ k_v = \frac{k_p}{\gamma} = \frac{0{\cdot}984}{1{\cdot}351} = 0{\cdot}728 \text{ kJ/kg K}$$

Heat supplied $= m \times k_v \times (t_2 - t_1) = 0{\cdot}5 \times 0{\cdot}728 \times (375 - 25) =$ **127·4 kJ**.

1.10 Perfect Gas Laws

There are three primary phases of a substance, namely, solid, liquid and gas.

Perfect gas (or ideal gas) is a gas which remains in gaseous phase at all pressures and temperatures. There are no perfect gases in nature, but from thermodynamic point of view, to somplify the formulae and calculations, some substances like oxygen, hydrogen, nitrogen, air, and carbon dioxide in gaseous phase are regarded as perfect gases.

A *gas* is a phase of a substance whose evaporation from its liquid phase is complete.

The behaviour of a perfect gas, undergoing changes of temperature and pressure is governed by certain laws. There are four such fundamental laws governing the behaviour of gases under the specific conditions, and these laws have been established from experimental results. These are not applicable to vapours for the reason stated above. Usual practice is to represent initial state of gas by the suffix 1 and final state by suffix 2.

A *vapour* is a secondary phase between liquid and gas. Thus, vapour is a phase of a substance whose evaporation from its liquid phase is partial. A vapour, therefore, contains particles of the liquid in suspension. Steam, carbon dioxide, sulphur dioxide and ammonia are some vapours used in engineering practice. Since a vapour is liable to further evaporation or condensation on changes of either its temperature or pressure, the laws of gases do not apply to vapours. A vapour becomes dry when it is completely evaporated. Any further heating of a dry vapour is termed superheating and such a state of the vapour is termed superheated state. The behaviour of a superheated vapour approaches that of a perfect gas.

1.10.1 Boyle's Law : Boyle experimentally established that *when a perfect gas is heated at constant temperature, the volume of a given mass of gas is inversely proportional to the absolute pressure.* Hence, volume increases as the absolute pressure decreases and vice versa.

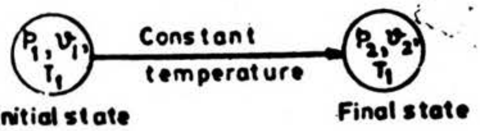


Fig. 1.7

Thus, $v \propto \frac{1}{p}$ $\therefore$ $v = \frac{C}{p}$

i.e. $pv = C$, where C = a constant

Applying the above law for initial and final states,

$$p_1v_1 = p_2v_2 = pv = \text{a constant} \qquad \text{...(1.16)}$$

where p and v are the corresponding pressure and volume for any instantaneous condition of a process in which temperature is constant. In other words, when we have constant temperature *(isothermal)* process, the product of pressure and volume at any state of the process is always equal to the same number as at any other state of the same process.

The expansion of a given mass of gas under the condition $pv = c$ is termed as *hyperbolic expansion.*

Problem - 12 : *4m^3 of air is compressed isothermally to a final volume of 0·6 m^3. If the initial pressure is 5 MPa, what will be the final pressure of the air ?*

Given, p_1 = 5 MPa = 5,000 kPa, v_1 = 4 m^3 and v_2 = 0·6 m^3.

Since for isothermal process, $p_1v_1 = p_2v_2$ i.e. 5,000 x 4 = p_2 x 0·6

$\therefore$ Final pressure, p_2 = 33,333 kPa or p_2 = **33·333 MPa.**

1.10.2 Charle's Law : It states that, *if a perfect gas is heated at constant pressure, its volume varies directly with the absolute temperature.* In other words, the Charle's law states that coefficient of expansion is constant at constant pressure, i.e. its change of volume per degree of temperature change is constant. This change in volume, he found to be same for all perfect gases. For each degree centigrade change in temperature, the change in volume he found was 1/273th of the initial volume of gas at 0°C when pressure is constant. If for example, the temperature of gas is changed from 0°C to 10°C, its volume will increase by 10/273th of the original volume no matter what the pressure is, so long the pressure is held constant.

Thus, when the pressure is constant, the volume varies directly as the absolute temperature, i.e. $v \propto T$ $\therefore$ $\frac{v}{T}$ = a constant

Applying the above law for initial and final states,

$$\frac{v_1}{T_1} = \frac{v_2}{T_2} \quad or \quad \frac{v_1}{v_2} = \frac{T_1}{T_2} \qquad \text{...(1.17)}$$

1.10.3 Gay-Lussac Law : This law expresses relationship between temperature and pressure of a perfect gas when the volume is kept constant. It is stated as

With volume remaining constant, the absolute pressure varies directly as the absolute temperature,

i.e. when v is constant,

$$p \propto T \quad \therefore \quad \frac{p}{T} = \text{a constant}$$

Applying the above law for initial and final states,

$$\frac{p_1}{T_1} = \frac{p_2}{T_2} \quad or \quad \frac{p_1}{p_2} = \frac{T_1}{T_2} \qquad ...(1.18)$$

Problem - 13 : *A quantity of gas at 0°C occupies 0·6 m³. What would be its volume at 400 °C if the pressure is same at both temperatures ?*

Given, $v_1 = 0{\cdot}6 \text{ m}^3$, $t_1 = 0°\text{C}$ and $t_2 = 400°\text{C}$.

$$\text{Increase in volume} = \frac{1}{273} \times 0{\cdot}6 \text{ m}^3 \text{ per degree centigrade rise}$$

$$\therefore \text{ Total increase in volume} = \frac{1}{273} \times 0{\cdot}6 \times 400 = 0{\cdot}88 \text{ m}^3$$

$$\therefore \text{ Final volume, } v_2 = 0{\cdot}6 + 0{\cdot}88 = \mathbf{1{\cdot}48 \text{ m}^3}$$

Alternatively, using eqn. (1.17), $\frac{v_1}{v_2} = \frac{T_1}{T_2}$,

$$\text{i.e. } v_2 = v_1 \times \frac{T_2}{T_1} = 0{\cdot}6 \times \frac{(400 + 273)}{(0 + 273)} = \mathbf{1{\cdot}48 \text{ m}^3 \text{(same as before)}}$$

1.11 Characteristic Equation for a Perfect Gas

An ideal or perfect gas can change its thermodynamic state without one of its characteristics p, v and T remaining constant. In other words, during the change of state, the pressure, volume and temperature may all be varied. This necessitates the derivation of another equation that will apply to such changes.

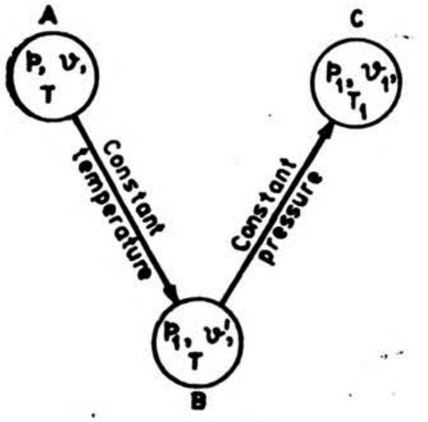


Fig. 1-8

This equation is known as the *characteristic gas equation* or *equation of state for a perfect gas.* It is derived by combining laws of Boyle and Charles.

Let a kilogram of gas be initially at pressure p, volume v and absolute temperature T at state represented by A (fig. 1-8). This gas is to be ultimately in the state represented by C where its characteristics have all changed becoming p_1, v_1 and T_1. Let us assume that it does not change directly from A to C, but passes through intermediate state B. From A to B the temperature is to remain constant so that the temperature at B is T. The pressure at B will be taken as p_1, the final pressure. Hence, the volume to permit this change is v'. The gas changes from state B to state C with its pressure held constant, so that the pressure at C is p_1. The volume and temperature will change to their final values v_1 and T_1.

Considering the change of state from A to B at constant temperature which follows the Boyle's law and using eqn. (1.16), $pv = p_1v'$

$$\therefore\ v' = \frac{pv}{p_1} \qquad \text{.. (i)}$$

The change from B to C being at constant pressure, then according to Charle's law and using eqn. (1.17),

$$\frac{v'}{v_1} = \frac{T}{T_1} \quad i.e.\ v' = \frac{Tv_1}{T_1} \qquad \text{...(ii)}$$

Comparing the values of v' from eqn. (i) and eqn. (ii),

$$\frac{pv}{p_1} = \frac{Tv_1}{T_1} \ or\ \frac{pv}{T} = \frac{p_1v_1}{T_1}$$

Thus, if mass remains unchanged, $\frac{pv}{T}$ = constant.

This constant is designated by R and is called the *Characteristics Gas Constant,* and its value is different for different gases.

Then, $pv = RT$ ...(1.19)

The above equation is called the Characteristic Gas Law or Equation of state of a perfect gas.

For any mass m kg of gas, v denotes valume of m kg of gas instead of specific volume and R is replaced by mR. Hence, the most general equation for all masses, volumes, pressures and temperatures of a perfect gas is

$pv = mRT$ ...(1.20)

It can be shown that the value of R represents work to be supplied when one kg of gas is raised in temperature by one degree centigrade at constant pressure. Thus, unit of R is N.m or joules per kg per degree centigrade temperature.

The value of R for air is 287 J/kg K or 0·287 kJ/kg K.

Problem - 14 : *The pressure of a gas supplied to an engine is measured as 100 mm of water gauge when barometer reads 756 mm of mercury. Determine the volume of 1·5 kg of this gas if its temperature is 85°C. The gas constant of the gas is 0·686 kJ/kg K.*

Pressure of gas, $p = 756 + (100/13{\cdot}6) = 763{\cdot}36$ mm of Hg ab.

or $= 763{\cdot}36 \times 0{\cdot}1333 = 101{\cdot}756$ kPa

Applying characteristic gas eqn. (1.20), $pv = mRT$

$$\text{Volume of gas, } v = \frac{mRT}{p} = \frac{1{\cdot}5 \times 0{\cdot}686 \times (85+273)}{101{\cdot}756} = 3{\cdot}62\ \text{m}^3.$$

It should be noted that if the pressure is expressed in kPa or kN/m^2, the unit of R would be kJ/kg K.

Problem - 15 : *A gas at a temperature of 20°C and pressure of 150 kPa occupies a volume of 0·105 m^3. If the gas is compressed to a pressure of 750 kPa and volume of 0·04 m^3, what would be the final temperature of gas ?*

Given, p_1 = 150 kPa, v_1 = 0·105 m^3, t_1 = 20°C, p_2 = 750 kPa and v_2 = 0·04 m^3.

Applying characteristic gas eqn. (1.20) for initial and final states,

$$p_1v_1 = mRT_1 \text{ and } p_2v_2 = mRT_2$$

Dividing one by the other and re-arranging, $\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$

$$\text{i.e. } \frac{150 \times 0{\cdot}105}{(20 + 273)} = \frac{750 \times 0{\cdot}04}{T_2}$$

For which, $T_2 = 558$ K $\quad\therefore$ Final temperature, $t_2 = 558 - 273 = 285°$C.

1.12 Avogadro's Law

A mol of gas has a mass numerically equal to the molecular weight of the gas. If M represents the molecular weight, 1 kg-mol = M kg.

Since, the molecular weight of Hydrogen is 2, one kg-mol of Hydrogen is equal to 2 kilograms. The molecular weight of Nitrogen (N_2) is 28, therefore, one kg-mol of Nitrogen is equal to 28 kg. It will be shown a little later that the volume of one kilogram-mole or kg-mol of any gas at N.T.P. (0°C and 760 mm of Hg) is 22·4 cubic metres.

Avogadro's law states that *equal volumes of all gases at the same temperature and pressure contain the same number of molecules.* Hence, under identical conditions of the temperature and pressure, one cubic metre of oxygen (O_2) contains the same number of molecules as one cubic metre of hydrogen (H_2). Hydrogen has a molecular weight of 2, while the molecular weight of oxygen is 32. Therefore, molecules of O_2 has a weight which is 32/2 = 16 times the weight of the molecules of H_2. Since, one cubic metre of these gases contain the same number of molecules and a molecule of O_2 weighs 16 times as much as a molecule of H_2, it is evident that the density of O_2 is 16 times the density of hydrogen. So this law indicates that *the specific weight or density of any of the two gases are directly proportional to their molecular weights if the gases are at the same temperature and pressure.*

The density of O_2 is 1·429 kg/m^3 at N.T.P. (0°C and 101·325 kPa). Since, the specific volume is reciprocal of the density, the specific volume of O_2 at N.T.P. = $\frac{1}{1{\cdot}429}$ m^3/kg. If one kilogram of O_2 has volume of $\frac{1}{1{\cdot}429}$ m^3, 32 kg or 1 kg-mol will have a volume of $\frac{1}{1{\cdot}429} \times 32$ m^3, which is equal to 22·4 m^3 at N.T.P. (0°C and 101·325 kPa),

i.e. at N.T.P. the *volume of one kg-mol is 22·4 m^3.*

Similarly, it can be proved that at N.T.P., volume of one kg-mol of any gas is 22·4 m^3.

One g-mol (gram-mol) of all gases occupies a volume of 22·4 litres at N.T.P.

Avogadro's law may also be expressed as :

"The volume of one kg-mol (molar volume) is the same for all gases provided that temperature and pressure are equal".

Density of a gas (kg/m^3) at 0°C and 760 mm of Hg

$$= \frac{\text{Molecular weight of gas in kg}}{22{\cdot}4} \qquad \text{...(1.21)}$$

Equation (1.21), will enable us to determine the density of gases at N.T.P. (at 0°C and 760 mm of Hg).

1.12.1 Universal Gas Constant : In the equation $pv = mRT$, let $p = 760$ mm of Hg or 101.325 kPa, $T = 273$ K and $m = M$, the number of kilograms in one kg-mol. Then, v in equation $pv = mRT$ becomes the volume of one kg-mol which under the N.T.P. conditions equals 22.4 m^3. Substituting these values in the equation $pv = mRT$, we get

$$101{\cdot}325 \times 22{\cdot}4 = MR \times 273$$

$$\therefore\ MR = 8{\cdot}314 \text{ kJ/kg-mol K or } = 8{,}314 \text{ J/kg-mol K} \qquad ...(1.22)$$

Equation (1.22) states that the product of molecular weight, M and the gas constant, R of any gas is equal to 8·314 kJ/kg-mol K. This constant MR is known as *Universal Gas Constant* and its value is constant for all perfect gases. It is usually denoted by the letter G.

$$\therefore\ G = MR \qquad \therefore\ R = \frac{G}{M} = \frac{8{\cdot}314}{M} \text{ or } M = \frac{G}{R} = \frac{8{\cdot}314}{R} \qquad ..(1.23)$$

where R is gas constant in kJ/kg K, and

G is in kJ/kg-mol K.

Since, the molecular weights of all gases are available, the gas constant R can be calculated from the eqn. (1.23).

The values of molecular weight of some common gases are listed in Table 7-1.

If we use the symbol $\overline{\upsilon}$ as molar volume (m^3/kg-mol), then the characteristic gas equation can be written as

$$p\overline{\upsilon} = GT \qquad ...(1.24)$$

Problem - 16 : *0·16 kg of gas at a pressure of 100 kPa and a temperature of 300 K occupies a volume of 0·14 m^3. Calculate the value of the gas constant and the molecular weight of the gas.*

As $pv = mRT$,

(where p is absolute pressure of gas in kPa, v is volume of gas in m^3, T is temperature of gas in K, R is characteristic gas constant in kJ/kg K, and m = mass of gas in kg)

Gas constant, $R = \frac{pv}{mT} = \frac{100 \times 0{\cdot}14}{0{\cdot}16 \times 300} =$ **0·2917 kJ/kg K**

Using eqn. (1.23), Molecular weight of the gas, $M = \frac{G}{R} = \frac{8{\cdot}314}{0{\cdot}2917} =$ **28·5**

1.13 Regnault's Law and Joule's Law

Regnault's law hints that *the specific heat of a perfect gas at constant pressure is constant and its value at constant volume is also constant.*

This is assumed to hold good for many thermodynamic calculations. However, actually a variation in the values of specific heat at constant pressure and constant volume for any gas is found. The value of the specific heat may increase to a certain extent with the increase in temperature. However, the law is assumed to hold good within small range of variation of temperature from whcih it follows that the ratio of the two specific heats k_p and k_v of any given gas is a constant. Thus,

$$\gamma = \frac{k_p}{k_v} = \text{a constant} \qquad ...(1.25)$$

Joule's Law states that the *internal energy of a gas is a function of the temperature only and is independent of the pressure or volume of the gas.* Thus, internal energy of a gas is proportional to absolute temperature.

Joule's experiment was carried out with two insulated pressure vessels, connected by a pipe and a valve. One vessel was at a higher pressure than the other. The valve was opened and the gas was allowed to have free unresisted expansion.

Applying the law of conservation of energy,

$$Q = \Delta u + W$$

$Q = 0$ because no heat was supplied or rejected, and

$W = 0$ because no work was done.

$\therefore \Delta u = 0$ i.e. internal energy did not change.

After free expansion it was found that the temperature did not change and hence it was concluded that the internal energy of a given mass of gas depends upon its temperature only.

1.13.1 Change of Internal Energy of a gas : Any convenient process of thermodynamics change may be selected, but the change of internal energy evaluated between any two temperatures will be the same.

Consider that heat is added to a gas at constant volume. Since, the volume does not change in such a process, no external work is done, that is, $p.dv = 0$. Hence, the energy equation for the process is,

Transferred heat, $Q = u_2 - u_1$ where $u_2 - u_1$ is change of internal energy.

Considering m kg of gas, the heat transferred in constant volume process,

$$Q = m \int_{T_1}^{T_2} k_v \, dT \quad \text{But, } Q = u_2 - u_1$$

$$\therefore u_2 - u_1 = m \int_{T_1}^{T_2} k_v \, dT$$

The equation for a constant value of k_v becomes

$$u_2 - u_1 = m\, k_v \int_{T_1}^{T_2} dT = m\, k_v (T_2 - T_1) \qquad ..(1.26)$$

This equation applies to all processes of a perfect gas between the temperature range from T_1 to T_2. This also shows that the heat added during a constant volume process goes merely to increase the internal energy of the gas.

A perfect of ideal gas is one which obeys the laws, namely, Boyle's, Charles', Gay-Lussac's, Regnault's and Joule's. It is assumed that these laws are very closely obeyed by gases such as hydrogen, oxygen, nitrogen and air, but not by vapours. A perfect gas obeys characteristic gas equation $pv = RT$ and has constant specific heats k_p and k_v.

1.14 Relation between Gas Constant and Two Specific Heats of a Gas

Let one kilogram of gas at initial condition p_1, v_1 and T_1 be heated at constant pressure. Let the final condition be p_1, v_2 and T_2.

The external work done in increasing the volume under constant pressure heating,

$$W = \int_1^2 p\,dv = p_1\,(v_2 - v_1)$$

The heat transferred to the gas during this heating process,

$$Q = k_p\,(T_2 - T_1)$$

Applying the law of conservation of energy, $Q = \Delta u + W$

i.e. change in internal energy, $u = Q - W = k_p\,(T_2 - T_1) - p_1\,(v_2 - v_1)$ ..(i)

Now, if the same one kilogram of the gas be heated at constant volume from temperature T_1 to T_2, and as there is no change in volume, there is no external work done.

$$\therefore\ \Delta u = k_v\,(T_2 - T_1) \qquad \text{..(ii)}$$

Since, the gas is heated through same range of temperature, the change in internal energy is the same in both cases. Hence, by equating eqns. (i) and (ii), we have

$k_p\,(T_2 - T_1) - p_1\,(v_2 - v_1) = k_v\,(T_2 - T_1)$

$\therefore\ p_1\,(v_2 - v_1) = k_p\,(T_2 - T_1) - k_v\,(T_2 - T_1)$

Since $p_1\,(v_2 - v_1) = R\,(T_2 - T_1)$ $\quad (\because\ p_2v_2 = RT_2$ and $p_1v_1 = RT_1)$

$R(T_2 - T_1) = k_p\,(T_2 - T_1) - k_v\,(T_2 - T_1)$

Dividing both sides by $(T_2 - T_1)$, we have

$$R = k_p - k_v \qquad \text{...(1.27)}$$

This proves that *the numerical difference between values of specific heat at constant pressure and constant volume is equal to gas constant (R).*

Problem - 17 : *A gas of certain mass is expanded from an initial state of 400 kPa and 0.04 m^3 to another condition of 120 kPa and 0·1 m^3. The temperature fall was observed to be 146°C. If the values of k_p and k_v are 1·0216 kJ/kg K and 0·7243 kJ/kg K respectively, calculate the change in the internal energy of the gas.*

Here, $p_1 = 400$ kPa, $v_1 = 0{\cdot}04\ \text{m}^3$, $p_2 = 120$ kPa, $v_2 = 0{\cdot}1\ \text{m}^3$ and $T_1 - T_2 = 146$.

From eqn. (1.27), $R = k_p - k_v = 1{\cdot}0216 - 0{\cdot}7243 = 0{\cdot}2973$ kJ/kg K

Applying characteristic eqn. (1.20) for initial and final states to evaluate mass of the gas.

$p_1v_1 = mRT_1$ and $p_2v_2 = mRT_2$

i.e. $mRT_1 - mRT_2 = p_1v_1 - p_2v_2$

$$\therefore\ m = \frac{p_1v_1 - p_2v_2}{R(T_1 - T_2)} = \frac{(400)\,0{\cdot}4 - (120)\,0{\cdot}1}{0{\cdot}2973 \times 146} = 0{\cdot}0915\ \text{kg}$$

Using eqn. (1.26), Change in internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1)$

$= 0{\cdot}0915 \times 0{\cdot}7243\,(-146) = -$ **9·745 kJ(decrease).**

Tutorial-1

1. (a) Explain the term thermodynamic medium.

(b) Explain the terms "systems and surroundings."

(c) Classify systems.

2. (a) Explain isolated, closed, and open systems.

(b) Explain the following terms as related to thermodynamics : System, State, Property, Process and Cycle.

3. State and explain basic thermodynamic properties.

4. (a) What is the concept of temperature ? How is temperature measured ?

(b) Explain gauge pressure, vacuum and absolute pressure. State and explain devices used to measure pressure.

5. Write a brief note on 'energy' and its various forms.

6. (a) Differentiate between :

(i) stored energy and energy in transition, and

(ii) mechanical energy and internal energy.

(b) Explain the energy forms : Work and Heat.

(c) Explain the term 'enthalpy'.

7. (a) Explain thermal equilibrium and Zeroth law of thermodynamics.

(b) Write brief note on thermodynamic equilibrium.

8. (a) State the first law of thermodynamics and discuss its importance.

(b) Show that internal energy is the consequence of the first law of thermodynamics.

9. (a) Explain the second law of thermodynamics with reference to (i) Kelvin-Planck and (ii) Clausius statements.

(b) State some important statements of the second law of thermodynamics formulated by different themodynamits.

10. (a) Explain reversible and irreversible processes.

(b) Explain the term 'entropy'.

11. (a) Define specific heat, and explain specific heat at constant volume and specific heat at constant pressure.

(b) Why specific heat of a gas at constant pressure is always greater than its specific heat at constant volume.

12. (a) Gas in a tank is at a pressure of 450 mm of Hg gauge. Barometer reads 750 mm of Hg. What is the absolute pressure of gas in the tank in kPa ?

(b) A vacuum gauge on a condenser reads 600 mm of mercury and at the same time barometer reads 740 mm of mercury. What is the absolute pressure in the condenser in kPa ?

[(a) 160 kPa, (b) 18·665 kPa]

13. The pressure of gas supplied to an engine is measured as 100 mm of water gauge. If the barometer reads 730 mm of mercury, what is the absolute pressure of gas in mm of mercury and in kPa ?

[737·353 mm of Hg, 98·304 kPa]

14. (a) Convert the following readings of pressure to kPa :

(a) 1 MPa, (b) 5 atm, (c) 1,50,000 n/m^2, (d) 10 bar, (e) 100 kN/m^2, (f) 1,200 mm of Hg, (g) 1,500 mm H_2O, and (h) 0·01 N/mm^2.

(b) Convert the following readings of pressure to bar :

(i) 800 kPa, (ii) 10 atm, (ii) 2 MPa, (iv) 1,12,000 N/m^2, (v) 200 kN/m^2, (vi) 1,520 mm of Hg, and (vii) 1,520 mm of H_2O.

(c) A pressure gauge reads 1.6 MPa and the barometer reads 95 kPa. Calculate the absolute pressure.

[(a) 1,000, (b) 506·625, (c) 150, (d) 1,000, (e) 100, (f) 160, (g) 14·706, (h) 10];
[(b) (i) 8, (ii) 10·1325 (iii) 20, (iv) 1·12, (v) 2, (vi) 2·0265 (vii) 0·149];
[(c) 1·695 MPa]

15. (a) 5 kg of gas occupy 4 m^3. What is the specific volume and density of the gas ?

(b) Find work done in kJ when a weight of 2 tons ascends through a height of 100 metres.

[(a) 0·8 m^3/kg; 1·25 kg/m^3, (b) 962 kJ]

16. (a) Find the energy given out when one kW power is maintained for 30 minutes.

(b) One kg of steam at 8 bar (0·8 MPa) contains 2,700 kJ of heat. Express this heat in newton-metres (N.m)

[(a) 1,800 kJ; (b) 27,00,000 N.m]

17. When a body was heated, 5·0 kJ were used in doing external work, 6·7 kJ were used in increasing the internal potential energy and 14·7 kJ were used in increasing the internal kinetic energy of the body. Determine : (a) the increase in internal energy and (b) the heat added to the body.

[(a) 21·4 kJ; (b) 26·4 kJ]

18. (a) Under certain conditions, 142·77 kJ are required to raise 2 kg of hydrogen from a temperature of 10 °C to a temperature of 15°C. Find the mean specific heat of hydrogen under these conditions.

(b) Mass of a copper vessel is 0·5 kg. Calculate the quantity of heat required to raise its temperature from 20°C to 810°C. Take specific heat of copper as 0·377 kJ/kg K.

[(a) 14·277 kJ/kg K; (b) 7·54 kJ]

19. (a) The value of γ for a certain gas is 1·381 and its specific heat at constant pressure is 1.05 kJ/kg K. Find the heat given out when 0·3 kg of this gas is cooled at constant volume from 180 °C to 10 °C.

(b) The values of specific heat at constant pressure and constant volume for a gas are 0.984 kJ/kg K and 0·729 kJ/kg K respectively. Find the value of γ for the gas.

If two kg mass of this gas is heated from 30 °C to 180 °C, find the heat absorbed if the heating takes place at constant volume and at constant pressure.

[(a) 38·775 kJ; (b) γ = 1·35, 218·7 kJ, 295·2 kJ]

20. Define perfect gas and state and explain the following laws :

(i) Boyle's law, (ii) Charles' law, and (iii) Gay-Lussac's law.

21. Using Boyle's and Charles' laws, derive the characteristic equation for a perfect gas. What is its utility in thermodynamics ? What is a gas constant ?

22. (a) What is kg-mol ?

(b) State and explain Avogadro's law.

(c) What is universal gas constant ? What is the relatioinship between universal gas constant and gas constant ?

(d) Write characteristic equation for a perfect gas using universal gas constant.

23. (a) State and explain Regnault's law and Joule's law.

(b) How is change in internal energy of a gas determined ?

24. Establish the relation between gas constant R and two specific heats, k_p and k_v of a gas.

2

GAS PROCESSES

2.1 Introduction

As described in art. 1.5, whenever one or more of the properties (such a pressure, volume, temperature, internal energy etc.) of a system change, we may say that a change in state has occurred. The path of the succession of states through which the system passes is called the *process.* The change in one or more properties takes place during the process as a result of flow of energy (in the form of work and/or heat) across the boundary of the system. The different thermodynamic processes of gases may be classified as under :

– Constant volume (or Isochloric or Isometric),

– Constant pressure (or Isothermal),

– Constant temperature (or Isothermal),

– Frictionless adiabatic (or Reversible adiabatic or Isentropic),

– Polytropic, and

– Throttling or Restricted flow.

Processes stated above, except throttling, are considered to be reversible non flow processes, and throttling process is an irreversible flow process.

Many of the above thermodynamic processes can be represented graphically by plotting simultaneous values of pressure and volume. Curves, thus, obtained from a diagram showing the changing states of the working substance is called *p-v* diagram, and it can be proved that the area under the curve plotted on such a diagram represents work done during the process if friction in the process is negligible, i.e. if the process is assumed to be reversible.

Consider a certain quantity of gas of volume v_1 in a cylinder at a pressure p_1. Now, if this substance is allowed to expand, work will be done. As the substance expands, the pressure normally drops and the state of the substance changes along some path 1–2 as shown in the fig. 2-1. Since the pressure, and therefore the force acting on the face of the piston is variable, it will be necessary to integrate to find the work done.

Consider a small change of state (fig. 2-1), so small that the pressure is essentially constant during the change. The force acting on the piston will be the pressure times the area of the piston ($p \times A$). The distance that the piston moves is differential quantity dL, and the work is force times distance $= (p \times A) \cdot dL = p \cdot (A \times dL) = p \cdot dv$.

Thus, the work done for this small displacement of the piston, $dW = p \cdot dv$.

If changes of pressure and volume are plotted as shown in fig. 2-1, the area of the elementary strip shown dotted is $p \cdot dv$. If the pressure and volume changes from state 1

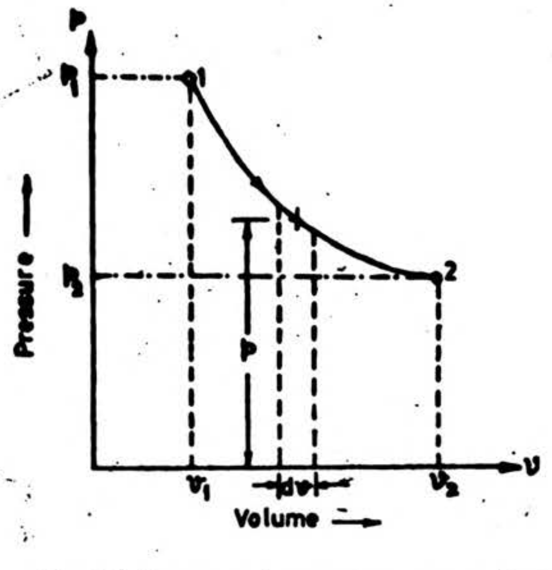


Fig. 2-1. An expansion process on p–v plane.

to state 2, then the area under the curve = $\int_1^2 p \cdot dv$.

Thus, it can be concluded that the area under the curve on p–v diagram gives work done by the gas.

$$\text{Work done, } W = \int_1^2 p \cdot dv \qquad .. (2.1)$$

The expression for work done derived above is applicable to both expansion and compression, but during expansion, work is done by the system while during compression, work is done on the system. Work done during expansion is considered to be positive work and work done during compression is considered to be negative work.

2.2 Constant Volume Process

When a gas is heated in an enclosed vessel, it does not undergo change in volume but the process will result in an increase of pressure. The process is represented graphically on the p-v plane by a vertical straight line 1–2 (fig. 2-2).

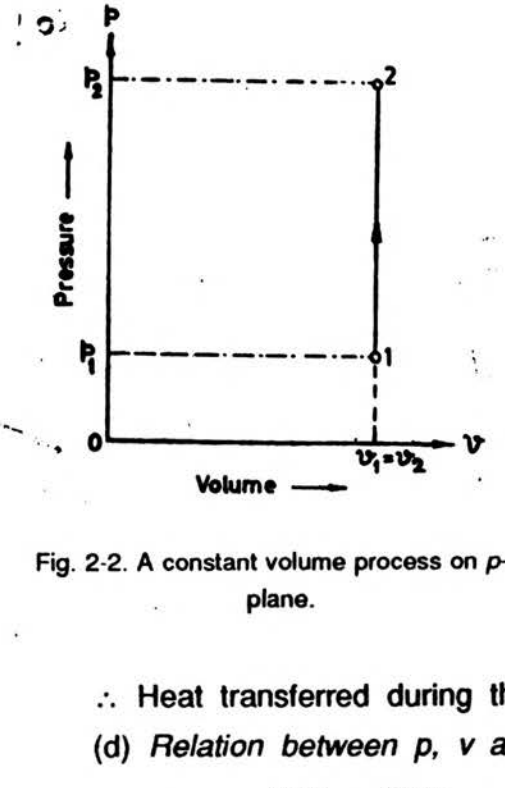


Fig. 2-2. A constant volume process on p–v plane.

(a) Work : Since there is no change in volume, $dv = 0$ and there is no area under the line 1-2, the work done,

$$dW = p \cdot dv = 0$$

(b) *Change in internal energy,*

$$\Delta u = u_2 - u_1 = mk_v(T_2 - T_1) \qquad .. (2.2)$$

(c) *Heat transferred,*

$$Q = W + (u_2 - u_1)$$

$$= 0 + mk_v(T_2 - T_1)$$

i.e. $Q = u_2 - u_1 = mk_v(T_2 - T_1)$

All the heat supplied or transferred during this porcess will be stored in the gas as internal energy.

$\therefore$ Heat transferred during the process, $Q = mk_v(T_2 - T_1)$ .. (2.3)

(d) *Relation between p, v and T :*

In general, $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$. But, here volume remains constant, i.e. $v_1 = v_2$.

$$\text{Hence, } \frac{p_1}{p_2} = \frac{T_1}{T_2} \qquad .. (2.4)$$

Problem-1 : *0·3 m^3 of a gas is filled in a closed tank at an initial condition of 2·75 bar and 40°C. The gas is heated until the pressure in the tank becomes 4 bar. Find the* *change in internal energy of the gas and heat added. Assume $R = 0{\cdot}287$ kJ/kg K and $k_v = 0{\cdot}711$ kJ/kg K for gas.*

Given : $p_1 = 2{\cdot}75$ bar = 275 kPa, $T_1 = 40 + 273 = 313$ K, $p_2 = 4$ bar = 400 kPa and $v_1 = v_2 = 0{\cdot}3\ \text{m}^3$.

Taking $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$ and during the change, $v_1 = v_2$,

$$\frac{T_2}{T_1} = \frac{p_2}{p_1} \qquad \therefore\ T_2 = \frac{p_2}{p_1} \times T_1 = \frac{400}{275} \times 313 = 455\ \text{K}$$

Again for initial state of gas, $p_1 v_1 = mRT_1\ \therefore\ m = \frac{p_1 v_1}{RT_1} = \frac{275 \times 0{\cdot}3}{0{\cdot}287 \times 313} = 0{\cdot}92$ kg

Change in internal energy, $u_2 - u_1 = mk_v(T_2 - T_1)$

$= 0{\cdot}92 \times 0{\cdot}711\ (455 - 313) =$ **92·89 kJ (increase)**

Since no external work is done, heat added is equal to the change (increase) in internal energy = **92·89 kJ.**

Problem-2 : *An air receiver has a volume of 0·45 m^3 and contains air at a pressure of 200 kPa and at a temperature of 15 °C. If 0·9 kg of additional air is pumped into the receiver at constant temperature, what will be the pressure in the receiver ? Assume that specific volume of air at N.T.P. (O °C and 101·325 kPa) is 0·7734 m^3 per kg.*

Applying characteristic equation at N.T.P.,

$$p_0 v_0 = mRT_0 \quad \text{i.e.}\ 101{\cdot}325 \times 0{\cdot}7734 = 1 \times R \times 273$$

$$\therefore\ R = \frac{101{\cdot}325 \times 0{\cdot}7734}{273} = 0{\cdot}287\ \text{kJ/kg K}$$

Initial mass of air in the receiver can be estimated by applying characteristic equation for initial state of air in the receiver,

$$p_1 v_1 = m_1 RT_1 \ \text{i.e.}\ m_1 = \frac{p_1 v_1}{RT_1} = \frac{200 \times 0{\cdot}45}{0{\cdot}287 \times (273 + 15)} = 1{\cdot}089\ \text{kg.}$$

Final mass of air (m_2) in the receiver after pumping = 1·089 + 0·9 = 1·989 kg.

Final pressure of air in the receiver after pumping additional air can be estimated by applying gas equation for final state,

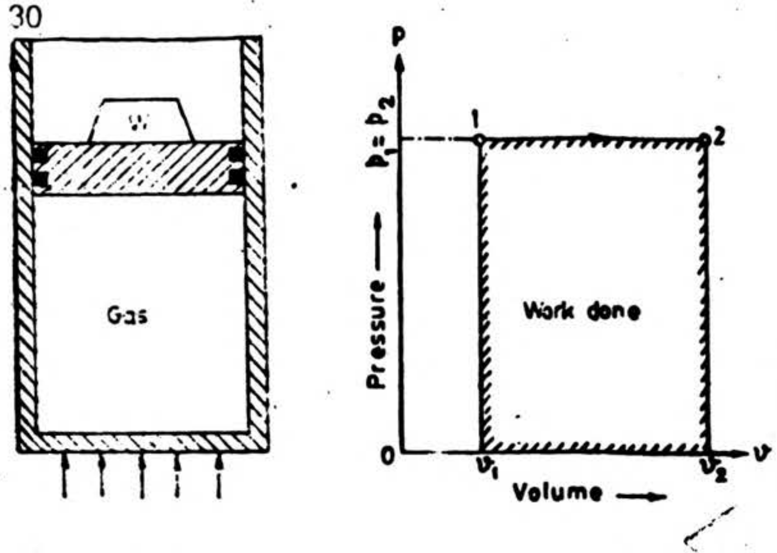
$$\text{i.e.}\ p_2 v_2 = m_2 RT_2 \ \text{i.e.}\ p_2 = \frac{m_2 RT_2}{v_2} = \frac{1{\cdot}989 \times 0{\cdot}287 \times 288}{0{\cdot}45} = \mathbf{365{\cdot}34\ kPa}$$

2.3 Constant Pressure Process

A gas may be heated in an enclosed cyclinder beneath a sliding piston loaded to produce a desired pressure p on the gas as shown in fig. 2-3(a). The heating will thus take place under a constant pressure p.

The heating of the substance under a constant pressure causes an increase in volume and temperature. External work is done owing to increase in volume. The process is represented on the p-v plane by a horizontal line 1-2 as shown in the fig. 2-3(b).

(a) *Work done* : If the state point moves from 1 to 2, the process is an expansion, and if it moves in the reverse direction from 2 to 1, the process is a compression.



(a) Heating (b) p-v plane

Fig 2.3. Constant pressure process by heating gas.

Work done by the gas,

$$W = \int_{v_1}^{v_2} p \cdot dv = p \int_{v_1}^{v_2} dv$$

$$= p(v_2 - v_1) \quad .. (2.5)$$

As pressure remains constant, the initial and final state equations can be written as

$pv_1 = mRT_1$ and $pv_2 = mRT_2$.

$$\text{Since, } W = pv_2 - pv_1$$

$$= mRT_2 - mRT_1$$

$$= mR(T_2 - T_1) \quad .. (2.6)$$

(b) *Change of internal energy* :

$$\Delta u = u_2 - u_1 = mk_v(T_2 - T_1) \quad .. (2.7)$$

(c) *Transferred heat* :

$$\text{Heat transferred, } Q = mk_p(T_2 - T_1) \quad .. (2.8)$$

(d) *Relation between p, v and T* :

Now, $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$ But, pressure remains constant, i.e. $p_1 = p_2$.

$$\text{Hence, } \frac{v_1}{v_2} = \frac{T_1}{T_2} \quad .. (2.9)$$

Problem-3 : *A quantity of air at an initial condition of 14 bar, 0·08 m³ and 450 K expands at constant pressure to 0·25 m³. Find : (a) the temperature at the end of the process, (b) the change of internal energy, and (c) the value of k_p. Given that R = 0·287 kJ/kg K and k_v = 0·711 kJ/kg K for air.*

Given : $p_1 = p_2 = 14$ bar = 1,400 kPa, $v_1 = 0{\cdot}08$ m³, $T_1 = 450$ K and $v_2 = 0{\cdot}25$ m³.

(a) Taking $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$ and $p_1 = p_2$, $\frac{T_2}{T_1} = \frac{v_2}{v_1}$

$$\therefore\ T_2 = \frac{v_2}{v_1} \times T_1 = \frac{0{\cdot}25}{0{\cdot}08} \times 450 = 1{,}406 \text{ K (final temperature)}$$

(b) Mass of the gas, $m = \frac{p_1 v_1}{RT_1} = \frac{1{,}400 \times 0{\cdot}08}{0{\cdot}287 \times 450} = 0{\cdot}8672$ kg.

Now, change in internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1)$

$$= 0{\cdot}8672 \times 0{\cdot}711\ (1{,}406 - 450) = \mathbf{589{\cdot}45\ kJ} \text{ (increase)}$$

(c) $k_p - k_v = R$ $\quad \therefore\ k_p = R + k_v = 0{\cdot}287 + 0{\cdot}711 = $ **0·998 kJ/kg K.**

Problem-4 : *Two kilograms of air at 700 kPa and having a volume of 0·3 m³ is expanded to a volume of 1·5 m³. If the expansion takes place at constant pressure, find the final temperature, the work done and the heat absorbed or rejected by the air during the process. Assume R = 0.287 kJ/kg K and k_v = 0·711 kJ/kg K for air.*

Given : $p_1 = p_2 = 700$ kPa, $v_1 = 0{\cdot}3$ m³, $v_2 = 1{\cdot}5$ m³ and $m = 2$ kg.

From $p_1v_1 = mRT_1$, $T_1 = \frac{p_1v_1}{mR} = \frac{700 \times 0{\cdot}3}{2 \times 0{\cdot}287} = 366$ K

Similarly, $T_2 = \frac{p_2v_2}{mR} = \frac{700 \times 1{\cdot}5}{2 \times 0{\cdot}287} = \mathbf{1{,}829\ K}$ or $t_2 = \mathbf{1{,}556^\circ C}$

Work done, $W = p_1(v_2 - v_1) = 700\ (1{\cdot}5 - 0{\cdot}3) = \mathbf{840\ kJ}$

Change in internal energy, $u_2 - u_1 = mk_v(T_2 - T_1)$

$= 2 \times 0{\cdot}711\ (1{,}829 - 366) = 2{,}080$ kJ (increase)

Heat transferred, $Q = (u_2 - u_1) + W = 2{,}080 + 840 = \mathbf{+\ 2{,}920\ kJ}$

Positive sign indicates that 2,920 kJ are received or absorbed by the air.

Problem-5 : *One kg of air is expanded at constant pressure of 300 kPa from a volume of 0·3 m^3 to 0·5 m^3. Find the value of : (a) the heat transferred, (b) the increase in internal energy, and (c) the external work done by the air during the process. Take* k_v *= 0·7165 kJ/kg K and* k_p *= 1·0035 kJ/kg K for air.*

(a) Given : $p_1 = p_2 = 300$ kPa, $v_1 = 0{\cdot}3\ m^3$, $v_2 = 0{\cdot}5\ m^3$.

Now, $R = k_p - k_v = 1{\cdot}0035 - 0{\cdot}7165 = 0{\cdot}287$ kJ/kg K

As $p_1v_1 = mRT$, $T_1 = \frac{p_1v_1}{mR} = \frac{300 \times 0{\cdot}3}{1 \times 0{\cdot}287} = 313{\cdot}6$ K

Similarly, $T_2 = \frac{p_2v_2}{mR} = \frac{300 \times 0{\cdot}5}{1 \times 0{\cdot}287} = 522{\cdot}6$ K

Heat transferred, $Q = mk_p(T_2 - T_1) = 1 \times 1{\cdot}0035 \times (522{\cdot}6 - 313{\cdot}6) = \mathbf{209{\cdot}73\ kJ.}$

(b) Change in internal energy, $u_2 - u_2 = mk_v(T_2 - T_1)$

$= 1 \times 0{\cdot}7165 \times (522{\cdot}6 - 313{\cdot}6) = \mathbf{149{\cdot}75\ kJ.}$

(c) Work done, $W = p(v_2 - v_1) = 300 \times (0{\cdot}5 - 0{\cdot}3) = \mathbf{60\ kJ.}$

Problem-6 : *The diameter of the cylinder of a Diesel engine is 200 mm and the stroke is 300 mm. The clearance volume is 1,500* cm^3*. At the commencement of the working stroke, the clearance volume is full of gas at pressure of 4.2 MPa* and *285°C. During the first one-tenth of the working stroke the gas expands at constant pressure. Assuming R as 0·287 kJ/kg K and* γ *as 1·4, find : (a) the temperature of gas, (b) the work done by the gas, and (c) the heat absorbed by the gas at the end of this period.*

(a) Clearance volume, $v_1 = 1{,}500\ cm^3 = 1{,}500 \times 10^{-6}\ m^3$,

$p_1 = 4{\cdot}2\ MPa = 4{,}200\ kPa = p_2$.

Considering the condition in the clearance space,

Mass of the gas, $m = \frac{p_1v_1}{RT_1} = \frac{4{,}200 \times 1{,}500 \times 10^{-6}}{0{\cdot}287 \times (285 + 273)} = 0{\cdot}0393$ kg.

Stroke volume $= \frac{\pi}{4} d^2 \times l = \frac{\pi}{4} (20)^2 \times 30 = 9{,}420\ cm^3$.

Total volume at the end of one-tenth of the working stroke,

v_2 = clearance volume + one-tenth of stroke volume

$= \frac{1{,}500}{10^6} + \frac{1}{10} \times \frac{9{,}420}{10^6} = \frac{2{,}442}{10^6}\ m^3.$

Temperature at the end of one-tenth stroke can be evaluated by using characteristic equation $p_2v_2 = mRT_2$.

$$\therefore\ T_2 = \frac{p_2v_2}{mR} = \frac{4{,}200 \times 2{,}442 \times 10^{-6}}{0{\cdot}0393 \times 0{\cdot}287} = 909{\cdot}3 \text{ K or } t_2 = 636{\cdot}3^\circ\text{C}$$

(b) Work done by the gas during constant pressure expansion,

$$W = p_1\ (v_2 - v_1) = 4{,}200\ (2{,}442 \times 10^{-6} - 1{,}500 \times 10^{-6}) = 3{\cdot}96 \text{ kJ}$$

(c) Since $\gamma = \frac{k_p}{k_v}$, $k_v = \frac{k_p}{\gamma}$ and also $R = k_p - k_v = k_p - \frac{k_p}{\gamma} = k_p\left(1 - \frac{1}{\gamma}\right)$

$$\therefore\ k_p = R\left(\frac{\gamma}{\gamma - 1}\right) = 0{\cdot}287\left(\frac{1{\cdot}4}{1{\cdot}4 - 1}\right) = 1{\cdot}0045 \text{ kJ/kg K}$$

Thus, heat received, $Q = mk_p(T_2 - T_1)$

$$= 0{\cdot}0393 \times 1{\cdot}0045 \times (909{\cdot}3 - 558) = 13{\cdot}87 \text{ kJ}$$

Problem-7 : *0·12 m^3 of gas at 1·1 bar and 92°C is compressed to a volume of 0·03 m^3, pressure of 6 bar and a temperature of 240°C. The work done on the gas during the compression process is 20·1 kJ.*

Calculate : (a) the mass of gas compressed, (b) the increase in internal energy of the gas, and (c) the heat received or rejected by the gas during compression. Assume $R = 0{\cdot}295$ kJ/kg K and $\gamma = 1{\cdot}4$ for the gas.

Given : $p_1 = 1{\cdot}1$ bar $= 110$ kPa, $v_1 = 0{\cdot}12$ m^3, $t_1 = 92^\circ$C, $v_2 = 0{\cdot}03$ m^3,
$p_2 = 6$ bar $= 600$ kPa, $t_2 = 240^\circ$C, $W = 20{\cdot}1$ kJ, and $R = 0{\cdot}295$ kJ/kg K.

(a) Using characteristic gas equation,

$$\text{Mass of the gas, } m = \frac{p_1v_1}{RT_1} = \frac{110 \times 0{\cdot}12}{0{\cdot}295 \times (92 + 273)} = 0{\cdot}1226 \text{ kg.}$$

(b) Combining the relation, $\gamma = \frac{k_p}{k_v}$ and $R = k_p - k_v$, we have,

$$k_v = R \times \frac{1}{(\gamma - 1)} = 0{\cdot}295 \times \frac{1}{(1{\cdot}4 - 1)} = 0{\cdot}7375 \text{ kJ/kg K.}$$

Now, change in internal energy, $u_2 - u_1 = m \times k_v(T_2 - T_1)$

$$= 0{\cdot}1226 \times 0{\cdot}7375 \times (513 - 365) = 13{\cdot}38 \text{ kJ (increase).}$$

(c) As per data given, work done on the gas, $W = -20{\cdot}1$ kJ.

Note : W is negative when work is done on the gas.

Thus, heat transferred, $Q = (u_2 - u_1) + W = 13{\cdot}38 - 20{\cdot}1 = -6{\cdot}72$ kJ.

Negative sign indicates that 6·72 kJ are *rejected* by the gas.

2.4 Isothermal Process

If a gas receives heat in such a manner that its temperature remains constant during expansion, it will do external work equal to the heat supplied. Such an expansion at constant temperature is known as isothermal expansion (fig. 2-4). On the other hand if work is done on the gas (during isothermal compression), heat should be removed from the gas in order to maintain its temperature constant. Gas is said to be expanded or compressed isothermally when its temperature remains constant during the process. For *perfect gas* it can be shown, using characteristic equation, that *if temperature remains*

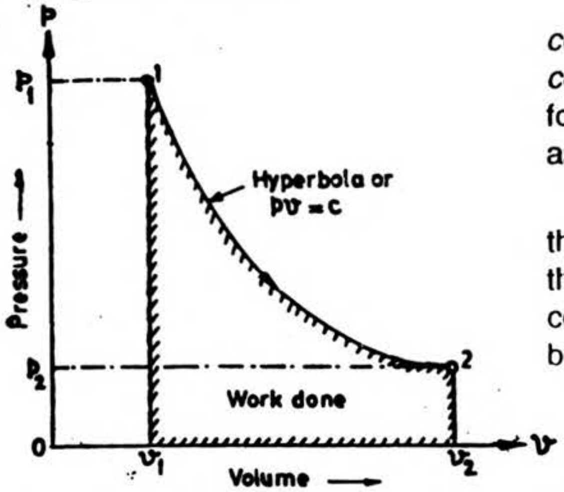


Fig. 2-4. Isothermal or Hyperbolic expansion process.

constant, product of pressure and volume also remains constant. Thus, the law of expansion or compression for isothermal process on p–v diagram is hyperbolic as p varies inversely as v.

(a) Work done : When heat is added or received, the process is an expansion and work is done by the gas, and when heat is rejected, the process is compression and work is done on the gas. Work done by the gas,

W = shaded area under the curve 1–2 (fig. 2-4)

$$= \int_{v_1}^{v_2} p \cdot dv \quad \text{But, } pv = p_1v_1 = C \quad \text{i.e. } p = \frac{p_1v_1}{v}$$

$$\text{Hence, } W = \int_{v_1}^{v_2} \frac{p_1 v_1}{v} \cdot dv = p_1 v_1 \int_{v_1}^{v_2} \frac{dv}{v} = p_1 v_1 (\log_e v_2 - \log_e v_1)$$

$$= p_1 v_1 \log_e \left(\frac{v_2}{v_1} \right) = p_1 v_1 \log_e (r) \qquad \text{.. (2.10)}$$

where, r = expansion ratio.

(b) Change of internal energy : Since the temperature is constant during the process, change in internal energy is zero,

i.e. $\Delta u = u_2 - u_1 = 0$ .. (2.11)

(c) Transferred heat :

Q = change in internal energy + work done = 0 + $p_1v_1 \log_e (r)$

Hence, transferred heat, $Q = p_1v_1 \log_e(r)$ .. (2.12)

(d) Relation between p, v and T :

As $\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$ and temperature remains constant, i.e. = $T_1 = T_2$,

$p_1v_1 = p_2v_2$ .. (2.13)

Problem-8 : *0·1 m^3 of air at 6 bar is expanded isothermally to 0·5 m^3. Calculate the final pressure and the heat supplied during the expansion process.*

Given : p_1 = 6 bar = 600 kPa, v_1 = 0·1 m^3, v_2 = 0·5 m^3.

During isothermal process in the case of perfect gases, the relationship between pressure and volume is given by $p_1v_1 = p_2v_2$, i.e. 600 × 0·1 = p_2 × 0·5.

∴ Final pressure, p_2 = 120 kPa or **1·2 bar**.

Using eqn. (2.10), the work done during isothermal expansion 1-2,

$$W = p_1v_1 \log_e \left(\frac{v_2}{v_1} \right) = 600 \times 0{\cdot}1 \times \log_e 5 = 96{\cdot}57 \text{ kJ}.$$

Since, for isothermal process internal energy does not change,

Heat supplied to air during isothermal expansion, Q = work done = **96·57 kJ.**

Problem-9 : *0·4 kg of air at a pressure of 100 kPa and a temperature of 288 K, is compressed isothermally to 500 kPa. Calculate the final volume and the heat rejected during the compression process. Take R = 0·287 kJ/kg K for air.*

Given : $p_1 = 100$ kPa, $T_1 = 288$ K, $p_2 = 500$ kPa, $m = 0{\cdot}4$ kg, $R = 0.287$ kJ/kg K.

As $p_1v_1 = mRT_1$, $v_1 = \frac{mRT_1}{p_1} = \frac{0{\cdot}4 \times 0{\cdot}287 \times 288}{100} = 0{\cdot}3306 \text{ m}^3$

Now, for isothermal process, $p_1v_1 = p_2v_2$

$\therefore$ Final volume, $v_2 = \frac{p_1}{p_2} \times v_1 = \frac{100}{500} \times 0{\cdot}3306 = \mathbf{0{\cdot}066 \ m^3}$.

Work done on the gas during isothermal compression from state 1 to state 2,

$$W = p_1v_1\log_e\left(\frac{v_1}{v_2}\right) = p_1v_1\log_e\left(\frac{p_2}{p_1}\right) = 100 \times 0{\cdot}3306 \times \log_e(5) = 53{\cdot}21 \text{ kJ.}$$

Heat rejected (transferred), Q = work done + change in internal energy.

Since for isothermal process, internal energy does not change,

Heat rejected, Q = Work done = **53·21 kJ**.

2.5 Adiabatic Process

Adiabatic process is one during which there is no heat transfer between the system and surroundings. The *adiabatic expansion* may take place in an engine cylinder if no transfer of heat takes place through the cylinder walls. Work is done on the piston by the gas when it expands during the expansion stroke and no transfer of heat takes place through the cylinder walls. Similarly, when work is done on the gas (during compression) without receiving heat from or rejecting heat to some external source (surrounding objects), the process is known as *adiabatic compression.*

As work is done by the gas during adiabatic expansion, the internal energy will diminish, and the fall in its internal energy value is equivalent to the work done by the gas. Since heat does not flow into the gas from any external source during the operation, its temperature will fall. Thus, internal energy of the gas is converted into work. During an adiabatic compression process, the amount of internal energy in the gas will increase and so also its temperature, as the work is done on the gas.

For an ideal adiabatic process three conditions must be satisfied, namely,

.. No heat is added or rejected during the process (i.e. heat transfer, Q is zero),

.. Work of some nature must be done by the gas during expansion or work should be done on the gas during its compression, and

.. The process must be frictionless.

A better name of this ideal adiabatic process is **isentropic process,** as entropy remains constant during this process.

In practice, an isentropic operation is not possible because of the exchange of heat between the gas and the cylinder walls and friction which takes place during the process. The amount of heat transferred can be reduced by increasing the speed of operation and providing good lagging on the cylinder walls in order to stop the heat flow.

It can be shown that equation of frictionless adiabatic (isentropic) process for a perfect gas can be written as $pv^{\gamma} = p_1v_1^{\gamma} = p_2v_2^{\gamma}$ = *a constant.* Here γ is the ratio of the two specific heats k_p and k_v as defined earlier.

For a small change in one kilogram of the gas, it can be written that

$$dQ = du + p \cdot dv$$

But, for adiabatic process, $dQ = 0$ and $du = k_v\,dT \quad \therefore\ 0 = k_v\,dT + p \cdot dv$

Putting, $p = \dfrac{RT}{v}, \qquad k_v\,dT + \dfrac{RT}{v}\,dv = 0 \qquad \therefore\ \dfrac{dT}{T} + \dfrac{R \cdot dv}{k_v\,v} = 0$

Integrating, we have, $\log_e T = \dfrac{R}{k_v}(\log_e v) = \text{a constant}$

But, $\dfrac{R}{k_v} = \dfrac{k_p - k_v}{k_v} = \gamma - 1 \qquad \therefore\ T(v)^{\gamma-1} = \text{a constant}$

Again putting $T = \dfrac{pv}{R}, \qquad \dfrac{pv}{R}(v)^{\gamma-1} = \text{a constant}$

$$\therefore\ pv = \text{constant or } p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \qquad ..\ (2.14)$$

The curve corresponding to this eqn. (2.14) is represented in fig. 2-5 on $p - v$ plane.

(a) *Work done* : Let p_1 and v_1 refer to the initial condition and p_2 and v_2 to the final condition of the gas. Consider any point on the curve 1-2 and let the pressure and the volume at this point be p and v respectively. Let the volume increase by dv, at a pressure p.

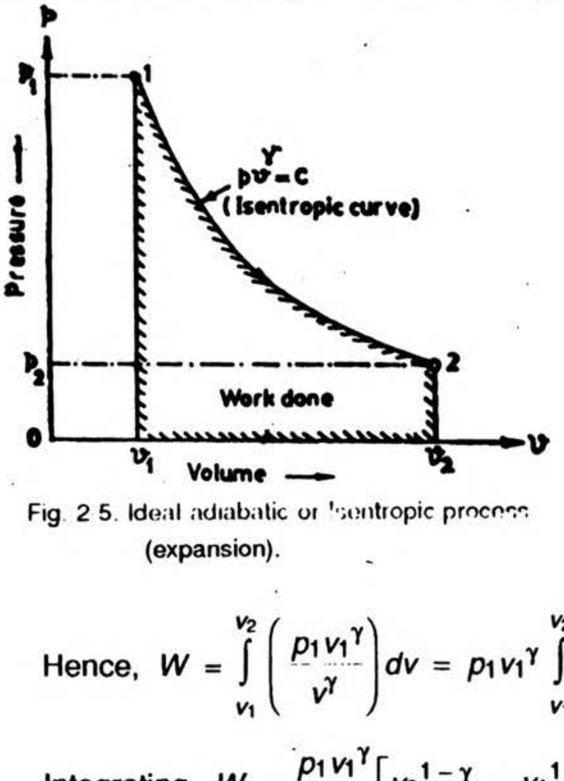


Fig. 2-5. Ideal adiabatic or isentropic process (expansion).

Small amount of work done during this small change in volume $= p \cdot dv$.

$\therefore\ W$ = shaded area under the curve 1–2

$$= \int_{v_1}^{v_2} p \cdot dv$$

But, since $pv^{\gamma} = p_1 v_1^{\gamma}$ = a constant,

$$\therefore\ p = p_1\left(\frac{v_1}{v}\right)^{\gamma}$$

Hence, $W = \displaystyle\int_{v_1}^{v_2}\left(\frac{p_1 v_1^{\gamma}}{v^{\gamma}}\right)dv = p_1 v_1^{\gamma}\int_{v_1}^{v_2} v^{-\gamma}\,dv$

Integrating, $W = \dfrac{p_1 v_1^{\gamma}}{1-\gamma}\left[v_2^{1-\gamma} - v_1^{1-\gamma}\right] = \dfrac{p_1 v_1^{\gamma}(v_2)^{1-\gamma}}{1-\gamma} - \dfrac{p_1 v_1^{\gamma} \cdot (v_1)^{1-\gamma}}{1-\gamma}$

But, $p_1 v_1^{\gamma} = p_2 v_2^{\gamma}\ \therefore\ p_1 v_1^{\gamma}$ may be replaced by $p_2 v_2^{\gamma}$ in the above equations.

Then, $W = \dfrac{p_1 v_1^{\gamma}(v_2)^{1-\gamma}}{1-\gamma} - \dfrac{p_1 v_1^{\gamma} \cdot (v_1)^{1-\gamma}}{1-\gamma} = \dfrac{p_2 v_2 - p_1 v_1}{1-\gamma} = \dfrac{p_1 v_1 - p_2 v_2}{\gamma - 1} \qquad ..\ (2.15)$

Since, $pv = mRT$, the last expression of W (work done) can be written as

$$W = \frac{mRT_1 - mRT_2}{\gamma - 1} = \frac{mR}{\gamma - 1}(T_1 - T_2) \qquad ..\ (2.16)$$

(b) *Change of internal energy* is equal to work done, since no heat is entering or leaving the gas during the adiabatic process, the work is done at the expense of internal energy,

$$\text{i.e. } \Delta u = -\frac{p_1v_1 - p_2v_2}{(\gamma - 1)}$$

$$= -\frac{mR}{(\gamma-1)}(T_1 - T_2) = -mk_v(T_1 - T_2) = mk_v(T_2 - T_1) \quad \text{.. (2.17)}$$

(c) *Transferred heat* during the process, Q = zero, since no heat is enterting or leaving the gas.

(d) *Relation between p, v and T :* (For adiabatic compression)

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} ; \text{ and } \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} \quad \text{.. (2.18)}$$

The eqn. (2.18) may be put in words :

$$\frac{\text{Final temperature}}{\text{Initial temperature}} = \left[\frac{\text{Initial volume}}{\text{Final volume}}\right]^{\gamma-1} = \left[\frac{\text{Final pressure}}{\text{Initial pressure}}\right]^{\frac{\gamma-1}{\gamma}}$$

$$\text{or } \frac{\text{Final pressure}}{\text{Initial pressure}} = \left[\frac{\text{Initial volume}}{\text{Final volume}}\right]^{\gamma}.$$

Problem-10 : *0·45 kilogram of gas occupies 0·35 m³ at 15 °C and 110 kPa. After adiabatic compression to 0·13 m³, the pressure is 440 kPa. Find, (a) the constant for this gas, (b) the molecular weight of this gas, (c) the ratio of the specific heats, (d) the specific heat at constant pressure, and (e) the specific heat at constant volume.*

(a) As $p_1v_1 = mRT_1$, $R = \frac{p_1v_1}{mT_1} = \frac{110 \times 0{\cdot}35}{0{\cdot}45 \times 288} = \mathbf{0{\cdot}297\ kJ/kg\ K}$.

(b) Using eqn. (1.23), Molecular weight,

$$M = \frac{G}{R} = \frac{8{\cdot}314}{0{\cdot}297} = 28$$

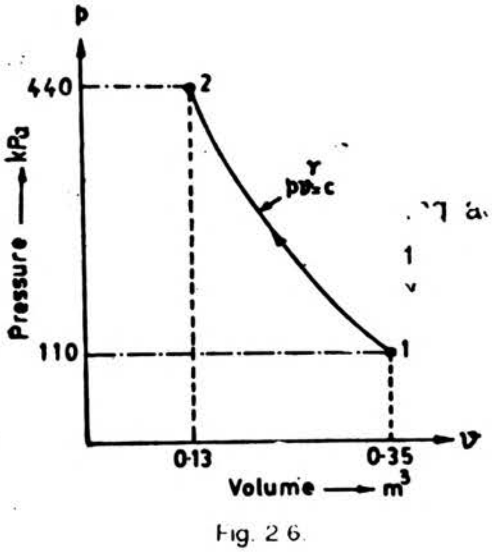


Fig. 2·6.

(c) Referring to fig.2-6, for adiabatic compression 1-2,

$$p_1v_1^{\gamma} = p_2v_2^{\gamma} \quad \text{or} \quad \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma}$$

$$\therefore \gamma = \frac{\log\frac{p_2}{p_1}}{\log\frac{v_1}{v_2}} = \frac{\log\frac{440}{110}}{\log\frac{0{\cdot}35}{0{\cdot}13}} = \frac{0{\cdot}6021}{0{\cdot}4298} = \mathbf{1{\cdot}4}$$

$$\text{(d) } \gamma = \frac{k_p}{k_v} \quad \therefore k_v = \frac{k_p}{\gamma}$$

$$\text{But, } R = k_p - k_v = k_p - \frac{k_p}{\gamma} = k_p\left(1 - \frac{1}{\gamma}\right)$$

$$\therefore 0{\cdot}297 = k_p(1 - 1/1{\cdot}4) = 0{\cdot}2857k_p$$

$\therefore$ Specific heat at constant pressure, $k_p = \frac{0{\cdot}297}{0{\cdot}2857} = $ **1·03955 kJ/kg K.**

(e) $\gamma = \frac{k_p}{k_v} \quad \therefore k_v = \frac{k_p}{\gamma} = \frac{1{\cdot}03955}{1{\cdot}4} = $ **0·7425 kJ/kg K.**

Problem-11 : *One kilogram of gas expands adiabatically and its temperature is observed to fall from 240°C to 115°C to while the volume is doubled. The gas produces 89 kJ of work in the process. Determine the specific heats of the gas at constant pressure and at constant volume.*

Given : $T_1 = 513$ K, $T_2 = 388$ K, $\frac{v_2}{v_1} = 2$, and $W = 89$ kJ.

Using eqn. (2.16), work done per kg during adiabatic expansion 1-2,

$$W = R\left(\frac{T_1 - T_2}{\gamma - 1}\right) = (k_p - k_v)\left(\frac{T_1 - T_2}{k_p/k_v - 1}\right) = k_v(T_1 - T_2)$$

$$\left(\because R = k_p - k_v \text{ and } \gamma = \frac{k_p}{k_v}\right)$$

i.e. $89 = k_v\,(513 - 388) \quad \therefore k_v = $ **0·712 kJ/kg K.**

Now, for adiabatic expansion 1–2, $\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma - 1}$ i.e. $\frac{513}{288}$ or $1{\cdot}322 = (2)^{\gamma - 1}$

Taking logs of both sides, $\log(1{\cdot}322) = (\gamma - 1)\log(2)$

i.e. $0{\cdot}1213 = (\gamma - 1)\,0{\cdot}301 \quad \therefore \gamma = 1{\cdot}4$

$\therefore k_p = \gamma \times k_v = 1{\cdot}4 \times 0{\cdot}712 = $ **0·9968 kJ/kg K.**

Problem-12 : *0·65 m^3 of hydrogen at 100 kPa and 20°C is compressed adiabatically to 1,400 kPa and then expanded isothermally to the original volume. Determine :*

(a) the pressure of the gas at the end of isothermal expansion,

(b) the heat which must be added to the gas during isothermal expansion, and

(c) the heat which must be extracted from the gas after isothermal expansion to reduce it to its original state of pressure at constant volume.

Take R = 4·124 kJ/kg K and k_p = 14·256 kJ/kg K for Hydrogen.

(a) Now, $k_p = k_v - R = 14{\cdot}256 - 4{\cdot}124 = 10{\cdot}132$ and $\gamma = \frac{k_p}{k_v} = \frac{14{\cdot}256}{10{\cdot}132} = $ **1·407**

Referring to fig. 2-7 and considering adiabatic compression 1-2,

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \text{ or } \frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}$$

$$\therefore v_2 = \frac{v_1}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}} = \frac{0{\cdot}65}{\left(\frac{1400}{100}\right)^{\frac{1}{1{\cdot}407}}} = \frac{0{\cdot}65}{6{\cdot}512} \approx 0{\cdot}1 \text{ m}^3$$

Now, gas expands isothermally (at constant temperature) from an initial pressure (p_2) of 1,400 kPa and volume 0·1 m^3, to a final presure (p_3) and volume 0·65 m^3.

For isothermal expansion 2-3, $p_2v_2 = p_3v_3$

i.e. $1{,}400 \times 0{\cdot}1 = p_3 \times 0{\cdot}65$

$$\therefore\ p_3 = \frac{1{,}400 \times 0{\cdot}1}{0{\cdot}65} = \mathbf{215\ kPa.}$$

(b) Now, work done during isothermal expansion 2-3,

$$W = p_2v_2 \log_e \frac{v_3}{v_2} = 1{,}400 \times 0{\cdot}1 \times \log_e\left(\frac{0{\cdot}65}{0{\cdot}1}\right) = \mathbf{262\ kJ}$$

Heat added during isothermal expansion 2–3

= heat equivalent of work done during isothermal expansion 2–3 = **262 kJ.**

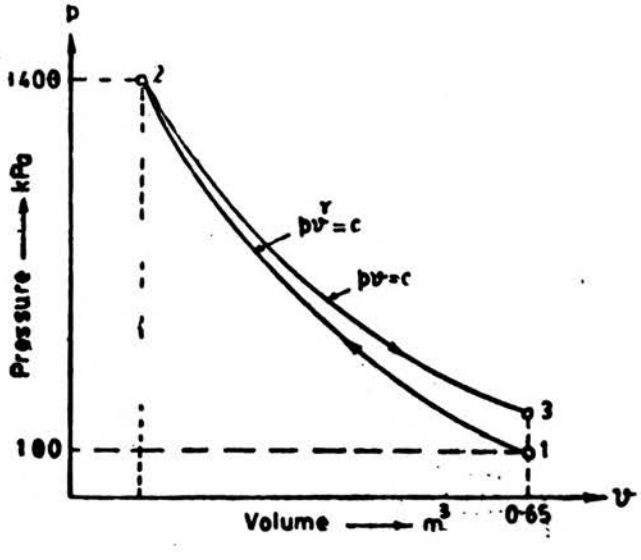


Fig. 2·7.

(c) For adiabatic compression 1–2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1{,}400}{100}\right)^{\frac{0{\cdot}407}{1{\cdot}407}} = 2{\cdot}147$$

$$\therefore\ T_2 = T_1 \times 2{\cdot}147 = 293 \times 2{\cdot}147 = \mathbf{629\ K}$$

As $p_1v_1 = mRT_1$,

$$m = \frac{p_1v_1}{RT_1} = \frac{100 \times 0{\cdot}65}{4{\cdot}124 \times 293} = 0{\cdot}0538\ \text{kg.}$$

The heat extracted from the gas in cooling it at constant volume from 3 to 1

$= m \times k_v(T_3 - T_1)$

$= 0{\cdot}0538 \times 10{\cdot}132 \times (629 - 293) = \mathbf{183{\cdot}15\ kJ}$

Problem-13 : *One kg of air at an initial condition of 350 kPa and 450 K expands adiabatically and without friction to a final condition of 130 kPa. Calculate the amount of external work done and the change in internal energy of the gas during the expansion. Take $\gamma = 1{\cdot}4$ and $R = 0{\cdot}287$ kJ/kg K for air.*

Given : $p_1 = 350$ kPa, $T_1 = 450$ K, $p_2 = 130$ kPa, $\gamma = 1{\cdot}4$, $R = 0{\cdot}287$ kJ/kg K.

As $p_1v_1 = RT_1$, $\quad v_1 = \dfrac{RT_1}{p_1} = \dfrac{0{\cdot}287 \times 450}{350} = 0{\cdot}369\ \text{m}^3$

Considering adiabatic expansion 1–2, $p_1v_1^{\gamma} = p_2v_2^{\gamma}$

$$\therefore\ \text{Final volume, } v_2 = v_1 \times \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = 0{\cdot}369 \times \left(\frac{350}{130}\right)^{\frac{1}{1{\cdot}4}} = 0{\cdot}369 \times 2{\cdot}208 = 0{\cdot}748\ \text{m}^3$$

Using eqn. (2.15),

$$\text{Work done, } W = \frac{p_1v_1 - p_2v_2}{\gamma - 1} = \frac{350 \times 0{\cdot}369 - 130 \times 0{\cdot}748}{1{\cdot}4 - 1} = \mathbf{79{\cdot}76\ kJ}$$

During expansion, work will be done by the gas and is positive.

Since the work done at the expense of internal energy of the gas, the *decrease* in internal energy is 79·76 kJ.

i.e. Change in internal energy = **– 79·76 kJ.**

Alternatively, using eqn. (2.17),

$$\text{Change in internal energy, } u_2 - u_1 = -\frac{p_1 v_1 - p_2 v_2}{\gamma - 1} = -79{\cdot}76 \text{ kJ.}$$

Problem-14 : *0·25 m^3 of gas at 288 K and 100 kPa (0·1 MPa) is compressed adiabatically to 700 kPa (0·7 MPa). Calculate : (i) the final temperature of the gas, and (ii) the work done on the gas. Take k_p = 1.001 kJ/kg K and k_v = 0·715 kJ/kg K for the gas.*

(i) Now, $\gamma = \frac{k_p}{k_v} = \frac{1{\cdot}001}{0{\cdot}715} = 1{\cdot}4.$

From eqn. (2.18),

For ideal adiabatic compression 1–2, $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$

$$\therefore\ T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 288 \times \left(\frac{700}{100}\right)^{\frac{1{\cdot}4-1}{1{\cdot}4}} = 502{\cdot}5\text{ K} \quad or \quad t_2 = 229{\cdot}5^\circ\text{C}$$

(ii) $R = k_p - k_v = 1{\cdot}001 - 0{\cdot}715 = 0{\cdot}286$ kJ/kg K

As $p_1 v_1 = mRT_1$, $m = \frac{p_1 v_1}{RT_1} = \frac{100 \times 0{\cdot}25}{0{\cdot}286 \times 288} = 0{\cdot}3035$ kg

Using equation (2.15), work done,

$$W = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1} = \frac{mR(T_1 - T_2)}{\gamma - 1} = \frac{0{\cdot}3035 \times 0{\cdot}286\,(288 - 502{\cdot}5)}{1{\cdot}4 - 1} = -46{\cdot}55\text{ kJ}$$

Problem-15 : *One kg of a gas occupying 0·1 m^3 at a pressure of 1,000 kPa is expanded at constant pressure to 0·2 m^3. The gas is then expanded adiabatically to 0·6 m^3.*

Calculate :

(a) the temperature at the end of constant pressure expansion,

(b) the temperature and pressure at the end of adiabatic expansion,

(c) the heat supplied and work done during each portion of the process, and

(d) the total work done during the whole process.

Take k_p = 1·008 kJ/kg K and k_v = 0·72 kJ/kg K for the gas.

Given : $p_1 = p_2 = 1{,}400$ kPa, $v_1 = 0{\cdot}1\ m^3$, $v_2 = 0{\cdot}2\ m^3$, and $v_3 = 0{\cdot}6\ m^3$.

(a) $R = k_p - k_v = 1{\cdot}008 - 0{\cdot}72 = 0{\cdot}288$ kJ/kg K

Now, $p_1 v_1 = mRT_1 \quad \therefore\ T_1 = \frac{p_1 v_1}{mR} = \frac{1{,}400 \times 0{\cdot}1}{1 \times 0{\cdot}288} = 486$ K

Now, $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$ and $p_1 = p_2 \quad \therefore\ \frac{v_1}{T_1} = \frac{v_2}{T_2}$

i.e. $T_2 = \frac{v_2}{v_1} \times T_1 = \frac{0{\cdot}2}{0{\cdot}1} \times 486 = 972$ K or $t_2 = 699^\circ$C

(b) $\gamma = \frac{k_p}{k_v} = \frac{1{\cdot}008}{0{\cdot}72} = 1{\cdot}4$

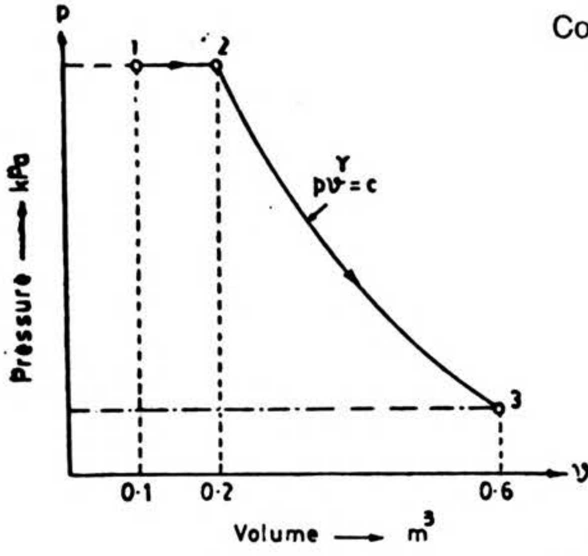


Fig. 2-8.

Considering adiabatic expansion 2-3 of fig. 2-8,

$$\frac{T_2}{T_3} = \left(\frac{v_3}{v_2}\right)^{\gamma - 1} = \left(\frac{0{\cdot}6}{0{\cdot}2}\right)^{1{\cdot}4 - 1} = (3)^{0{\cdot}4}$$

$$\therefore\ T_3 = \frac{T_2}{(3)^{0{\cdot}4}} = \frac{792}{1{\cdot}522} = \mathbf{626\ K\ or\ t_3 = 353^{\circ}C}$$

Also $p_2 v_2^{\gamma} = p_3 v_3^{\gamma}$

$$\therefore\ p_3 = \frac{p_2}{\left(\dfrac{v_3}{v_2}\right)^{\gamma}} = \frac{1{,}400}{(3)^{1{\cdot}4}} = \frac{1{,}400}{4{\cdot}655} = \mathbf{300\ kPa.}$$

(c) Heat supplied during constant pressure expansion 1-2 (fig. 2-8),

$$Q = mk_p\ (T_2 - T_1) = 1 \times 1{\cdot}008 \times (972 - 486) = \mathbf{490\ kJ.}$$

Heat supplied during adiabatic expansion 2 - 3, Q = 0.

Work done during constant pressure expansion 1 - 2,

$$W = P_1\ (v_2 - v_1) = 1{,}400\ (0{\cdot}2 - 0{\cdot}1) = \mathbf{140\ kJ.}$$

Work done during adiabatic expansion 2 - 3,

$$W = \frac{p_2 v_2 - p_3 v_3}{\gamma - 1} = \frac{1{,}400 \times 0{\cdot}2 - 300 \times 0{\cdot}6}{1{\cdot}4 - 1} = \mathbf{250\ kJ.}$$

(d) Total work done during the whole process = 140 + 250 = **390 kJ.**

2.6 Polytropic Process

The polytropic process is a name usually applied to any process that can be described by the general equation pv^n= constant, where n is a constant having any value between zero and infinity. Many actual processes have expansion and compression curves defined by the relation pv^n= constant.

Polytropic processes in general cover all the equations for the change of state of the substance for all the processes considered before. A particular process can be defined from the general law $pv^n = C$, by choosing proper value of n as follows :

Process	Value of index n	Equation
Constant volume	$n = \propto$	$pv^{\propto} = C$ or $v = C$
Constant pressure	$n = 0$	$pv^0 = C$ or $p = C$
Isothermal or hyperbolic	$n = 1$	$pv = C$
Ideal adiabatic or isentropic	$n = \gamma$	$pv^{\gamma} = C$

The polytropic process of a perfect gas is mathematically similar to the adiabatic process except that the index n is used in place of the index γ.

Therefore, the results obtained before for the adiabatic process may be used for polytropic process by substituting n for γ. Thus,

(a) Work done, $$W = \frac{p_1 v_1 - p_2 v_2}{n - 1} = \frac{mR(T_1 - T_2)}{n - 1} \qquad ..\ (2.19)$$

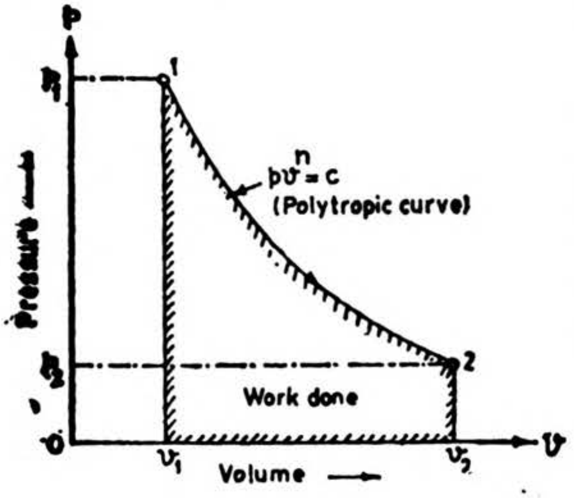


Fig. 2·9.

(b) Change in internal energy

$$\Delta u = mk_v(T_2 - T_1) \qquad \text{.. (2.20)}$$

(c) Expression for heat received or rejected (transferred heat) during an expansion of the form Pv^n = constant is derived as follows :

If a gas expands according to the law pv^n= constant, when n is not equal to γ, the heat received or rejected can be calculated as follows :

Let p_1, v_1 and T_1 be the initial conditions and p_2, v_2 and T_2 the final conditions.

From the first law of thermodynamics,

Heat received = Work done + Change in internal energy

$$\text{i.e. } Q = W + \Delta u \qquad \text{.. (i)}$$

But, $\Delta u = mk_v(T_2 - T_1)$

$$= \frac{mR}{(\gamma - 1)}(T_2 - T_1) \qquad \left\{ \because k_v = \frac{R}{\gamma - 1} \right\}$$

$$= -\frac{p_1v_1 - p_2v_2}{(\gamma - 1)} \qquad (\because mRT_1 = p_1v_1 \text{ and } mRT_2 = p_2v_2) \qquad \text{.. (ii)}$$

$$\text{Also, } W = \frac{p_1v_1 - p_2v_2}{n - 1} \qquad \text{.. (iii)}$$

Substituting values from eqns. (ii) and (iii) in eqn. (i), transferred heat,

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

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

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

$$\text{or } Q = \frac{mR(T_1 - T_2)}{(n-1)} \times \left(\frac{\gamma - n}{\gamma - 1} \right) = \text{Work done} \times \left(\frac{\gamma - n}{\gamma - 1} \right) \qquad \text{.. (2.21)}$$

This equation gives the amount of heat which passes into the gas, through the cylinder walls, as the gas expands, *This is true when n is less than γ and more than 1.* If however, $n > \gamma$ heat is rejected by the gas or gas is losing heat during expansion.

In case of the compression, heat equivalent of work done will be negative, i.e. work will be done on the gas. Hence, Q becomes –ve or negative heat is received, i.e. heat is rejected by the gas during compression. This is true only when $\gamma > n$ and $n > 1$. If, however, n is greater than γ, gas will be receiving heat during compression.

(b) *Relationship between p, v and T* : The relationship between p, v and T for polytropic process may be expressed by changing the index n for γ in the relationship between p, v and T for the adiabatic compression (as given in eqn. 2.18), i.e. $$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} ; \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{n} \qquad .. (2.22)$$

where p_1, v_1 T_1 refer to initial state and p_2, v_2, T_2 refer to the final state for polytropic expansion or compression.

Determination of the value of n

If the initial and final volumes and pressures are known for an expansion of the form $PV^n = C$, the value of n can be obtained from the logarithms of these values.

Now, $p_1 v_1^n = p_2 v_2^n$

Taking logs of both sides of this equation,

$$\log p_1 + n \log v_1 = p_2 + n \log v_2$$

$$\therefore\ n \log v_2 - n \log v_1 = \log p_1 - \log p_2$$

$$\therefore\ n (\log v_2 - \log v_1) = \log p_1 - \log p_2$$

$$\therefore\ n = \frac{\log p_1 - \log p_2}{\log v_2 - \log v_1} = \frac{\log \frac{p_1}{p_2}}{\log \frac{v_2}{v_1}} \qquad .. (2.23)$$

For an adiabatic expansion, the value of γ may be obtained from equation (2.23) by substituting γ for n.

Problem-16 : *0·3 m^3 of air of mass 1 kg at an initial pressure of 5·5 bar expands to a final volume of 0·5 m^3. If the expansoin is according to the law $pv^{1.3} = C$, find the work done, the change in internal energy and heat received or rejected during the process.*

Assume k_v = 0·708 kJ/kg K and R = 0·287 kJ/kg K for air.

Given : p_1 = 5·5 bar = 550 kPa, v_1 = 0·3 m^3, v_2 = 0·5 m^3, n = 1·3, R = 0·287 kJ/kg K, and k_v = 0·708 kJ/kg K.

Referring to fig. 2-10, and considering polytropic expansion 1–2,

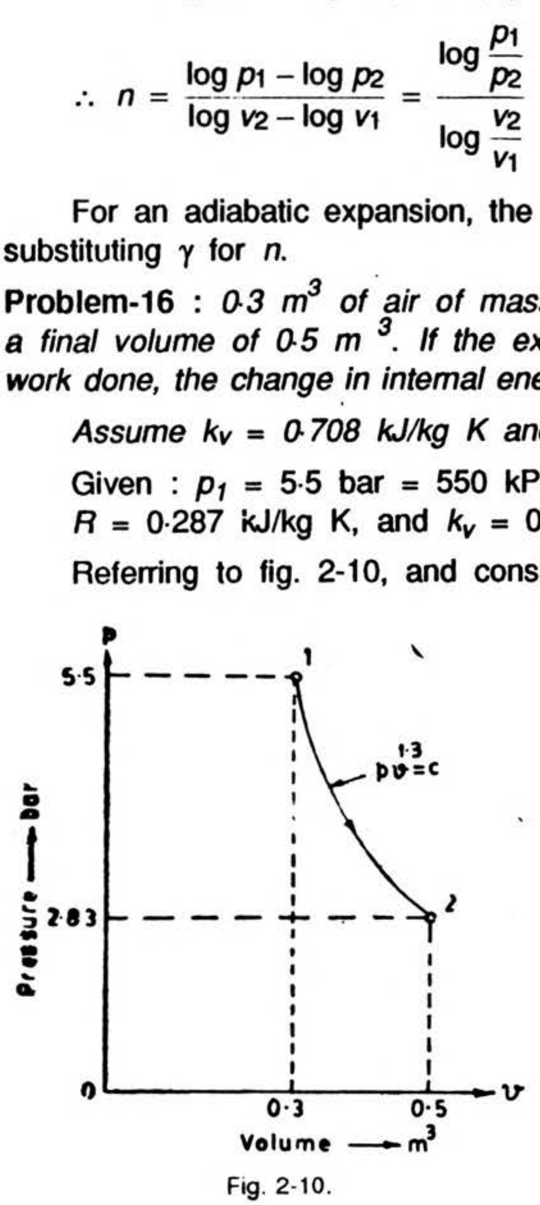


Fig. 2-10.

$$p_2 = p_1\left(\frac{v_1}{v_2}\right)^{n} = 550\left(\frac{0{\cdot}3}{0{\cdot}5}\right)^{1{\cdot}3} = \mathbf{283\ kPa.}$$

Work done during the polytropic expansion 1-2,

$$W \approx \frac{p_1 v_1 - p_2 v_2}{n-1}$$

$$= \frac{550 \times 0{\cdot}3 - 283 \times 0{\cdot}5}{1{\cdot}3 - 1} = \mathbf{78{\cdot}3\ kJ}$$

$$T_1 = \frac{p_1 v_1}{mR} = \frac{550 \times 0{\cdot}3}{1 \times 0{\cdot}287} = 574{\cdot}9\ \text{K}$$

and $$T_2 = \frac{p_2 v_2}{mR} = \frac{283 \times 0{\cdot}5}{1 \times 0{\cdot}287} = 493\ \text{K}$$

Change in internal energy, $u_2 - u_1$

$$= m k_v (T_2 - T_1)$$

$$= 1 \times 0{\cdot}708\ (493 - 574{\cdot}9) = \mathbf{-\ 58\ kJ.}$$

The negative sign indicates that the internal energy has decreased as a result of external work done by the air during the expansion.

Heat transferred, $Q = (u_2 - u_1) + W = -58 + 78{\cdot}3 = 20{\cdot}3$ kJ.

i.e. heat received by the air during expansion is **20·3 kJ.**

Problem-17 : *One kg of gas is compressed polytropically from a pressure of 150 kPa and 290 K to 750 kPa. If the compression is according to the law* $pv^{1.3} = C$, *find : (a) the initial volume, (b) the final temperature, (c) the work done, (d) the change in internal energy, (e) the heat received or rejected during the process, and (f) the change in enthalpy.*

Take $R = 0{\cdot}28$ kJ/kg K *and* $k_p = 1{\cdot}001$ kJ/kg K *for the gas.*

(a) From the characteristic gas equation $p_1v_1 = mRT_1$,

$$v_1 = \frac{mRT_1}{p_1} = \frac{1 \times 0{\cdot}28 \times 290}{150} = \mathbf{0{\cdot}541\ m^3}$$

(b) Considering polytropic compression 1-2 (fig. 2-9),

$$T_2 = T_1 \left\{\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 290\left(\frac{750}{150}\right)^{\frac{1{\cdot}3-1}{1{\cdot}3}} = 290 \times 1{\cdot}45 = 420{\cdot}5\ \text{K or } t_2 = \mathbf{147{\cdot}5^\circ C.}$$

(c) The work done during polytropic compression 1-2,

$$W = \frac{p_1v_1 - p_2v_2}{n-1} = \frac{mR(T_1 - T_2)}{n-1}$$

$$= \frac{1 \times 0{\cdot}28 \times (290 - 420{\cdot}5)}{1{\cdot}3 - 1} = -\mathbf{121{\cdot}83\ kJ}\ \text{(work is done on the gas)}$$

(d) $k_v = k_p - R = 1{\cdot}001 - 0{\cdot}28 = 0{\cdot}721$ kJ/kg K,

Change of internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1)$

$= 1 \times 0{\cdot}721 \times (420{\cdot}5 - 290) = \mathbf{94{\cdot}09\ kJ.}$

(e) Transferred heat, $Q = (u_2 - u_1) + W = 94{\cdot}09 + (-121{\cdot}83) = -\mathbf{27{\cdot}74\ kJ.}$

Negative sign indicates that 27·74 kJ are rejected by the gas.

(f) Change in enthalpy, $H_2 - H_1 = mk_p\,(T_2 - T_1)$

$= 1 \times 1{\cdot}001 \times (420{\cdot}5 - 290) = \mathbf{130{\cdot}63\ kJ}$

Problem-18 : *A cylinder contains 0·25* m^3 *of gas at 100 kPa and 100°C. The gas is compressed according to the law* $pv^n = C$ *to a volume of 0·05* m^3, *the final pressure being 750 kPa. Determine : (a) the value of index "n" for the compression, (b) the increase in internal energy, and (c) the heat received or rejected by the gas during the compression process. Take* $\gamma = 1{\cdot}4$ *and* $R = 0{\cdot}298$ kJ/kg K *for the gas.*

(a) Considering polytropic compression 1-2,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n \text{ i.e. } \frac{750}{100} = \left(\frac{0{\cdot}25}{0{\cdot}05}\right)^n \quad \text{or} \quad 7{\cdot}5 = (5)^n \text{ or } \log(7{\cdot}5) = n\log(5)$$

$$\therefore\ n = \frac{\log(7{\cdot}5)}{\log(5)} = \frac{0{\cdot}8751}{0{\cdot}699} = \mathbf{1{\cdot}252}$$

(b) $\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1}$ $\quad \therefore\ T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1}$

$$\therefore\ T_2 = 373\left(\frac{0{\cdot}25}{0{\cdot}05}\right)^{1{\cdot}252-1} = 373 \times 1{\cdot}5 = 559{\cdot}5\ \text{K}$$

As $p_1v_1 = mRT_1$, $\quad m = \frac{p_1v_1}{RT_1} = \frac{100 \times 0{\cdot}25}{0{\cdot}298 \times 373} = 0{\cdot}225$ kg

Combining the relations $\gamma = k_p/k_v$ and $R = k_p - k_v$, we obtain,

$$k_v = \frac{R}{\gamma - 1} = \frac{0{\cdot}298}{0{\cdot}4} = 0{\cdot}745\ \text{kJ/kg K}$$

Increase in internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1)$

$$= 0{\cdot}225 \times 0{\cdot}745\ (559{\cdot}5 - 373) = \mathbf{31{\cdot}26\ kJ}$$

(c) Work done during polytropic compression 1-2,

$$W = \frac{p_1v_1 - p_2v_2}{n-1} = \frac{100 \times 0{\cdot}25 - 750 \times 0{\cdot}05}{1{\cdot}252 - 1} = -\frac{12{\cdot}5}{0{\cdot}252}$$

$= -49{\cdot}61$ kJ (work done on the gas).

Heat transferred, $Q = (u_2 - u_1) + W = 31{\cdot}26 + (-49{\cdot}61) = -$ **18·35 kJ** (rejected).

Problem-19 : *Two kilograms of air at a presure of 875 kPa occupy a volume of 0·35 m^3. This air is then expanded to a volume of 2 m^3. Find the final temperature, the work done, and the heat received or rejected by this air for each of the following methods of expansion : (a) at constant pressure, (b) isothermally, (c) adiabatically, and (d) according to the law $PV^{1.25}$= constant. Assume R = 0.287 kJ/kg K, k_v= 0·718 kJ/kg K and γ = 1·4 for air.*

(a) *At constant pressure :* $p_1 = p_2 = 875$ kPa

As $p_1v_1 = mRT_1$, $\quad T_1 = \frac{p_1v_1}{mR} = \frac{875 \times 0{\cdot}35}{2 \times 0{\cdot}287} = 533{\cdot}5$ K

As $p_2v_2 = mRT_2$, $T_2 = \frac{p_2v_2}{mR} = \frac{875 \times 2}{2 \times 0{\cdot}287} = 3{,}048{\cdot}6$ K or t_2 = **2,775·6° C**

Work done, $W = p_1(v_2 - v_1) = 875\ (2 - 0.35) =$ **1,443·75 kJ.**

Change in internal energy, $u_2 - u_1 = mk_v\,(T_2 - ,T_1)$

$$= 2 \times 0{\cdot}718\ (3{,}048{\cdot}6 - 533{\cdot}5) = 3{,}611{\cdot}7\ \text{kJ}$$

Now, heat transferred, $Q = (u_2 - u_1) +$ W $= 3{,}611{\cdot}7 + 1{,}443{\cdot}75 =$ **5,055.45 kJ.**

i.e. Heat received by air during expansion is 5,055·45 kJ.

(b) *Isothermally :* $T_2 = T_1 = 533{\cdot}5$ K or t_2 = **260·5°C**

As the temperature remains constant, change in internal energy, $u_2 - u_1 = 0$.

Work done, $W = p_1v_1 \log_e\left(\frac{v_2}{v_1}\right) = 875 \times 0{\cdot}35 \log_e\left(\frac{2}{0{\cdot}35}\right)$

$$= 875 \times 0{\cdot}35 \times 1{\cdot}744 = \mathbf{534\ kJ}$$

Heat transferred, $Q = (u_1 - u_2) + W = 0 + 534 =$ **534 kJ.**

i.e. Heat received by air during expansion is 534 kJ.

1 × 0·708 (493 − 574·9) = − **58 kJ.**

(c) *Adiabatically i.e.* $pv^{\gamma} = C$: As $p_1v_1^{\gamma} = p_2v_2^{\gamma}$

$$p_2 = \frac{p_1}{\left(\frac{v_2}{v_1}\right)^{\gamma}} = \frac{875}{\left(\frac{2}{0{\cdot}35}\right)^{1{\cdot}4}} = \frac{875}{11{\cdot}48} = 76{\cdot}2 \text{ kPa}$$

Now, $p_2v_2 = mRT_2 \quad \therefore \; T_2 = \frac{p_2v_2}{mR} = \frac{(76{\cdot}2) \times 2}{2 \times 0{\cdot}287} = 265{\cdot}5 \text{ K}$ or $t_2 = -7{\cdot}5°\text{C}$

Work done, $W = \frac{p_1v_1 - p_2v_2}{\gamma - 1} = \frac{(875)\,0{\cdot}35 - (76{\cdot}2)\,2}{1{\cdot}4 - 1} = \mathbf{384{\cdot}6 \text{ kJ}}.$

During adiabatic expansion, **no heat is received or rejected by air.**

(d) *According to the law* $pv^{1.25}$ *= constant (Polytropically) : As* $p_1v_1^n = p_2v_2^n$,

$$p_2 = \frac{p_1}{\left(\frac{v_2}{v_1}\right)^{1{\cdot}25}} = \frac{875}{\left(\frac{2}{0{\cdot}35}\right)^{1{\cdot}25}} = \frac{875}{8{\cdot}835} = 99 \text{ kPa}$$

Now, $p_2v_2 = mRT_2 \quad \therefore \; T_2 = \frac{p_2v_2}{mR} = \frac{99 \times 2}{2 \times 0{\cdot}287} = 345 \text{ K}$ or $t_2 = 72°\text{C}$.

Work done, $W = \frac{p_1v_1 - p_2v_2}{n - 1} = \frac{875 \times 0{\cdot}35 - 99 \times 2}{1{\cdot}25 - 1} = \mathbf{433 \text{ kJ}}.$

During the expansion, work will be done by the gas and is positive.

Change in internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1) = 2 \times 0{\cdot}718\,(345 - 533{\cdot}5)$
$= -\,270{\cdot}7$ kJ.

Negative sign indicates decrease in internal energy.

Heat transferred, $Q = (u_2 - u_1) + W = -\,270{\cdot}7 + 433 = \mathbf{162{\cdot}3 \text{ kJ}}$

i.e. Heat received by air during expansion is **162·3 kJ.**

Problem-20 : *Prove that the transferred heat during a polytropic process,* pv^n = *constant, is given by* $Q = mk_n(T_2 - T_1)$, *where* $k_n = k_v\,\frac{n-\gamma}{n-1}$.

From the first law, expression

$$Q = W + (u_2 - u_1) = \frac{p_1v_1 - p_2v_2}{(n-1)} + mk_v(T_2 - T_1)$$

$$= \frac{mR\,(T_1 - T_2)}{(n-1)} + mk_v(T_2 - T_1) = m\left\{k_v - \frac{R}{(n-1)}\right\}(T_2 - T_1)$$

But, $R = k_v\,(\gamma - 1)$

$$\therefore\; Q = \text{m}\left\{k_v - \frac{k_v(\gamma - 1)}{n-1}\right\}(T_2 - T_1) = mk_v\left(1 - \frac{\gamma - 1}{n - 1}\right)(T_2 - T_1)$$

$$= mk_v\frac{n-\gamma}{n-1}\,(T_2 - T_1)$$

Putting, $k_v\,\frac{n-\gamma}{n-1} = k_n$, Transferred heat, $Q = mk_n\,(T_2 - T_1)$.

Problem-21 : *A certain quantity of air in a closed vessel of fixed volume of 0·14 m^3, exerts a pressure of 10 bar at 250°C. If the vessel is cooled so that the pressure falls to 3·6 bar, determine : (a) the final temperature of air, (b) the change in internal energy, and (c) the heat transferred. Take k_v = 0·712 kJ/kg K and R = 0·287 kJ/kg K for air.*

(a) Applying the state equation, $p_1v_1 = mRT_1$.

$$\therefore\ m = \frac{p_1v_1}{RT} = \frac{1{,}000 \times 0{\cdot}14}{0{\cdot}287 \times (250 + 273)} = 0{\cdot}933 \text{ kg.}$$

As this is constant volume process, $v_1 = v_2$ and hence, the final temperature T_2 is

$$T_2 = \frac{p_2}{p_1} \times T_1 = \frac{360}{1{,}000} \times 523 = 188{\cdot}3 \text{ K or } t_2 = -84{\cdot}7^\circ\text{C}$$

(b) Change in internal energy $= mk_v\,(T_2 - T_1) = 0{\cdot}993 \times 0{\cdot}712\ (1{\cdot}883 - 523)$

$$= 0{\cdot}933 \times 0{\cdot}712 \times (-\ 334{\cdot}7) = -\mathbf{222{\cdot}3\ kJ.}$$

Negative sign signifies decrease in internal energy.

(c) As there is no work done during this constant volume process, the *heat transferred* is equal to change in internal energy, i.e. **– 222·3 kJ**. In this case the negative sign signifies rejection of heat.

2.7 Curves for Isothermal, Polytropic and Adiabatic Expansion

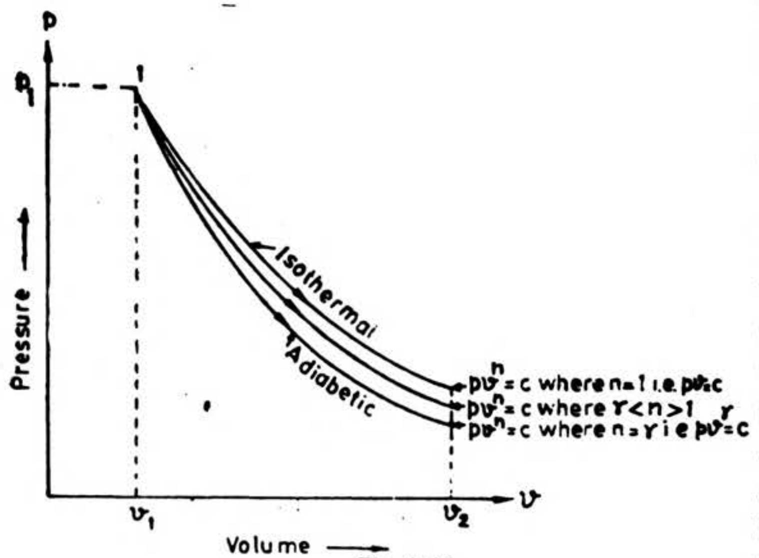


Fig. 2-11.

If the expansion curve representing the expansion, pv^n = constant is plotted on a $p - v$ diagram, the slope of the curve will depend upon the value of the index *n*. Larger the value of *n*, the steeper will be the curve.

For the expansion curve of an isothermal or hyperbolic expansion of air, the law is pv = constant, the index *n* is equal to unity. For the expansion curve of an adiabatic expansion of air, the law is pv^γ = constant; as the value of γ for air is approximately equal to 1·4, the adiabatic curve will be steeper than the isothermal. For the expansion curve of a polytropic process, the law is $pv^n = C$. When the value of *n* is more than unity and less than k_p/k_v, the expansion curve will be steeper than the isothermal and lie in between the isothermal and adiabatic curves.

The expansion curves for isothermal, adiabatic and polytropic ($pv^n = C$) when *n* is more than unity and less than k_p/k_v processes are shown plotted in fig. 2-11. In each case the same mass of air has been expanded from a volume of v_1 to v_2. Area under these curves represent the work done during expansion. It will be noticed that more work is done when the expansion is isothermal.

2.8 Throttling Process

This process occurs when a gas or vapour is expanded through a restricted passage such as a narrow throat or a partly opened value in the line of flow. When the fluid passes through a narrow aperture or opening, there is an appreciable drop of pressure of the fluid. The pressure drop should result in an increase in the kinetic energy of the fluid stream as it passes through the restricted passage. However, the opening is so narrow that the frictional resistance between the fluid and surface of the opening reduces the fluid velocity on leaving to a negligible amount. Consequently, the fluid escapes through the narrow opening with only a small amount of kinetic energy. The friction converts the kinetic energy into heat, which results in warming of the fluid to its original temperature. An expansion like this is termed *throttling.*

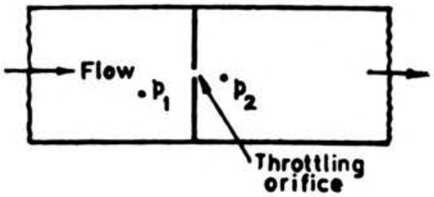


Fig. 2-12.

Figure 2-12 shows a fluid flowing along a pipe from a higher pressure p_1 to a lower pressure p_2 and passing through a small orifice.

During the throttling process there is no external work done and no transfer of heat occurs, and in the case of a perfect gas there is no change in temperature. The enthalpy or total heat of the fluid must, therefore, remain constant.

Problem - 22 : *A vessel of volume* 0·28 m^3 *initially contains air at 1·05 bar and 5°C. A valve is opened to admit air from a line in which the pressure and temperature are respectively 3·5 bar and 60°C and is closed when the vessel pressure has risen to 2·8 bar.*

Assuming the process to be adiabatic and the air to be a perfect gas with $\gamma = 1{\cdot}4$ *and* $R = 0{\cdot}287$ *kJ/kg K, find : (i) the initial mass of air in the vessel, (ii) the mass of air admitted, and (iii) the final temperature in the vessel.*

(i) Initial mass of air, $m_1 = \dfrac{p_1 v_1}{RT_1} = \dfrac{105 \times 0{\cdot}28}{0{\cdot}287 \times (273+5)}$ = **0·3685 kg.**

(ii) $k_v = \dfrac{R}{(\gamma - 1)} = \dfrac{0{\cdot}287}{(1{\cdot}4 - 1)} = 0{\cdot}7175$ and $k_p = \gamma k_v = 1{\cdot}4 \times 0{\cdot}7175 = 1{\cdot}0045$ kJ/kgK.

The process of charging the vessel is a combination of non-flow and flow processes. *The process is adiabatic.* Hence, there is no heat transfer across the boundaries of the system. As *volume is constant,* no external work is done.

During the process 1-3, mass of air is added to the system. Hence, initial mass (at point 1) and the final mass (at point 3) are different, as indicated at points 1 and 3 in fig. 2-13.

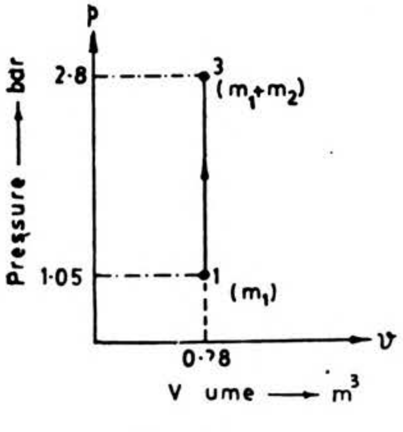


Fig. 2-13.

Let 1 - represents initial condition,

2 - represents condition of air flowing, and

3 - represents final condition.

Then, energy equation can be written as,

$$m_1u_1 + m_2h_2 = m_3u_3$$

Taking absolute zero as datum for internal energy,

$$m_1k_vT_1 + m_2k_pT_2 = (m_1 + m_2)k_vT_3$$

i.e. $0{\cdot}3685 \times 0{\cdot}7175 \times 278 + m_2 \times 1{\cdot}0045 \times 333$

$= (m_1 + m_2) \times 0{\cdot}7175 \times T_3$

But, $m_3 = m_1 + m_2 = \dfrac{p_3 v_3}{RT_3} = \dfrac{280 \times 0{\cdot}28}{0{\cdot}287 \times T_3} = \dfrac{273}{T_3}$

$$\therefore\ 73{\cdot}5 + 334{\cdot}5\ m_2 = \frac{273}{T_3} \times 0{\cdot}7175 \times T_3 = 195{\cdot}9$$

$$\therefore\ m_2 = \frac{195{\cdot}9 - 73{\cdot}5}{334{\cdot}5} = \mathbf{0{\cdot}366\ kg.}$$

(iii) $m_3 = m_1 + m_2 = 0{\cdot}3685 + 0{\cdot}366 = 0{\cdot}7345$ kg,

and $T_3 = \dfrac{273}{m_3} = \dfrac{273}{0{\cdot}7345} = 371{\cdot}7$ K or $t_3 = \mathbf{98{\cdot}7^\circ C}$ (final temperature).

Problem-23 : *Atmospheric air at 102 kPa and 27°C is compressed to a pressure of 300 kPa, the law of compression being* $pv^{1{\cdot}3}$ = *constant, and stored in a vessel of volume 0·6* m^3. *Now a valve on a line, connecting source of air at 500 kPa and 100°C and the vessel, is opened and 0·6 kg of air is allowed to flow into the vessel. Determine : (a) the temperature at the end of polytropic compression, (b) the initial mass and volume of air, (c) the change in internal energy during compression, and (d) the final temperature and pressure of air at the end of air flow into the vessel, assuming air flow operation to be adiabatic. Assume air to be a perfect gas for which* $\gamma = 1{\cdot}4$ *and* $R = 0{\cdot}287$ *kJ/kg K.*

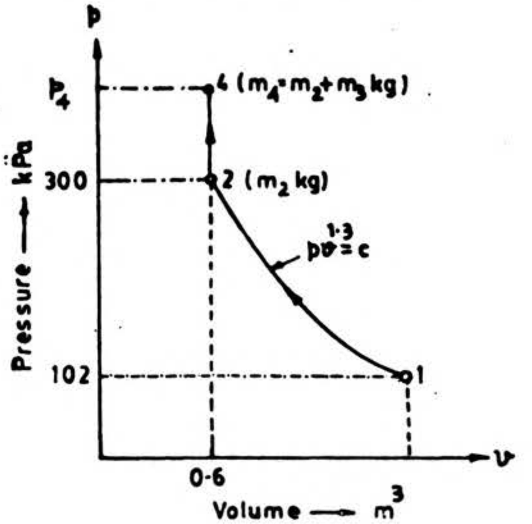


Fig. 2·14

The polytropic compression and air flow operation are shown by 1-2 and 2-4 processes repsectively in fig. 2-14.

(a) For process 1-2, $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\dfrac{300}{102}\right)^{\frac{1{\cdot}3-1}{1{\cdot}3}} = (2{\cdot}94)^{0{\cdot}231} = 1{\cdot}2825$

$$\therefore\ T_2 = (27 + 273) \times 1{\cdot}2825 = 300 \times 1{\cdot}2825 = 385 \text{ K or } t_2 = \mathbf{112^\circ C}$$

(b) Mass of air compressed and stored $m_1 = m_2$ kg or initial mass of air can be obtained from

$p_2 v_2 = m_2 R T_2$

i.e. $m_2 = \dfrac{300 \times 0{\cdot}6}{0{\cdot}287 \times 385} = \mathbf{1{\cdot}629\ kg.}$

Initial volume (v_1) may be obtained from $p_1 v_1 = m_1 R T_1$

i.e. $v_1 = \dfrac{1{\cdot}629 \times 0{\cdot}287 \times 300}{102} = \mathbf{1{\cdot}375\ m^3.}$

(c) $k_v \doteq \dfrac{R}{\gamma - 1} = \dfrac{0{\cdot}287}{1{\cdot}4 - 1} = 0{\cdot}7175$ kJ/kg K and

$k_p = \gamma\ k_v = 1{\cdot}4 \times 0{\cdot}7175 = 1{\cdot}0045$ kJ/kg K.

Change in internal energy during polytropic compression is given by

$$u_2 - u_1 = m_1 k_v (T_2 - T_1) = 1{\cdot}629 \times 0{\cdot}7175 \times (385 - 300) = \mathbf{99{\cdot}35\ kJ}$$

(d) During the operation of air admission 2-4, work and heat transfer are absent.

For a combination of non-flow and flow process,

Internal energy at 2 + Enthalpy of admitted air = Internal energy at 4

Thus, $m_2u_2 + m_3h_3 = m_4u_4$

where 2 and 4 represent initial and final conditions of operations, and 3 represents condition of air flowing.

Taking absolute zero as datum for internal energy,

$$m_2k_vT_2 + m_3k_pT_3 = (m_2 + m_3)k_vT_4$$

i.e. $1{\cdot}629 \times 0{\cdot}7175 \times 385 + 0{\cdot}6 \times 1{\cdot}0045 \times 373 = (1{\cdot}629 + 0{\cdot}6) \times 0{\cdot}7175 \times T_4$

$$\therefore\ T_4 = \frac{674{\cdot}8}{1{\cdot}5993} = 422 \text{ K or } t_4 = 149^\circ\text{C}.$$

Now, $m_4 = 1{\cdot}629 + 0{\cdot}6 = 2{\cdot}229$ kg.

Applying the characteristic equation for ideal gas at point 4,

$$p_4 = \frac{m_4RT_4}{v_4} = \frac{2{\cdot}229 \times 0{\cdot}287 \times 422}{0{\cdot}6} = 450 \text{ kPa.}$$

2.9 Collection of the Formulae

Characteristic equation of perfect gas, $pv = mRT$

Adiabatic index, $\gamma = k_p / k_v$; Gas constant, $R = k_p - k_v$

The polytropic index n or adiabatic index γ is given by,

$$n \text{ or } \gamma = \frac{\log (p_1/p_2)}{\log (v_2/v_1)} \qquad \text{[eqn. (2·23)]}$$

Expressions for different items for various thermodynamic processes are given in the following table 2-1.

Table 2-1 Formulae for Perfect Gas Processes

Type of process	Constant volume $v = C$	Constant Pressure $p = C$	Isothermal $T = C$	Isentropic or Frictionless Adiabatic $pv^\gamma = C$	Polytropic $pv^n = C$
p, v and T relations	$\frac{T_2}{T_1} = \frac{p_2}{p_1}$	$\frac{T_2}{T_1} = \frac{v_2}{v_1}$	$p_1v_1 = p_2v_2$ $T_1 = T_2$	$p_1v_1^\gamma = p_2v_2^\gamma$ $\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1}$ $\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$	$p_1v_1^n = p_2v_2^n$ $\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{n-1}$ $\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}$
Work done, W kJ/kg	0	$p\,(v_2 - v_1)$	$p_1v_1 \log_e \frac{v_2}{v_1}$	$\frac{p_1v_1 - p_2v_2}{(\gamma - 1)}$	$\frac{p_1v_1 - p_2v_2}{(n - 1)}$
Change in Internal Energy, $u_2 - u_1$ kJ/kg	$k_v\,(T_2 - T_1)$	$k_v\,(T_2 - T_1)$	0	$k_v\,(T_2 - T_1)$	$k_v\,(T_2 - T_1)$
Transferred Heat, Q kJ/kg	$k_v\,(T_2 - T_1)$	$k_p\,(T_2 - T_1)$	$p_1v_1 \log_e \frac{v_2}{v_1}$	0	$\frac{p_1v_1 - p_2v_2}{(n - 1)} \times \frac{\gamma - n}{\gamma - 1}$
Change in Enthalpy (total heat), kJ/kg	$k_p\,(T_2 - T_1)$	$k_p\,(T_2 - T_1)$	0	$k_p\,(T_2 - T_1)$	$k_p\,(T_2 - T_1)$

Note : p_1, v_1 and T_1 refere to initial state and p_2, v_2 and T_2 to the final state of the gas.

Tutorial - 2

1. (a) What is a gas process ? Classify and explain basic gas processes. How are they shown graphically on a $p - v$ plane ?
   (b) What is the significance of the area under a process line on $p - v$ plane ?
2. Justify the following statements :
   (i) "Isothermal process of an ideal gas is also the hyperbolic process".
   (ii) "The processes - isochoric, isobaric, isothermal and isentropic are particular cases of the polytropic process".
3. Find the values of k_p and γ for a gas whose gas constant is 0·324 kJ/kg K and K_v = 0·84 kJ/kg K. If 2·25 kg of this gas with an initial volume of 1·15 m^3 undergoes a process during which its pressure remains constant at 7 bar and its temperature becomes 280°C at the end of the process, calculate : (a) the change in internal energy, (b) the change in enthalpy, (c) the transferred heat, and (d) the work done.
   [1·164 kJ/kg K; 1·3857; (a) − 1041·86 kJ; (b) − 1443·72 kJ; (c) − 1443·72 kJ; (d) − 401·86 kJ]
4. Four kg of gas occupying 0·5 m^3 is heated from 60 ° to 115 °C at constant pressure of 774 kPa. Find the gas constant and final volume after heating. If the initial temperature remains unchanged and the volume increases four times its initial volume, what will be the final pressure of the gas ?
   [0·29 kJ/kg K, 0·583 m^3; 193·5 kPa]
5. Two kg of gas at 50 °C is heated at constant volume until the pressure is doubled. Determine : (a) the final temperature, (b) the change in internal energy, and (c) the change in enthalpy. Take k_p = 1·005 kJ/kg K and γ = 1·69 for the gas.
   [(a) 373°C; (b) 384·37 kJ; (c) 649·23 kJ]
6. A certain quantity of air in a closed vessel of fixed volume of 0·15 m^3 exerts a pressure of 1 MPa at 500 °K. If the vessel is cooled so that the pressure falls to 350 kPa, determine : (a) the final temperature of air, (b) the change in internal energy, and (c) the heat transferred. Take k_v = 0·7165 kJ/kg K and R = 0·287 kJ/kg K for air.
   [(a) − 98°C; (b) − 243·33 kJ (decrease); (c) −243·33 kJ (rejected)]
7. (a) Explain the statement "during constant pressure process, heat added is distributed to change in internal energy and work done".
   (b) One kg of air at 150 kPa and 30 °C undergoes a constant pressure process until the volume is trippled. Determine : (i) the change in internal energy, and (ii) the change in enthalpy. Take k_p = 1·0035 kJ/kg K and k_v = 0·7165 kJ/kg K for air.
   [(i) 434·2 kJ; (ii) 608·12 kJ]
8. "Isothermal process of a perfect gas is also a hyperbolic process". Justify the statement.
   The pressure of 0·15 m^3 of air increases from 500 kPa to 1·5 MPa while undergoing isothermal compression. Determine the value of transferred heat.
   [− 82·4 kJ (rejected)]
9. A quantity of gas is expanded isothermally from initial condition of 0·1 m^3 and 735 kPa, to a final pressure of 118 kPa. Find : (a) the final volume, (b) the work done, and (c) the heat supplied during the expansion process.
   [(a) 0·623 m^3; (b) 134·55 kJ; (c) 134·55 kJ]
10. When 2·25 kg of a perfect gas at an initial temperature of 400 °C is heated at constant pressure, 523 kJ are added to it. If the gas has a value of R = 2·079 kJ/kg K and γ = 1·66, calculate : (a) the final temperature, (b) the change in enthalpy, (c) the change in internal energy, and (d) the work done.
    [(a) 84·4°C; (b) 523 kJ; (c) 315 kJ; (d) 208 kJ]
11. State the difference between an adiabatic process and an isothermal process.
    A perfect gas is compressed adiabatically from a state of 1·93 MPa, volume 0·1 m^3 and temperature −4°C, to a pressure of 5.84 MPa. Find : (a) temperature and volume at the end of compression, and (b) the change in enthalpy. Take for gas γ = 1·4, k_p = 1·005 kJ/kg K and k_v = 0·718 kJ/kg K.
    [(a) 96°C, 0·0454 m^3; (b) 251·24 kJ]
12. Prove that the index γ in pv^{γ} = constant for adiabatic expansion of a gas, is the ratio of the specific heat at constant pressure to the specific heat at constant volume.
    0·12 m^3 of air at 1·5 MPa and 1,500°C expands adiabatically to pressure of 175 kPa. Find : (a) the final temperature, and (b) the work done. Take k_p = 1·0035 kJ/kg K and k_v = 0·7165 kJ/kg K for air.
    [(a) 686·4°C; (b) 206·75 kJ]
13. In the characteristic equation $pv = RT$ for unit mass of a gas, prove that, R is the difference between the specific heats at constant pressure and at constant volume.
    0·5 kg of air at 179 °C expands adiabatically to three times its original volume and in the process falls in temperature from 179 °C to 18 °C. The work done during expansion is 57 kJ. Determine the specific heats of air at constant pressure and at constant volume.
    [k_p = 0·9917 kJ/kg K, k_v = 0·7081 kJ/kg K]

14. (a) State what do you understand by :
(i) an isothermal process, and (ii) an adiabatic process.
(b)Two kg of air at a pressure of 700 kPa occupies a volume of 0·3 m^3. This air is expanded adiabatically to a volume of 1·5 m^3. Find : (a) the final temperature and pressure, (b) the work done, and (c) the heat transferred. Take k_p = 1·0035 kJ/kg K, and k_v = 0·7165 kJ/kg K for air.

[(a) – 81°C, 73·5 kPa; (b) 249·25 kJ; (c) No heat transferred]

15. Show that if a gas is expanded according to the law pv^n = constant from initial condition p_1, v_1 to final condition p_2, v_2, the work done by the gas $= \dfrac{p_1v_1 - p_2v_2}{n-1}$.
1·5 kilograms of a gas at 93·2 kPa and 30°C are compressed according to the law $pv^{1.32}$ = C. If the transferred heat is – 75 kJ, find : (a) the final temperature and pressure, (b) the change in internal energy and enthalpy, and (c) the work done.
Take γ = 1·4 and k_v = 0·712 kJ/kg K for the gas.

[(a) 311°C, 1395·2 kPa; (b) 300·11 kJ, 420·15 kJ; (c) 375·14 kJ]

6. What is the significance of the area under a process line of perfect gas on $p - v$ diagram ?

7. Show that if a quantity of gas expands according to the law pv^n = C, then the heat supplied during expansion is given by $Q = \dfrac{\gamma - n}{\gamma - 1} \times W$ kJ
where γ is the adiabatic index and W is the work done in kJ.
The pressure and temperature of the air in the cylinder are 93·5 kPa and 45°C. The air is compressed according to the law $pv^{1.24}$ = constant until the pressure is 640 kPa. The volume of air initially is 0·035 m^3. Find : (a) the mass of air in the cylinder, (b) the temperature of air at the end of compression, (c) the work done on the air during compression, and (d) the heat rejected by the air during compression. Take γ = 1·4 and R = 0·287 kJ/kg K for air.

[(a) 0·0357 kg; (b) 188°C; (c) – 6·11 kJ; (d) – 2·45 kJ]

18. Derive an expression for interchange of heat from the cylinder walls to the gas undergoing expansion according to the law pv^n = constant.
A gas initially at 603 K expands until its volume is 5·2 times the initial volume according to the law pv^n = constant. If the intial and final pressures are observed to be 834 kPa and 93·5 kPa respectively, and assuming that the specific heat of gas at constant volume is 0·711 kJ/kg K and that the ratio of specific heats is 1·41, calculate : (a) the value of index n, (b) the work done per kilogram of gas, and (c) the heat received by gas per kilogram of gas.

[(a) 1·33; (b) 224 kJ; (c) 43·6 kJ]

19. A quantity of oxygen occupies 0·9 m^3 at 5 °C and 147 kPa. It is compressed according to law $pv^{1.25}$ = constant until the pressure is 2,060 kPa. If the gas constant is 0·26 kJ per kg per degree Kelvin, determine : (a) the final volume and temperature of the oxygen, (b) the heat interchange between the oxygen and its surroundings, stating whether this is a heat reception or rejection. Take k_v for oxygen = 0·712 kJ/kg K.

[(a) 0·109 m^3, 198·2°C; (b) – 116 kJ (rejected)]

20. 0·3 m^3 of gas at 1375 kPa and 15°C expands according to the law $pv^{1.36}$ = constant until the pressure is 345 kPa. Working from first principles, find the heat supplied to the gas during expansion. Take R for gas = 0·647 kJ/kg K and the ratio of specific heats γ = 1·39.

[27 kJ]

21. What are basic gas processes ? How are they shown graphically on $p - v$ diagram ?
Two kilograms of air at a pressure of 690 kPa occupy a volume of 0·3 m^3. This air is then expanded to a volume of 1·5 m^3. Find the final temperature, the work done, and the heat received or rejected by this air for each of the following methods of expansion :
(a) at constant pressure, (b) isothermally, and (c) according to the law $pv^{1.2}$ = constant.
Take R = 0.287 kJ/kg K, k_p = 1·0035 kJ/kg K and k_v = 0·7165 kJ/kg K for air,

[(a) 1,530·1°C, 828 kJ, 2,895·1 kJ received; (b) 87·6°C, 333·15 kJ, 333·15 kJ received;
(c) – 11·7°C, 285 kJ, 142·7 kJ received.]

22. 1 kg of gas initially at 1,195 kPa and 320°C expands adiabatically until its pressure is 102 kPa. The gas is then heated at constant volume until its temperature becomes 320 °C and its pressure is found to 207 kPa. Finally, the gas is compressed isothermally to the original pressure of 1,195 kPa. Determine : (a) the ratio of specific heats, (b) the temperature of the gas at the end of adiabatic expansion, and (c) the change in internal energy of the gas during adiabatic expansion. Take k_p for gas = 1·005 kJ/kg K.

[(a) 1·4037; (b) 19·2 °C; (c) 215·34 kJ (loss of internal energy)]

23. (a) Prove that specific heat for polytropic process k_n can be expressed as

$$k_n = k_v \cdot \frac{n - \gamma}{n - 1}$$

where k_v is specific heat at constant volume, n is polytropic index, and γ is adiabatic index.

(b) A certain gas, having a volume of 0·075 m^3 at 16 °C and 690 kPa, expands to 0·35 m^3 according to the law $pv^{1.2} = C$. Determine : (a) the final temperature, (b) the work done, and (c) the heat received or rejected during the process. Take $\gamma = 1{\cdot}4$ and $R = 0{\cdot}286$ kJ/kg K for the gas.

[(a) – 60·7 °C; (b) 68·65 kJ; (c) 34·33 kJ (received)]

24. What is an isentropic process ?
One kg of gas occupying 0·15 m^3 at a pressure of 1·5 MPa is expanded at constant pressure to 0·25 m^3. The gas is then expanded adiabatically to 0·9 m^3. Calculate :
(a) the temperature at the end of constant pressure expansion,
(b) the temperature and pressure at the end of adiabatic expansion,
(c) the heat supplied during each portion of the process, and
(d) the total work done during the whole process.
Take $k_p = 1{\cdot}013$ kJ/kg K and $k_v = 0{\cdot}72$ kJ/kg K for gas.

[(a) 1,007°C; (b) 487°C, 247·4 kPa; (c) 518·66 kJ, No heat is supplied during adiabatic expansion; (d) 150 + 374·3 = 524·3 kJ]

25. One kilogram of air initially at 108 kPa and 20°C is compressed through a volume of 6 to 1, to a final pressure of 1,475 kPa. Heat is then added at constant volume to the temperature of 605°C. Determine : (a) the change in internal energy during the compression process, and (b) the amount of heat added at constant volume.
Take k_v for air = 0·712 kJ/kg K.

[(a) 266·29 kJ; (b) 150·23 kJ]

26. One kilogram of nitrogen gas occupying 0·1 m^3 at a pressure of 1,375 kPa is expanded at constant pressure to 0·2 m^3. The gas is then expanded adiabatically to 0·6 m^3. Calculate : (a) the temperature at the end of constant pressure process, (b) the temperature and pressure at the end of adiabatic expansion, and (c) the total work done during the whole process. Take $k_p = 1{\cdot}038$ kJ/kg K and $k_v = 0{\cdot}741$ kJ/kg K for nitrogen.

[(a) 653°C; (b) 323·65°C, 295·5 kPa; (c) 382 kJ]

27. Two kilograms of oxygen are compressed polytropically from a pressure of 148 kPa and 17°C, to 740 kPa. If the compression is according to the law $pv^{1.3}$ = constant, find : (a) the intial volume, the final temperature and the final volume of the gas, (b) the work done, (c) the change in internal energy, and (d) the transferred heat.
Take gas constant $R = 0{\cdot}26$ kJ/kg K and $k_p = 0{\cdot}913$ kJ/kg K for oxygen.

[(a) 0·5095 m^3/kg, 147.5°C, 0·1477 m^3/kg; (b) 226·2 kJ; (c) 170·43 kJ; (d) –55·77 kJ]

28. Five kilograms of sulphur dioxide expand from the initial state $p_1 = 1{,}176$ kPa and $t_1 = 250$ °C, to the final pressure $p_2 = 196$ kPa. Find the final parameters and the work done by the gas if the expansion is (a) isothermal, (b) polytropic (n = 1·2), and (c) isentropic. Take $R = 0{\cdot}13$ kJ/kg K and $k_p = 0{\cdot}645$ kJ/kg K for sulphur dioxide.

Process	Final temp., t_2°C	Final volume, v_2 m^3/kg	Work done, kJ
(a) Isothermal	250	0·3469	608·16
(b) Polytropic	115	0·2573	438·9
(c) Isentropic	92	0·2418	408·16

29. A cylinder fitted with a movable piston contains a quantity of a gas at a presure of 345 kPa and temperature of 90 °C. The piston is allowed to move slowly outwards, no heat being lost or gained by the gas during this process, and at the end of expansion the pressure is 68·7 kPa. The piston is then fixed and heat is added until the gas reaches a state of 90 °C and 109 kPa. Determine the specific heats k_p and k_v of the gas.

[14·4 kJ/kg K, 10·3 kJ/kg K]

30. One kilogram of nitrogen is compressed from the initial state $p_1 = 196$ kPa and $t_1 = 40$°C, to the final pressure $p_2 = 1{,}176$ kPa. Find the final parameters, the work done, the change in internal energy, and the heat transferred if the expansion is isothermal, polytropic (n = 1·25), and isentropic. Take $R = 0{\cdot}297$ kJ/kg K and $k_p = 1{\cdot}038$ kJ/kg K for nitrogen.

Process	Final temp. t_2, °C	Final volume, v_2 m^3/kg	Work done, kJ/kg	Change in internal energy, kJ/kg	Heat transferred, kJ/kg
Isothermal	40	0·079	– 166·8	0	–166·8
Polytropic	175	0·1131	– 160·16	100·03	– 60·13
Isentropic	249	0·1318	– 155·1	155·1	0

31. 0·65 m^3 of hydrogen at 1 bar and 20°C is compressed isentropically to 14 bar and then expanded isothermally to the original volume of 0·65 m^3. Determine :
(a) the temperature of the gas at the end of isentropic compression,

(b) the pressure of the gas at the end of isothermal expansion,
(c) the heat which must be added to the gas during isothermal expansion, and
(d) the heat which must be extracted from the gas at constant volume after isothermal expansion to reduce it to its original state.

Take $R = 4{\cdot}125$ kJ/kg K and $k_p = 14{\cdot}26$ kJ/kg K.

[(a) 356°C; (b) 2·15 bar; (c) 262 kJ; (d) – 183·14 kJ]

32. One m^3 of air at 690 kPa pressure and 50 °C temperature is expanded isothermally, polytropically ($n = 1{\cdot}25$) and isentropically to 4 m^3. Calculate : the pressure and temperature of air at the end of expansion in each case, and the work done in each case. Take $\gamma = 1{\cdot}4$ for air.

Process	Final temp., t_2 °C	Final pressure, p_2 kPa	Work done, kJ
Isothermal	50	172·5	956·55
Polytropic	– 44·56	122	808
Isentropic	– 88	99·1	734·3

33. A certain gas is compressed isentropically until the pressure becomes five times its original value. During the process the temperature increases from 27°C to 202·5°C and 130 kJ of work is done on 1 kg of gas. Find the specific heats k_p and k_v, the gas constant R, and the molecular weight of the gas.

[1·0378 kJ/kg K, 0·7407 kJ/kg K, 0·2971 kJ/kg K, 27·984]

34. A vessel of 0·4 m^3 initially contains air at 1 bar and 17°C. A valve is opened to admit air from a line in which the pressure and temperature are respectively 4 bar and 67°C and is closed when the vessel pressure has risen to 3 bar. Assuming the process to be adiabatic and the air to be a perfect gas for which γ and R are 1·4 and 0·287 kJ/kg K respectively, find : (a) the initial mass of air in the vessel, (b) the mass of air admitted, and (c) the final temperature of air in the vessel.

[(a) 0·48 kg; (b) 0·586 kg; (c) 119°C]

35. One m^3 of atmospheric air at 98 kPa and 17°C is compressed to a pressure of 392 kPa, the law of compression being $pv^{1.25}$ = constant, and stored in a vessel. A valve on a line, connecting a source of compressed air at 784 kPa and 80°C and the vessel, is opened and 0·5 kg of air is allowed to flow into the vessel. Determine : (a) the initial mass of air in the vessel, (b) the volume of the vessel, (c) the temperature at the end of polytropic compression, (d) the change in internal energy during polytropic compression, and (e) the final temperature and pressure of air at the end of air flow into the vessel, assuming this air flow operation to be adiabatic and air to be a perfect gas for which k_p and k_v are 1·005 kJ/kg K and 0·716 kJ/kg K respectively.

[(a) 1·1693 kg; (b) 0·3297 m^3; (c) 109·5°C; (d) 77·44 kJ;
(e) 143·35°C, 609·22 kPa]

36. Explain the throttling process. Why is it essentially an irreversible process ?

3
PROPERTIES OF STEAM

3.1 Introduction

Steam is a vapour. It is used as the working substance in the operation of steam engines and steam turbines. As stated in chapter 2, a vapour is a partially evaporated liquid carrying in it particles of liquid and it can be liquefied by minor changes in temperature or pressure. Steam as a vapour would not obey the laws of perfect gases unless it is in a highly dried condition. Steam in such a dried state is known as ***superheated steam*** and it is assumed to behave like a perfect gas when highly superheated.

Although steam is considered as a perfect gas on account of it being a mixture of dry steam (gas) and moisture (water), it possesses properties like those of gases : namely, pressure, volume, temperature, internal energy, enthalpy and entropy. But the pressure, volume and temperature of steam as a vapour are not connected by any simple relationship such as is expressed by the characteristic equation for a perfect gas.

Properties of steam were first investigated experimentally by Regnault and subsequently investigated by Prof. Callender by carrying out extensive thermodynamic experiments by means of electrical calorimeters. Tables giving the properties of steam in each condition were compiled from study of the equations derived experimentally. These properties are also represented on charts. The quantities tabulated in steam tables are :

- Pressure,
- Temperature,
- Specific Volume,
- Enthalpy, and
- Entropy.

Pressure, temperature and volume can be given their actual absolute values; whereas, enthalpy and entropy are purely relative quantities. They are measured relatively from convenient datum condition and calculated per 1 kg of steam. For steam, datum point is arbitrarily fixed as the condition of the water at 0°C. Thus, the enthalpy, the internal energy and the entropy of water at 0°C are assumed to be zero. All their values measured above this temperature are considered positive and those measured below are taken as negative.

The general conservation of energy equation is applicable to steam in the same manner as it is applicable to gases. The properties of steam and the changes in the properties can be determined by using standard steam tables or steam charts.

3.2 Formation of Steam at Constant Pressure

Consider a cylinder fitted with frictionless piston which may be loaded to any desired pressure p bar as shown in fig. 3-1(a). Now, assume for convenience that there is one kilogram of water initially at temperature 0°C in the cylinder under the piston and the piston exerts a constant pressure p bar. Let the area of the piston be one square metre and the volume of one kilogram of water be v_w m^3, the length of the cylinder occupied by water will be v_w m. Now, let heat be supplied to the water in the cylinder. The temperature of water will rise when sensible enthalpy be supplied. The rise in temperature will continue until the boiling point is reached, where it will remain constant. The temperature at which water boils depends upon the pressure on it. For any given pressure, there is one *definite* boiling point. The boiling point is called the saturation temperature (t_s) or the temperature of steam formation. Water boils at 99·63°C when the pressure on it is 1 bar, and at 184·09°C when the pressure on it is 11 bar (these values are taken from steam tables).

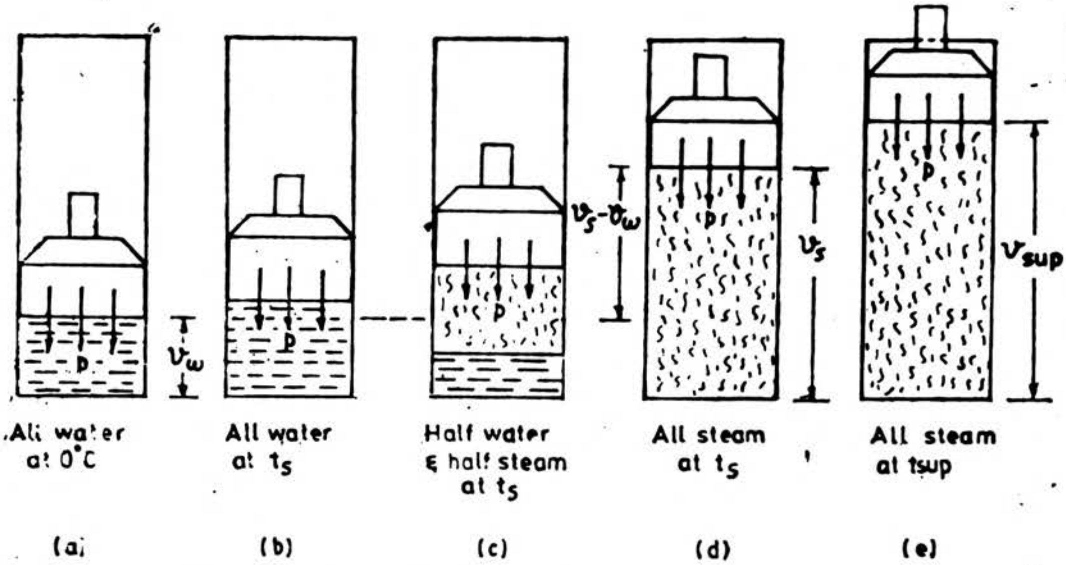


Fig. 3-1. Formation of steam at constant pressure.

The water will expand slightly during the rise of temperature. The increase in volume of water causes the piston to move up slightly as shown in the fig. 3-1(b), thus, work is done in moving the piston against this pressure p. This work, however, is only a small portion of the heat added to water during the rise in temperature and may be neglected in general. Figure 3-1(a) represents the condition before the first stage commences and fig. 3-1(b) represents the condition at the end of the first stage. It will be noted that the piston is at slightly higher level at the end of operation.

The next stage, as shown in fig. 3-1(c), is the actual production of steam. If the heating of this one kilogram of water is continued after the boiling point is reached, it will be noticed that there is no further increase in temperature, as the pressure is maintained constant but steam begins to form and piston commences to ascend (rise) in the cylinder, rising higher and higher as more and more steam is formed. The heat absorbed is now utilised in converting water into steam and is known as *evaporation enthalpy* or *latent heat*. As long as there is some water left unevaporated in the cylinder, the steam formed will not be pure (dry) steam, but will have some water mixed with it. In fig. 3-1(c), part of water is evaporated and the cylinder has in it, mixture of water and steam (about half steam and half water). A mixture of steam and water is called ***wet steam***.

If the heating of this *wet steam* is further continued and as soon as last particle of water in suspension in wet steam disappears (evaporates), the steam produced is known as ***dry saturated steam***.

Figure 3-1(d) shows that the process of formation of steam is completed. The water is entirely evaporated and the volume of steam below the piston is v_s m^3 (v_s is the volume of one kilogram of dry saturated steam at a given pressure p). The piston will rise to accommodate this increased volume, and will occupy length v_s m of the cylinder. The work done on the piston during the change of volume of the steam from v_w to v_s

$$= \frac{10^5\, p\,(v_s - v_w)}{10^3} \text{ kJ} \quad \text{where } p \text{ is the pressure in bar.} \qquad \text{.. (3.1a)}$$

If p is in kPa, then the above expression is written as

$$\text{Work done} = p(v_s - v_w) \text{ kJ} \qquad \text{.. (3.1b)}$$

Finally, if further heat is added to one kilogram of dry steam in the cylinder at the constant pressure, the temperature of steam will rise and there will be further increase in volume of steam as shown in fig. 3-1(e). The steam produced is known as ***superheated steam.***

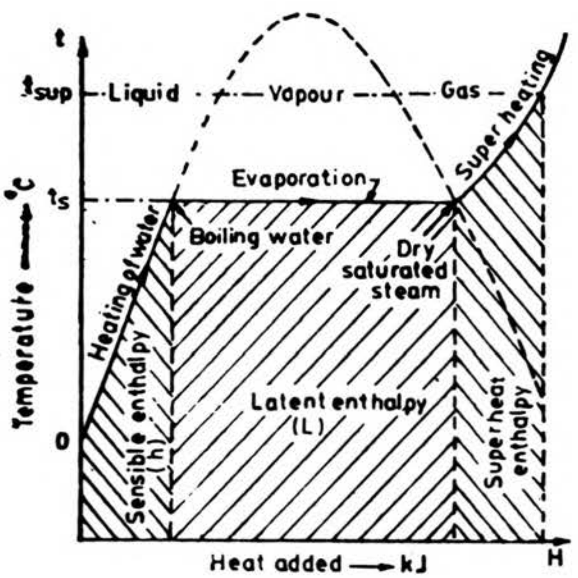


Fig. 3-2. Formation of steam at constant pressure.

Figure 3-2 shows graphically that what happens when heat is added to one kilogram of water initially at 0°C. The heat added is plotted along the horizontal axis in kJ units while the corresponding temperatures are plotted along the vertical axis in centigrade units.

During the first stage of heating, the temperature of water will begin to rise until the water boils at a temperature known as **saturation temperature** which depends upon the pressure in the cylinder. After the boiling temperature is reached, steam begins to be formed during which time the temperature remains constant. Until the point is reached at which all water is converted into steam, the contents of mixture will be steam and water known as ***wet steam.*** When all the water including those particles of water held in suspension will be evaporated, the steam is said to be dry and is known as ***dry saturated steam.*** As heating continues further, the temperature of steam begins to rise again and steam is now known as ***superheated steam*** and behaves more or less as a perfect gas.

3.2.1 Enthalpy : For reasons too long to explain here, the term heat content is not recommended for use in thermodynamics. The terms formerly known as "heat of water" (sensible heat), "heat of evaporation" (latent heat) and "total heat of steam" are now known as the "enthalpy of (saturated) water", "enthalpy of evaporation" and "enthalpy of saturated steam" respectively. Similarly, the term total heat of superheated steam is now known as "enthalpy of superheated steam". Enthalpy of evaporation is the difference between enthalpy of dry saturated steam and enthalpy of (saturated) water, i.e,

Enthalpy of evaporation = Enthalpy of dry saturated steam – Enthalpy of boiling water

3.2.2 Enthalpy of Water : The amount of heat absorbed by one kilogram of water in being heated from the freezing point (0°C) to the boiling point t_s, is known as the *enthalpy of the saturated water (sensible heat of water)* and is denoted by the symbol

h. It is customary to reckon the enthalpy of water from 0°C at which temperature the enthalpy of water is said to be zero. To raise the temperature of one kilogram of water from 0°C to 100°C requires $4{\cdot}187 \times 100 = 418{\cdot}7$ kJ; hence this number is the enthalpy of one kg of water at 100°C. If the temperature of water to begin with is say 20°C instead of 0°C, then number of heat units required to raise 1 kg of water at 20°C to water at 100°C is equal to $4{\cdot}187\ (100 - 20) = 334{\cdot}96$ kJ, where 4·187 kJ/kg K is the specific heat water.

If the specific heat of water were 4·187 kJ/kg K at all temperatures, then the enthalpy of water would be $4{\cdot}187\ (t_s - 0)$ kJ/kg. For engineering purpose, it is of sufficient accuracy to assume the value of specific heat of water (k) to be 4·187 kJ/kg K at low pressures in which case,

Enthalpy of (saturated) water, h at any temperature t_s above 0°C is

$$h = 4{\cdot}187 \times (t_s - 0) \text{ kJ/kg} \quad \text{.. (3.2)}$$

The error in results calculated by this formula becomes larger as the temperature rises. The value of enthalpy of saturated water, h given in column 4 of the steam tables (given at the end of this book) should be used in preference to the results calculated from the above formula.

3.2.3 Enthalpy of Evaporation : The enthalpy of evaporation (or latent heat) is defined as the amount of heat required to convert one kilogram of water at a given temperature t_s and pressure p into steam at the same temperature and pressure. The value of enthalpy of evaporation varies with the pressure. It is usually expressed by the symbol L and its value at 1 bar is 2,258 kJ per kg. The value of enthalpy of evaporation, L of 1 kg of dry saturated steam can be directly obtained from the steam tables.

The value of enthalpy of evaporation of dry saturated steam is given in column 5 of the steam tables.

3.2.4 Enthalpy of Dry Saturated Steam : It is the sum of enthalpy of saturated water and enthalpy of evaporation and is defined as the quantity of heat required to raise the temperature of one kilogram of water from freezing point to the temperature of evaporation t_s (corresponding to given pressure p) and then convert it into dry saturated steam at that temperature and pressure. It is denoted by the symbol H_s. The *enthalpy* of one kg of dry saturated steam, H_s = enthalpy of (saturated) water + enthalpy of evaporation,

$$\text{i.e.,} \quad H_s = h + L \text{ kJ/kg} \quad \text{... (3.3)}$$

The value of enthalpy (total heat), H_s of 1 kg of dry saturated steam can be directly obtained from the steam tables corresponding to given value of pressure or temperature. The value of enthalpy of dry saturated steam is given in column 6 of the steam tables.

Enthalpy of evaporation is the enthalpy difference between dry saturated steam and saturated water.

3.2.5 Wet Steam : The steam in the steam space of a boiler generally contains water mixed with it in the form of a mist (fine water particles). Such a steam is termed as *wet steam.* The quality of steam as regards its dryness is termed as ***dryness fraction.*** Dryness fraction is usually expressed by the symbol x or q. Dryness fraction is often spoken as the *quality of wet steam.*

If m_s = mass of dry steam contained in the steam considered, and

m = mass of water in suspension in the steam considered.

Then, dryness fraction, $x = \frac{m_s}{m_s + m}$ ... (3.4)

Thus, if drynes fraction of wet steam, $x = 0{\cdot}8$, then one kg of wet steam contains 0·2 kg of moisture (water) in suspension and 0·8 kg of dry steam.

When the steam produced is not dry steam, but a mixture of steam and water as is often the case, the enthalpy of one kilogram of mixture of water and steam (wet steam) is less than enthalpy of the same quantity of pure or dry steam. To generate steam of dryness fraction 0·8, takes only 0·8 of the enthalpy of evaporation necessary for the formation of dry steam at the same temperature. Hence, enthalpy of evaporation of one kg of wet steam is the value of the enthalpy of evaporation given in steam tables for dry steam, at the required pressure, multiplied by its dryness fraction x.

i.e. Enthalpy of evaporation of 1 kg of wet steam = xL kJ/kg ... (3.5)

Enthalpy of wet steam is the amount of heat required to raise the temperature of one kilogramof water from freezing point (0°C) to the boiling point t_s (corresponding to given pressure, p) and then to convert the boiling water into wet steam,

i.e. Enthalpy of one kg of wet steam = enthalpy of saturated water + enthalpy of evaporation,

i.e. $H_{wet} = h + xL$ kJ/kg ... (3.6)

3.2.6 Superheated Steam : If the steam remains in intimate contact with water during its formation, interchange of molecules between the water and steam will result. This interchange of molecules will continue as long as there is any water in the cylinder of fig. 3-1(d). This will not allow the steam to become dry.

If the water is entirely evaporated and further heat is then supplied, the first effect on the steam is to make it dry if it is not already dry. The temperature of steam will then begin to increase with a corresponding increase in volume. Steam in this condition, heated out of contact with water, is said to be superheated. Superheating is assumed to take place at constant pressure. The amount of superheating is measured by the rise in temperature of the steam above its saturation temperature (t_s). Greater the amount of superheating, the more will the steam acquire the properties of a perfect gas.

In practice, steam is superheated after it has left the boiler by passing it through a nest of tubes known as *superheater.* The superheater is heated externally by the furnace gases of the boiler. The additional heat supplied to steam during the process of superheating is known as the heat of superheat.

Heat absorbed per kg of dry steam during superheating = $k_p(t_{sup} - t_s)$ kJ/kg ...(3.7)

where t_{sup} = temperature of superheated steam,

t_s = saturation temperature at the given pressure, and

k_p = mean specific heat of superheated steam at constant pressure.

The temperature rise ($t_{sup} - t_s$) is termed as *dgree of superheat.* It means the number of degrees of temperature by which steam at a given pressure is raised above the saturation temperature corresponding to that pressure. The condition of superheated steam is given either by its degree of superheat or by its temperature. The value of specific heat of superheated steam (k_p) varies with the pressure and degree of superheating. It increases with the increase of pressure and decreases with the increase of degree of superheating. The value of specific heat at constant pressure of superheated steam (k_p) varies from 2 to 2·5 kJ/kg K.

The enthalpy (or total heat) of one kg of superheated steam may be defined as the number of heat units required to convert one kg of water at freezing temperature (0°C) into superheated steam at a given pressure and temperature, assuing that the heat is added at constant pressure, which is the case in practice. The enthalpy of 1 kg of superheated steam is, therefore, the sum of enthalpy of water, the enthalpy of evaporation, and the heat of superheat. It is denoted by symbol H_{sup}. Hence, enthalpy of one kg of superheated steam,

$$H_{sup} = h + L + k_p(t_{sup} - t_s) = H_s + k_p(t_{sup} - t_s) \text{ kJ/kg} \quad \text{... (3.8)}$$

3.2.7 Specific Volume of Steam : The volume of one kilogram of dry saturated steam at all pressures is given in column 3 of the steam tables. The value in cubic metre per kg of dry saturated steam (m^3/kg) is known as the *specific volume of dry saturated steam* and its symbol is v_s.

Density of dry steam is the mass of one cubic metre of steam (kg/m^3) and is the reciprocal of specific volume and, therefore, is equivalent to $1/v_s$ kg per cubic metre.

If the steam is *wet*, having a dryness fraction of x, one kg of wet steam will consist of x kg of dry (pure) steam and $(1 - x)$ kg of water held in suspension.

Specific volume of wet steam, having a dryness fraction of x

= volume of dry steam + volume of water particles

$= xv_s + (1 - x)v_w$

where, v_s and v_w denote the specific volume of steam and water respectively.

As the volume of water at low pressure is very small compared with the steam, the term $(1 - x)v_w$ will become still smaller and can be neglected.

Hence, the specific volume of wet steam, neglecting volume of water particles

$= xv_s$ m^3/kg, and density of wet steam $= 1/xv_s$ kg/m^3.

For some problems of thermodynamics, it is sufficiently accurate that, at a given pressure, the volume of superheated steam is proportional to its absolute temperature. This assumption is made on the ground that superheated steam behaves like a perfect gas. The approximate specific volume of *superheated steam* may be calculated from the expression,

$$v_{sup} = v_s \times \frac{T_{sup}}{T_s} \text{ m}^3\text{/kg} \quad \text{... (3.9)}$$

where T_{sup} is the absolute temperature of superheated steam, T_s is the absolute saturation temperature of steam and v_s is the specific volume of dry saturated steam.

3.3 Steam Tables

Reference is now made to the steam tables which give the properties of 1 kg of dry saturated steam. The information given in steam tables (given at the end of this book) has been obtained by performing experiments on dry saturated steam at various pressures and temperatures. In the steam tables, the initial temperature of water is taken as 0°C.

Absolute pressures given in column 1 of the steam tables are in bar. Before the steam tables can be used, it is necessary to convert boiler gauge pressure (shown by the steam pressure gauge fitted on the boiler) to absolute pressure. This is obtained by adding atmospheric (barometric) pressure to boiler gauge pressure. See solved problem no.5. The information contained in steam tables is as under :

Absolute pressure of steam	.. p bar
Saturation temperature of steam (temperature of formation of steam)	.. t_s°C
Specific volume of dry saturated steam	.. v_s m^3/kg
Enthalpy of saturated water (reckoned above 0°C)	.. h kJ/kg
Enthalpy of evaporation of dry steam	.. L kJ/kg
Enthalpy of dry saturated steam (reckoned above 0°C)	.. H_s kJ/kg
Entropy of saturated water	.. Φ_w kJ/kg K
Entropy of dry saturated steam	.. Φ_s kJ/kg K

It is important to note that the data given in steam tables is for *dry saturated steam only* and therefore, the effect of dryness fraction (x) must be taken into account in the manner already explained. The enthalpy of wet steam and superheated steam can be calculated by using eqns. (3.6) and (3.8).

The properties of 1 kg of dry saturated steam at various absolute pressures are given in steam tables, at the end of this book.

Note : In this text, the pressures stated in bar, kPa and MPa will indicate absolute pressures. Gauge pressure will be indicated as gauge pressure or pressure gauge reading.

Problem-1 : *Determine the dryness fraction of steam if 0·8 of water is in suspension with 45 kg of dry steam.*

Mass of wet steam

$$= \begin{Bmatrix} \text{mass of dry steam in} \\ \text{certain mass of wet steam} \end{Bmatrix} + \begin{Bmatrix} \text{mass of water in suspension} \\ \text{in the same mass of wet steam} \end{Bmatrix}$$

$$= 45 + 0{\cdot}8 = 45{\cdot}8 \text{ kg.}$$

Thus, in 45·8 kg of wet steam, the dry steam is 45 kg.

The dryness fraction of wet steam is the ratio of the mass of actual dry steam to the mass of wet steam containing it.

$\therefore$ Dryness fraction, $x = 45/45{\cdot}8 = \mathbf{0{\cdot}982}$

i.e. Steam is **98·2 per cent dry or** 1·8 per cent wet.

Problem-2 : *How much heat is needed to convert 1 kg of feed water at 20°C into dry saturated steam at 10 bar (1 MPa) ? Take specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 10 bar, enthalpy of saturated water, h = 762·81 kJ/kg and enthalpy of evaporation, L = 2,015·3 kJ/kg.

Enthalpy of dry saturated steam at 10 bar (above 0°C), $H_s = h + L$

$$= 762{\cdot}81 + 2{,}015{\cdot}3 = 2{,}778{\cdot}11 \text{ kJ/kg.}$$

Enthalpy of 1 kg of feed water at 20°C above 0°C $= k\,(t_1 - 0)$

$$= 4{\cdot}187\ (20 - 0) = 83{\cdot}74 \text{ kJ/kg.}$$

$\therefore$ Heat supplied to convert 1 kg of feed water at 20°C into dry saturated steam at 10 bar = enthalpy of dry saturated steam − enthalpy of feed water

$$= 2{,}778{\cdot}11 - 83{\cdot}74 = \mathbf{2{,}694{\cdot}37 \text{ kJ/kg.}}$$

Problem-3 : *How much heat is needed to convert 5 kg of water at 40°C into 90 per cent dry (or 10 per cent wet) steam at 5 bar (500 kPa) ? Take specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 5 bar, h = 640·23 kJ/kg and L = 2,108·5 kJ/kg.

Enthalpy of 1 kg of wet steam at 5 bar and 0·9 dry above 0°C,

$$H_{wet} = h + xL = 640{\cdot}23 + 0{\cdot}9 \times 2{,}108{\cdot}5 = 2{,}537{\cdot}88 \text{ kJ/kg.}$$

Enthalpy of 1 kg of feed water at 40°C above 0°C

$$= k(t_1 - 0) = 4{\cdot}187\ (40 - 0) = 167{\cdot}48 \text{ kJ/kg.}$$

∴ Heat needed to convert 1 kg of feed water at 40°C into wet steam at 5 bar and 0·9 dry = Enthalpy of wet steam – Enthalpy of feed water

$$= 2{,}537{\cdot}88 - 167{\cdot}48 = 2{,}370{\cdot}4 \text{ kJ/kg.}$$

∴ Heat supplied to convert 5 kg of water at 40°C into steam at 5 bar and 0·9 dry

= 5 × 2,370·4 = **11,852 kJ.**

Problem-4 : *How much heat is needed to convert 4 kg of water at 20°C into steam at 8 bar (800 kPa) and 200°C. Take* k_p *of superheated steam as 2·1 kJ/kg K and specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 8 bar, t_s = 170·43°C, h = 721·11 kJ/kg and L = 2,048 kJ/kg. Since the temperature (200°C) of given steam is more than saturation temperature, t_s (170·43°C), the steam is superheated.

Degree of Superheat = $t_{sup} - t_s$ = 200 – 170·43 = 29·57°C.

Enthalpy of 1 kg of superheated steam (above 0°C),

$$H_{sup} = h + L + k_p\ (t_{sup} - t_s) = 721{\cdot}11 + 2{,}048 + 2{\cdot}1\ (200 - 170{\cdot}43)$$
$$= 2{,}831{\cdot}21 \text{ kJ/kg}$$

Enthalpy of 1 kg of water above 0°C = $k\ (t_1 - 0)$ = 4·187 (20 – 0) = 83·74 kJ/kg.

Heat needed to convert 1 kg of water at 20°C into superheated steam at 8 bar and 200°C = Enthalpy of superheated steam – Enthalpy of water

= 2,831·21 – 83·74 = 2,747·47 kJ/kg

∴ Heat needed to convert 4 kg of water into steam = 4 × 2,747·47 = **10,990 kJ.**

Problem-5 : *A boiler is supplied with feed water at a temperature of 44°C and converted into steam at a gauge pressure of 4·493 bar and a temperature of 188°C. If 900 kg of steam is generated by the boiler in one hour, find the quantity of heat supplied per minute to feed water to produce steam assuming barometric pressure of 750 mm of Hg and* k_p *of superheated steam as 2·1 kJ/kg K.*

Now, 760 mm of Hg = 1·01325 bar.

$$\therefore\ 750 \text{ mm of Hg} = \frac{1{\cdot}01325 \times 750}{760} = 1{\cdot}007 \text{ bar (barometric pressure)}$$

Absolute pressure of steam = Boiler gauge pressure + Barometric pressure

= 4·493 + 1·007 = 5·5 bar.

Referring to steam tables, at 5·5 bar h = 655·93 kJ/kg, L = 2,097 kJ/kg and t_s = 155·48°C. Since the temperature of given steam (188°C) is more than the saturation temperature (155·48°C), the steam produced is superheated.

Enthalpy of 1 kg of feed water at 44°C above 0°C = $k\ (t_1 - 0)$

$$= 4{\cdot}187\ (44 - 0) = 184{\cdot}23 \text{ kJ/kg.}$$

$\therefore$ Enthalpy of (saturated) water $= h - k(t_1 - 0) = 655{\cdot}93 - 184{\cdot}23 = 471{\cdot}7$ kJ/kg.

Heat supplied/kg $= [h - k\,(t_1 - 0) + L + k_p(t_{sup} - t_s)]$

$= 471{\cdot}7 + 2{,}097 + 2{\cdot}1\,(188 - 155{\cdot}48) = 2{,}636{\cdot}99$ kJ/kg

$\therefore$ Heat supplied per min. $= \dfrac{900}{60} \times 2{,}636{\cdot}99 =$ **39,554·85 kJ/min.**

Problem-6 : *Find the volume of one kilogram of steam at a pressure of 15 bar (1·5 MPa) in each of the following cases :*

(i) when steam is dry saturated,

(ii) when steam is wet having dryness fraction of 0·9, and

(iii) when steam is superheated, the degree of superheat being 40°C.

At 15 bar, $v_s = 0{\cdot}13177$ m^3/kg and $t_s = 198{\cdot}32$°C (from steam tables).

(i) When steam is dry saturated :

Specific volume (v_s) of dry saturated steam at 15 bar is **0·13177 m^3/kg.**

(ii) When steam is wet, dryness fraction 0·9 :

Specific volume of wet steam of dryness fraction x, neglecting the volume of water $= xv_s$ m^3/kg.

$\therefore$ Volume of wet steam at a pressure of 15 bar and 0·9 dry $= xv_s$

$= 0{\cdot}9 \times 0{\cdot}13177 =$ **0·1186 m^3/kg**

(iii) When steam is superheated by 40°C :

Absolute temperature of saturated steam, $T_s = 198{\cdot}32 + 273 = 471{\cdot}32$ K

Absolute temperature of superheated steam, $T_{sup} = 198{\cdot}32 + 40 + 273 = 511{\cdot}32$ K

The specific volume of superheated steam is given by eqn. (3·9),

$$v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}13177 \times \frac{511{\cdot}32}{471{\cdot}32} = \mathbf{0{\cdot}1426\ m^3/kg.}$$

Problem-7 : *A closed vessel contains 1·2 m^3 of dry saturated steam at 14 bar (1·4 MPa). What is the temperature and mass of this steam ?*

From steam tables, saturation temperature corresponding to 14 bar = **195·7°C.**

Specific volume (v_s) at 14 bar $= 0{\cdot}14084$ m^3/kg (from steam tables).

$$\therefore \text{Mass of steam} = \frac{\text{Volume of steam}}{\text{Specific volume of steam }(v_s)} = \frac{1{\cdot}2}{0{\cdot}14084} = \mathbf{8{\cdot}52\ kg.}$$

Problem-8 : *Determine the condition of steam in each of the following cases :*

(i) at a pressure of 10 bar and temperature 200°C,

(ii) at a pressure of 8 bar and volume 0·22 m^3/kg, and

(iii) at a pressure of 12 bar, if 2,688 kJ/kg are required to produce it from water at 0°C.

(i) Saturation temperature (t_s) corresponding to pressure of 10 bar from steam tables is 179·91°C. Since the temperature of given steam is 200°C, the steam is superheated. Its degree of superheat, $t_{sup} - t_s = 200 - 179{\cdot}91 =$ **20·09°C.**

(ii) Specific volume of dry saturated steam (v_s) at 8 bar is 0·2404 m^3/kg. Since the volume of given steam (0·22 m^3) is less than the specific volume of dry saturated steam, it is obvious that the steam with a volume of 0·22 m^3/kg is wet.

∴ Dryness fraction, $x = 0{\cdot}22/0{\cdot}2404 = 0{\cdot}915$ i.e. **91·5% dry.**

(iii) Enthalpy of dry saturated steam (H) at 12 bar from steam tables is 2,784·8 **kJ/kg**. Since the enthalpy of given steam(2,688 kJ/kg) is less than the enthalpy of dry saturated steam, we conclude that the given **steam is wet.**

At 12 bar, $h = 798{\cdot}65$ kJ/kg, $L = 1{,}986{\cdot}2$ kJ/kg (from steam tables).

Using eqn. (3·6) for determining quality of steam, $H_{wet} = h + xL$

i.e. $2{,}688 = 798{\cdot}65 + x\ 1{,}986{\cdot}2$

∴ Dryness fraction of steam, $x = \dfrac{2{,}688 - 798{\cdot}65}{1{,}986{\cdot}2} = 0{\cdot}9513$ i.e. **95·13% dry**

Problem-9 : *A boiler is supplying steam at a pressure of 9 bar and 90 percent dry. Find : (i) the mass of 5 m^3 of this steam, and (ii) the enthalpy per m^3 of this steam above 0°C.*

(i) From steam tables, at 9 bar,

$h = 742{\cdot}83$ kJ/kg, $L = 2{,}031{\cdot}1$ kJ/kg, and $v_s = 0{\cdot}215$ m^3/kg.

Mass of one m^3 of wet steam $= \dfrac{1}{xv_s} = \dfrac{1}{0{\cdot}9 \times 0{\cdot}215} = 5{\cdot}168$ kg

∴ Mass of 5 m^3 of wet steam = 5 × 5·168 = **25·84 kg**

(ii) Enthalpy of 1 kg of wet steam at a pressure of 9 bar and dryness fraction 0·9,

$H_{wet} = h + xL = 742{\cdot}83 + 0{\cdot}9 \times 2{,}031{\cdot}1 = 2{,}570{\cdot}83$ kJ/kg

∴ Enthalpy of one m^3 of wet steam = mass of steam × $(h + xL)$

$= 5{\cdot}168 \times 2{,}570{\cdot}83 =$ **13,286 kJ.**

Problem-10 : *Determine from steam tables the following :*

(i) Enthalpy and volume of 1 kg of steam at 12·1 bar and dryness fraction 0·9, and

(ii) Enthalpy and volume of 1 kg of steam at 12·1 bar and 225°C.

Take the specific heat at constant pressure for superheated steam as 2·1 kJ/kg K.

By using steam tables and *interpolating* to find the values for a pressure of 12·1 bar, a table is constructed as shown below :

p bar	t_s °C	h kJ/kg	L kJ/kg	H kJ/kg	v_s m^3/kg	
12·0	187·99	798·65	1986·2	2784·8	0·16333	from steam tables
12·2	188·74	801·98	1983·4	2785·4	0·16077	from steam tables
0·2	0·75	3·33	2·8	0·6	0·00256	for 0·2 bar
0·1	0·38	1·67	1·4	0·3	0·00128	for 0·1 bar
12·1	188·37	800·32	1984·8	2785·1	0·16205	by interpolating

(i) Enthalpy of 1 kg of steam at 12·1 bar and 0·9 dry,

$H_{wet} = h + xL = 800{\cdot}32 + 0{\cdot}9 \times 1{,}984{\cdot}8 =$ **2,463·52 kJ/kg.**

Volume of 1 kg of steam at 12·1 bar and 0·9 dry

$= x \times v_s = 0{\cdot}9 \times 0{\cdot}16205 = \mathbf{0{\cdot}14585\ m^3/kg}$

(ii) Enthalpy of 1 kg of steam at 12·1 bar and 225°C,

$$H_{sup} = H + k_p(t_{sup} - t_s) = 2{,}785{\cdot}1 + 2{\cdot}1(225 - 188{\cdot}37) = \mathbf{2{,}862{\cdot}02\ kJ/kg}$$

Using eqn. (3·9), volume of 1 kg of superheated steam at 12·1 bar and 225°C,

$$v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}16205 \times \frac{498}{461{\cdot}37} = \mathbf{0{\cdot}1751\ m^3/kg.}$$

Problem-11 : *One kg of steam initially at 8 bar (800 kPa) and 0·7 dry, receives 410 kJ heat at constant pressure. Determine the enthalpy, condition, and density of steam at the end of constant pressure heat supply.*

From steam tables, at 8 bar, $h = 721{\cdot}11$ kJ/kg, $L = 2{,}048$ kJ/kg, $v_s = 0{\cdot}2404\ m^3/kg$.

Enthalpy initially, $H_1 = h_1 + x_1L_1 = 721{\cdot}11 + 0{\cdot}7 \times 2{,}048 = 2{,}154{\cdot}71$ kJ/kg

Enthalpy at the end of constant pressure process, $H_2 = 2{,}154{\cdot}71 + 410 = \mathbf{2{,}564{\cdot}71\ kJ/kg}$

Let the final condition of steam be x_2. Then, $H_2 = h_2 + x_2L_2$

From which, $x_2 = \frac{H_2 - h_2}{L_2} = \frac{2{,}564{\cdot}71 - 721{\cdot}11}{2{,}048} = $ **0·9 dry** (final condition of steam)

Density of steam $= \frac{1}{x_2v_s} = \frac{1}{0{\cdot}9 \times 0{\cdot}2404} = \mathbf{4{\cdot}62\ kg/m^3}$.

Problem-12 : *Two boilers A and B, are delivering steam in equal proportion to a common mains steam pipe, both at a pressure of 14 bar. Boiler A has a superheater and boiler B is without a superheater. The temperature of steam supplied by the boiler A is 300°C. The temperature of the resulting mixture of steam in the steam mains is 235°C. Assuming k_p for the superheated steam as 2·1 kJ/kg K, estimate the quality of steam supplied by the boiler without the superheater, i.e., boiler B.*

From steam tables, at 14 bar, $L = 1{,}959{\cdot}7$ kJ/kg and $t_s = 195{\cdot}07$°C.

The wet steam from boiler *B* gains heat from the superheated steam produced by boiler *A*. As a result of which it (wet steam) becomes superheated. During the process of its mixing with superheated steam, first its moisture content is evaporated and then it is superheated. Let the dryness fraction of wet steam generated by boiler *B* be *x*.

Then, moisture in 1 kg of wet steam = $(1 - x)$ kg. Heat required to evaporate this moisture and to make it dry = $(1 - x)L$ kJ/kg.

Heat required to superheat this dry steam = $k_p(t_{sup2} - t_s)$ kJ/kg.

$\therefore$ Heat gained by the wet steam of boiler $B = (1 - x)L + k_p(t_{sup2} - t_s)$ kJ/kg.

Heat lost by the superheated steam of boiler $A = k_p(t_{sup1} - t_{sup2})$

Heat gained by wet steam of boiler *B* = Heat lost by superheated steam of boiler *A*

i.e. $(1 - x)\ L + k_p(t_{sup2} - t_s) = k_p(t_{sup1} - t_{sup2})$

i.e. $(1 - x)\ 1{,}959{\cdot}7 + 2{\cdot}1(235 - 195{\cdot}07) = 2{\cdot}1(300 - 235)$

$\therefore x = \frac{1{,}907{\cdot}05}{1{,}959{\cdot}7} = 0{\cdot}9732$ i.e. **97·32 % dry** (quality of steam supplied by boiler *B*)

Problem-13 : *A boiler generates 1,400 kg of steam per hour at a pressure of 18 bar (1·8 MPa) and 0·95 dry. The steam after leaving the boiler stop valve, passes through a superheater where its temperature is raised to 305°C. If the feed water supplied to the* *boiler is at 40°C, calculate : (a) the heat supplied to feed water per hour to produce wet steam, and (b) the percentage of the heat absorbed in the boiler and in the superheater. Take k_p of superheated steam as 2·3 kJ/kg K and specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 18 bar, $H = 884{\cdot}79$ kJ/kg, $L = 1{,}912{\cdot}4$ kJ/kg and $t_s = 207{\cdot}15$°C.

(a) Heat supplied to feed water to produce wet steam in the boiler per hour

$= W_s\ [h - k(t_1 - 0) + xL]$

$= 1{,}400\ [884{\cdot}97 - (4{\cdot}187 \times 40) + 0{\cdot}95 \times 1{,}912{\cdot}4] =$ **35,47,700 kJ/hr.**

(b) The heat absorbed by superheater is utilized to remove the moisture content of wet steam and then to superheat the dry saturated steam from 207·15°C to 305°C.

Heat absorbed by the superheater per hour $= W_s\ [(1 - x)L + k_p(t_{sup} - t_s)]$

$= 1{,}400\ [(1 - 0{\cdot}95)\ 1{,}912{\cdot}4 + 2{\cdot}3\ (305 - 207{\cdot}15)] =$ **4,48,950 kJ/hg.**

Total heat absorbed per hour = 35,47,700 + 4,48,950 = 39,96,650 kJ/kg

Percentage of heat absorbed in boiler $= \dfrac{35{,}47{,}700}{39{,}96{,}650} \times 100 =$ **88·76 %**

Percentage of heat absorbed in superheater $= \dfrac{4{,}48{,}950}{39{,}96{,}650} \times 100 =$ **11·24 %**

Problem-14 : *Wet steam of mass 2·5 kg and occupying a volume of 0·49 m^3 at 7·5 bar has a total heat (enthalpy) increase of 1,500 kJ when superheated at constant pressure. Determine :*

(i) Initial quality of steam,

(ii) Final quality (degree of superheat) of steam, and

(iii) Increase in volume of steam after superheating.

Assume k_p for the superheated steam to be 2·1 kJ/kg K.

(i) From steam tables, at 7·5 bar,

$h = 709{\cdot}47$ kJ/kg, $t_s = 167{\cdot}78$°C, $L = 2{,}057$ kJ/kg, $v_s = 0{\cdot}2556$ m^3/kg.

Total volume of 2·5 kg of dry saturated steam $= 2{\cdot}5 \times 0{\cdot}2556 = 0{\cdot}639$ m^3

As the volume of 2·5 kg of steam in data is given as 0·49 m^3 at 7·5 bar (which is less than 0·639 m^3), the steam is wet.

∴ Initial dryness fraction (quality) of steam = 0·49/0·639 = 0·767 i.e. **76·7 % dry.**

(ii) 1,500 kJ of heat supplied (added) at constant pressure is utilized in evaporating the moisture content of wet steam and then superheating it. Let t_{sup} be the temperature of superheated steam. Then,

$W_s\ [(1 - x)L + k_p(t_{sup} - t_s)] =$ Heat supplied to superheat 2·5 kg of wet steam

i.e. $2{\cdot}5\ [(1 - 0{\cdot}767)\ 2{,}057 + 2{\cdot}1\ (t_{sup} - 167{\cdot}78)] = 1{,}500$

Solving the equation, $t_{sup} = 225{\cdot}2$°C.

∴ Degree of superheat $= t_{sup} - t_s = 225{\cdot}2 - 167{\cdot}78 = 57{\cdot}42$°C i.e. steam is superheated by **57·42°C.**

(iii) Now, $T_s = 167{\cdot}78 + 273 = 440{\cdot}78$ K; and $T_{sup} = 225{\cdot}2 + 273 = 498{\cdot}2$ K

Let v_{sup} = Specific volume of superheated steam,

(specific volume of superheated steam is proportional to its absolute temperature)

Then, volume of superheated steam is given by eqn. (3·9),

$$\therefore\ v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}2556 \times \frac{498{\cdot}2}{440{\cdot}78} = 0{\cdot}2891 \text{ m}^3\text{/kg}$$

$\therefore$ Total volume of 2·5 kg of superheated steam = 2·5 ×0·2891

= 0·7228 m^3 (final volume)

$\therefore$ Increase in volume of steam after superheating = 0·7228 − 0·49 = **0·2328 m^3.**

Problem-15 : *Steam enters a steam engine at a pressure of 12 bar with 67°C of superheat and is exhausted at 0·15 bar and 0·94 dry. Calculate the drop in enthalpy from admission to exhaust, and volume of 1 kg of steam at admission and exhaust conditions. Take k_p of superheated steam as 2·1 kJ/kg K.*

From steam tables, at 12 bar,

h = 798·65 kJ/kg, L = 1,986·2 kJ/kg, t_s = 187·99°C and v_s = 0·16333 m^3/kg.

From steam tables, at 0·15 bar,

h = 225·94 kJ/kg, L = 2,373·1 kJ/kg and v_s = 10·022 m^3/kg.

Enthalpy of superheated steam at 12 bar (*at admission*),

$$H_{sup} = h + L + k_p \times \text{degree of superheat}$$
$$= 798{\cdot}65 + 1{,}986{\cdot}2 + 2{\cdot}1\ (67) = 2{,}925{\cdot}55 \text{ kJ/kg}$$

Enthalpy of wet steam at 0·15 bar (*at exhaust*),

$$H_{wet} = h + xL = 225{\cdot}94 + 0{\cdot}94 \times 2{,}373{\cdot}1 = 2{,}456{\cdot}65 \text{ kJ/kg.}$$

Hence, enthalpy difference (or drop) from admission to exhaust (during expansion)

$$= H_{sup} - H_{wet} = 2{,}925{\cdot}55 - 2{,}456{\cdot}65 = \mathbf{468{\cdot}9\ kJ/kg}$$

At 12 bar, t_s = 187·99°C, and T_s = 187·99 + 273 = 460·99 K,

t_{sup} = 187·99 + 67 = 254·99°C, and

T_{sup} = 254·99 + 273 = 527·99 K.

Since, specific volume of steam is assumed to be proportional to its absolute temperature for approximate calculations. Using eqn. (3·9),

$$v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}16333 \times \frac{527{\cdot}99}{460{\cdot}99} = \mathbf{0{\cdot}1871\ m^3/kg}\ (\textit{at admission})$$

Volume of wet steam (neglecting the volume of water) at 0·15 bar

$$= xv_s = 0{\cdot}94 \times 10{\cdot}022 = \mathbf{9{\cdot}421\ m^3/kg}\ (\textit{at exhaust}).$$

Problem-16 : *A vessel having a capacity of 1 m^3 contains steam at 10 bar and 0·92 dry. The steam is blown off from the vessel by opening the valve until the pressure drops to 5 bar. The valve is then closed and the vessel is cooled until the pressure is 4 bar. Assuming that the enthalpy per kg of steam in the vessel remains constant during the blowing off period, determine :*

(a) the mass of steam blown off, (b) the dryness fraction of the steam in the vessel after cooling, and (c) the heat lost by the steam per kg during cooling.

From steam tables :

p bar	v_s m^3/kg	h kJ/kg	L kJ/kg
10	0·19444	762·81	2,015·3
5	0·3749	640·23	2,108·5
4	0·4625	604·74	2,133·8

(a) *Before blowing-off,* p_1 = 10 bar, v = 1 m^3, x_1 = 0·92 and v_{s1} = 0·19444 m^3.kg.

Initial mass of steam in the vessel *before blowing-off,*

$$m_1 = \frac{1}{x_1 v_{s1}} = \frac{1}{0{\cdot}92 \times 0{\cdot}19444} = 5{\cdot}642 \text{ kg}$$

After blowing of, p_2 = 5 bar, v = 1 m^3 and v_{s2} = 0·3749 m^3/kg.

As given in the data,

Enthalpy before blowing off (at 10 bar) = Enthalpy after blowing off (at 5 bar)

i.e. $h_1 + x_1 L_1 = h_2 + x_2 L_2$

i.e. $762{\cdot}81 + 0{\cdot}92 \times 2{,}015{\cdot}3 = 640{\cdot}23 + x_2 \times 2{,}108{\cdot}5 \quad \therefore\ x_2 = 0{\cdot}9373$

Final mass of steam in the vessel *after blowing-off,*

$$m_2 = \frac{1}{x_2 v_{s2}} = \frac{1}{0{\cdot}9373 \times 0{\cdot}3749} = 2{\cdot}846 \text{ kg}$$

$\therefore$ Mass of steam blown off = $m_1 - m_2$ = 5·642 − 2·846 = **2·796 kg.**

(b) *After cooling,* p_3 = 4 bar, v_{s3} = 0·4625 m^3/kg, and m_3 = 2·846 kg.

Dryness fraction of steam in the vessel after cooling,

$$x_3 = \frac{1}{v_{s3} \times m_3} = \frac{1}{0{\cdot}4625 \times 2{\cdot}846} = 0{\cdot}76 \text{ or } \mathbf{76\%\ dry}$$

(c) Enthalpy *before* cooling, $H_2 = h_2 + x_2 L_2$

$= 640{\cdot}23 + 0{\cdot}9373 \times 2{,}108{\cdot}5 = 2{,}615{\cdot}23$ kJ/kg

Now, Enthalpy *after* cooling, $H_3 = h_3 + x_3 L_3$

$= 604{\cdot}74 + 0{\cdot}76 \times 2{,}133{\cdot}8 = 2{,}226{\cdot}43$ kJ/kg

Enthalpy difference or heat lost per kg of steam during cooling,

$H_2 - H_3 = 2{,}615{\cdot}23 - 2{,}226{\cdot}43 =$ **388·8 kJ/kg**

Problem-17 : *Steam at a pressure of 6 bar (600 kPa) is passed into a tank containing water where it is condensed. Before the steam is introduced in the tank, the mass and temperature of water in the tank is 74 kg and 15°C respectively. The water equivalent of tank is one kg. Calculate the dryness fraction of steam as it enters the tank if 2·5 kg of steam is condensed in the tank and the resulting temperature of the mixture is 35°C. Assume no losses and specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 6 bar, h = 670·56 kJ/kg, and L = 2,086·3 kJ/kg.

Mass of steam condensed, M_s = 2·5; total mass of water, M_w = 1 + 74 = 75 kg.

Assuming that all heat lost by steam is gained by water

i.e. Heat lost by steam = Heat gained by water i.e. $M_s\ [h + xL - k \times t_2] = M_w \times k \times (t_2 - t_1)$

(where, t_2 = temp. of mixture, and t_1 = initial temp. of water)

$$\text{i.e. } h + xL - k \times t_2 = \frac{M_w}{M_s} \times k \times (t_2 - t_1)$$

$$\text{i.e. } 670{\cdot}56 + 2{,}086{\cdot}3\ x - 4{\cdot}187 \times 35 = \frac{75}{2{\cdot}5} \times 4{\cdot}187 \times (35 - 15)$$

$\therefore$ **x = 0·95 (dryness fraction of steam)**

Problem-18 : *Exhaust steam at a pressure of 0·5 bar and 0·8 dry enters a surface condenser. the water resulting from the condensation leaves the condenser at a temperature of 56°C. Assuming that all the heat lost by steam is taken up by the condensing water, find the heat removed from the steam per kg and also the mass of condensing water required per kg of steam, if the temperature rise of condensing water is 28°C. Take specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 0·5 bar, h = 340·49 kJ/kg and L = 2,305·4 kJ/kg.

Enthalpy of exhaust steam per kg as it enters the condenser

$= h + xL = 340{\cdot}49 + 0{\cdot}8 \times 2{,}305{\cdot}4 = 2{,}184{\cdot}81$ kJ/kg

Enthalpy of water formed by condensation of steam per kg

$= 4{\cdot}187 \times (56 - 0) = 234{\cdot}47$ kJ/kg.

$\therefore$ Heat removed by cooling or condensing water from 1 kg of exhaust steam in condenser or heat lost by 1 kg of exhaust steam = 2,184·81 − 234·47 = **1,950·34 kJ/kg**

Each kg of condensing water removes (from data) 28 × 4·187 = 117·24 kJ

$\therefore$ Mass of condensing water required per kg of exhaust steam,

$$m_w = \frac{1{,}950{\cdot}34}{117{\cdot}24} = \mathbf{16{\cdot}63\ kg}$$

Alternatively, assuming that all the heat lost by exhaust steam is gained by condensing water,

Heat lost by 1 kg of exhaust steam = Heat gained by m_w kg of condensing water

i.e. $1\ [(h + xL) - 56 \times 4{\cdot}187] = m_w \times 28 \times 4{\cdot}187$

i.e. $1\ [340{\cdot}49 + (0{\cdot}8 \times 2{,}305{\cdot}4) - 234{\cdot}47] = m_w \times 117{\cdot}24$

$$\therefore m_w = \frac{1{,}950{\cdot}34}{117{\cdot}24} = 16{\cdot}63 \text{ kg of water/kg of exhaust steam.}$$

3.4 External Work Done during Evaporation

The enthalpy of evaporation (latent heat) absorbed by the steam during evaporation is utilized in two ways :

Firstly, in overcoming internal molecular resistance of water in changing its condition from water to steam and secondly, in overcoming the external resistance of the piston to its increasing volume during evaporation.

The first of these effects of enthalpy of evaporation (latent heat) is called *internal work*, because changes have been brought within the body itself, and the second is called *external work* because work has been done on the bodies external to itself. The first represents the energy stored in the steam and is known as *internal latent heat* and the second represents the energy which has passed out of the steam having been utilized in doing work on the piston and is known as *external work of evaporation.* The value of enthalpy of evaporation (latent heat) = internal latent heat + external latent heat, can be directly obtained from the steam tables. In evaporating water to steam the volume increases from v_w to v_s (fig. 3-1) under a constant pressure p; external work is thus done and the energy for performing this work is obtained during the absorption of enthalpy of evaporation. This work is known as external work of evaporation.

External work done per kg of dry saturated steam

$$= \frac{10^5 \times p \times (v_s - v_w)}{10^3} \quad \text{kJ/kg} \qquad \text{...(3.10)}$$

where, p = absolute pressure of steam in bar, v_s = volume of 1 kg of dry saturated steam in m^3 at presure p, and v_w = volume of 1 kg of water in m^3.

It is sufficiently accurate to neglect the term v_w in eqn. (3.10) as at low pressure, the volume of 1 kg of water is very small compared with the volume of the steam it forms.

The eqn. (3.10) may be written as

$$\text{Work of evaporation of dry saturated steam} = \frac{10^5 p v_s}{10^3} \quad \text{kJ/kg} \qquad \text{... (3.11)}$$

If the steam is wet having a dryness fraction of x, the final volume of wet steam is xv, then

$$\text{External work done per kg of wet steam} = \frac{10^5 p\,(x v_s)}{10^3} \quad \text{kJ/kg} \qquad \text{... (3.12)}$$

$$\text{External work done per kg of superheated steam} = \frac{10^5 p v_{sup}}{10^3} \quad \text{kJ/kg}$$

The above expressions for external work of evaporation give the portion of latent heat which has been transformed into mechanical work or represent the heat energy which has passed out of the steam. The remainder of the heat energy supplied remains as internal energy or energy stored in the steam and is known as *internal latent heat.* It is found by subtracting the external work of evaporation from the full latent heat.

$$\text{i.e. Internal latent heat} = L - \frac{10^5 p v_s}{10^3} \quad \text{kJ/kg (for dry saturated steam)} \qquad \text{... (3.13)}$$

For wet steam having dryness fraction x,

$$\text{Internal latent heat} = xL - \frac{10^5 p\,(x v_s)}{10^3} \quad \text{kJ/kg} \qquad \text{...(3.14)}$$

where p = absolute pressure in bar, v_s = volume of 1 kg of dry saturated steam in m^3 at pressure p, x = dryness fraction of steam, and L = enthalpy of evaporation in kJ/kg.

3.5 Internal Engery of steam

The internal energy of steam is the actual heat energy above the freezing point of water stored in the steam, and is the sum of internal latent enthalpy and sensible enthalpy reckoned from 0°C. It has already been shown that the latent enthalpy of steam consists of the internal latent enthalpy plus the external work of evaporation; hence the total enthalpy of steam consists of the sensible enthalpy, plus the internal latent enthalpy and the external work of evaporation. But the external work of evaporation is utilized in doing external work and represents the heat energy which has passed out of the steam. Hence, the internal energy of steam denoted by symbol u consists of two terms only, namely, the internal latent enthalpy and the sensible enthalpy.

Thus, internal energy is found by substracting external work of evaporation from the enthalpy of steam.

If the steam is dry saturated, Enthalpy, $H_s = h + L$ kJ/kg, and

$$\text{Internal energy, } u = H_s - \frac{10^5 p v_s}{10^3} \text{ kJ/kg} \quad \dots (3.15)$$

If the steam is wet having dryness fraction x, Enthalpy, $H_{wet} = h + xL$ kJ/kg, and

$$\text{Internal energy, } u = H_{wet} - \frac{10^5 p\,(x v_s)}{10^3} \text{ kJ/kg} \quad \dots (3.16)$$

If the steam is superheated to temperature t_{sup} and volume v_{sup} m^3 per kg,

Enthalpy of superheated steam, $H_{sup} = H_s + k_p\,(t_{sup} - t_s)$ kJ/kg, and

$$\left.\begin{aligned}\text{Internal energy, } u &= H_{sup} - \frac{10^5 p v_{sup}}{10^3} \text{ kJ/kg} \\ &= H_{sup} - \frac{10^5\, p\left(v_s\,\dfrac{t_{sup}+273}{t_s+273}\right)}{10^3} \text{ kJ/kg}\end{aligned}\right] \quad \dots (3.17)$$

Problem-19 : *Find the external work done during evaporation, internal latent enthalpy and internal energy per kg of steam at a pressure of 15 bar (1,500 kPa) when the steam is (i) 0·9 dry, and (ii) dry saturated.*

(i) For wet steam (0·9 dry):

From steam tables, at 15 bar,

$h = 844{\cdot}89$ kJ/kg; $L = 1{,}947{\cdot}3$ kJ/kg; $H_s = 2{,}792{\cdot}2$ kJ/kg; $v_s = 0{\cdot}13177$ m^3/kg.

Specific volume of wet steam neglecting the volume of water,

$= x v_s = 0{\cdot}9 \times 0{\cdot}13177$ m^3/kg

External work done during evaporation per kg of steam

$$= \frac{10^5\, p\,(x\, v_s)}{1{,}000} = \frac{10^5 \times 15(0{\cdot}9 \times 0{\cdot}13177)}{1{,}000} = \mathbf{177{\cdot}89 \text{ kJ/kg}}.$$

Internal latent enthalpy

= enthalpy of evaporation – heat absorbed in doing work of formation of steam

$$= xL - \frac{10^5\, p\,(x\, v_s)}{1{,}000} = (0{\cdot}9 \times 1{,}947{\cdot}3) - 177{\cdot}89 = \mathbf{1{,}574{\cdot}68 \text{ kJ/kg}}.$$

Enthalpy of wet steam, $H_{wet} = h + xL = 844{\cdot}89 + 0{\cdot}9 \times 1{,}947{\cdot}3 = 2{,}597{\cdot}46$ kJ/kg.

Using eqn. (3·16), internal energy, u

= Enthalpy of 1 kg of wet steam–External work of evaporation

$$\therefore\ u = H_{wet} - \frac{10^5\, p\,(x\, v_s)}{1{,}000} = 2{,}597{\cdot}46 - 177{\cdot}89 = \mathbf{2{,}419{\cdot}57\ kJ/kg}$$

(ii) For dry saturated steam :

External work done during evaporation per kg of dry saturated steam

$$= \frac{10^5\, p \times v_s}{1{,}000} = \frac{10^5 \times 15 \times 0{\cdot}13177}{1{,}000} = \mathbf{197{\cdot}66\ kJ/kg}$$

Internal latent enthalpy $= L - \frac{10^5\, p \times v_s}{1{,}000} = 1{,}947{\cdot}3 - 197{\cdot}66 = \mathbf{1{,}749{\cdot}64\ kJ/kg}$

Internal energy, $u = H_s - \frac{10^5\, p \times v_s}{1{,}000} = 2{,}792{\cdot}2 - 197{\cdot}66 = \mathbf{2{,}594{\cdot}54\ kJ/kg}$.

Problem-20 : *0·025 m^3 of steam at 3·5 bar and dryness fraction 0·8 is converted into dry saturated steam at 11 bar. By how much are the enthalpy and internal energy changed ?*

From steam tables :

p bar	h kJ/kg	L kJ/kg	H kJ/kg	v_s m^3/kg
3·5	584·33	2,148·1	—	0·5243
11	—	—	2,781·7	0·17753

Volume of one kg of wet steam (at 3·5 bar and 0·8 dry) $= xv_s$

$= 0{\cdot}8 \times 0{\cdot}5243 = 0{\cdot}41944$ m^3/kg,

$\therefore$ Mass of 0·025 m^3 of steam (at 3·5 bar and 0·8 dry) = 0·025/0·41944 = 0·0597 kg

Enthalpy of one kg of *wet* steam (at 3·5 bar and 0·8 dry), $H_{wet} = h + xL$

$= 584{\cdot}33 + 0{\cdot}8 \times 2{,}148{\cdot}1 = 2{,}302{\cdot}81$ kJ/kg.

$\therefore$ Enthalpy of 0·0597 kg of *wet* steam (at 3·5 bar and 0·8 dry), H_{wet}

$= 0{\cdot}0597 \times 2{,}302{\cdot}81 = 137{\cdot}48$ kJ

Enthalpy of 0·0597 kg of dry saturated steam (at 11 bar), H_s

$= 0{\cdot}0597 \times 2{,}781{\cdot}7 = 166{\cdot}07$ kJ

$\therefore$ Change in Enthalpy = 166·07 − 137·48 = **28·59 kJ**

Using eqn. (3·16), initial internal energy of 0·025 m^3 of steam at 3·5 bar and 0·8 dry,

$$u_1 = H_{wet} - \frac{10^5\, pv}{10^3} = 137{\cdot}48 - \frac{10^5 \times 3{\cdot}5 \times 0{\cdot}025}{10^3} = 128{\cdot}73\ \text{kJ}$$

Using eqn. (3·15), final internal energy of *dry saturated* steam at 11 bar,

$$u_2 = H_s - \frac{10^5\, pv}{10^3} = 166{\cdot}07 - \frac{10^5 \times 11 \times (0{\cdot}0579 \times 0{\cdot}17753)}{10^3} = 154{\cdot}41\ \text{kJ}$$

$\therefore$ Change in internal energy, $u_2 - u_1 = 154{\cdot}41 - 128{\cdot}73 = \mathbf{25{\cdot}68\ kJ}$

Problem-21 : *The internal energy of 1 kg of steam at a pressure of 14 bar (1·4 MPa) is 2,420 kJ. Calculate the dryness fraction of this steam. Find the increase in internal energy if this steam is superheated at constant pressure to a temperature of 295°C. Take k_p of superheated steam as 2·3 kJ/kg K.*

From steam tables, at 14 bar,

$h = 830{\cdot}3$ kJ/kg, $L = 1{,}959{\cdot}7$ kJ/kg, $v_s = 0{\cdot}14084$ m^3/kg and $t_s = 195{\cdot}07$°C.

Internal energy of 1 kg of *wet* steam,

$$u_1 = H_{wet} - \text{External work done during evaporation}$$

$$= (h + xL) - \frac{10^5 p(xv_s)}{1{,}000}$$

$$\text{i.e. } 2{,}420 = (830{\cdot}3 + 1{,}959{\cdot}7\,x) - \frac{10^5 \times 14\ (x \times 0{\cdot}14084)}{1{,}000}$$

$$\text{i.e. } 2{,}420 = (830{\cdot}3 + 1{,}959{\cdot}7x) - 197{\cdot}18\ x$$

$$\therefore\ x = \frac{2{,}420 - 830{\cdot}3}{1{,}959{\cdot}7 - 197{\cdot}18} = 0{\cdot}9 \text{ (dryness fraction)}$$

Using eqn. (3·17), Internal engery of 1 kg of superheated steam,

$$u_2 = \{h + L + k_p(t_{sup} - t_s\} - \left\{\frac{10^5 p\left(v_s \times \frac{t_{sup} + 273}{t_s + 273}\right)}{1{,}000}\right\}$$

$$= \{830{\cdot}3 + 1{,}959{\cdot}7 + 2{\cdot}3\,(295 - 195{\cdot}07)\} - \left\{\frac{10^5 \times 14 \times \left(0{\cdot}14084 \times \frac{295 + 273}{195{\cdot}07 + 273}\right)}{1{,}000}\right\}$$

$$= 3{,}020 - 239{\cdot}2 = 2{,}780{\cdot}8 \text{ kJ/kg}$$

Increase in internal energy, $u_2 - u_1 = 2{,}780{\cdot}8 - 2{,}420 =$ **360·8 kJ/kg**

Problem-22 : *What fraction of enthalpy of 1 kg of steam at 10 bar and 0·9 dry represents the internal energy ? What is the change in internal energy when the pressure and temperature of this steam is raised to 13 bar and 250°C ? Take k_p of superheated steam as 2·1 kJ/kg K.*

For wet steam at 10 bar and 0·9 dry :

From steam tables, at 10 bar,

$h = 762{\cdot}81$ kJ/kg, $L = 2{,}015{\cdot}3$ kJ/kg and $v_s = 0{\cdot}19444$ m^3/kg.

Enthalpy of 1 kg of wet steam (at 10 bar and 0·9 dry), $H_{wet} = h + xL$

$$= 762{\cdot}81 + 0{\cdot}9 \times 2{,}015{\cdot}3 = 2{,}576{\cdot}58 \text{ kJ/kg}$$

External work done during evaporation $= \dfrac{10^5 \times p(xv_s)}{10^3}$

$$= \frac{10^5 \times 10(0{\cdot}9 \times 0{\cdot}19444)}{10^3} = 175 \text{ kJ/kg}$$

Internal energy, u_1 = Enthalpy − External work done during evaporation

$$= 2{,}576{\cdot}58 - 175 = 2{,}401{\cdot}58 \text{ kJ/kg}$$

$\therefore$ Required fraction of enthalpy $= \dfrac{2{,}401{\cdot}58}{2{,}576{\cdot}58} = 0{\cdot}9324$

For superheated steam at 13 bar and 250°C :

At 13 bar (from steam tables), $v_s = 0{\cdot}15125$ m^3/kg, $H_s = 2{,}787{\cdot}6$ kJ/kg,

$$t_s = 191{\cdot}64^\circ\text{C or } T_s = 191{\cdot}64 + 273 = 464{\cdot}64 \text{ K};$$

$$t_{sup} = 250^\circ\text{C, or } T_{sup} = 250 + 273 = 523 \text{ K}.$$

Enthalpy of 1 kg of superheated steam (at 13 bar and at 250°C),

$$H_{sup} = H_s + k_p\,(t_{sup} - t_s) = 2{,}787{\cdot}6 + 2{\cdot}1\,(250 - 191{\cdot}64) = 2{,}910{\cdot}16 \text{ kJ/kg}$$

Using eqn. (3·9), volume of 1 kg of superheated steam,

$$v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}15125 \times \frac{523}{464{\cdot}64} = 0{\cdot}1702 \text{ m}^3\text{/kg}$$

External work done during evaporation

$$= \frac{10^5 \times p \times v_{sup}}{10^3} = \frac{10^5 \times 13 \times 0{\cdot}1702}{10^3} = 221{\cdot}26 \text{ kJ/kg}$$

Internal energy, u_2 = Enthalpy − External work done during evaporation

$$= 2{,}910{\cdot}16 - 221{\cdot}26 = 2{,}688{\cdot}9 \text{ kJ/kg}$$

∴ Change in internal energy = $u_2 - u_1$ = 2,688·9 − 2,401·58 = **287·32 kJ/kg.**

Problem-23 : *A certain quantity of steam in a closed vessel of fixed volume of 0·14 m^3 exerts a pressure of 10 bar at 250°C. If the vessel is cooled so that the pressure falls to 3·6 bar, determine : (a) the final quality of steam, (b) the final temperature, (c) the change in internal energy, and (d) the heat transferred during the process. Take k_p for superheated steam as 2·1 kJ/kg K.*

(a) From steam tables, at 10 bar,

$$t_s = 179{\cdot}91^\circ\text{C},\ v_s = 0{\cdot}19444 \text{ m}^3\text{/kg},\ H_s = 2{,}778{\cdot}1 \text{ kJ/kg}$$

Given steam is superheated with $t_{sup} = 250^\circ$C, or $T_{sup} = 250 + 273$ K

Initial enthalpy of superheated steam,

$$H_1 = H_{sup} = H_s + k_p\,(t_{sup} - t_s) = 2{,}778{\cdot}1 + 2{\cdot}1\,(250 - 179{\cdot}91)$$
$$= 2{,}925{\cdot}29 \text{ kJ/kg}$$

$$v_1 = v_{sup} = v_s \times \frac{T_{sup}}{T_s} = 0{\cdot}19444 \times \frac{250 + 273}{179{\cdot}91 + 273} = 0{\cdot}2246 \text{ m}^3\text{/kg}$$

$$m_s = \text{mass of steam} = \frac{\text{Volume of steam}}{\text{Sp. volume of steam}} = \frac{0{\cdot}14}{0{\cdot}2246} = 0{\cdot}6233 \text{ kg.}$$

Initial internal energy, $u_1 = m_s\left(H_1 - \dfrac{10^5\,p_1 v_1}{10^3}\right)$

$$= 0{\cdot}6233\left(2{,}925{\cdot}29 - \frac{10^5 \times 10 \times 0{\cdot}2246}{10^3}\right)$$

$$= 0{\cdot}6233\,(2{,}925{\cdot}29 - 224{\cdot}6) = 1{,}683{\cdot}34 \text{ kJ}$$

At 3·6 bar, $t_s = 139{\cdot}87^\circ$C, $v_s = 0{\cdot}5106$ m^3/kg. $h = 588{\cdot}59$ kJ/kg, $L = 2145{\cdot}1$ kJ/kg.

If steam is dry saturated at 3·6 bar, its volume

$$= m_s \times \text{specific volume of dry saturated steam} = 0{\cdot}6233 \times 0{\cdot}5106 = 0{\cdot}3183 \text{ m}^3$$

But the volume of steam is equal to the volume of vessel, i.e. 0·14 m^3.

Therefore, steam is wet with dryness fraction, $x_2 = \frac{0{\cdot}14}{0{\cdot}3183} = \mathbf{0{\cdot}44}$

Thus, finally steam is wet having dryness fraction of 0·44.

(b) As steam is wet at 3·6 bar, its temperature is equal to saturation temperature corresponding to this pressure, i.e. **139·87°C** (from steam tables).

(c) Enthalpy at final state, $H_2 = h_2 + x_2L_2 = 588{\cdot}59 + 0{\cdot}44 \times 2{,}145{\cdot}1 = 1{,}502{\cdot}43$ kJ/kg and $v_1 = v_2 = 0{\cdot}2246$ m^3/kg.

Final internal energy, $u_2 = m_s\left(H_2 - \frac{10^5\, p_2v_2}{10^3}\right)$

$$= 0{\cdot}6233\left(1{,}502{\cdot}43 - \frac{10^5 \times 3{\cdot}6 \times 0{\cdot}2246}{10^3}\right) = 885{\cdot}82 \text{ kJ}$$

Change in internal energy $= u_2 - u_1 = 885{\cdot}82 - 1{,}683{\cdot}34 =$ **−797·52 kJ.**

(d) As this is a constant volume process, the work done is zero and hence, heat transferred is equal to change in internal energy, i.e., **−797·52 kJ.**

3.6 Summary of Formulae

Following are the important formulae derived in this chapter :

Enthalpy of 1 kg of wet steam, $H_{wet} = h + xL$ kJ

Enthalpy of 1 kg of dry saturated steam, $H_s = h + L$ kJ

Enthalpy of 1 kg of superheated steam, $H_{sup} = H_s + k_p\,(t_{sup} - t_s)$ kJ

Volume of 1 kg of dry saturated steam = v_s m^3

Volume of 1 kg of wet steam = xv_s m^3

Volume of 1 kg of superheated steam, $v_{sup} = v_s \times \frac{T_{sup}}{T_s}$ m^3 eqn. (3·9)

External work done per kg of dry saturated steam = $\frac{10^5\, pv_s}{10^3}$ kJ eqn. (3·11)

External work done per kg of wet steam = $\frac{10^5\, p\,(xv_s)}{10^3}$ kJ eqn. (3·12)

External work done per kg of superheated steam = $\frac{10^5\, pv_{sup}}{10^3}$ kJ

Internal latent enthalpy per kg of dry saturated steam = $L - \frac{10^5\, pv_s}{10^3}$ kJ eqn. (3·13)

Internal latent enthalpy per kg of wet steam = $xL - \frac{10^5\, p\,(xv_s)}{10^3}$ kJ eqn. (3·14)

Internal energy of 1 kg of dry saturated steam = $H_s - \frac{10^5\, pv_s}{10^3}$ kJ eqn. (3·15)

Internal energy of 1 kg of wet steam = $H_{wet} - \frac{10^5\, p\,(xv_s)}{10^3}$ kJ eqn. (3·16)

Internal energy of 1 kg of superheated steam = $H_{sup} - \frac{10^5\, pv_{sup}}{10^3}$ kJ eqn. (3·17)

Tutorial - 3

1. What are phases of a substance ? Explain the importance of vapour as a working medium in thermal engineering.

2. Explain the following terms :

(i) Saturated steam, (ii) Dry saturated steam, (iii) Wet steam, (iv) superheated steam, (v) Dryness fraction of steam, (vi) Specific volume of steam, and (vii) Saturated water.

3. Explain the term dryness fraction of stteam.

Determine the dryness fraction of steam if 0·6 kilogram of water is in suspension with 24 kilograms of dry steam.

[0·976]

4.(a) Explain the following terms as referred to steam :

(i) Enthalpy of water. (ii) Enthalpy of evaporation, (iii) Superheat, (iv) Specific volume, and (v) Enthalpy of dry saturated steam.

(b) Determine from steam tables the enthalpy of one kg of steam above 0°C :

(i) at a pressure of 10 bar and 0·9 dry, and

(ii) at a pressure of 15 bar and with 140°C of superheat.

Take specific heat at constant pressure of superheated steam as 2·1 kJ/kg K.

[(i) 2,576·58 kJ; (ii) 3,086·2 kJ]

5. (a) Differentiate between saturated steam and dry saturated steam.

(b) Estimate the amount of heat needed to convert 5 kg of water at 50°C into steam at a pressure of 9 bar and with 110°C of superheat. Take specific heat at constant pressure for superheated steam as 2·1 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

[13,978 kJ]

6. (a) How much heat is needed to convert one kg of feed water at 20°C into dry saturated steam at a pressure of 9 bar ? Take specific heat of water as 4·187 kJ/kg K.

(b) How much heat is needed to convert 5 kg of water at 45°C into steam at pressure of 4 bar and 80 per cent dry. Take specific heat of water as 4·187 kJ/kg K.

[(a) 2,690·19 kJ; (b) 10,616·8 kJ]

7. What information can be found in steam tables ? What is the effect of increase in pressure on saturation temperature, enthalpy of saturated water, enthalpy of evaporation and enthalpy of dry saturated steam.

8. (a) What is meant by critical point ? What are critical parameters of H_20 ?

(b) Find the heat required to produce 0·9 kg of steam at 7 bar and 240°C from water at 28°C. Take k_p of superheated steam as 2·26 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

[(b) 2,534·27 kJ]

9. Temperature of feed water entering a boiler is 50°C and the pressure of steam in the boiler is 13 bar. How much heat will be required to produce one kilogram of steam if the steam produced is (a) 0·97 dry, (b) dry saturated, and (c) superheated with 40°C of superheat ? Take the specific heat at constant pressure for superheated steam as 2·1 kJ/kgK and specific heat of water as 4·187 kJ/kg K.

[(a) 2,519·1 kJ; (b) 2,578·28 kJ; (c) 2,662·25 kJ]

10. Distinguish clearly between the various qualities of steam.

Find the heat required to convert one kilogram of steam at 8 bar and 80 per cent dry into steam at 8 bar and 200°C. Assume k_p for superheated steam as 2·1 kJ/kg K.

[471·74 kJ]

11. A boiler is supplied with feed water at 65°C and produces steam at 17 bar and 260°C. If 75% of the heat of the coal of calorific value of 29,300 kJ/kg is utilised in producing steam, calculate the mass of coal consumed in producing 5,000 kg of steam per hour. Take specific heat at constant pressure for superheated steam as 2·1 kJ/kg K.

[600·8 kg]

12. find the volume of one kilogram of steam at a pressure of 10 bar in each of the following cases : (i) when the steam is dry saturated, (ii) when the steam is wet, having a dryness fraction of 0·8 (neglecting the volume of water), and (iii) when the steam is superheated and temperature of steam is 240°C. Take specific heat at constant pressure for superheated steam as 2·1 kJ/kg K.

[(i) 0·19444 m^3/kg; (ii) 0·15555 m^3/kg; (iii) 0·2202 m^3/kg]

13. Calculate the specific volume of superheated steam at 11 bar and 250°C, treating the superheated steam as a perfect gas. Take k_p for superheated steam = 2·1 kJ/kg K.

[0·2031 m^3/kg]

14. Find the volume of one kg of steam at a pressure of 9·8 bar (980 kPa) in each of the following cases :

(i) when the steam is dry saturated,

(ii) when the steam is wet, having a dryness fraction of 0·8, and

(iii) when the steam is superheated and the temperature of steam is 240°C.

Take specific heat at constant pressure of superheated steam as 2·1 kJ/kg K.

[(i) 0·1982 m^3/kg; (ii) 0·15856 m^3/kg; (iii) 0·22496 m^3/kg]

15. Use the steam tables to find the following :

(a) Enthalpy and volume per kg of steam of dryness fraction 0·85 at a pressure of 8·5 bar.

(b) Enthalpy and volume per kg of steam at 16 bar and at 230°C. Take specific heat at constant pressure for superheated steam as 2·1 kJ/kg K.

(c) Dryness fraction of steam at 4·5 bar having enthalpy of 2,580 kJ/kg.

(d) Density of 1 kg of steam at 7 bar, having a dryness fraction of 0·9.

(e) Enthalpy of one cubic metre of steam at 9 bar and 0·9 dry.

[(a) 2,465·71 kJ/kg, 0·193 m^3/kg; (b) 2,854 kJ/kg, 0·1312 m^3/kg; (c) 0·923; (d) 4·071 kg/m^3; (e) 13,280 kJ]

16. Determine the condition of steam in each of the following cases :

(a) at a pressure of 10 bar, if 2,640 kJ/kg are required to produce it from water at 0°C.

(b) at a pressure of 8 bar and temperature 200°C, and

(c) at a pressure of 12 bar and volume 0·144 m^3/kg.

[(a) wet, dryness fraction 0·9313; (b) superheated by 29·57°C; (c) wet, dryness fraction 0·8816]

17. (a) What is meant by saturation temperature ? Differentiate between saturated water and dry saturated steam.

(b) Determine the condition of steam in each of the following cases :

(i) At a pressure of 10 bar (1 MPa) and temperature 200°C,

(ii) At a pressure of 8 bar and volume 0·22 m^3/kg, and

(iii) At a pressure of 12 bar, if 2,688 kJ/kg are required to produce it from water at 0°C.

[(i) steam is superheated by 20·09°C; (ii) wet, 91·5% dry; (iii) wet, 95·15% dry]

18. Use the steam tables and find the following :

(i) Dryness fraction of steam at a pressure of 5 bar (500 kPa) having enthalpy of 2,500 kJ/kg.

(ii) Density of 1 kg of steam at a pressure of 6 bar, having a dryness fraction of 0·8.

(iii) Enthalpy and volume per kg of steam at a temperature of 250°C and at a pressure of 15 bar. Take k_p of superheated steam as 2·1 kJ/kg K.

(iv) Enthalpy and volume per kg of steam at a pressure of 9 bar and 0·9 dry.

[(i) 0·882 dry; (ii) 3·96 kg/m^3; (iii) 2,900·72 kJ/kg, 0·14626 m^3/kg; (iv) 2,573·81 kJ/kg, 0·1935 m^3/kg]

19. (a) A closed vessel contains 0·9 m^3 of dry saturated steam at a pressure of 7·5 bar. How many kg of steam does the vessel contain ?

(b) A closed vessel contains 1·5 m^3 of dry saturated steam at 12 bar. What is the temperature and mass of this steam ?

[(a) 3·521 kg; (b) 187·99°C, 9·184 kg]

20. (a) A boiler supplies steam at a pressure of 11 bar and 0·8 dry. Find the mass of 0·5m^3 of this steam and its enthalpy per cubic metre above 0°C.

(b) A closed drum of 0·15 m^3 capacity contains 1·25 kg of wet steam at a pressure of 10 bar. Find the quality of steam.

[(a) 3·52 kg, 16,770 kJ; (b) 0·617 dry]

21. Two boilers discharge equal amount of steam into the main steam pipe. The steam from one is at a pressure of 14 bar and temperature of 290°C and from the other at the same pressure and 92% dry. What is the resulting condition of the steam after mixing ? Take k_p of superheated steam as 2·1 kJ/kg K.

[superheated by 10·13°C]

22. A tank containing 180 kg of water of 16°C is to be heated to 66°C by blowing dry saturated steam at 2 bar into the tank. Find the mass of steam that should enter the tank to heat the water to the required temperature. Assume no losses. Take specific heat of water as 4·187 kJ/kg K.

[13·92 kg]

23. Dry saturated steam enters a surface condenser at a pressure of 1·1 bar and the condensate leaves at 85°C. The cooling water enters at 15°C and leaves at 60°C. Find the amount of cooling water required to cndense 1 kg of steam. Assume that all the heat lost by steam is taken up by the cooling water. Take specific heat of water as 4·187 kJ/kg K.

[12·31 kg]

24. Exhaust steam at a pressure of 0·5 bar (50 kPa) and 0·8 dry enters a surface condenser. The water resulting from the condensation leaves the condenser at a temperature of 56°C. Assuming that all heat lost by steam is taken up by condensing water, find the mass of condensing water required per kg of steam, if the temperature rise of condensing water is 28°C. Take specific heat of water as 4·187 kJ/kg K.

[16·63 kg]

25. Steam is exhausted from a steam engine into a condenser at the rate of 5 kg/min. The pressure inside the condenser is 0·15 bar. The steam from the condenser leaves as water at 40°C. Cooling water is circulated through the condenser tubes at the rate of 115 kg/min. with a rise in temperature from inlet to outlet of 20°C. Calculate the dryness fraction of the steam entering the condenser. Take specific heat of water as 4·187 kJ/kg K.

[0·7954 dry]

26. (a) Explain the following terms as applied to steam :

(i) Internal latent heat, (ii) External work of evaporation, and (iii) Internal energy.

(b) Find the external work of evaporation, internal latent heat and internal energy of one kg of steam at a pressure of 12 bar (1·2 MPa) :

(i) when the steam is dry saturated, and (ii) when the steam is 0·8 dry.

[(i) 196 kJ/ig, 1,790·2 kJ/kg, 2,588 kJ/kg; (ii) 156·8 kJ/kg, 1,432·16 kJ/kg, 2,230·81 kJ/kg]

27. Calculate the internal energy of 1 kg of steam at a pressure of 8 bar (800 kPa) : (a) when the steam is dry saturated, (b) when the steam is 0·8 dry, and (c) when the steam is superheated and temperature of steam is 270°C. Take specific heat of superheated steam at constant pressure as 2·2 kJ/kg K.

[(a) 2,576·79 kJ/kg; (b) 2,205·65 kJ/kg; (c) 2,752·56 kJ/kg]

28. Distinguish clearly between enthalpy and internal energy of steam.

1 kg of steam at 10 bar occupies a volume of 0·18 m^3. Determine the condition of steam, and the enthalpy and internal energy of steam.

[0·9258 dry; 2,627·81 kJ; 2,447·81 kJ]

29. Use the steam tables to calculate the following :

(a) The internal energy of 1 kg of steam of dryness fraction 0·85, at a pressure of 6 bar.

(b) The internal energy of 1 kg of steam at 10 bar and temperature 200°C. Take k_p of superheated steam as 2·1 kJ/kg K.

(c) Internal energy of 0·125 m^3 of steam at 12 bar and dryness fraction 0·75.

[(a) 2,282·94 kJ; (b) 2,617·09 kJ; (c) 2,184·13 kJ]

30. The internal energy of 1 kg of steam at 10 bar is 2,400 kJ. Calculate the dryness fraction of this steam. Also find the increase in internal energy when the above steam is superheated at constant pressure to a temperature of 250°C. Take k_p of superheated steam as 2·1 kJ/kg K.

[0·899 dry, 300·92 kJ/kg]

31. (a) Find the external work of evaporation and internal energy of 1 kg of steam at a temperature of 200°C and at a pressure of 10 bar (1 MPa). Take k_p of superheated steam as 2·1 kJ/kg K.

(b) What fraction of enthalpy of 1 kg of steam at a temperature of 200°C and at a pressure of 10 bar represents the internal energy ?

[(a) 203 kJ/kg, 2,617·3 kJ/kg; (b) 0·929]

32. A certain quantity of steam in a closed vessel of fixed volume of 0·3 m^3 exerts a pressure of 15 bar at 280°C. If the vessel is cooled so that the pressure falls to 10 bar, determine : (a) the final temperature , (b) the final quality of steam, (c) the change in internal energy, and (d) the heat transferred during the process. Take k_p for superheated steam = 2·1 kJ/kg K.

[(a) 179·91°C; (b) 0·7953 dry; (c) – 1,010 kJ; (d) – 1,010 kJ]

4

ENTROPY

4.1 Introduction

The term entropy means "transformation". Entropy is a thermodynamic property of a working substance *which increases with the addition of heat and decreases with removal of heat.* It may be regarded as a thermodynamic variable (parameter of thermodynamic state like pressure, temperature etc.) introduced to facilitate the study of working fluids (working substances) when they are passing through reversible cycle (cycle consisting of only reversible operations). It is used by engineers as a means of providing quick solutions for problems dealing with isentropic operation. Entropy is usually represented by the symbol Φ (phai). The small increase of entropy, $d\Phi$ of a substance is defined as the ratio of small addition of heat, dQ to the absolute temperature, T of the substance at which the heat is supplied.

$$\text{i.e. } d\Phi = \frac{dQ}{T} \quad \text{or} \quad dQ = T \times d\Phi \qquad \text{...(4.1)}$$

The source of heat may be internal such as friction. The heating process represented on a diagram having absolute temperature and entropy as the two coordinates, is known as *temperature-entropy or T–Φ diagram* as shown in fig. 4-1. Let a substance be supplied with small amount of heat, dQ, during which the mean absolute temperature is T.

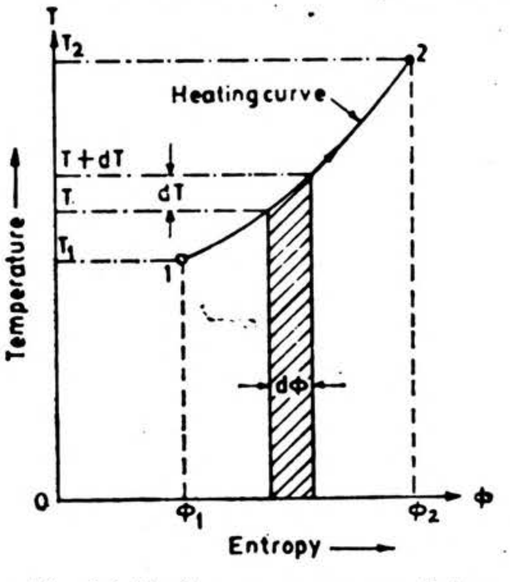


Fig. 4-1. Heating process represented on temperature-entropy diagram.

Then, area of the shaded strip (fig. 4-1) is given by $T \times \Phi = dQ$, by definition from eqn. (4.1).

$\therefore$ Area under the curve 1-2

$$= \int T \times d\Phi = \int dQ$$

= total heat supplied or transferred to the substance.

Thus, the area under the *T–Φ curve represents the heat supplied to the substance during the process, i.e. if a substance is heated from temperature T_1 to T_2,* the amount of heat supplied is given by the area under the curve 1-2 as shown in fig. 4-1.

Entropy is not the physical property of a substance in the same sense as pressure, volume, temperature etc. and therefore, it cannot be measured directly by instruments. It is a derived thermal property of a substance. It is used to indicate the thermodynamic *state* of a substance in the same sense as pressure, volume, temperature etc. The property, entropy obviously depends upon the mass of the system and hence it is an extensive property.

Entropy is measured per unit mass, and therefore, it would be more precise to speak of it as *specific entropy* as in case of specific volume etc. Absolute values of entropy cannot be determined, but the change in its value can be determined by reckoning the change from an arbitrary datum level of temperature. For steam, this arbitrary datum level is taken as 0°C, while for refrigerants (ammonia, carbon dioxide, Freon etc.,) this arbitrary datum level (reference point) is taken as −40°C. At this temperature of reference or datum, the entropy of the substance is assumed to be zero.

The *unit of entropy* is heat unit per degree Kelvin per kilográm mass of fluid. Entropy is expressed as kJ/kg K. Since, this unit of entropy has no important meaning, it is customary to speak of it as so many units of entropy, without referring to the dimensional units. Entropy is commonly used with absolute temperature (K) and enthalpy or total heat (H) kJ per kg to show graphically the state of a working stance. The H–Φ (*enthalpy-entropy*) and T–Φ (*temperature–entropy*) charts are available for steam, air, carbon dioxide, ammonia, Freon etc.

The reason for introducing this thermodynamic variable will be clearly understood by considering the Carnot cycle. The effeciency of the Carnot cycle, $\eta = \frac{T_1 - T_2}{T_1}$ (which will be proved later in chapter 9).

For temperature range of one degree (i.e. $T_1 - T_2 = 1$), $\eta = \frac{1}{T_1}$.

But efficiency, $\eta = \frac{dW}{dQ}$ (by definition)

$$\therefore \quad \frac{1}{T_1} = \frac{dW}{dQ} \qquad \text{i.e.} \qquad dW = \frac{dQ}{T_1} = d\,\Phi \qquad \text{...(4.2)}$$

Since, the Carnot cycle has maximum efficiency, the importance of entropy which represents the greatest amount of work obtainable per degree fall in temperature, will be at once evident from eqn. (4.2).

4.2 Clausius Theorem

Clausius formulated that "*all processes in the nature are irreversible, and the entropy of universe tends to become a maximum,*"

Let Q be the quantity of heat flowing from a hot body at absolute temperature T_1 to a cold body at absolute temperature T_2. The hot body temperature T_1 looses entropy by $\frac{Q}{T_1}$, while the cold body gains entropy by $\frac{Q}{T_2}$ which is greater than $\frac{Q}{T_1}$, so the *gain of entropy* of the system as a whole is $\frac{Q}{T_2} - \frac{Q}{T_1} = \frac{Q\,(T_1 - T_2)}{T_1\,T_2}$.

Thus, in all natural processes heat flows from a body at higher temperature to one at lower temperature. This process of heat flow cannot be reversed. It is obvious, that *entropy* increases and tends to become maximum as the temperature falls.

4.3 Entropy of Perfect Gases (General Expression)

Since entropy is a function of the thermodynamic state of substance, its change can be expressed in the same manner as temperature, pressure and volume. From the law of conservation of energy,

Heat supplied = Work done + Change in internal energy i.e. $dQ = dW + du$

Now, $dW = p \cdot dv$

Then, for a small change in the state of a system of one kg of perfect gas,

$$dQ = p \cdot dv + du$$

But, $du = k_v \cdot dT \qquad \therefore \; dQ = p \cdot dv + k_v \cdot dT$

Diving throughout by T, $\dfrac{dQ}{T} = \dfrac{p}{T} \times dv + k_v \times \dfrac{dT}{T}$

But, $\dfrac{dQ}{T} = d\Phi$ (definition). $\qquad \therefore \; d\Phi = \dfrac{p}{T} \times dv + k_v \times \dfrac{dT}{T}$

But, $pv = RT$ for one kg of perfect gas, i.e. $\dfrac{p}{T} = \dfrac{R}{v} \qquad \therefore \; d\Phi = R \times \dfrac{dv}{v} + k_v \times \dfrac{dT}{T}$

Integrating both sides, $\int d\Phi = \int R \cdot \dfrac{dv}{v} + \int k_v \cdot \dfrac{dT}{T}$

If we regard k_v as constant for the temperature range of T_1 to T_2, then integral becomes

$$\int_1^2 d\Phi = R \int_1^2 \frac{dv}{v} + k_v \int_1^2 \frac{dT}{T}$$

$$\therefore \; \Phi_2 - \Phi_1 = k_v \log_e \left(\frac{T_2}{T_1} \right) + R \log_e \left(\frac{v_2}{v_1} \right) \text{ per kg} \qquad \text{...(4.3)}$$

Expression (4.3), can also be expressed *in terms of pressures and temperatures.* From characteristic equation for perfect gas,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \text{ or } \frac{v_2}{v_1} = \frac{p_1}{p_2} \times \frac{T_2}{T_1}$$

Substituting this value of $\dfrac{v_2}{v_1}$ in eqn. (4.3), we have,

$$\Phi_2 - \Phi_1 = k_v \log_e \left(\frac{T_2}{T_1} \right) + R \log_e \left(\frac{p_1}{p_2} \times \frac{T_2}{T_1} \right)$$

$$= k_v \log_e \left(\frac{T_2}{T_1} \right) + R \log_e \left(\frac{p_1}{p_2} \right) + R \log_e \left(\frac{T_2}{T_1} \right)$$

$$= \left(k_v + R \right) \log_e \left(\frac{T_2}{T_1} \right) + R \log_e \left(\frac{p_1}{p_2} \right)$$

$$\therefore \; \Phi_2 - \Phi_1 = k_p \log_e \left(\frac{T_2}{T_1} \right) - R \log_e \left(\frac{p_2}{p_1} \right) \text{ per kg} \qquad \text{... (4.4)}$$

To express eqn. (4.3) in *terms of pressures and volumes,* we have from characteristic equation for perfect gas,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \quad \text{or} \quad \frac{T_2}{T_1} = \frac{v_2}{v_1} \times \frac{p_2}{p_1}$$

$$\text{Now, } \Phi_2 - \Phi_1 = k_v \log_e\left(\frac{T_2}{T_1}\right) + R \log_e\left(\frac{v_2}{v_1}\right) \qquad \text{[from eqn. (4.3)]}$$

$$\therefore\ \Phi_2 - \Phi_1 = k_v \log_e\left(\frac{v_2}{v_1} \times \frac{p_2}{p_1}\right) + R \log_e\left(\frac{v_2}{v_1}\right)$$

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

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

$$\therefore\ \Phi_2 - \Phi_1 = k_v \log_e\left(\frac{p_2}{p_1}\right) + k_p \log_e\left(\frac{v_2}{v_1}\right) \text{ per kg} \qquad \text{... (4.5)}$$

It is convenient to regard lower limit (reference point) of entropy, Φ, as zero at N.T.P. (0°C and 1·01325 bar).

Problem-1: *Air at an initial condition of 1·4 bar, 0·3 m^3 and 20°C is compressed to a pressure of 4·5 bar, and its volume reduces to 0·12 m^3. Calculate the change of entropy during the process. Assume air as perfect gas. Take R = 0·287 kJ/kg K and k_v = 0·7165 kJ/kg K for air.*

Using characteristic equation for perfect gas,

$$p_1 v_1 = mRT_1 \text{ i.e. } m = \frac{p_1 v_1}{RT_1} = \frac{(10^5 \times 1{\cdot}4)\ 0{\cdot}3}{0{\cdot}287 \times 10^3 \times (20 + 273)} = 0{\cdot}488 \text{ kg}$$

$$\text{Again, } p_2 v_2 = mRT_2 \text{ i.e. } T_2 = \frac{p_2 v_2}{mR} = \frac{(10^5 \times 4{\cdot}5)\ 0{\cdot}12}{0{\cdot}488 \times 0{\cdot}287 \times 10^3} = 385 \text{ K}$$

Using eqn. (4.3) for one kg of air, change of entropy ,

$$\Phi_2 - \Phi_1 = R \log_e\left(\frac{v_2}{v_1}\right) + k_v \log_e\left(\frac{T_2}{T_1}\right) = 0{\cdot}287 \log_e\left(\frac{0{\cdot}12}{0{\cdot}3}\right) + 0{\cdot}7165 \log_e\left(\frac{385}{293}\right)$$

$$= -0{\cdot}2627 + 0{\cdot}1809 = -0{\cdot}0818$$

$\therefore$ Total change of entropy = 0·488 × (− 0·0818) = − **0·0399 unit of entropy**.

Negative value of change of entropy shows that heat is abstracted or removed from air, i.e. entropy decreases.

Problem-2 : *One kg of air at 700 kPa and 27°C is expanded to a pressure of 600 kPa and temperature of 120°C. Assuming the value of R = 0·287 kJ/kg K and k_p = 1·0035 kJ/kg K, calculate the change of entropy during the process.*

Now, T_1 = 27 + 273 = 300 K; T_2 = 120 + 273 = 393 K.

Using eqn. (4.4), the change of entropy per kg of air,

$$\Phi_2 - \Phi_1 = k_p \log_e\left(\frac{T_2}{T_1}\right) - R \log_e\left(\frac{p_2}{p_1}\right)$$

$$= 1{\cdot}0035 \log_e\left(\frac{393}{300}\right) - 0{\cdot}287 \log_e\left(\frac{600}{700}\right)$$

$$= [1{\cdot}0035 \times 0{\cdot}27] - [0{\cdot}287 \times (-0{\cdot}1541)] = \mathbf{0{\cdot}315184 \text{ units}}$$

Positive value of change of entropy shows that heat is added i.e. entropy has increased.

4.4 Changes of Entropy for Perfect Gas

Expressions are derived for change of entropy during various thermodynamic processes for perfect gas.

4.4.1 Constant Volume Process : Let one kilogram of gas be heated from temperature T_1 to temperature T_2 at constant volume. Let k_v be the specific heat of the gas at constant volume. The value of k_v is assumed to be constant between the temperature range of T_1 to T_2.

Consider a small element of area of the diagram shown shaded in fig. 4-1 where the mean absolute temperature is T.

Left dT = a very small increase of the temperature, and

$d\Phi$ = corresponding small increase of entropy during this change of temperature.

As heat is transferred at constant volume and the increase of temperature is dT,

$$dQ = k_v \cdot dT$$

Then, by definition of specific entropy, $T \times d\Phi = dQ = k_v \cdot dT$

Dividing throughout by T, we have, $d\Phi = \frac{dQ}{T} = \frac{k_v \cdot dT}{T}$

Then, total change of entropy, $\Phi_2 - \Phi_1 = \int_{T_1}^{T_2} \frac{dQ}{T} = \int_{T_1}^{T_2} \frac{k_v \cdot dT}{T} = k_v \int_{T_1}^{T_2} \frac{dT}{T}$

$$= k_v \Big[\log_e (T) \Big]_{T_1}^{T_2}$$

$$= k_v \log_e \left(\frac{T_2}{T_1} \right) = k_v \log_e \left(\frac{p_2}{p_1} \right) \text{ per kg} \quad ...(4.6)$$

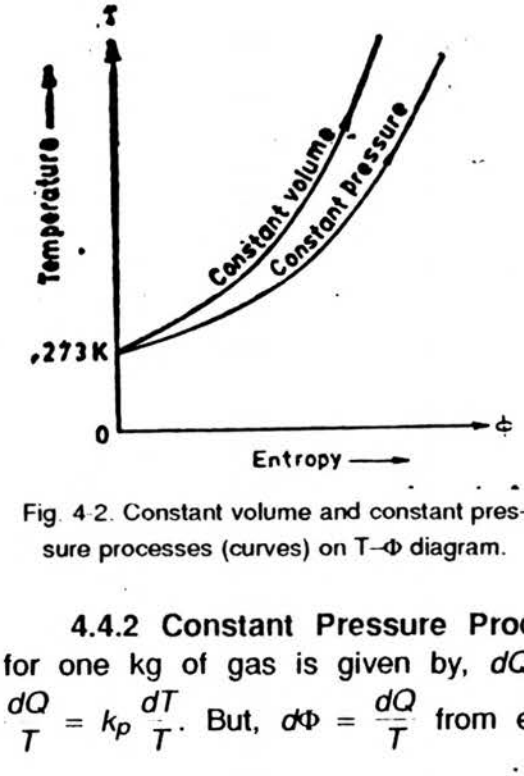


Fig. 4-2. Constant volume and constant pressure processes (curves) on T–Φ diagram.

Alternatively, eqn. (4.6) can be obtained directly from eqn. (4.3) by putting $v_2 = v_1$.

Figure 4-2 shows the plot of the eqn. (4.3) for constant volume process assuming value of Φ_1 as zero at 273 K, i.e. datum is taken as 0°C.

Positive value of change of entropy will show that heat is added and entropy has increased, and negative value will show that heat is abstracted (removed) and entropy has decreased.

4.4.2 Constant Pressure Process : In constant pressure process, heat supplied for one kg of gas is given by, $dQ = k_p \cdot dT$. Dividing both the sides by T, we have, $\frac{dQ}{T} = k_p \frac{dT}{T}$. But, $d\Phi = \frac{dQ}{T}$ from eqn. (4.1). $\therefore\ d\Phi = k_p \frac{dT}{T}$

Then, total change in entropy, $\Phi_2 - \Phi_1 = k_p \int_{T_2}^{T_2} \frac{dT}{T} = k_p \Big[\log_e (T) \Big]_{T_1}^{T_2}$

$$\therefore\ \Phi_2 - \Phi_1 = k_p \log_e \left(\frac{T_2}{T_1} \right) = k_p \log_e \left(\frac{v_2}{v_1} \right) \text{ per kg} \quad ...\ (4.7)$$

In this case also k_p is assumed constant between the temperature range of T_1 to T_2. The eqn. (4.7) can be obtained directly from eqn. (4.4) by putting $p_2 = p_1$. Figure 4-2 shows the plot of entropy change for constant pressure process, taking datum as 0°C or 273 K.

Problem-3 : *Two kg of gas are heated at constant volume from a temperature of 4°C to a temperature of 95°C. What is the change of entropy ? Assume* k_v *for gas as 1·415 kJ/kg K.*

Using eqn. (4.6), for 1 kg of gas,

$$\text{Change of entropy, } \Phi_2 - \Phi_1 = k_v \log_e\left(\frac{T_2}{T_1}\right) = 1{\cdot}415 \times \log_e\left(\frac{95+273}{4+273}\right)$$

$$= 1{\cdot}415 \times 0{\cdot}2839 = 0{\cdot}4017 \text{ units of entropy (increase)}$$

$\therefore$ Total change of entropy = 2 × 0·4017 = **0·8034 units of entropy (increase).**

Problem-4 : *Two kg of gas increases in temperature from 4°C to 95°C at constant pressure. Assuming* k_p *for gas = 1 kJ/kg K, calculate the change of entropy.*

Using eqn. (4.7), for one kilogram of gas,

$$\text{Change in entropy } \Phi_2 - \Phi_1 = k_p \log_e\left(\frac{T_2}{T_1}\right) = 1 \times \log_e\left(\frac{95+273}{4+273}\right)$$

$$= 1 \times 0{\cdot}2839 = 0{\cdot}2839 \text{ units of entropy.}$$

$\therefore$ Total change of entropy = 2 × 0·2839 = **0·5678 units of entropy (increase)**

4.4.3 Isothermal Process : Isothermal expansion process is represented on the $T-\Phi$ diagram (fig. 4-3) by horizontal line 1-2, as the temperature remains constant during the process.

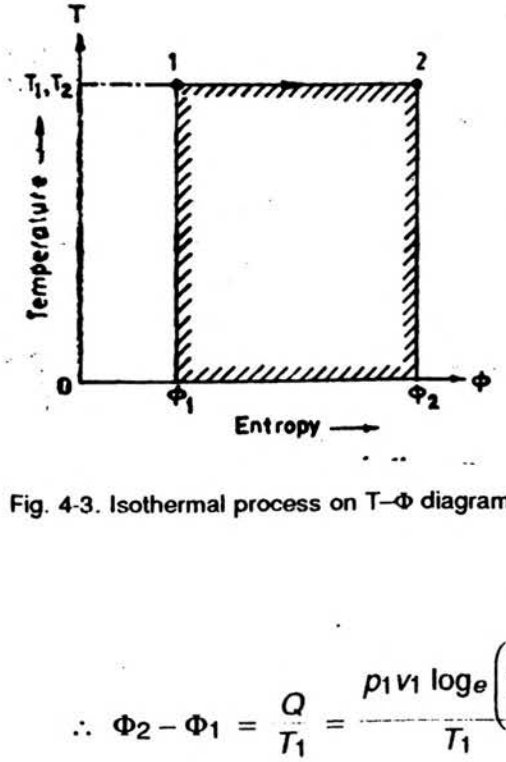


Fig. 4-3. Isothermal process on T–Φ diagram.

The heat transferred is represented by the area under the curve 1-2. During an isothermal process, the transferred heat is equal to the work done as the change of internal energy is zero. Therefore, the shaded area under the curve 1-2 will also represent the work done by the gas.

$$\therefore \text{Change of entropy, } \Phi_2 - \Phi_1 = \frac{Q}{T_1} \text{ or } \frac{Q}{T_2}$$

As mentiioned above, for isothermal process no change of internal energy occurs and thus, heat supplied (Q) is equal to work done (W) by the gas. But, work done during isothermal process,

$$W = p_1 v_1 \log_e\left(\frac{v_2}{v_1}\right)$$

$$\therefore \Phi_2 - \Phi_1 = \frac{Q}{T_1} = \frac{p_1 v_1 \log_e\left(\frac{v_2}{v_1}\right)}{T_1} = \frac{mRT_1}{T_1} \log_e\left(\frac{v_2}{v_1}\right) \qquad (\because p_1 v_1 = mRT_1)$$

$$= mR \log_e\left(\frac{v_2}{v_2}\right)$$

$$\therefore\ \Phi_2 - \Phi_1 = R\log_e\left(\frac{v_2}{v_1}\right) = R\log_e\left(\frac{p_1}{p_2}\right)\ \text{per kg} \quad \text{... (4.8)}$$

Alternatively, eqn. (4.8) can be obtained directly from eqn. (4.3) by putting $T_1 = T_2$.

Problem-5 : *Two kg of air at pressure of 10 bar and a volume of 0·03 m^3 is expanded at constant temperature until its pressure is 3·5 bar. Calculate the change of entropy during the expansion, taking R for air as 0·287 kJ/kg K.*

Using equation for isothermal process, $p_1v_1 = p_2v_2$,

$$\text{i.e. } v_2 = \frac{p_1v_1}{p_2} = \frac{10\times10^5\times0{\cdot}03}{3{\cdot}5\times10^5} = 0{\cdot}0857\ \text{m}^3$$

Using eqn. (4.8), change of entropy per kg of air,

$$\Phi_2 - \Phi_1 = R\log_e\left(\frac{v_2}{v_1}\right) = 0{\cdot}287\times\log_e\left(\frac{0{\cdot}0857}{0{\cdot}03}\right) = 0{\cdot}3012\ \text{units of entropy}$$

∴ Total change of entropy = 2 × 0·3012 = **0·6024 units of entropy (increase)**

4.4.4 Frictionless Adiabatic Process: During frictionless adiabatic process heat is not transferred to or from the working substance both from internal and external sources and the work is done at the expense of the internal energy of the substance, hence, $dQ = 0$.

$$\text{i.e. } d\Phi = \frac{dQ}{T} = 0$$

This means that the entropy remains constant during frictionless adiabatic process. Many authors define this process as an adiabatic process. The better name for this process is isentropic process. *In an adiabatic process, if friction is present there is going to be increase in entropy, i.e. it is not an isentropic process.*

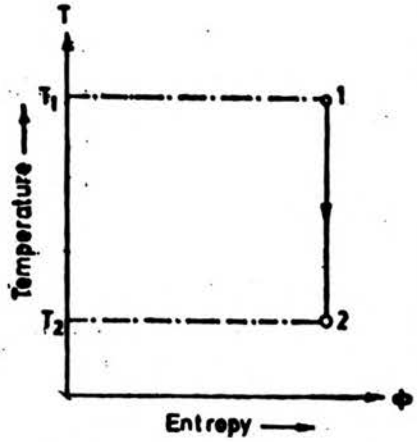


Fig. 4-4. Isentropic process on T–Φ diagram.

The isentropic process on T–Φ or H–Φ diagram is represented by vertical straight line 1-2 as shown in fig. 4-4.

For an isentropic process from eqn. (4.5),

$$\Phi_2 - \Phi_1 = k_v\log_e\left(\frac{p_2}{p_1}\right) + k_p\log_e\left(\frac{v_2}{v_1}\right) = 0$$

$$\therefore\ \log_e\left(\frac{p_2}{p_1}\right) = -\frac{k_p}{k_v}\log_e\left(\frac{v_2}{v_1}\right) = -\gamma\log_e\left(\frac{v_2}{v_1}\right) = \gamma\log_e\left(\frac{v_1}{v_2}\right) = \log_e\left(\frac{v_1}{v_2}\right)^{\gamma}$$

Taking antilog of both sides, $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma}$ i.e. $p_1v_1^{\gamma} = p_2v_2^{\gamma}$

This is the law of isentropic process as derived in Chapter 2.

4.4.5 Polytropic Process : In a polytropic process, pressure, volume, temperature and also entropy changes take place. Taking the expression of change of entropy from eqn. (4.3),

$$\Phi_2 - \Phi_1 = k_v \log_e\left(\frac{T_2}{T_1}\right) + R\log_e\left(\frac{v_2}{v_1}\right)$$

Using the polytropic law as pv^n = constant, $p_1v_1^n = p_2v_2^n$ i.e. $\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$

But, from characteristic equation for perfect gas, $\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$ i.e. $\frac{p_1}{p_2} = \frac{v_2}{v_1} \times \frac{T_1}{T_2}$

Equating these two values of $\frac{p_1}{p_2}$,

we have, $\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{n-1}$ from whcih $\frac{v_2}{v_1} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{n-1}}$

Hence,

$$\begin{aligned}\Phi_2 - \Phi_1 &= k_v \log_e\left(\frac{T_2}{T_1}\right) + R \times \frac{1}{n-1}\log_e\left(\frac{T_1}{T_2}\right)\\ &= k_v\log_e\left(\frac{T_2}{T_1}\right) - \frac{k_p - k_v}{n-1}\log_e\left(\frac{T_2}{T_1}\right)\\ &= \left(k_v - \frac{k_p - k_v}{n-1}\right)\log_e\left(\frac{T_2}{T_1}\right) = k_v\left[1 - \frac{\gamma - 1}{n-1}\right]\log_e\left(\frac{T_2}{T_1}\right)\\ &= k_v\left[\frac{n-\gamma}{n-1}\right]\log_e\left[\frac{T_2}{T_1}\right] \text{ per kg} \end{aligned} \quad ...(4.9a)$$

The change of entropy in a polytropic process can also be expressed in terms of pressures and volumes by using characteristic equations

$$\Phi_2 - \Phi_1 = k_v\left(\frac{n-\gamma}{n}\right)\log_e\left(\frac{p_2}{p_1}\right) \text{ per kg and} \quad ...(4.9b)$$

$$\Phi_2 - \Phi_1 = k_v(\gamma - n)\log_e\left(\frac{v_2}{v_1}\right) \text{ per kg} \quad ...(4.9c)$$

Problem-6 : *One kg of a perfect gas is compressed according to the law $pv^{1.3}$ = constant and the temperature is raised from 5°C to 157°C. Assuming the characteristic gas constant $R = 0.275$ kJ/kg K and value of $k_p = 1.045$ kJ/kg K, evaluate the change of entropy.*

Now $T_1 = 5 + 273 = 278$ K; $T_2 = 157 + 273 = 430$ K.

As $k_v = k_p - R = 1.045 - 0.275 = 0.77$ kJ/kg K, $\gamma = \frac{k_p}{k_v} = \frac{1.045}{0.77} = 1.357$.

Using eqn.(4.9a) for polytropic process, change of entropy,

$$\Phi_2 - \Phi_1 = k_v\left(\frac{n-\gamma}{n-1}\right)\log_e\left(\frac{T_2}{T_1}\right)$$

$$\therefore\ \Phi_2 - \Phi_1 = 0.77\left(\frac{1.3 - 1.357}{1.3 - 1}\right)\log_e\left(\frac{430}{278}\right)$$

$= -0.0638$ **units of entropy (decrease)**

Problem-7 : *The air in the cylinder of an internal combustion engine at the beginning of compression stroke occupies 0.013 m^3, the pressure is 100 kPa and the temperature is 100°C. It is compressed to 0.001 m^3 according to the law pv^n = constant and the pressure then is 2,800 kPa. Heat is now added at constant volume until the pressure reaches 5,600 kPa. Find the change of entropy during each operation, and state whether it is an increase or decrease. Assume that specific heats of air remain constant and k_p = 1.0035 kJ/kg K and k_v = 0.7165 kJ/kg K.*

Here, $T_1 = 100 + 273 = 373$ K.

For polytropic compression, $\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n$ i.e. $\frac{2,800}{100} = \left(\frac{0.013}{0.001}\right)^n$ or $28 = (13)^n$

$$\therefore\ n = \frac{\log(28)}{\log(13)} = \frac{1.4472}{1.1139} = 1.3 \text{ and } \gamma = \frac{k_p}{k_v} = \frac{1.0035}{0.7165} = 1.4$$

and $R = k_p - k_v = 1.0035 - 0.7165 = 0.287$ kJ/kg K

Now, $p_1v_1 = mRT_1 \quad \therefore\ m = \frac{p_1v_1}{RT_1} = \frac{100 \times 0.013}{0.287 \times 373} = 0.0121$ kg

Also, $\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} \quad \therefore\ \frac{T_2}{T_1} = \frac{p_2v_2}{p_1v_1} = \frac{2,800 \times 0.001}{100 \times 0.013} = 2.15$

Using eqn. (4.9a), change of entropy during polytropic compression,

$$\Phi_2 - \Phi_1 = m \times k_v\left(\frac{n-\gamma}{n-1}\right)\log_e\left(\frac{T_2}{T_1}\right) = 0.0121 \times 0.7165 \times \left(\frac{1.3-1.4}{1.3-1}\right)\log_e(2.15)$$

$$= -\ \mathbf{0.0022 \text{ units of entropy (decrease)}}$$

Using eqn. (4.6), change of entropy during constant volume heat addition process,

$$\Phi_2 - \Phi_1 = m \times k_v \log_e\left(\frac{p_3}{p_2}\right) = 0.0121 \times 0.7165 \log_e\left(\frac{5,600}{2,800}\right)$$

$$= \mathbf{0.006 \text{ units of entropy (increase)}}$$

4.5 Estimation of Heat Supplied

The curve on a T–Φ diagram representing the heating of gas is usually of a very small curvature, as shown in fig. 4-5. In fact it is so slightly curved that for a small range of temperature, it can be assumed to be a straight line for approximate calculations.

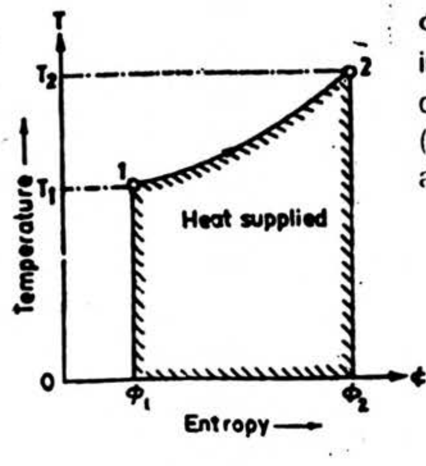


Fig. 4-5.

Let one kg of a gas at temperature T_1 and entropy Φ_1 be heated to a temperature T_2 when the entropy is increased to Φ_2. This operation is represented on the T–Φ diagram (fig. 4-5) by the curve 1-2. The heat absorbed (added) during the process will be represented by the area under the curve 1-2.

For approximate calculations, heat absorbed (added)

= area under the curve 1-2

= Change of entropy × Mean absolute temperature

$$= (\Phi_2 - \Phi_1)\left(\frac{T_1 + T_2}{2}\right) \quad \text{... (4.10)}$$

Thus, heat supplied in the process from state point 1 to state point 2 is approximately equal to change of entropy multiplied by the mean absolute temperature. The approximation is sometimes used when dealing with superheated steam.

This approximation is also sometimes used for calculating the coefficient of performance of vapour compression refrigerating plants by assuming liquid line on T–Φ diagram to be a straight line.

Problem-8 : *0·3 m^3 of air of mass 1 kg at an initial pressure of 5·5 bar expands to a final volume of 0·5 m^3. If the expansioin is according to the law $pv^{1{\cdot}3} = C$, find; (i) the change in internal energy, and (ii) the heat absorbed (added) during the process, and show that it is approximately equal to change of entropy multiplied by the mean absolute temperature. Assume $k_v = 0{\cdot}7165$ kJ/kg K, $\gamma = 1{\cdot}4$ and $R = 0{\cdot}287$ kJ/kg K for air.*

(i) Considering polytropic expansion,

$$p_1 v_1^{\,n} = p_2 v_2^{\,n} \text{ i.e. } p_2 = p_1 \left(\frac{v_1}{v_2}\right)^n \text{ i.e. } p_2 = 5{\cdot}5 \left(\frac{0{\cdot}3}{0{\cdot}5}\right)^{1{\cdot}3} = 2{\cdot}83 \text{ bar}$$

Further, $T_1 = \dfrac{p_1 v_1}{mR} = \dfrac{(10^5 \times 5{\cdot}5) \times 0{\cdot}3}{1 \times 0{\cdot}287 \times 10^3} = 575$ K

and $T_2 = \dfrac{p_2 v_2}{mR} = \dfrac{(10^5 \times 2{\cdot}83) \times 0{\cdot}5}{1 \times 0{\cdot}287 \times 10^3} = 493$ K

Change in internal energy, $u_2 - u_1 = mk_v\,(T_2 - T_1)$

$$= 1 \times 0{\cdot}7165\ (493 - 575) = -\ \mathbf{58{\cdot}75\ kJ}$$

(ii) Heat transferred, $Q = \dfrac{\gamma - n}{\gamma - 1} \times \text{work done} = \dfrac{\gamma - n}{\gamma - 1} \times \dfrac{p_1 v_1 - p_2 v_2}{(n-1)}$

$$= \frac{\gamma - n}{\gamma - 1} \times R\,(T_1 - T_2) \times \frac{1}{n-1}$$

$$= \frac{1{\cdot}4 - 1{\cdot}3}{1{\cdot}4 - 1} \times 0{\cdot}287\,(575 - 493) \times \frac{1}{1{\cdot}3 - 1}$$

$$= \mathbf{19{\cdot}53\ kJ\ (heat\ added)}$$

Using eqn. (4.9a), change of entropy during polytropic expansion per kg of air,

$$\Phi_2 - \Phi_1 = k_v \left(\frac{n - \gamma}{n - 1}\right) \log_e \left(\frac{T_2}{T_1}\right) = 0{\cdot}7165 \times \left(\frac{1{\cdot}3 - 1{\cdot}4}{1{\cdot}3 - 1}\right) \log_e \left(\frac{493}{595}\right)$$

$$= 0{\cdot}0365 \text{ units of entropy}$$

Mean absolute temperature $= \dfrac{T_1 + T_2}{2} = \dfrac{575 + 493}{2} = 534$ K

Heat absorbed = Change of entropy × Mean absolute temperature

$$= (\Phi_2 - \Phi_1)\left(\frac{T_1 + T_2}{2}\right) = 0{\cdot}0365 \times 534 = \mathbf{19{\cdot}479\ kJ}.$$

This approximate result is very close to the correct value of heat transferred (Q) calculated above.

4.6 Entropy of Steam

Steam is a vapour used as working fluid or working substance in the operation of steam engines and steam turbines which transform some of the heat energy supplied to the steam in its formation into mechanical work. Although steam is not considered as a perfect gas on account of it being a mixture of dry steam (gas) and moisture (water), it possesses properties like those of gases, namely, pressure, volume, temperature, internal energy, enthalpy or total heat content and entropy. *The pressure, volume and temperature of steam as a vapour are not connected by any simple relationship such as is expressed by the characteristic equation for a perfect gas.* Properties of steam were first investigated experimentally by Regnault and subsequently investigated by Prof. Callender by carrying out extensive thermodynamic experiments by means of electrical calorimeters. Tables giving the properties of steam in each condition were compiled from the study of the equations derived experimentally. These properties are also represented on charts.

The quantities tabulated in the *steam tables* are the pressure, saturation temperature, specific volume, enthalpy (total heat) and entropy. Pressure, temperature and volume can be given their actual absolute values; whereas, enthalpy and entropy are purely relative quantities. They are measured relatively from convenient datum condition and calculated for one kg of steam. For steam, datum point is fixed as the condition of the water at a temperature of 0°C. Thus, the enthalpy, internal energy and entropy of water at 0°C are taken as zero. All their values measured above 0°C are considered as positive and those measured below 0°C are taken as negative. *The general conservation of energy equation is applicable to steam in the same manner as it is applicable to gases.* The properties of steam and the changes in the properties can be determined by using steam tables and enthalpy-entropy (H–Φ) chart for steam.

The entropy of steam is only considered here but the formulae deduced for steam are equally applicable to other vapours. The process of generation of steam takes place in a boiler where the heating of the water and its evaporation takes place at constant pressure. If the steam so formed is superheated by passing it through a superheater, the process of superheating also takes place at constant pressure, that is, at the steam pressure existing in the boiler.

4.6.1 Entropy of Water : In general, addition of heat, dQ to one kg of water will cause its temperature to rise by dT.

Then, $dQ = k \cdot dT$ where k = specific heat of water.

By definition, $d\Phi = \frac{dQ}{T}$ $\therefore\ dQ = \frac{k \cdot dT}{T} = k \cdot \frac{dT}{T}$

Integrating between temperature limits of T_1 and T_2,

$$\Phi_2 - \Phi_1 = k \int_{T_1}^{T_2} \frac{dT}{T} = k\Big[\log_e(T)\Big]_{T_1}^{T_2} = k \log_e\left(\frac{T_2}{T_1}\right)$$

As explained earlier, the heating of water in the boiler takes place at constant pressure. Therefore, k may be replaced by k_p. Entropy of water at 0°C is taken as zero. The value of entropy of water reckoned from freezing temperature (T_o), and upto the temperature of formation (boiling temperature T_s) is written as Φ_w or Φ_l.

Then, entropy of water, $\Phi_w = k_p \log_e \left(\frac{T_s}{T_o} \right)$ ...(4.11)

where kp is specific heat of water ≈ 4·187 kJ/kg K.

Value of Φ_w can also be obtained directly from the steam tables. By varying value of T_s, a logarithmic curve shown in fig. 4-6 is obtained for heating of water at constant pressure. The curve is known as *water line.*

4.6.2 Evaporation Entropy : Heat added during evaporation takes place at constant temperature if pressure is maintained constant. If the steam formed is wet having dryness fraction *x*,

Entropy of evaporation, $\Phi_e = \frac{xL}{T_s}$ ... (4.12a)

If the steam formed is dry saturated, $x = 1$, then

Entropy of evaporation, $\Phi_e = \frac{L}{T_s}$ ... (4.12b)

The value of evaporation entropy, $\Phi_e = \Phi_s - \Phi_w$ may also be obtained from the steam tables.

4.6.3 Wet Steam : If the steam formed is wet having dryness fraction *x*, total entropy of wet steam reckoned from freezing point of water (0°C),

$$\Phi_{wet} = \Phi_w + \frac{xL}{T_s} = \Phi_w + x\Phi_e$$

$$= \Phi_w + x(\Phi_s - \Phi_w) \quad ...(4.13)$$

4.6.4 Dry Saturated Steam : The total entropy (Φ_s) of dry saturated steam reckoned from the freezing point of water (0°C) is equal to the sum of water entropy (Φ_w) and evaporation entropy (Φ_e),

i.e. $\Phi_s = k_p \log_e \left(\frac{T_s}{T_o} \right) + \frac{L}{T_s} = \Phi_w + \Phi_e$ ...(4.14)

The value of Φ_s may also be obtained directly from the steam tables.

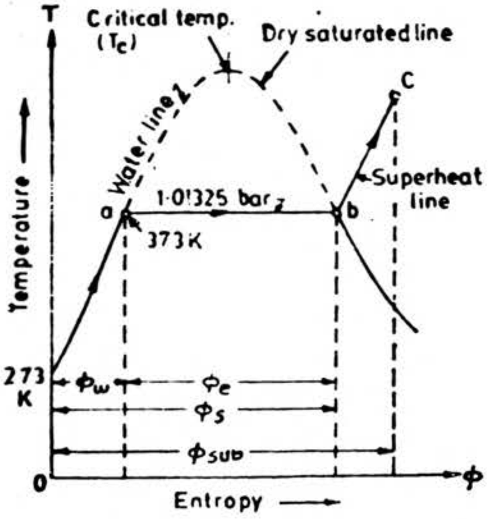


Fig. 4 6. T Φ curve for heating of water and steam at constant pressure.

By varying value of pressure of steam, i.e. value of temperature of steam, a curve is obtained as shown in fig. 4-6. This curve is known as *dry saturated steam line.* Horizontal distance *ab* between water and dry saturation lines is equal to the evaporation entropy, Φ_e. Since value of enthalpy of evaporation decreases as the pressure or saturation temperature is increased, evaporation entropy decreases with the increase in saturation temperature. Thus, water and dry saturation lines come nearer to each other as saturation temperature is increased. The point at which they meet is known as *critical point* and the corresponding temperature and pressure are known as critical values. Critical temperature point is shown in fig.4-6. For steam, critical temperature is 374·14°C and critical pressure is 220·9 bar.

4.6.5 Superheated Steam : During superheating, the heat is supplied to the steam at constant pressure. If the saturated steam at T_s degree absolute, be superheated to T_{sup} degree absolute and the specific heat of superheated steam over this range of temperature is k_p.

Then, increase of entropy during superheating $= \Phi_{sup} - \Phi_s = k_p \log_e\left(\frac{T_{sup}}{T_s}\right)$

And the total entropy of 1 kg of superheated steam reckoned above the freezing temperature of water,

$$\Phi_{sup} = k_p \log_e \frac{T_s}{T_o} + \frac{L}{T_s} + k_p \log_e\left(\frac{T_{sup}}{T_s}\right)$$

$$= \Phi_w + \Phi_e + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) = \Phi_s + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) \quad \ldots (4.15)$$

The value of increase in entropy during superheating is usually to be calculated. The total entropy of superheated steam may also be taken directly from the steam tables.

Specific heat of superheated steam (k_p) varies with the pressure and with the degree of superheating, increasing with the pressure and diminishing with the degree of superheating. The value of k_p varies between 2 kJ/kg K and 2·3 kJ/kg K.

For different values of T_{sup} and fixed value of T_s, i.e. for varying degrees of superheat, various points are obtained on T–Φ chart. The curve is then drawn through the points so obtained; then this curve represents the entropy of superheated steam at the chosen pressure, for all temperatures of superheat. This curve is a constant pressure line and is known as *superheat line* (line *b-c* in fig. 4-6). Similar superheat lines can be drawn for superheated steam at other pressures.

It should be noted that the area between the water line and the dry saturation line on the T–Φ chart represents wet steam. The area of the chart to the right of the dry saturated steam line represents superheated steam, and is known as *area of superheat.* The area of the chart to the left of water line is known as *water region.*

Problem-9 : *Calculate from first principles the entropy of 1 kg of boiling water at a pressure of 2 bar. Compare the value with the corresponding value obtained directly from steam tables.*

From steam tables, at 2 bar, t_s = 120·23°C and Φ_w = 1·5301

Using eqn. (4.11), entropy of water,

$$\Phi_w = k_p \log_e\left(\frac{T_s}{T_o}\right) = 4{\cdot}187 \times \log_e\left(\frac{120{\cdot}23 + 273}{0 + 273}\right)$$

$$= 4{\cdot}187 \times 0{\cdot}3649 = \mathbf{1{\cdot}5281 \text{ units of entropy}}.$$

The corresponding value obtained from steam tables is 1·5301 which is very close to value calculated.

Problem-10 : *Calculate from first principles the evaporation entropy of 1 kg of dry saturated steam at a pressure of 7 bar and compare it with the value obtained from the steam tables.*

From steam tables, at 7 bar,

$$t_s = 164{\cdot}97^\circ\text{C},\ L = 2{,}066{\cdot}3 \text{ kJ/kg},\ \Phi_w = 1{\cdot}9922 \text{ and } \Phi_s = 6{\cdot}7080$$

Using eqn. (4.12), entropy of evaporation,

$$\Phi_e = \frac{L}{T_s} = \frac{2{,}066{\cdot}3}{164{\cdot}97 + 273} = 4{\cdot}7197 \textbf{ units of entropy.}$$

The corresponding value obtained from steam tables is $\Phi_s - \Phi_w = 6{\cdot}708 - 1{\cdot}9922 = 4{\cdot}7158$ units of entropy which is very close to the value calculated.

Problem-11 : *Using the steam tables, determine the entropy of one kg of steam at a pressure of 15 bar under the following conditions : (i) steam is dry saturated, (ii) steam is wet, the dryness fraction being 0·9, and (iii) steam is superheated, the degree of superheat being 100°C. Take k_p of superheated steam at 15 bar as 2·3 kJ/kg K.*

From steam tables, at 15 bar, $\Phi_w = 2{\cdot}315$, $\Phi_s = 6{\cdot}4448$ and $t_s = 198{\cdot}32^\circ\text{C}$.

(i) Entropy of dry saturated steam, $\Phi_s = 6{\cdot}4448$ **units of entropy** (from steam tables)

(ii) Entropy of wet steam, $\Phi_{wet} = \Phi_w + x\Phi_e = \Phi_w + x(\Phi_s - \Phi_w)$

$$= 2{\cdot}315 + 0{\cdot}9\,(6{\cdot}4448 - 2{\cdot}315) = 6{\cdot}0318 \textbf{ units of entropy}$$

(iii) Using eqn. 4.15, entropy of superheated steam,

$$\Phi_{sup} = \Phi_s + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) = 6{\cdot}4448 + 2{\cdot}3 \log_e\left[\frac{198{\cdot}32 + 100 + 273}{198{\cdot}32 + 273}\right]$$

$$= 6{\cdot}8869 \textbf{ units of entropy.}$$

Problem-12 : *Using the steam tables, estimate the change of entropy of 1 kg of steam at a pressure of 7·5 bar and 0·8 dry when heated at constant pressure to superheat it by 15°C. Assume k_p for superheated steam as 2·1 kJ/kg K.*

From steam tables, at 7·5 bar, $t_s = 167{\cdot}78^\circ\text{C}$, $\Phi_w = 2{\cdot}02$ and $\Phi_s = 6{\cdot}6847$

Entropy of wet steam, $\Phi_{wet} = \Phi_w + x(\Phi_s - \Phi_w)$

$$= 2{\cdot}02 + 0{\cdot}8\,(6{\cdot}6847 - 2{\cdot}02) = 5{\cdot}7678 \text{ units of entropy}$$

Now, $T_{sup} = 167{\cdot}78 + 273 + 15 = 455{\cdot}78$ K and $T_s = 167{\cdot}78 + 273 = 440{\cdot}78$ K

Using eqn. (4.15), entropy of superheated steam,

$$\Phi_{sup} = \Phi_s + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) = 6{\cdot}6847 + 2{\cdot}1 \log_e\left(\frac{455{\cdot}78}{440{\cdot}78}\right)$$

$$= 6{\cdot}7561 \text{ units of entropy}$$

$\therefore$ Change of entropy $= 6{\cdot}7561 - 5{\cdot}7678 = 0{\cdot}9883$ **units of entropy (increase).**

Problem-13 : *Calculate from first principles the entropy of 3 kg of steam at 14 bar and 360°C. Take k_p of superheated steam as 2·1 kJ/kg K,*

From steam tables, at 14 bar, $t_s = 195{\cdot}07^\circ\text{C}$ and $L = 1{,}959{\cdot}7$ kJ/kg.

Entropy of water, $\Phi_w = k_p \log_e\left(\dfrac{T_s}{T_o}\right) = 4{\cdot}187 \log_e\left(\dfrac{195{\cdot}07 + 273}{0 + 273}\right) = 2{\cdot}256$

Entropy of evaporation, $\Phi_e = \dfrac{L}{T_s} = \dfrac{1{,}959{\cdot}7}{195{\cdot}07 + 273} = 4{\cdot}186$

Increase of entropy due to superheating,

$$\Phi_{sup} - \Phi_s = k_p \log_e \frac{T_{sup}}{T_s} = 2{\cdot}1 \log_e \left(\frac{360 + 273}{195{\cdot}07 + 273} \right) = 0{\cdot}6348$$

Total entropy of 1 kg of superheated steam (at 14 bar and 360°C),

$$\Phi_{sup} = \Phi_w + \Phi_c + (\Phi_{sup} - \Phi_s) = 2{\cdot}256 + 4{\cdot}186 + 0{\cdot}6348$$

$$= 7{\cdot}0768 \text{ units of entropy.}$$

∴ Total entropy of 3 kg of steam = 3 × 7·0768 = **21·2304 units of entropy.**

4.7 Temperature-Entropy Diagram for Steam

This diagram is very useful and convenient for the graphical solution of problems concerning the properties of steam. The curves shown in fig. 4-7 represent changes in temperature and the corresponding change in entropy. The ordinate represents the absolute temperature and the abscissa represents the units of entropy.

The state path *o-a-b-c* shows the variation in temperature and entropy of a kilogram of water when it is heated from 0°C or 273 K to the saturation temperature corresponding to the given pressure, then vaporised at the same given pressure into dry saturated steam, and finally superheated by adding further heat. The curves are drawn by obtaining entropy values from the steam tables and plotting these values against the corresponding absolute temperature.

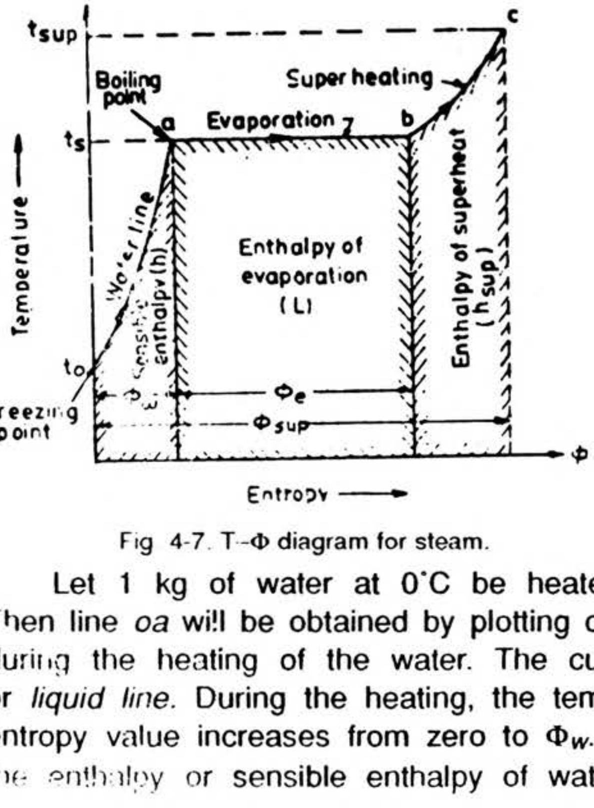


Fig 4-7. T–Φ diagram for steam.

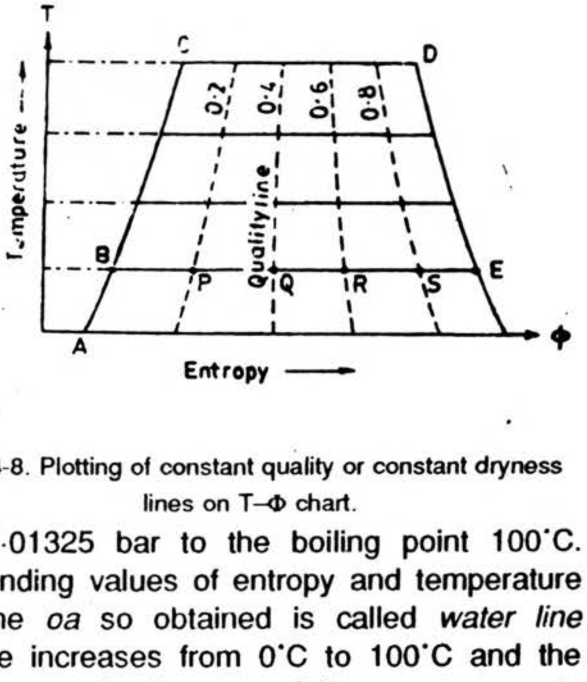


Fig. 4-8. Plotting of constant quality or constant dryness lines on T–Φ chart.

Let 1 kg of water at 0°C be heated at 1·01325 bar to the boiling point 100°C. Then line *oa* will be obtained by plotting corresponding values of entropy and temperature during the heating of the water. The curved line *oa* so obtained is called *water line* or *liquid line*. During the heating, the temperature increases from 0°C to 100°C and the entropy value increases from zero to Φ_w. The area under the curved line *oa* represents the enthalpy or sensible enthalpy of water (*h*).

Further addition of heat will cause evaporation of the water. The evaporation begins at point *a* and is completed at point *b*. The evaporation takes place at constant pressure and the heat added during the evaporation is the enthalpy of evaporation (*L*) which is represented by the area under the line *ab*. During the evaporation of the water the value of entropy increases from Φ_w to Φ_s. The line *ab* is known as the *evaporation line*.

Since the evaporation is completed at point *b*, the steam has received all the heat it can take up (enthalpy of evaporation) at the given saturation temperature. If entropy values are plotted against corresponding absolute temperature of evaporation (T_s) obtained from steam tables, a curve will be obtained which is known as *saturation line* or *dry steam line* (not shown in fig. 4-7). The water and the dry steam lines ultimately meet at the critical point, where the temperature is the critical temperature (374·14°C for steam). The dry saturation line and critical temperature point is shown in fig. 4-6.

Further addition of heat after the water is completely evaporated (at point *b*), will cause the temperature and entropy of steam to increase further as shown by the curved line *bc*. The curved line *bc* is known as *superheated line* and the area under the curved line *bc* represents the enthalpy of superheat. During the superheating of the steam the value of entropy increases from Φ_s to Φ_{sup}.

Constant quality or constant dryness lines may be drawn on T–Φ chart in the following manner. Referring to fig. 4-8, *ABC* is the water line and *DEF* is the dry saturation line on a T–Φ chart. As already explained, a horizontal line BE between the water line and dry saturation line represents the process of evaporation of one kilogram of water at a definite temperature. Point B represents one kilogramof water at the same temperature, while point E represents one kilogram of dry saturated steam at that temperature. It thus follows that any intermediate point on BE will represent one kilogram of wet steam.

Length BE represents the increase in entropy during evaporation (Φ_e), and since temperature is constant during evaporation, the heat added is proportional to the increase of entropy. But after evaporation begins, the amount of water evaporated is proportional to the heat added. thus, if BE be divided equally into five parts at point P, Q, R and S, these points will represent 0·2, 0·4, 0·6 and 0·8 of a kilogram of saturated steam respectively. In other words, the point P, Q, R and S, represent a kilogram of wet steam at a definite temperature and of dryness fraction 0·2, 0·4, 0·6 and 0·8 respectively. Simiarly, point B represents 0·0 (zero) dryness fraction and point E represents 1·0 dryness fraction. Dividing CD and other horizontal distances between the water line and dry saturation line into the same number of equal parts, and joining corresponding points by smooth curves as shown in fig. 4-8, lines of quality of dryness fraction 0·2, 0·4, 0·6 and 0·8 are obtained.

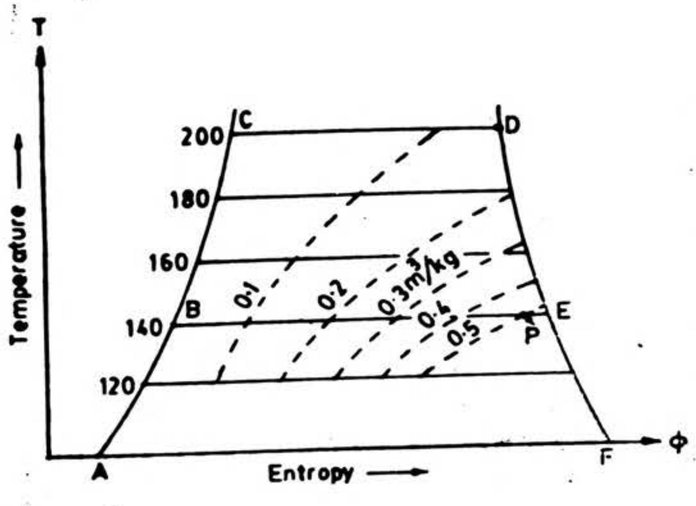


Fig. 4-9. Plotting of constant volume lines on T–Φ chart.

Constant volume lines may be drawn on T–Φ chart in the following manner. referring to fig. 4-9, ABC and DEF are portions of water and dry saturation lines on a T–Φ chart. BE is the horizontal distance between the water and dry saturation lines at temperature 140°C. Since distances from B along BE are proportional to mass of water evaporated, these distances will also be proportional to the volume of steam produced, the temperature and therefore the pressure being constant. Thus, the length *BE* may be taken to represent specific volume of steam at 140°C viz. 0·5089 m^3. If *P* is a point on *BE* such that *BP* represents 0·5 m^3 of steam, then $BP = 0{\cdot}5/0{\cdot}5089 \times BE$. Thus, *BP* is calculated. Dividing *BP* into five equal parts, points representing 0·1, 0·2, 0·3 and 0·4 m^3 are found. Repeating the above method for different horizontal distance between water and dry saturation lines, other points representing 0·1, 0·2, 0·3, 0·4, 0·5 m^3, etc. are found, and joining corresponding points by smooth curves as shown in fig. 4-9, lines of constant volume are drawn.

A fairly complete *T–Φ* chart for water and steam is given in fig. 4-10, but a *T–Φ* chart suitable for use in practice should be much larger and the quality lines and horizontal temperature lines should also be more numerous.

Isothermal changes : On a *T–Φ* chart, an isothermal expansion for wet steam is represented by horizontal line. the line *ab* (fig. 4-7) is an isothermal expansion representing the vaporisation of the water to steam. Similarly, the horizontal line in the superheat region represents the isothermal expansion of superheated steam, which behaves like a gas.

Isentropic changes : A frictionless adiabatic (isentropic) expansion is an expansion at constant entropy, since heat (external or internal) is not added or taken away. It is therefore, called isentropic. such an expansion will be represented by a vertical line on the *T–Φ* chart as shown in fig. 4-10. Vertical lines AB and CDE are two isentropics on the temperature-entropy diagram of fig. 4-10. Vertical line AB denotes isentropic expansion of wet steam, while line CDE denotes the isentropic expansion of superheated steam. It illustrates the fact that superheated steam when expands isentropically will become dry saturated as at D and wet as at E. If a kilogram of wet steam having dryness fraction x_1, pressure p_1 and temperature t_1, expands isentropically to pressure p_2 and temperature t_2, its final quality x_2 at the end of expansion can be found by equating the values of the entropies for the two conditions and solving for x_2 in the equation

$$\Phi_{w1} + x_1\,\Phi_{e1} = \Phi_{w2} + x_2\,\Phi_{e2}$$

$$\text{or } \Phi_{w1} + x_1\,(\Phi_{s1} - \Phi_{w1}) = \Phi_{w2} + x_2\,(\Phi_{s2} - \Phi_{w2}) \qquad \text{...(4.16)}$$

The value of x_1 being known, the other quantities except x_2 may be taken directly from the steam tables for the given pressures p_1 and p_2. However, the same result can be obtained easily and speedily by using a complete *T–Φ* chart and drawing a vertical line from the point representing initial condition of steam to meet pressure line p_2.

4.8 Mollier chart or Enthalpy-Entropy (H–Φ) Chart

Another entropy chart which is commonly used by engineers is the Mollier chart (fig. 4-11). This is plotted by taking enthalpy (total heat) and entropy as the two co-ordinates. The portion of the chart below the dry saturation line is the wet steam area, while the upper portion represents the superheated steam area. Constant pressure lines are plotted over the whole chart; they are straight lines in the wet region and curved lines in the superheat area. Lines of constant dryness fraction are plotted in the wet region and lines of constant temperature are plotted in the superheat region.

As the base of the *H–Φ* chart diagram represents the entropy, ideal adiabatic (isentropic) process will be represented by a vertical line on the *H–Φ* chart. Isentropic expansion is therefore also known as constant entropy expansion.

A throttling expansion will be represented by a horizontal line on *H–Φ* chart as

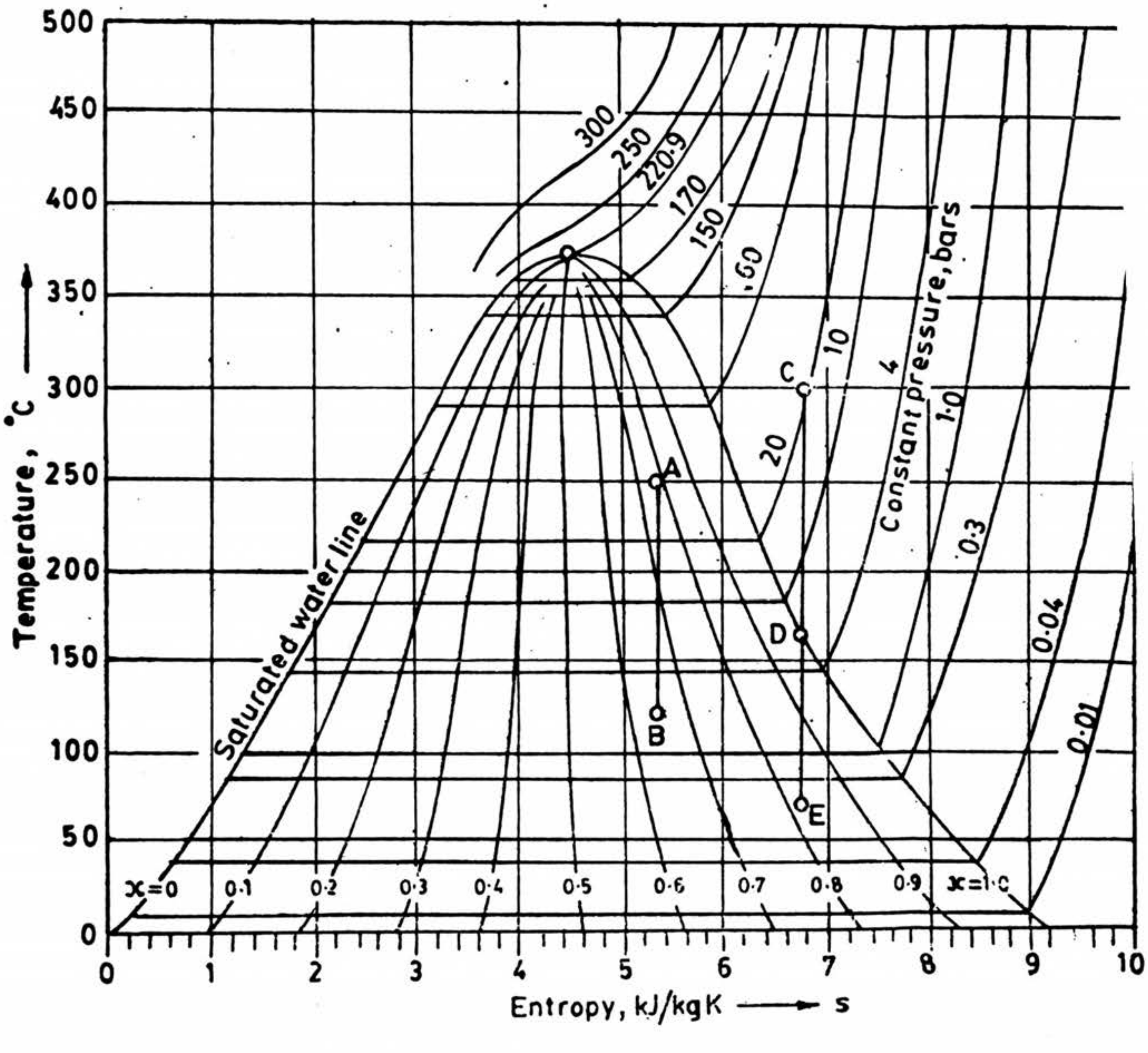


FIG. 4-10 TEMPERATURE ENTROPY DIAGRAM FOR STEAM

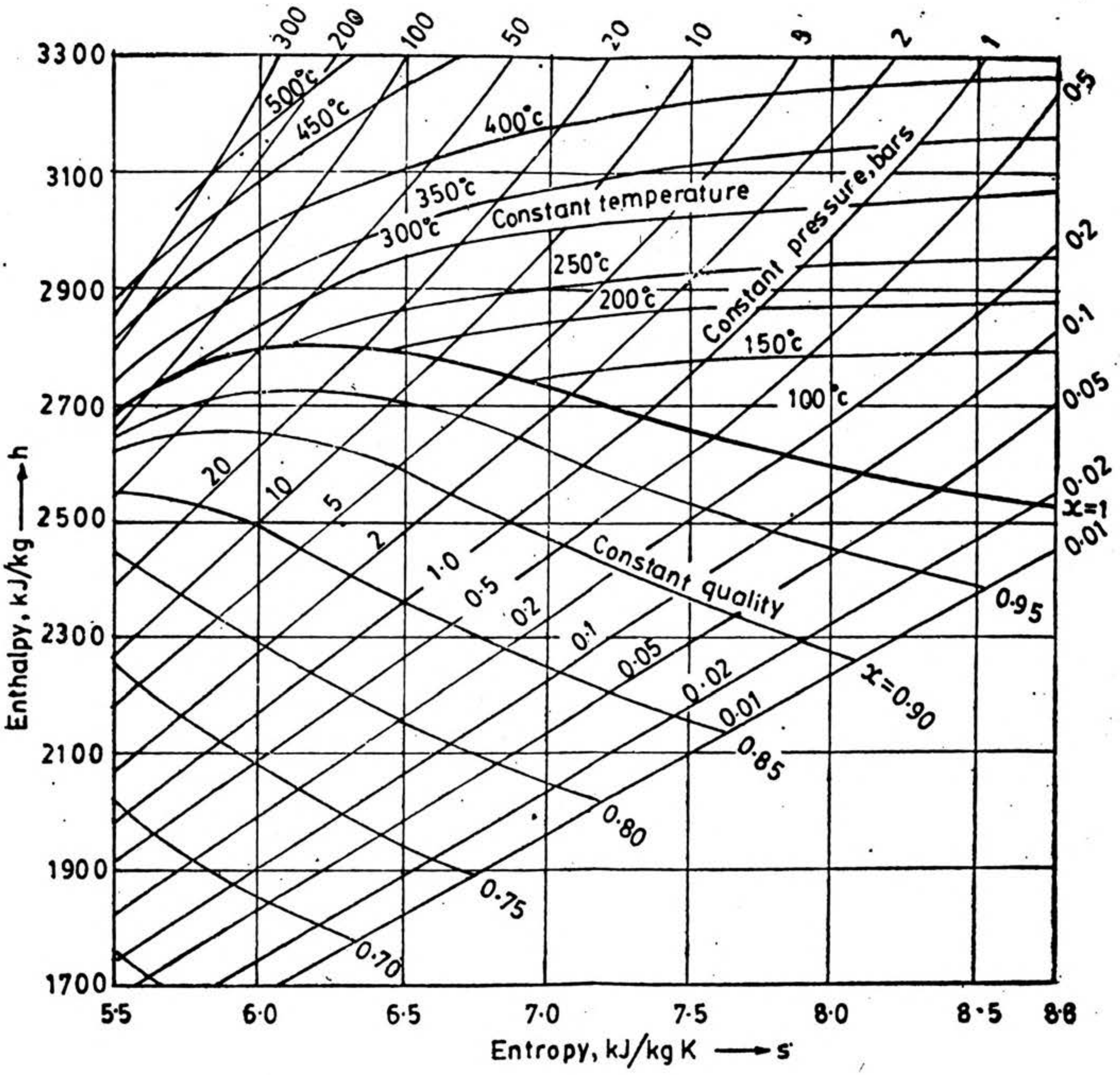


FIG. 4-11 ENTHALPY ENTROPY CHART FOR STEAM
(MOLLIER DIAGRAM FOR STEAM)

enthalpy remains constant during this process.

The main use of H–Φ chart is for finding out the drop in the enthalpy of steam during isentropic expansion. The final condition of the steam after isentropic expansion can also be read off directly from the H–Φ chart.

4.9 Summary of Formulae

Expressions for change of entropy of a perfect gas :

(ia) General case in terms of temperature and volumes

$$\Phi_2 - \Phi_1 = k_p \log_e\left(\frac{T_2}{T_1}\right) + R \log_e\left(\frac{v_2}{v_1}\right) \text{ per kg}$$

eqn. (4.3)

(ib) General case in terms of temperatures and pressures

$$\Phi_2 - \Phi_1 = k_p \log_e\left(\frac{T_2}{T_1}\right) - R \log_e\left(\frac{p_2}{p_1}\right) \text{ per kg}$$

eqn. (4.4)

(ic) General case in terms of pressures and volumes

$$\Phi_2 - \Phi_1 = k_v \log_e\left(\frac{p_2}{p_1}\right) + k_p \log_e\left(\frac{v_2}{v_1}\right) \text{ per kg}$$

eqn. (4.5)

(ii) Constant volume process

$$\Phi_2 - \Phi_1 = k_v \log_e\left(\frac{T_2}{T_1}\right) = k_v \log_e\left(\frac{p_2}{p_1}\right) \text{ per kg}$$

eqn. (4.6)

(iii) Constant pressure process

$$\Phi_2 - \Phi_1 = k_p \log_e\left(\frac{T_2}{T_1}\right) = k_p \log_e\left(\frac{v_2}{v_1}\right) \text{ per kg}$$

eqn. (4.7)

(iv) Isothermal or constant temperature process

$$\Phi_2 - \Phi_1 = R \log_e\left(\frac{v_2}{v_1}\right) = R \log_e\left(\frac{p_1}{p_2}\right) \text{per kg}$$

eqn. (4.8)

(v) Isentropic or frictionless adiabatic process, i.e. pv^γ = constant

$$\Phi_2 - \Phi_1 = 0$$

(vi) Polytropic process, i.e. pv^n = constant

$$\Phi_2 - \Phi_1 = k_v\left(\frac{n-\gamma}{n-1}\right)\log_e\left(\frac{T_2}{T_1}\right) \text{ per kg}$$

eqn. (4.9a)

$$\Phi_2 - \Phi_1 = k_v\left(\frac{n-\gamma}{n}\right)\log_e\left(\frac{p_2}{p_1}\right) \text{ per kg}$$

eqn. (4.9b)

$$\Phi_2 - \Phi_1 = k_v(\gamma - n) \log_e\left(\frac{v_2}{v_1}\right) \text{ per kg}$$

eqn. (4.9c)

Expressions for entropy of steam :

Entropy of water above freezing temperature, $\Phi_w = k_p \log_e\left(\frac{T_s}{T_o}\right)$ kJ/kg K

eqn. (4.11)

Entropy of evaporation of wet steam, $\Phi_s = \frac{xL}{T_s}$ kJ/kg K

eqn. (4.12a)

Entropy of evaporation of dry saturated steam, $\Phi_e = \frac{L}{T_s}$ kJ/kg K

eqn. (4.12b)

Total entropy of wet steam, $\Phi_{wet} = k_p \log_e\left(\frac{T_s}{T_o}\right) + \frac{xL}{T_s} = \Phi_w + x\,\Phi_e$

$$= \Phi_w + x(\Phi_s - \Phi_w) \text{ kJ/kg K}$$

eqn. (4.13)

Total entropy of dry saturated steam,

$$\Phi_s = k_p \log_e\left(\frac{T_s}{T_o}\right) + \frac{L}{T_s} = \Phi_w + \Phi_e \text{ kJ/kg K}$$

eqn. (4.14)

Total entropy of superheated steam, $\Phi_{sup} = \Phi_s + k_p \log_e\left(\frac{T_{sup}}{T_s}\right)$ kJ/kg K

eqn. (4.15)

where T_o is the freezing temperature of water in absolute scale (273 K),

T_s is the temperature of formation of steam in absolute scale,

T_{sup} is the absolute temperature of superheated steam, and

L is enthalpy of evaporation in kJ/kg.

Tutorial – 4

1. (a) Define the term 'entropy'.

   (b) Explain Clausius theorem.

2. Derive an expression for the change of entropy of a gas receiving heat at constant volume.

   One kg of air is heated at constant volume from a temperature of 40°C to temperature of 150°C. What is the change of entropy ? Assume $k_v = 0{\cdot}7165$ kJ/kg K for air.

   [0·2155 units of entropy (increase)]

3. Derive an expression for the change of entropy of a perfect gas receiving heat at constant pressure.

   0·2 kg of gas is heated from 727°C to 1,127°C at constant pressure. Assuming $k_p = 1{\cdot}0465$ kJ/kg K for gas, calculate the change of entropy.

   [0·07042 units of entropy (increase)]

4. Two kg of air at a pressure of 1,000 kPa and a volume of 0·03 m^3 is expanded at constant temperature until its pressure is 350 kPa. Calculate the change of entropy during the expansion, taking $R = 0{\cdot}287$ kJ/kg K for air,

   [0·6025 units of entropy (increase)]

5. 2 kg of gas increase in temperature from 727°C to 1,127°C at constant pressure. Assuming $k_p = 1{\cdot}05$ kJ/kg K for gas, calculate the change of entropy.

   [0·7066 units of entropy (increase)]

6. Show that the area under heating curve on temperature-entropy diagram represents heat supplied during the process.

   Find the change of entropy when 1 kg of air is compressed to one-fifth of the initial volume from an initial temperature and pressure of 7°C and 1 bar respectively, when

   (i) the compression is isothermal,

   (ii) the compression is ideal adiabatic (isentropic), and

   (iii) the compression is at constant pressure.

   Take $k_v = 0{\cdot}7165$ kJ/kg K and R = 0·287 kJ/kg K for air.

   [(i) – 0·4619 units (decrease); (ii) zero; (iii) – 1·165 units (decrease)]

7. 0·6 m^3 of gas at pressure of 7 bar and a temperature of 275°C is compressed at constant pressure to a volume of 0·35 m^3. Calculate the change of entropy of this quantity of gas. Take $R = 0{\cdot}287$ kJ/kg K and $k_v = 0{\cdot}7165$ kJ/kg K for the gas.

   [– 1·44 units (decrease)]

8. Two kg of air at 2 bar and 100°C changes its state to 1 bar and 37°C by polytropic expansion. Calculate the value of index of polytropic expansion and the change of entropy. Take k_p and k_v for air as 1·013 kJ/kg K and 0·7179 kJ/kg K respectively.

   [1·36, 0·04116 units]

9. A gas engine has a compression ratio of 6 to 1. The maximum pressure is limited to 28 bar. The pressure and temperature of the charge at the beginning of polytropic compression are 1 bar and 100°C respectively. The index of polytropic compression is 1·35. Calculate per kg of charge the change of entropy (i) during the compression and (ii) during the constant volume combustion process.

   Take $k_p = 1{\cdot}038$ kJ/kg K and $k_v = 0{\cdot}752$ kJ/kg K for the charge.

   [(i) –0·04042 units; (ii) 0·68605 units]

10. Derive an expression for the change of entropy of a gas during polytropic expansion.

    Two kilograms of oxygen are compressed polytropically from a pressure of 1 bar and 20°C to 7 bar. The gas constant R is 0·26 kJ/kg K, k_p is 0·908 kJ/kg K and the compression is according to the law $pv^{1.3}$ = constant. Determine the final temperature and the change of entropy.

    [186°C, – 0·1938 units]

11. A quantity of air having a volume of 0·12 m^3 at 1 bar and 20°C, is compressed according the law $pv^{1.15}$ = constant until the pressure is 8 bar. Find the change in internal energy and also the change of entropy of this quantity of air. Specific heats of air at constant pressure and constant volume are 0·9963 kJ/kg K and 0·7074 kJ/kg K respectively.

[9·21 kJ, – 0·6622 units]

12. One kg of air at 5°C and 1 bar undergoes a change until the temperature and volume become 275°C and 0·13 m^3 respectively. Determine the decrease of entropy. Take k_p = 1·0035 kJ/kg K and k_v = 0·7165 kJ/kg K for air.

[0·0343 units]

13. Develop from first principles an expression, in terms of T and v, for gain of entropy of a perfect gas during a change from condition p_1, v_1, T_1 to condition p_2, v_2, T_2.

14. Derive from first principles a general expression in terms of initial and final pressures and volumes, for the change of entropy of 1 kg of perfect gas when heated.

Calculate the increase of entropy when a quantity of air which occupies 0·05 m^3 at a pressure of 35 bar and 1,000°C, expands to 1 bar, the index of expansion being 1·35. Take R = 0·287 kJ/kg K and k_v = 0·7165 kJ/kg K for air.

[0·04673 units]

15. Derive a general expression in terms of the initial and final temperatures and volumes, for the increase in entropy of a perfect gas when heated.

Air at an initial condition of 1·2 bar, 0·35 m^3 and 25°C, is compressed to a final condition of 5 bar and 0·11 m^3. Calculate the increase of entropy during the process. Take R = 0·287 kJ/kg K and k_v = 0·7165 kJ/kg K for air.

[0·07661 units]

16. Explain the term entropy and show by means of neat sketches how the lines of constant volume and constant dryness fraction (quality) can be drawn on $T - \Phi$ chart for water and steam.

17. Calculate from first principles the entropy of 1 kg of steam reckoned above 0°C at a pressure of 13 bar and 0·8 dry and compare it with the value obtained by using the steam tables.

[5·615 units; 5·6385 units by using steam tables]

18. Calculate from first principles the entropy of 1 kg of steam reckoned above 0°C at a pressure of 10 bar and compare it with the value obtained by using the steam tables, when (i) the steam is wet, having 0·9 dryness fraction, (ii) the steam is dry saturated, and (iii) the steam is superheated, the temperature of steam being 200°C. Take k_p of superheated steam as 2·3 kJ/kg K.

[(i) 6·123, (ii) 6·569, (iii) 6·6679; by using steam tables (i) 6·1416, (ii) 6·5865, (iii) 6·6854]

19. Using steam tables, calculate the entropy of 1 kg of superheated steam reckoned above 0°C at a pressure of 9 bar and with 20°C of superheat. Take k_p for superheated steam as 2·1 kJ/kg K.

[6·7129 units]

20. Explain construction and applications of the following :

(i) $T - \Phi$ chart, and

(ii) $H - \Phi$ chart or Mollier chart.

5

VAPOUR PROCESSES

5.1 Introduction

A vapour is a mixture of gas and particles of liquid in suspension and it can be liquified by moderate changes of pressure or temperature. When a vapour undergoes the process of heating or expanding, its dryness fraction changes continuously due to the evaporation of its liquid content or due to the condensation of the gaseous part of the vapour. *Since the state (proportion of liquid and gas) of a vapour may be changing during a thermodynamic process, it does not obey the laws of perfect gases.*

The pressure, volume and temperature of vapour are not connected by any simple relationship such as is expressed in the characteristic equation for a perfect gas. Equations have been developed from experiental results, from which, tables giving the properties of vapours are compiled. Since experimental results somewhat vary, the tables for the properties of a given vapour, such as steam, as prepared by different investigators do not exactly agree. The quantities tabulated in vapour (steam) tables are : the pressure, saturation temperature, specific volume, enthalpy and entropy.

The process of vapour (steam) formation from liquid state has the following important characteristics :

.. If the process of vapourization is carried out at constant pressure, the temperature will remain constant until vapourization is complete, that is, until the liquid is completely evaporated. The vapour is then said to be *dry saturated.*

.. The heat supplied during the process of vapourization changes its state gradually from the liquid state to the gaseous state.

.. If further heat is added after the dry saturated vapour state is reached, the temperature will again rise gradually and the vapour is then said to be *superheated.*

.. During the process of superheating a vapour at constant pressure, the volume will increase approximately in proportion to the absolute temperature, which indicates that the vapour is approaching the state of a perfect gas.

.. If the pressure on the surface of the liquid is increased, the temperature at which evaporation is taking place also increases and vice versa.

.. If the temperature and consequently the pressure of vapourization increases, the enthalpy of evaporation required to obtain the change of state decreases. In other words, the enthalpy of evaporation *decreases* with the increase of pressure and temperature but the enthalpy of saturated water increases under this condition. The specific volume decreases with the increase of pressure and temperature.

.. The enthalpy (total heat) of dry saturated vapour increases if the saturation temperature is increased. This is, however, true at lower pressures and temperatures as in this range of pressures and temperatures, the rate of increase in sensible enthalpy is greater than the rate of decrease in latent enthalpy with the increase in pressure. After certain saturation temperature (e.g. for steam at 235·72°C), the rate of increase in enthalpy of saturated water is less than the rate of decrease in enthalpy of evaporation and hence, *the enthalpy of the dry saturated vapour decreases.* This will continue to happen until a saturation temperature is reached when the enthalpy of evaporation becomes zero and the enthalpy of dry saturated vapour becomes equal to the enthalpy of saturated water. This will be well understood by referring to tables of properties of dry saturated steam. This saturation temperature is called *critical temperature* and the corresponding pressure is called *critical pressure.* The state of the substance (vapour) at the critical temperature and pressure is called its *critical state.*

.. Each vapour has its own critical temperature and pressure. For example, the critical temperature of steam is 374·14°c and its critical pressure is 220·9 bar. Steam cannot exist as saturated vapour, above this critical temperature. At any temperature higher than the critical temperature, this substance (vapour) can only exist as a gas, whatever the pressure may be. At critical temperature, the specific volume of dry saturated vapour becomes equal to the specific volume of the liquid from which it is being formed.

5.2 Thermodynamic Processes

The general energy equations applicable to perfect gases are also applicable to vapour and the procedure of finding the change of internal energy is also the same as was adopted in case of gases. The different processes of expanding and compressing gases are also applicable to vapours but the results that will be obtained may be different. In each of the thermodynamic processes, some property of the vapour will remain constant from which its final state may be determined. The equations for the work done by a vapour will be the same as those used for a perfect gas, since the equations for the work done are based on mathematical laws of expansion or compression.

The different thermodynamic processes for vapour may be classified as constants volume, constant pressure, constant temperature (isothermal), constant pv or hyperbolic, constant pv^n or polytropic, constant entropy or frictionless adiabatic (isentropic) and constant enthalpy or throttling.

In this chapter, in each of the processes, one kilogram of steam at an initial condition of absolute pressure p_1, specific volume v_{s1}, absolute temperature T_1 and dryness fraction x_1 will be considered but the same method can be applied to any other vapour.

5.2.1 Constant Volume Process : If one kg of wet steam at initial pressure p_1 and dryness fraction x_1 is heated at constant volume to final pressure p_2, its dryness fraction will change. Let x_2 be the final dryness fraction of the steam. Then, since volume remains constant,

Volume before heating = Volume after heating

i.e. $x_1 v_{s1} = x_2 v_{s2}$

$$\therefore \; x_2 = \frac{x_1 v_{s1}}{v_{s2}} \qquad \text{...(5.1)}$$

The values of the specific volumes (v_{s1} and v_{s2}) for the pressures p_1 and p_2 can be obtained from the steam tables and final state of the steam, i.e., x_2 can be determined from eqn. (5.1) for the given value of x_1.

$$\text{Internal energy} = \begin{bmatrix}\text{enthalpy of}\\ \text{1 kg of steam}\end{bmatrix} - \begin{bmatrix}\text{external work}\\ \text{of evaporation}\end{bmatrix}$$

i.e. $u_1 = H_1 - \frac{10^5 p_1 (x_1 v_{s1})}{10^3}$ kJ and $u_2 = H_2 - \frac{10^5 p_2 (x_2 v_{s2})}{10^3}$ kJ

where p_1 and p_2 are in bar.

If p_1 and p_2 are in kPa, then, $u_1 = H_1 - p_1 (x_1 v_{s1})$ kJ and $u_2 = H_2 - p_2 (x_2 v_{s2})$ kJ.

Hence, change in internal energy, $u_2 - u_1$ may be calculated. Since in this process there is no change in volume, no external work is performed, i.e. $W = 0$.

Applying law of conservation of energy,

Heat supplied, $Q = (u_2 - u_1) + W = u_2 - u_1 + 0 = u_2 - u_1$ kJ.

Problem-1 : *One kg of steam at an initial condition of 6 bar and 0·2 dry is heated at constant volume until the pressure is 20 bar. What is final state of steam and how much heat is added ?*

For steam tables, at 6 bar, $v_s = 0{\cdot}3157$ m^3/kg, $h = 670{\cdot}56$ kJ/kg and $L = 2{,}086{\cdot}3$ kJ/kg, and at 20 bar, $v_s = 0{\cdot}09963$ m^3/kg, $h = 908{\cdot}79$ kJ/kg, and $L = 1{,}890{\cdot}7$ kJ/kg.

Using eqn (5.1), $x_1 \times v_{s1} = x_2 \times v_{s2}$

$$\therefore x_2 = \frac{x_1 \times v_{s1}}{v_{s2}} = \frac{0{\cdot}2 \times 0{\cdot}3157}{0{\cdot}09963} = \mathbf{0{\cdot}634 \text{ dry}} \text{ (final state)}$$

Enthalpy at initial and final conditions,

$H_1 = h_1 + x_1 L_1 = 670{\cdot}56 + 0{\cdot}2 \times 2{,}086{\cdot}3 = 1{,}087{\cdot}82$ kJ/kg and

$H_2 = h_2 + x_2 L_2 = 908{\cdot}79 + 0{\cdot}634 \times 1{,}890{\cdot}7 = 2{,}107{\cdot}5$ kJ/kg

Initial internal energy, $u_1 = H_1 - \frac{10^5 p_1 \times (x_1 v_{s1})}{10^3}$

$$= 1{,}087{\cdot}82 - \frac{10^5 \times 6 \times 0{\cdot}2 \times 0{\cdot}3157}{10^3}$$

$$= 1{,}087{\cdot}82 - 37{\cdot}88 = 1{,}049{\cdot}94 \text{ kJ/kg,}$$

Final internal energy, $u_2 = H_2 - \frac{10^5 p_2 \times x_2 v_{s2}}{10^3}$

$$= 2{,}107{\cdot}5 - \frac{10^5 \times 20 \times 0{\cdot}634 \times 0{\cdot}09963}{10^3}$$

$$= 2{,}107{\cdot}5 - 126{\cdot}33 = 1{,}981{\cdot}17 \text{ kJ/kg}$$

Change in internal energy, $u_2 - u_1 = 1{,}981{\cdot}17 - 1{,}049{\cdot}17 = 931{\cdot}23$ kJ/kg.

Applying law of conservation of energy, heat supplied, $Q = (u_2 - u_1) + W$

No work is done during a constant volume process i.e. $W = 0$

$\therefore$ Heat supplied, $Q = (u_2 - u_1) + W = 931{\cdot}23 + 0 =$ **931·23 kJ/kg.**

5.2.2 Constant Pressure Process : If heat is supplied to steam at constant pressure, its dryness fraction and internal energy change, and the work is done against constant external resistance (pressure). This represents the evaporation (or drying) of wet steam.

Let the dryness fraction of one kg. of steam changes from x_1 to x_2 at pressure p bar by the addition of heat. If the specific volume of steam at presure p bar is v_s m^3/kg,

$$\text{External work done, } W = \frac{10^5 p \times x_2 v_s}{10^3} - \frac{10^5 p \times x_1 v_s}{10^3}$$

$$= \frac{10^5 p \times v_s (x_2 - x_1)}{10^3} \text{ kJ}$$

$$\text{Heat supplied, } Q = (u_2 - u_1) + W$$

$$= \left[H_2 - \frac{10^5 p \times x_2 v_s}{10^3}\right] - \left[H_1 - \frac{10^5 p \times x_1 v_s}{10^3}\right] + \frac{10^5 p \times v_s (x_2 - x_1)}{10^3}$$

$$= H_2 - H_1 \text{ kJ} \qquad \text{...(5.2)}$$

This means that the heat supplied to steam at constant pressure is equal to the change in enthalpy of steam.

As long as steam is wet its temperature has a definite value at a given pressure and as such the temperature remains constant in a constant pressure process. From this fact it follows that *for wet steam, a constant pressure process is also a constant temperature or isothermal process.*

If the wet steam is further heated after the dry saturation condition is reached, it will be superheated. Let the absolute temperature of superheated steam be T_{sup}. Then,

$$\text{External work done, } W = \frac{10^5 p\, v_s}{10^3}\left(\frac{T_{sup}}{T_s} - x_1\right) \text{ kJ}$$

Heat supplied, $Q = H_{sup} - H_1$ kJ

where, H_{sup} = Enthalpy of superheated steam, and H_1 = Enthalpy of wet steam.

It may be noted that once the steam is superheated, the expansion will no longer be isothermal.

Problem-2 : *One kilogram of steam at a pressure 7 bar and 0·7 dry is heated at constant pressure until it becomes dry saturated. How much heat is added ? What is the change in internal energy ? If the process is continued until the final temperature is 200°C, how many kJ are required to be added ? Take k_p for superheated steam as 2·1 kJ/kg K.*

At 7 bar, from steam tables,

h = 697·22 kJ/kg, L = 2,066·3 kg/kg, t_s = 164·97°C and v_s = 0·2729 m^3/kg.

Enthalpy for the initial state (wet),

$H_1 = h_1 + x_1 L_1$ = 697·22 + 0·7 × 2,066·3 = 2,143·63 kJ/kg

Enthalpy for the final state (dry saturated),

$H_2 = h_2 + L_2$ = 697·22 + 2,066·3 = 2,763·52 kJ/kg

Using eqn. (5.2),

Heat supplied, $Q = H_2 - H_1$ = 2,763·52 − 2,143·63 = **619·89 kJ/kg**

Initial volume, $v_1 = x_1 v_{s1}$ = 0·7 × 0·2729 = 0·19103 m^3.kg

Final volume, $v_2 = v_{s2}$ = 0·2729 m^3/kg (from steam tables).

Initial internal energy, $u_1 = H_1 - \dfrac{10^5 p \times v_1}{10^3}$

$$= 2{,}143{\cdot}63 - \frac{10^5 \times 7 \times 0{\cdot}19103}{10^3} = 2{,}009{\cdot}91 \text{ kJ/kg}$$

Final internal energy, $u_2 = H_2 - \dfrac{10^5 p \times v_2}{10^3}$

$$= 2{,}763{\cdot}52 - \frac{10^5 \times 7 \times 0{\cdot}2729}{10^3} = 2{,}572{\cdot}49 \text{ kJ/kg.}$$

$\therefore$ Change in internal energy, $u_2 - u_1 = 2{,}572{\cdot}49 - 2{,}009{\cdot}91 =$ **562·58 kJ/kg.**

Enthalpy of superheated steam at 7 bar and 200°C,

$H_{sup} = H_s + k_p(t_{sup} - t_s) = 2{,}763{\cdot}52 + 2{\cdot}1\ (200 - 164{\cdot}97) = 2{,}837{\cdot}08$ kJ per kg

$\therefore$ Heat supplied to superheat 1 kg of steam upto 200°C from dry saturated condition,

$H_{sup} - H_2 = 2{,}837{\cdot}08 - 2{,}763{\cdot}52$

$= 73{\cdot}56$ **kJ/kg**

5.2.3 Constant Temperature (Isothermal) Process : As explained in the constant pressure process, for wet steam, a constant temperature process is also a constant pressure process. As soon as the steam becomes superheated, it will approximately behave like a perfect gas and will follow hyperbolic law at constant temperature. *The isothermal process will, therefore, be hyperbolic process for superheated steam, i.e. product of pressure and volume will remain constant.*

5.2.4 Hyperbolic Process : For wet steam, the hyperbolic process, i.e. pv = constant, is not an isothermal process as in the case of perfect gases. In case of superheated steam, hyperbolic process may be regarded as an isothermal process since superheated steam behaves like a perfect gas.

Let 1 kg of wet steam at initial pressure p_1 and dryness fraction x_1 be expanded in a cylinder, according to the law pv = constant, to the final pressures p_2 and unknown dryness fraction x_2. Then,

Initial volume of steam, $v_1 = x_1 v_{s1}$, and

Final volume of steam, $v_2 = x_2 v_2$

The values of specific volumes (v_{s1} and v_{s2}) for pressure p_1 and p_2 can be obtained from steam tables.

Then, *as* pv = constant, $p_1 v_1 = p_2 v_2$ i.e. $p_1(x_1 v_{s1}) = p_2(x_2 v_{s2})$ ... (5.3)

From which the final state of steam, i.e. x_2 can be calculated. If x_2 is found to be greater than unity, it follows that the steam is superheated. Then, assuming the superheated steam to behave as a perfect gas, final volume, $v_2 = v_{s2} \times \dfrac{T_{sup}}{T_s}$

where, T_{sup} = absolute temperature of superheated steam (unknown), and

T_s = absolute temperature of formation of steam at pressure p_2 (from steam tables).

Thus, hyperbolic law is, $p_1(x_1v_{s1}) = p_2\left(v_{s2} \times \frac{T_{sup}}{T_s}\right)$ ... (5.4)

From eqn. (5.4), the value of T_{sup} can be calculated.

But, as $p_1(x_1v_{s1}) = p_2(x_2v_{s2})$ or $p_2\left(v_{s2} \times \frac{T_{sup}}{T_s}\right)$ i.e. $p_1v_1 = p_2v_2$,

it follows that the external work of evaporation is constant for a hyperbolic expansion.

Now, $H_1 = h_1 + x_1L_1$ kJ and $H_2 = h_2 + x_2L_2$ kJ, if the final condition of steam is wet,

or $H_2 = h_2 + L_2 + k_p\,(t_{sup} - t_s)$ kJ, if steam is superheated.

$$\therefore\ H_2 - H_1 = \left[u_2 + \frac{10^5\,p_2v_2}{10^3}\right] - \left[u_1 + \frac{10^5\,p_1v_1}{10^3}\right] = u_2 - u_1 \text{ kJ} \qquad \text{... (5.5)}$$

where p_1 and p_2 are in bar.

Thus, for hyperbolic expansion, change of internal energy is equal to change in enthalpy.

Work done on piston, $W = \frac{10^5\,p_1 \times v_1 \log_e(r)}{10^3} = \frac{10^5\,p_1\,(x_1v_{s1}) \log_e(r)}{10^3}$ kJ ... (5.6)

where, $r = \frac{v_2}{v_1} = \frac{p_1}{p_2}$

Applying law of conservation of energy, heat transferred during the process,

$$Q = W + (u_2 - u_1) = \frac{10^5\,p_1\,(x_1v_{s1}) \log_e(r)}{10^3} + (u_2 - u_1) \text{ kJ} \qquad \text{... (5.7)}$$

Problem-3 : *One kg of steam at a pressure of 8 bar and 90% dry is expanded to a pressure of 0·4 bar. If the expansion is hyperbolic, determine : (a) the final condition of steam, and (b) the quantity of heat transferred between the cylinder walls and steam, stating the direction. Take k_p of superheated steam as 2·1 kJ/kg K.*

From steam tables, at 8 bar, v_s 0·2404 m^3/kg, h = 721·11 kJ/kg, L = 2,048 kJ/kg and at 0·4 bar, v_s = 3·993 m^3/kg, H = 2,636·8 kJ/kg, t_s = 75·87°C.

(a) As the expansion is hyperbolic, $p_1v_1 = p_2v_2$

Then, $p_1(x_1v_{s1}) = p_2\,(x_2v_{s2})$ i.e. $8\,(0·9 \times 0·2404) = 0·4\,(x_2 \times 3·993)$

$$\therefore\ x_2 = \frac{1·7389}{1·5972} = 1·08$$

This shows that the steam is superheated at the end of hyperbolic expansion as x_2 is greater than unity.

Then, $p_1\,(x_1v_{s1}) = p_2\left(v_{s2} \times \frac{T_{sup}}{T_{s2}}\right)$

i.e. $8 \times 0·9 \times 0·2404 = 0·4 \times 3·993 \times \frac{T_{sup}}{75·87 + 273}$

$\therefore\ T_{sup} = 378·1$ K i.e. $t_{sup} = 378·1 - 273 = 105·1$°C i.e. Final condition of steam is superheated by $t_{sup} - t_{s2} = 105{\cdot}1 - 75{\cdot}87 =$ **29·23°C**

(b) Using eqn. (5.6), Work done on piston, $W = \dfrac{10^5 p_1 (xv_{s1}) \log_e (r)}{10^3}$

$$= \frac{10^5 \times 8 \times (0{\cdot}9 \times 0{\cdot}2404) \log_e \left(\frac{8}{0{\cdot}4}\right)}{10^3} = 519{\cdot}8 \text{ kJ/kg}$$

$H_1 = h_1 + x_1 L_1 = 721{\cdot}11 + 0{\cdot}9 \times 2{,}048 = 2{,}564{\cdot}31$ kJ/kg

$H_2 = H_{s2} + k_p(t_{sup} - t_{s2}) = 2{,}636{\cdot}8 + (2{\cdot}1 \times 29{\cdot}23) = 2{,}698{\cdot}18$ kJ/kg

Using eqn.(5.5), Change of internal energy, $u_2 - u_1$ = change in enthalpy, $H_2 - H_1$

$= 2{,}698{\cdot}18 - 2{,}564{\cdot}31 = 133{\cdot}87$ kJ/kg

Applying the law of conservation of energy, Heat transferred, $Q = W + (u_2 - u_1)$

$= 519{\cdot}8 + 133{\cdot}87 =$ **653·67 kJ/kg**

Positive value indicates that heat is received by the steam from cylinder walls during expansion.

Problem-4 : *One kilogram of steam at a pressure of 4 bar and 0·9 dry is compressed in cylinder according to the law pv = constant to a pressure of 12 bar. Calculate the final condition of steam and the heat removed during the process.*

From steam tables,

At 4 bar, $v_s = 0{\cdot}4625$ m³/kg, $h = 604{\cdot}64$ kJ/kg, $L = 2{,}133{\cdot}8$ kJ/kg;

and at 12 bar, $v_s = 0{\cdot}16333$ m³/kg, $h = 798{\cdot}65$ kJ/kg, $L = 1{,}986{\cdot}2$ kJ/kg.

As the compression is hyperbolic, $p_1 v_1 = p_2 v_2$ or $p_1(x_1 v_{s1}) = p_2(x_2 v_{s2})$

$$\therefore\ x_2 = \frac{p_1}{p_2} \times \frac{x_1 v_{s1}}{v_{s2}} = \frac{4 \times 10^5}{12 \times 10^5} \times \frac{0{\cdot}9 \times 0{\cdot}4625}{0{\cdot}16333} = 0{\cdot}849$$ **(final dryness fraction)**

Now, $H_2 = h_2 + x_2 L_2 = 798{\cdot}65 + 0{\cdot}849 \times 1{,}986{\cdot}2 = 2{,}484{\cdot}92$ kJ/kg

$H_1 = h_1 + x_1 L_1 = 604{\cdot}64 + 0{\cdot}9 \times 2{,}133{\cdot}8 = 2{,}525{\cdot}16$ kJ/kg

Using eqn.(5.5), change of internal energy, $u_2 - u_1$ = change of enthalpy, $H_2 - H_1$

$= 2{,}484{\cdot}92 - 2{,}525{\cdot}16 = -40{\cdot}24$ kJ/kg

Using eqn. (5.6), work done,

$$W = \frac{10^5 p_1 v_1 \log_e \frac{p_1}{p_2}}{10^3} = \frac{10^5 \times p_1 (x_1 v_{s1}) \log_e \frac{p_1}{p_2}}{10^3} = -\frac{10^5 \times p_1 (x_1 v_{s1}) \log_e \frac{p_2}{p_1}}{10^3}$$

$$= \frac{10^5 \times 4\,(0{\cdot}9 \times 0{\cdot}4625) \log_e 3}{10^3} = -\ 183 \text{ kJ/kg}$$

Applying the law of conservation of energy,
Heat transferred, $Q = W + (u_2 - u_1) = -183 - 40{\cdot}24 =$ **− 223·24** kJ/kg

Negative value of Q indicates that heat is removed from steam, i.e. 223·24 kJ/kg is rejected by the steam to the cylinder walls during compression.

5.2.5 Polytropic Process : This process is covered by the general law pv^n = constant. During this process there is a transfer of heat between steam and the cylinder walls and external work is done on the piston. The law is applicable to saturated steam and also to superheated steam.

Let 1 kg of wet steam be expanded in a cylinder so that the expansion follows the law pv^n = constant, the value of n being known. Let the initial condition of the steam be pressure p_1 and dryness fraction x_1, and let the steam be expanded to a pressure p_2 and unknown dryness fraction x_2.

Now, $p_1 v_1^n = p_2 v_2^n$, i.e. $p_1 (x_1 v_{s1})^n = p_2 (x_2 v_{s2})^n$

$$\therefore\ x_2 = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \times \frac{x_1 v_{s1}}{v_{s2}} \qquad \text{...(5.8)}$$

from which the value of x_2 may be calculated. If x_2 is found to be greater than unity, the steam will be superheated after the expansion. In that case,

$$p_1 (x_1 v_{s1})^n = p_2 \left(v_{s2} \times \frac{T_{sup}}{T_s}\right)^n \quad \therefore\ T_{sup} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \times \frac{x_1 v_{s1}}{v_{s2}} \times T_s \qquad \text{...(5.9)}$$

Now, $H_1 = h_1 + x_1 L_1$ kJ

$H_2 = h_2 + x_2 L_2$ kJ if steam is wet after expansion or

$H_2 = H_{s2} + k_p (t_{sup} - t_s)$ kJ if steam is superheated after expansion.

Also $u_1 = H_1 - \dfrac{10^5\, p_1 (x_1 v_{s1})}{10^3}$ kJ and $u_2 = H_2 - \dfrac{10^5\, p_2 (x_2 v_{s2})}{10^3}$ kJ

if wet after expansion, or

$$u_2 = H_2 - \frac{10^5\, p_2 \left(v_{s2} \times \dfrac{T_{sup}}{T_s}\right)}{10^3} \text{ kJ if superheated after expansion,}$$

where p_1 and p_2 are in bar

Work done on piston during expansion,

$$W = \frac{10^5}{10^3}\left[\frac{p_1 v_1 - p_2 v_2}{n-1}\right] = \frac{10^5}{10^3}\left[\frac{p_1 (x_1 v_{s1}) - p_2 (x_2 v_{s2})}{n-1}\right] \text{ kJ if wet after expansion, and}$$

$$W = \frac{10^5}{10^3}\left[\frac{p_1 (x_1 v_{s1}) - p_2 \left(v_{s2} \times \dfrac{T_{sup}}{T_s}\right)}{n-1}\right] \text{ kJ if steam is superheated after expansion.}$$

Applying the law of conservation of energy,

Heat transferred, $Q = W + (u_2 - u_1)$ kJ

This will give the heat absorbed from or rejected to the cylinder walls by the steam during expansion. Positive sign will indicate that heat is received by the steam from the cylinder walls and negative sign will indicate that heat is lost or rejected by the steam to the cylinder walls.

Problem-5 : *One kilogram of steam at a pressure of 7 bar and 0·9 dry expands according to the general law $pv^n = C$ in a cylinder to a pressure of 1 bar, the value* *of the index of expansion being 1·3. Find : (a) the quality of the steam at the end of expansion. (b) the work done, and (c) the heat transferred through the cylinder walls.*

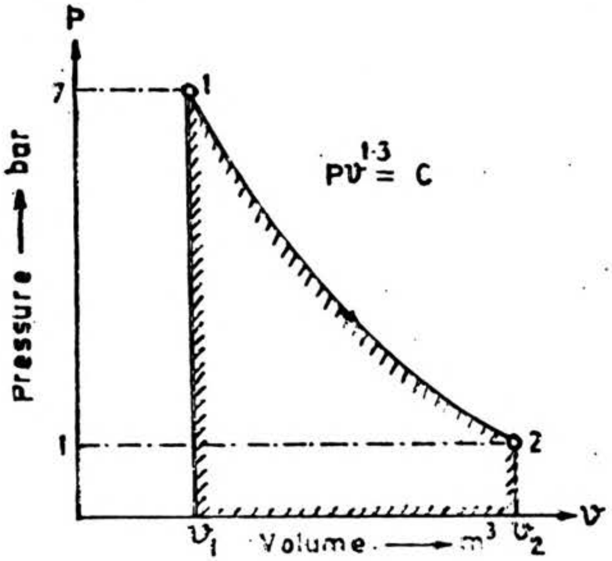


Fig. 5·1.

From steam tables, at ·7 bar,

h = 697·22 kJ/kg, L = 2,066·3 kJ/kg, v_s = 0·2729 m³/kg; and at 1 bar,

h = 417·46 kJ/kg, L = 2,258 kJ/kg, and v_s = 1·694 m³/kg.

(a) As $p_1v_1^n = p_2v_2^n$,

$$p_1(x_1v_{s1})^n = p_2(x_2v_{s2})^n$$

$$7 \times 10^5 \; (0{\cdot}9 \times 0{\cdot}2729)^{1{\cdot}3} = 1 \times 10^5 \, (x_2 \times 1{\cdot}694)^{1{\cdot}3}$$

$$\therefore \; 7 \times 10^5 \times 0{\cdot}1612 = 1 \times 10^5 \times (x_2)^{1{\cdot}3} \times 1{\cdot}985$$

$\therefore$ x_2 = **0·6424 dry (final state)**

(b) Work done during expansion according to the law $pv^n = C$,

$$W = \frac{10^5}{10^3}\left[\frac{p_1v_1 - p_2v_2}{n-1}\right] = \frac{10^5}{10^3}\left[\frac{7 \times 0{\cdot}9 \times 0{\cdot}2729 - 1 \times 0{\cdot}6424 \times 1{\cdot}694}{1{\cdot}3 - 1}\right] = \textbf{210·4 kJ per kg}$$

(c) $H_1 = h_1 + x_1L_1$ = 697·22 + 0·9 × 2,066·3 = 2,556·89 kJ/kg

$H_2 = h_2 + x_2L_2$ = 417·46 + 0·6424 × 2,258 = 1,868·46 kJ/kg

Initial internal energy, $u_1 = 2{,}556{\cdot}89 - \dfrac{10^5 \times 7\,(0{\cdot}9 \times 0{\cdot}2729)}{10^3}$ = 2,384·97 kJ/kg

Final internal energy, $u_2 = 1{,}868{\cdot}46 - \dfrac{10^5 \times 1\,(0{\cdot}6424 \times 1{\cdot}694)}{10^3}$ = 1,760·1 kJ/kg

Applying law of conservation of energy,

$Q = W + (u_2 - u_1)$ = 210·4 + (1,760·1 − 2,384·97) = **− 414·47 kJ per kg.**

Since value of Q is negative, heat is rejected by the steam to the cylinder walls, i.e., heat is extracted from the steam.

Problem-6 : *Expansion of the steam in a cylinder follows the law $pv^{1.1}$ = constant. The state of steam at the commencement of expansion is 10% wet and at 3 bar, and the expansion is continued until the volume is four times that at the commencement. Calculate the quantity of heat which passes into or out of 1 kg of steam during expansion.*

From steam tables, at 3 bar, h = 561·47 kJ/kg,

L = 2,163·8 kJ/kg, v_s = 0·6058 m³/kg.

Now, initial volume of steam,

$v_1 = x_1v_{s1}$ = 0·9 × 0·6058 = 0·5452 m³/kg

Final volume, v_2 = 4 × 0·5452 = 2·181 m³/kg.

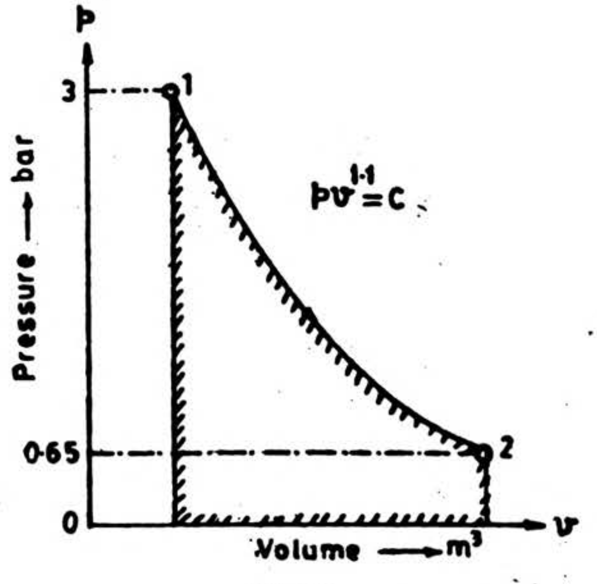


Fig. 5-2.

Now, $p_1 v_1^n = p_2 v_2^n$

$$\therefore\ p_2 = \frac{p_1}{\left(\frac{v_2}{v_1}\right)^n} = \frac{3}{(4)^{1\cdot1}} = 0{\cdot}65 \text{ bar}$$

The work done during expansion according to the law $pv^n = C$ is given by

Work done,

$$W = \frac{10^5 \times p_1 v_1 - 10^5 \times p_2 v_2}{10^3 (n-1)}$$

$$= \frac{10^5 \times 3 \times 0{\cdot}5452 - 10^5 \times 0{\cdot}65 \times 2{\cdot}181}{10^3 \times 0{\cdot}1}$$

$= 217{\cdot}9$ kJ/kg

From steam tables at 0·65 bar, $h = 368{\cdot}54$ kJ/kg, $L = 2{,}288{\cdot}3$ kJ/kg, $v_{s2} = 2{\cdot}535$ m³/kg

Dryness fraction, $x_2 = \frac{v_2}{v_{s2}} = \frac{2{\cdot}181}{2{\cdot}535} = 0{\cdot}86$

$H_1 = h_1 + x_1 L_1 = 561{\cdot}47 + 0{\cdot}9 \times 2{,}163{\cdot}8 = 2{,}508{\cdot}89$ kJ/kg

$H_2 = h_2 + x_2 L_2 = 368{\cdot}54 + 0{\cdot}86 \times 2{,}288{\cdot}3 = 2{,}336{\cdot}48$ kJ/kg

Initial internal energy, $u_1 = 2{,}508{\cdot}89 - \frac{10^5 \times 3 \times 0{\cdot}5452}{10^3} = 2{,}345{\cdot}33$ kJ/kg

Final internal energy, $u_2 = 2{,}336{\cdot}48 - \frac{10^5 \times 0{\cdot}65 \times 2{\cdot}181}{10^3} = 2{,}194{\cdot}71$ kJ/kg

Applying law of conservation of energy,

$Q = W + (u_2 - u_1) = 217{\cdot}9 + (2{,}194{\cdot}71 - 2{,}345{\cdot}33) = +\ 67{\cdot}28$ **kJ/kg**

Since Q is positive, heat is received by steam,

i.e. heat passes into the steam during expansion from an outside source.

Here, the steam source does 217·9 kJ/kg of work; the internal energy supplies only 150·62 kJ/kg, leaving 67·28 kJ/kg to be supplied from an outside source, which would probably be the steam jacket.

5.2.6 Isentropic Process : Since during this process there is no transfer of heat between the medium (steam) and the cylinder walls, external work will be done on the piston during the process of expansion at the cost of internal energy, if friction is assumed to be absent. *The exponent of law* pv^n *= constant no longer has a constant value as it has for a perfect gas. However, the external work can be calculated directly.*

Let 1 kg of steam at pressure p_1 and dryness fraction x_1, be expanded isentropically in a cylinder to a final pressure p_2. The final dryness fraction x_2 at the end of expansion (in which friction is totally absent) can be determined by applying the rule that entropy of the steam before expansion is equal to the entropy of the steam after expansion,

i.e. $\Phi_{w1} + x_1\ \Phi_{e1} = \Phi_{w2} + x_2\,\Phi_{e2}$

or $\Phi_{w1} + x_1(\Phi_{s1} - \Phi_{w1}) = \Phi_{w2} + x_2(\Phi_{s2} - \Phi_{w2})$ .. (5.10)

If the steam was initially superheated to an absolute temperature T_{sup}, eqn.(5.10) would become

$$\Phi_{w1} + \Phi_{e1} + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) = \Phi_{w2} + x_2\,\Phi_{e2}$$

or $\Phi_{s1} + k_p \log_e\left(\frac{T_{sup}}{T_s}\right) = \Phi_{w2} + x_2(\Phi_{s2} - \Phi_{w2})$ .. (5.11)

Isentropic (ideal adiabatic) expansion is represented by a vertical line on the T–Φ and H–Φ charts. The final condition of steam (x_2) can also be read off directly from the chart.

Having determined the final dryness fraction (x_2) of steam, the change in internal energy may be calculated as under :

$$u_1 = H_1 - \frac{10^5\,p_1\,(x_1 v_{s1})}{10^3}\text{ kJ} \qquad \text{and } u_2 = H_2 - \frac{10^5\,p_2\,(x_2 v_{s2})}{10^3}\text{ kJ}$$

where p_1 and p_2 are in bar

Heat transferred during isentropic (frictionless adiabatic) process, $Q = 0$

Applying law of conservation of energy,

Heat transferred, $Q = W + (u_2 - u_1)$ i.e. $0 = W + (u_2 - u_1)$

Hence, external work done, $W = u_1 - u_2$ kJ .. (5.12)

This means that *during an isentropic process, the external work done is equal to the change in internal energy.*

For approximate results, it is often assumed that the law for isentropic process for steam is pv^n = constant, where, $n = 1{\cdot}13$ for wet steam, and $n = 1{\cdot}3$ for superheated steam.

Dr. Zeuner has suggested the well known equation

$n = 1{\cdot}035 + 0{\cdot}1x$ where x is the initial dryness fraction of the steam.

Then, work done on piston $= \dfrac{10^5(p_1 \times x_1 v_{s1} - p_2 \times x_2 v_{s2})}{10^3(n-1)}$ kJ .. (5.13)

Problem-7 : *One kilogram of steam at a pressure of 14 bar and 90% dry, is expanded isentropically to a pressure of 1 bar. If the final quality of the steam is 0·785 dry, find, using steam tables only, the external work done and the value of the index of isentropic expansion, assuming that the law of isentropic expansion is according to the equation pv^n = constant.*

From steam tables, at 14 bar,

$h = 830{\cdot}3$ kJ/kg, $L = 1{,}959{\cdot}7$ kJ/kg, $v_s = 0{\cdot}14084$ m3/kg;

and at 1 bar, $h = 417{\cdot}66$ kJ/kg, $L = 2{,}258$ kJ/kg, $v_s = 1{\cdot}694$ m^3/kg.

Now, $H_1 = h_1 + x_1 L_1 = 830{\cdot}3 + 0{\cdot}9 \times 1{,}959{\cdot}7 = 2{,}594$ kJ/kg.

$H_2 = h_2 + x_2 L_2 = 417{\cdot}66 + 0{\cdot}785 \times 2{,}258 = 2{,}189$ kJ/kg.

Initial internal energy, $u_1 = H_1 - \frac{10^5 p_1 (x_1 v_{s1})}{10^3} = 2{,}594 - \frac{10^5 \times 14\,(0{\cdot}9 \times 0{\cdot}14084)}{10^3}$

$= 2{,}416{\cdot}54$ kJ/kg.

Final internal energy, $u_2 = H_2 - \frac{10^5 p_2 (x_2 v_{s2})}{10^3} = 2{,}189 - \frac{10^5 \times 1\,(0{\cdot}785 \times 1{\cdot}694)}{10^3}$

$= 2{,}056{\cdot}1$ kJ/kg

As the expansion is isentropic, external work done = change in internal energy, i.e. $W = u_1 - u_2 = 2{,}416{\cdot}54 - 2{,}056{\cdot}1 =$ **360·44 kJ/kg.**

Assuming the law for isentropic expansion as pv^n = constant,

$$p_1 v_1^n = p_2 v_2^n \qquad \therefore \frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$$

$$\therefore \text{ Index, } n = \frac{\log\left(\frac{p_1}{p_2}\right)}{\log\left(\frac{v_2}{v_1}\right)} = \frac{\log\left(\frac{14 \times 10^5}{1 \times 10^5}\right)}{\log\left\{\frac{0{\cdot}785 \times 1{\cdot}694}{0{\cdot}9 \times 0{\cdot}14084}\right\}} = \frac{1{\cdot}1461}{1{\cdot}0183} = \mathbf{1{\cdot}125}$$

Problem-8 : *One kilogram of dry saturated steam at 8 bar is contained in a cylinder. If it expands isentropically behind a piston to 1·1 bar, find, using the steam tables only, the external work done during expansion.*

From steam tables, at 8 bar, $\Phi_s = 6{\cdot}6628$ kJ/kg K, $v_s = 0{\cdot}2404$ m³/kg, $H = 2{,}769{\cdot}1$ kJ/kg and at 1·1 bar, $\Phi_w = 1{\cdot}3329$ kJ/kg K, $\Phi_s = 7{\cdot}3273$ kJ/kg K, $v_s = 1{\cdot}5495$ m³/kg, $h = 428{\cdot}79$ kJ/kg, $L = 2{,}250{\cdot}9$ kJ/kg.

Entropy before expansion, Φ_1 = Entropy after expansion, Φ_2

i.e. $\Phi_{s1} = \Phi_{w2} + x_2 (\Phi_{s2} - \Phi_{w2})$

i.e. $6{\cdot}6628 = 1{\cdot}3329 + x_2\,(7{\cdot}3273 - 1{\cdot}3329)$ $\qquad \therefore x_2 = 0{\cdot}893$

Now, (from steam tables), $H_1 = 2769{\cdot}1$ kJ/kg

$H_2 = h_2 + x_2 L_2 = 428{\cdot}79 + 0{\cdot}893 \times 2{,}250{\cdot}9 = 2{,}437{\cdot}79$ kJ/kg.

As the expansion is isentropic, work done W = change in internal energy, $u_1 - u_2$

$$\therefore W = (H_1 - H_2) - \frac{10^5 (p_1 \times v_{s1} - p_2 \times x_2 v_{s2})}{10^3}$$

$$= (2{,}769{\cdot}1 - 2{,}437{\cdot}79) - \frac{10^5 (8 \times 0{\cdot}2404 - 1{\cdot}1 \times 0{\cdot}893 \times 1{\cdot}5495)}{10^3}$$

$$= (2{,}769{\cdot}1 - 2{,}437{\cdot}79) - 40{\cdot}12 = \mathbf{291{\cdot}19 \text{ kJ/kg.}}$$

Problem-9 : *Steam at a pressure of 14 bar and with 105°C of superheat expands isentropically until it becomes dry and saturated. What will be the steam pressure after isentropic expansion ? Find the value of the index 'n', assuming that the law of isentropic expansion is according to the equation $pv^n = C$. Take k_p of superheated steam as 2·1 kJ/kg K.*

From steam tables, at 14 bar, $\Phi_s = 6{\cdot}4693$ kJ/kg K, $t_s = 195{\cdot}07^\circ C$ and $v_s = 0{\cdot}14084$ m^3/kg.

Entropy of steam at 14 bar and with 105°C of superheat (before expansion)

$$= \Phi_s + k_p \log_e \left(\frac{T_{sup}}{T_s}\right)$$

$$= 6{\cdot}4693 + 2{\cdot}1 \log_e \left\{\frac{195{\cdot}07 + 105 + 273}{195{\cdot}07 + 273}\right\} = 6{\cdot}8954 \text{ kJ/kg K}$$

During isentropic expansion, the entropy is constant.

∴ After isentropic expansion to dry saturated state, the entropy of steam is 6·8954 kJ/kg K (same as before expansion).

Referring to entropy column of steam tables for dry saturated steam (Φ_s), it will be found that the entropy of dry saturated steam at a pressure of 4 bar is 6·8959 kJ/kg K, which is almost the same as the entropy at 14 bar and with 105°C of superheat,

i.e. steam after isentropic expansion to dry saturated state is at a **pressure of 4 bar.**

Now, at 4 bar, $v_s = 0{\cdot}4625$ m^3/kg (from steam tables).

$\therefore\ v_2 = v_{s2} = 0{\cdot}4625$ m^3/kg.

Now, $v_1 = v_{s1} \times \dfrac{T_{sup}}{T_{s1}} = 0{\cdot}14084 \times \dfrac{195{\cdot}07 + 105 + 273}{195{\cdot}07 + 273} = 0{\cdot}17243$ m^3/kg

Assuming the law of isentropic expansion as pv^n = constant, $p_1 v_1^n = p_2 v_2^n$

$$\therefore\ \frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$$

$$\therefore\ \text{Index, } n = \frac{\log\left(\frac{p_1}{p_2}\right)}{\log\left(\frac{v_2}{v_1}\right)} = \frac{\log\left(\frac{14}{4}\right)}{\log\left(\frac{0{\cdot}4625}{0{\cdot}17243}\right)} = \mathbf{1{\cdot}3}$$

5.2.7 Throttling Process : When steam is forced through a small orifice under pressure, the steam is said to be throttled. For example, throttling process occurs when steam passes through a partially open valve. At the orifice, eddies are formed which eventually reconvert kinetic energy into heat at the lower pressure, so that throttling is always a wasteful operation which lowers the quality of heat. For a flow through a small orifice, there is no change of potential energy, no external work is done, and if the pipe is well lagged (covered with asbestos rope or any lagging material) there is no heat transferred. Furthermore, in practical cases, the change in velocity is very small, hence we many write,

$$u_1 + p_1 v_1 = u_2 + p_2 v_2 \quad \text{i.e.} \quad H_1 = H_2 \text{ kJ/kg} \qquad .. (5.14)$$

That is to say, the enthalpy remains constant.

This process is also called *constant enthalpy process.* Since no work is done during the expansion, the enthalpy remains constant. Some regard it as adiabatic expansion with friction which converts all the kinetic energy into heat. As there is no change in enthalpy of the steam, its final condition (x_2) can be found as follows :

$$h_1 + x_1L_1 = h_2 + x_2L_2$$

$$\therefore\ x_2 = \frac{h_1 + x_1L_1 - h_2}{L_2} \quad \text{.. (5.15)}$$

If x_2 is found to be greater than unity, steam is superheated after expansion. In that case, the temperature of superheated steam (t_{sup}) after throttling may be obtained from the expression,

$$h_1 + x_1L_1 = H_{s2} + k_p(t_{sup} - t_{s2}) \quad \text{.. (5.16)}$$

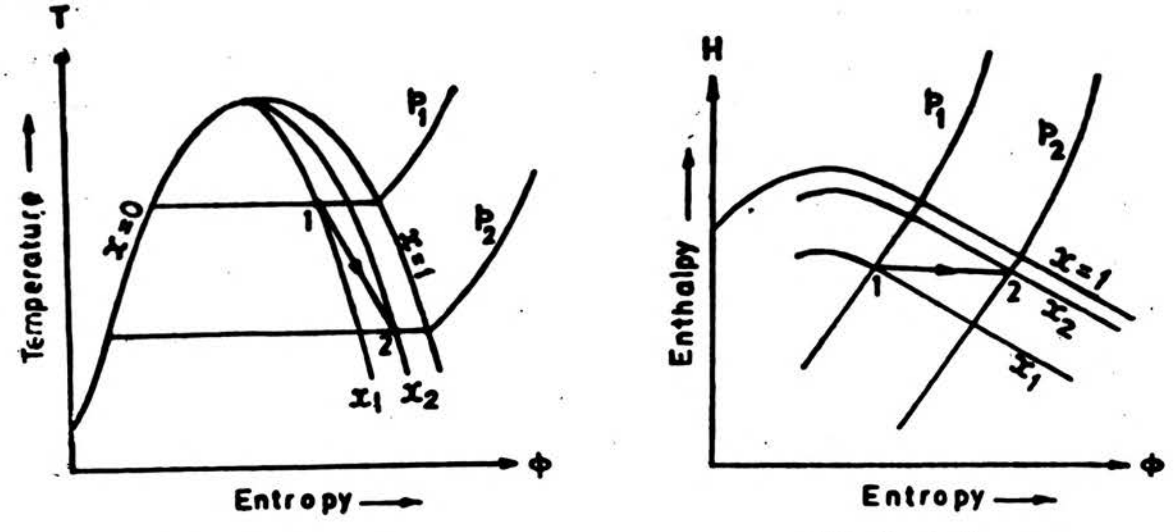
During the process of throttling, the temperature of the steam is slightly reduced but this is compensated by the heat produced due to the friction offered to the steam by the surface of the very small orifice through which the steam has to pass. This heat due to friction, dries the steam and once the steam becomes completely dry it will be superheated by further throttling.

This property whereby wet steam can be superheated by throttling, is used in steam calorimeters for measuring the dryness fraction of wet steam.

Throttling expansion is represented by a horizontal line on the Mollier ($H - \Phi$) chart. The final condition of steam after throttling may be read off from the chart.

As throttling is an irreversible process, the area under curve on $T - \Phi$ chart does not represent the heat supplied from an external source.

Problem-10 : *Steam at 10 bar and 0·9 dry, is throttled to a pressure of 2 bar. Using the steam tables only, calculate the final quality of steam. Also estimate the change of entropy during the process. Represent the throttling process on $T - \Phi$ and $H - \Phi$ diagrams.*



(a) T−Φ diagram (b) H−Φ diagram

Fig. 5-3.

From steam tables, at 10 bar,

h_1 = 762·81 kJ/kg, L_1 = 2,015·3 kJ/kg, Φ_{w1} = 2·1387 kJ/kg K

Φ_{s1} = 6·5865 kJ/kg K, and

At 2 bar, h_2 = 504·7 kJ/kg, L_2 = 2,201·9 kJ/kg, Φ_{w2} = 1·5301 kJ/kg K,

Φ_{s2} = 7·1271 kJ/kg K.

For throttling process (1–2) of fig. 5-3, Enthalpy before throttling = Enthalpy after throttling,

$$\text{i.e. } H_1 = H_2$$

$$\text{i.e. } h_1 + x_1L_1 = h_2 + x_2L_2$$

$$\text{i.e. } 762{\cdot}81 + 0{\cdot}9 \times 2{,}015{\cdot}3 = 504{\cdot}7 + x_2 \times 2{,}201{\cdot}9$$

$$\therefore\ x_2 = \frac{2{,}071{\cdot}88}{2{,}201{\cdot}9} = 0{\cdot}941 \quad \text{(final quality of steam)}$$

Initial entropy, $\Phi_1 = \Phi_{w1} + x_1(\Phi_{s1} - \Phi_{w1})$

$= 2{\cdot}1387 + 0{\cdot}9\ (6{\cdot}5865 - 2{\cdot}1387) = 6{\cdot}1417$ kJ/kg K

Final entropy, $\Phi_2 = \Phi_{w2} + x_2(\Phi_{s2} - \Phi_{w2})$

$= 1{\cdot}5301 + 0{\cdot}941\ (7{\cdot}1271 - 1{\cdot}5301) = 6{\cdot}7969$ kJ/kg K

$\therefore$ Change in entropy during throttling = 6·7969 − 6·1417 = **0·6552 kJ/kg K.**

Problem-11 : *The temperature and pressure in the main steam pipe from the boiler are 310°C and 11 bar respectively. Before the steam enters the engine cylinder it is throttled down to 7 bar. The steam is then expanded isentropically to 1·5 bar. Calculate the following, using the steam tables only :*

(i) the condition (temperature) of steam before isentropic expansion (or after throttling), and (ii) the condition of steam after isentropic expansion. Take k_p of superheated steam as 2·1 kJ/kg K.

From steam tables, at 11 bar, t_s = 184·09°C, H_s = 2,781·7 kJ/kg;

at 7 bar, t_s = 164·97°C, H_s = 2,763·5 kJ/kg, Φ_s = 6·708 kJ/kg K,

at 1·5 bar, t_s = 111·37°C, Φ_w = 1·4336 kJ/kg K, Φ_s = 7·2233 kJ/kg K.

(i) *Condition of steam after steam is throttled down to 7 bar :*

As the steam in the main pipe from the boiler is superheated, enthalpy of 1 kg of steam,

$$H_1 = H_{s1} + k_p(t_{sup1} - t_{s1}) = 2{,}781{\cdot}7 + 2{\cdot}1\ (310 - 184{\cdot}09) = 3{,}046{\cdot}1 \text{ kJ/kg.}$$

As the steam is throttled down to 7 bar before entering the cylinder, the enthalpy before and after throttling is same, i.e. steam after throttling is at 7 bar and has enthalpy of 3,046·1 kJ/kg. This shows that the steam is superheated after throttling as enthalpy of dry saturated steam at 7 bar is 2,763·5 kJ/kg (from steam tables).

$$H_1 = H_2 = H_{s2} + k_p(t_{sup2} - t_{s2})$$

$$\text{i.e. } 3{,}046{\cdot}1 = 2{,}763{\cdot}5 + 2{\cdot}1\ (t_{sup2} - 164{\cdot}97)$$

$$\therefore\ t_{sup2} - 164{\cdot}97 = \frac{3{,}046{\cdot}1 - 2{,}763{\cdot}5}{2{\cdot}1} = 134{\cdot}6 \qquad \therefore\ t_{sup2} = 299{\cdot}57^\circ\text{C}$$

This shows that the steam *after throttling* or before isentropic expansion is at 7 bar and at a temperature of 299·57°C, i.e., steam after throttling is superheated by 299·57 − 164·97 = **134·6°C or degree of superheat is 134·6°C.**

(ii) *Condition of steam after isentropic expansion to 1·5 bar :*

Entropy of 1 kg of superheated steam at 7 bar and 299·57°C,

$$\Phi_2 = \Phi_{s2} + k_p \log_e\left(\frac{T_{sup2}}{T_{s2}}\right) = 6{\cdot}708 + 2{\cdot}1 \log_e\left\{\frac{299{\cdot}57 + 273}{164{\cdot}97 + 273}\right\} = 7{\cdot}2714 \text{ kJ/kg K}$$

Entropy of 1 kg of wet steam at 1·5 bar,

$\Phi_3 = \Phi_{w3} + x_3\,(\Phi_{s3} - \Phi_{w3}) = 1{\cdot}4336 + x_3\,(7{\cdot}2233 - 1{\cdot}4336) = 1{\cdot}4336 + x_3 \times 5{\cdot}7897$

Now, as the expansion is isentropic, entropy is constant,

i.e. Φ_2 (entropy *before* expansion) = Φ_3 (entropy *after* expansion)

i.e. $7{\cdot}2714 = 1{\cdot}4336 + x_3 \times 5{\cdot}7897 \qquad \therefore\ x_3 = 1{\cdot}008$

This shows that the steam is superheated after isentropic expansion as the dryness fraction of steam (x_3) is greater than unity.

Again equating Φ_2 and Φ_3, i.e. $7{\cdot}2714 = \Phi_{s3} + k_p \log_e\left(\dfrac{T_{sup3}}{T_{s3}}\right)$

i.e. $7{\cdot}2714 = 7{\cdot}2233 + 2{\cdot}1 \log_e\left(\dfrac{T_{sup3}}{T_{s3}}\right)$

$$\therefore\ \log_e\left(\frac{T_{sup3}}{T_{s3}}\right) = \frac{7{\cdot}2714 - 7{\cdot}2233}{2{\cdot}1} = 0{\cdot}0239$$

$$\therefore\ \left(\frac{T_{sup3}}{T_{s3}}\right) = 1{\cdot}024 \qquad (\because \log_e 1{\cdot}024 = 0{\cdot}0239)$$

$\therefore\ T_{sup3} = 1{\cdot}024\ (111{\cdot}37 + 273) = 392{\cdot}6$ K or $t_{sup3} = 119{\cdot}6$°C

i.e. Temperature of steam *after isentropic expansion* to 1·5 bar is 119·6°C.

or Steam is superheated by 119·6 − 111·37 = **8.23°C** or **Degree of superheat is 8·23°C.**

Problem-12 : *Steam at 10 bar and 95 percent dry expands isentropically to 4 bar after which it is condensed at constant volume until the pressure is 0·4 bar. Determine the dryness fraction of steam in its final state.*

From steam tables, at 10 bar, $\Phi_w = 2{\cdot}1387$ kJ/kg K, $\Phi_s = 6{\cdot}5865$ kJ/kg K;

at 4 bar, $\Phi_w = 1{\cdot}7766$ kJ/kg K, $\Phi_s = 6{\cdot}8959$ kJ/kg K, $v_s = 0{\cdot}4625$ m^3/kg;

and at 0·4 bar, $v_s = 3{\cdot}993$ m^3/kg.

Since the expansion from 10 bar to 4 bar is isentropic,

Entropy before expansion, Φ_1 = Entropy after expansion, Φ_2

i.e. $\Phi_{w1} + x_1\,(\Phi_{s1} - \Phi_{w1}) = \Phi_{w2} + x_2\,(\Phi_{s2} - \Phi_{w2})$

i.e. $2{\cdot}1387 + 0{\cdot}95\,(6{\cdot}5865 - 2{\cdot}1387) = 1{\cdot}7766 + x_2\,(6{\cdot}8959 - 1{\cdot}7766) \quad \therefore\ x_2 = 0{\cdot}8962$

Let x_3 be the dryness fraction after the steam is condensed at constant volume to 0·4 bar.

Then, $x_2 v_{s2} = x_3 v_{s3}$ i.e. $0{\cdot}8962 \times 0{\cdot}4625 = x_3 \times 3{\cdot}993$

$\therefore\ x_3 =$ **0·1038 (final dryness fraction)**

Problem-13 : *Dry saturated steam at 18 bar is throttled down to 15 bar and then expanded isentropically to 0·3 bar. Using the steam tables (but not the Mollier chart), find : (a) temperature (condition) of the steam after throttling, (b) the dryness fraction of the steam at the end of isentropic expansion, (c) the gain of entropy during throttling, and (d) the work done per kilogram of steam during isentropic expansion. Draw the operations on the* $T-\Phi$ *diagram. Take* k_p *of superheated steam as 2·1 kJ/kg.*

From steam tables, at 18 bar, $H_s = 2{,}797{\cdot}1$ kJ/kg, $\Phi_s = 6{\cdot}3794$ kJ/kg K;

at 15 bar, H_s = 2,792·2 kJ/kg, t_s = 198·32°C, Φ_s = 6·4448 kJ/kg K, v_s = 0·13177 m^3/kg; at 0·3 bar, h = 289·23 kJ/kg, L = 2,336·1 kJ/kg, Φ_w = 0·9439 kJ/kg K, Φ_s = 7·7686 kJ/kg K, v_s = 5·229 m^3/kg.

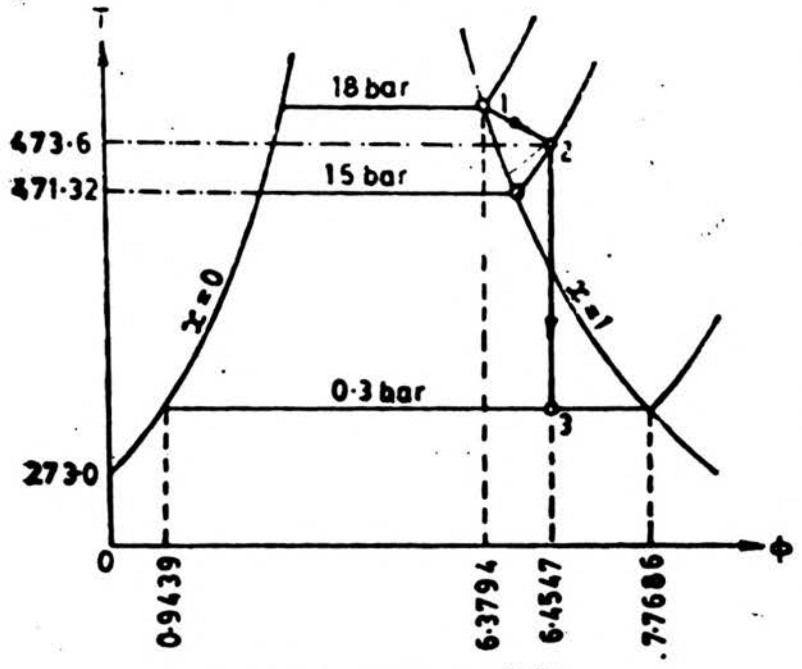


Fig. 5-4. Operations on T–Φ diagram.

Referring to fig. 5-4, 1-2 is throttling process and 2-3 is isentropic expansion.

(a) Enthalpy of dry saturated steam at 18 bar, H_1 = 2,797·1 kJ/kg (from steam tables).

During throttling process (1-2), enthalpy is constant i.e. $H_1 = H_2$

Enthalpy of dry saturated at 15 bar,

$$H_{s2} = 2{,}792{\cdot}2 \text{ kJ/kg}$$

$$H_2 = H_{s2} + k_p(t_{sup2} - t_{s2})$$

i.e. $2{,}797{\cdot}1 = 2{,}792{\cdot}2 + 2{\cdot}1\,(t_{sup2} - 198{\cdot}32)$

$\therefore$ t_{sup2} = **200·6°C.**

(b) During isentropic expansion (2-3), entropy is constant.

i.e. Entropy before expansion, Φ_2 = Entropy after expansion, Φ_3

$$\text{i.e. } \Phi_{s2} + k_p \log_e\left(\frac{T_{sup2}}{T_{s2}}\right) = \Phi_{w3} + x_3\,(\Phi_{s3} - \Phi_{w3})$$

$$\text{i.e. } 6{\cdot}4448 + 2{\cdot}1 \log_e\left(\frac{200{\cdot}6 + 273}{198{\cdot}32 + 273}\right) = 0{\cdot}9439 + x_3\,(7{\cdot}7686 - 0{\cdot}9439)$$

$\therefore$ Dryness fraction, x_3 = **0·8**

$$\text{(c) } \Phi_2 = \Phi_{s2} + k_p \log_e\left(\frac{T_{sup2}}{T_{s2}}\right) = 6{\cdot}4448 + 2{\cdot}1 \log_e\left(\frac{200{\cdot}6 + 273}{198{\cdot}32 + 273}\right) = 6{\cdot}4547 \text{ kJ/kg K}$$

Now, Entropy of dry saturated at 18 bar, Φ_1 = 6·3794 kJ/kg K (from steam tables).

$\therefore$ Gain of entropy during throttling, $\Phi_2 - \Phi_1$ = 6·4547 – 6·3794 = **0·0753 kJ/kg K**

(d) Enthalpy at 15 bar and 200·6°C, H_2 = 2,797·1 kJ/kg.

Initial internal energy at 15 bar and 200·6°C,

$$u_2 = H_2 - \frac{10^5\, p_2 \times v_{s2}\left(\frac{T_{sup2}}{T_{s2}}\right)}{10^3} = 2{,}797{\cdot}1 - \frac{10^5 \times 15 \times 0{\cdot}13177\left(\frac{200{\cdot}6 + 273}{198{\cdot}32 + 273}\right)}{10^3}$$

= 2,598·5 kJ/kg.

Enthalpy at 0·3 bar and 0·8 dry, $H_3 = h_3 + x_3L_3$ = 289·23 + 0·8 × 2,336·1

= 2,158·11 kJ/kg

Final internal energy at 0·3 bar and 0·8 dry,

$$u_3 = H_3 - \frac{10^5 p_3 \times x_3 v_{s3}}{10^3} = 2{,}158{\cdot}11 - \frac{(10^5 \times 0{\cdot}3) \times (0{\cdot}8 \times 5{\cdot}229)}{10^3} = 2{,}032{\cdot}6 \text{ kJ/kg}$$

As the expansion (2-3) is isentropic,

Work done, W = change in internal energy, $u_2 - u_3$

$= 2{,}598{\cdot}5 - 2{,}032{\cdot}6 =$ **565·9 kJ per kg.**

Problem-14 : *Steam at 15 bar and 305°C expands isentropically to 3·5 bar. The steam is the throttled until it is dry saturated and then reheated at constant pressure until the degree of superheat is 60°C. Finally, the steam expands isentropically to 0·07 bar. Draw this cycle of operations on the enthalpy-entropy (H – Φ) chart provided and calculate using the steam tables :(a) the final dryness fraction, (b) the heat supplied per kg of steam during constant pressure reheating process, and (c) the available work per kg of steam. Take k_p of superheated steam as 2·1 kJ/kg K.*

At 15 bar, H_s = 2,792·2 kJ/kg, t_s = 198·32°C, Φ_s = 6·4448 kJ/kg K;

at 3·5 bar, Φ_w = 1·7275 kJ/kg K, Φ_s = 6·9405 kJ/kg K, h = 584·33 kJ/kg, L = 2,148·1 kJ/kg;

at 0·07 bar, h = 163·4 kJ/kg, L = 2,409·1 kJ/kg, Φ_w = 0·5592 kJ/kg K, Φ_s = 8·2758 kJ/kg K.

(a) Referring to fig. 5-5, and *considering isentropic expansion 1-2*, $\Phi_1 = \Phi_2$

$$\text{i.e. } \Phi_{s1} + k_p \log_e\left(\frac{T_{sup1}}{T_1}\right) = \Phi_{w2} + x_2(\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e. } 6{\cdot}4448 + 2{\cdot}1 \log_e\left(\frac{305 + 273}{198{\cdot}32 + 273}\right) = 1{\cdot}7275 + x_2(6{\cdot}9405 - 1{\cdot}7275)$$

$$\therefore\ x_2 = 0{\cdot}9879$$

Enthalpy at 1, $H_1 = H_{s1} + k_p\ (t_{sup1} - t_{s1})$

$= 2{,}792{\cdot}2 + 2{\cdot}1\ (305 - 198{\cdot}32) = 3{,}016{\cdot}23$ kJ/kg

Enthalpy at 2, $H_2 = h_2 + x_2L_2 = 584{\cdot}33 + 0{\cdot}9879 \times 2{,}148{\cdot}1 = 2{,}705{\cdot}5$ kJ/kg.

Considering throttling expansion 2-3 (fig. 5-5):

Enthalpy at 2, H_2 = Enthalpy at 3, H_3 = 2,705·5 kJ/kg

i.e. Enthalpy of steam after throttling at 3 is 2,705·5 kJ/kg, and from the data given the steam becomes dry saturated after throttling.

Referring to the enthalpy (H) column of steam tables, it will be seen that enthalpy of 1 kg of dry saturated steam at 1·95 bar is 2,705·5 kJ/kg.

∴ The pressure of steam after throttling at 3 is 1·95 bar and the steam is dry saturated.

From steam tables, at 1·95 bar, H_s = 2,705·5 kJ/kg, Φ_s = 7·1356 kJ/kg K, t_s = 119·43°C.

Considering constant pressure reheating 3-4 (fig. 5-5):

At 3, i.e. after throttling the steam is dry saturated at **1·95 bar**. The steam is then re-heated at constant pressure until the steam is superheated by 60°C, i.e. at 4 the steam is at a pressure of **1·95 bar** and with 60°C of superheat.

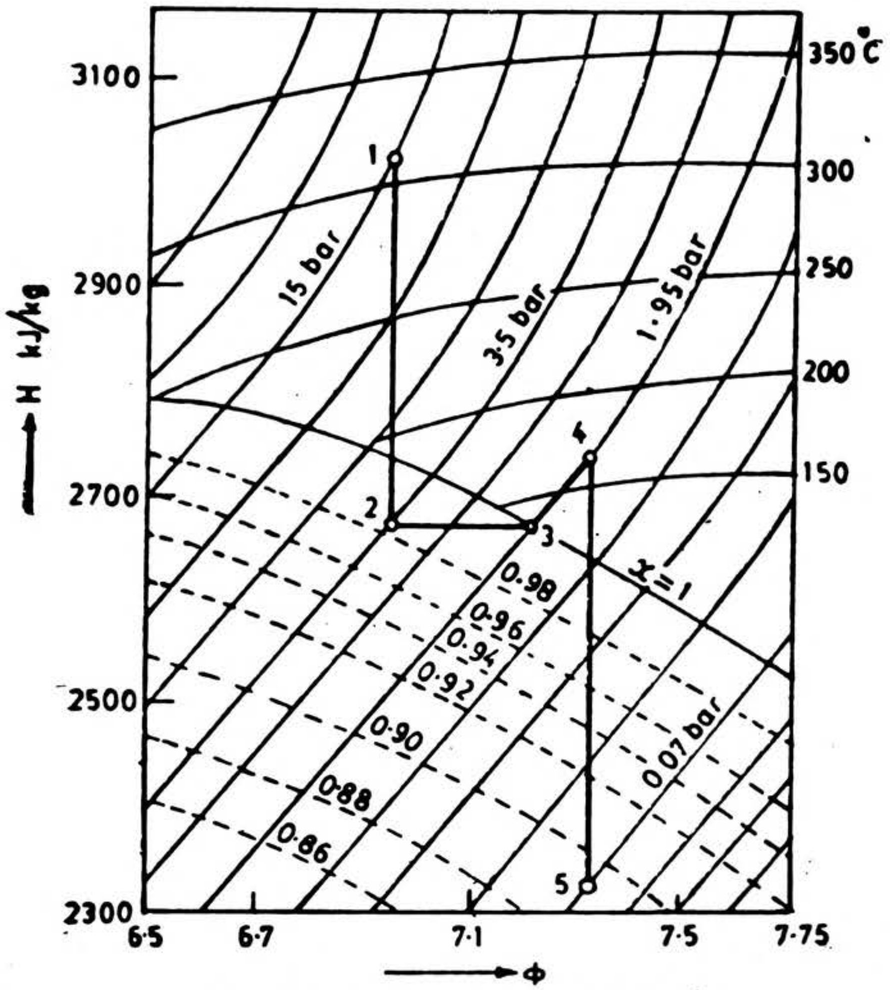


Fig. 5-5. Cycle of operations on H–Φ diagram.

Considering isentropic expansion 4-5 (fig. 5-5),

Φ_4 (entropy of steam at 1·95 bar and with 60°C superheat)

$= \Phi_5$ (entropy of wet steam at 0·07 bar)

$$\text{i.e. } \Phi_{s4} + k_p \log_e\left(\frac{T_{sup4}}{T_{s4}}\right)$$

$$= \Phi_{w5} + x_5\,(\Phi_{s5} - \Phi_{w5})$$

$$\text{i.e. } 7{\cdot}1356 + 2{\cdot}1 \log_e\left(\frac{119{\cdot}43 + 60 + 273}{119{\cdot}43 + 273}\right)$$

$$= 0{\cdot}5592 + x_5(8{\cdot}2758 - 0{\cdot}5592)$$

$\therefore\ x_5 =$ **0·891** (final dryness fraction)

(b) Enthalpy at 2, H_2 = enthalpy at 3, H_3 = 2,705·5 kJ/kg,

i.e. Enthalpy at 3, H_3 = 2,705·5 kJ/kg and steam is dry saturated.

As the dry saturated steam at 3 is re-heated at constant pressure until the steam is superheated by 60°C, the heat supplied during constant pressure re-heating process 3-4 will be given by $k_p(t_{sup} - t_s) = 2{\cdot}1\ (60) =$ **126 kJ/kg.**

(c) Enthalpy at 4, H_4 (i.e. after re-heating 3-4)

$$= H_3 + k_p(t_{sup} - t_s) = 2{,}705{\cdot}5 + 2{\cdot}1\ (60) = 2{,}831{\cdot}5 \text{ kJ/kg}$$

Enthalpy at 5, H_5 (i.e. after isentropic expansion 4-5)

$$= h_5 + x_5L_5 = 163{\cdot}4 + 0{\cdot}891 \times 2{,}409{\cdot}1 = 2{,}309{\cdot}91 \text{ kJ/kg}$$

Available work per kg of steam is equal to work available during isentropic expansion 1-2 *plus* work available during isentropic expansion 4-5.

$\therefore$ Available work per kg of steam $= (H_1 - H_2) + (H_4 - H_5)$

$$= (3{,}016{\cdot}23 - 2{,}705{\cdot}5) + (2{,}831{\cdot}5 - 2{,}309{\cdot}91) = \mathbf{832{\cdot}32 \text{ kJ/kg.}}$$

Problem-15 : *What is the final condition of the steam in each of the following operations, if the initial condition is 0·95 dry and pressure is 8 bar ?*

(i) The steam loses 293 kJ/kg at constant pressure, (ii) the temperature of steam is reduced at constant volume to 158·85°C, and (iii) steam receives 84 kJ/kg at constant pressure and is then throttled to a pressure of 3 bar.

Take k_p of superheated steam = 2·1 kJ/kg K.

(i) At constant pressure, the loss in heat will increase the wetness of steam. From steam tables, the enthalpy of evaporation of steam at 8 bar is 2,048 kJ/kg.

$\therefore$ Increase in wetness due to removal of 293 kJ/kg $= \frac{293}{2{,}048} = 0{\cdot}07$ or 7%

$\therefore$ Final dryness fraction of steam = 95 − 7 = 88% or 0·88 dry.

(ii) Since the saturation temperature at 8 bar (from steam tables) is 170·43°C, the reduction in temperature at constant volume will lower the steam pressure to saturation pressure at 158·85°C. This saturation pressure from steam tables is 6 bar.

From steam tables, at 8 bar, $v_s = 0{\cdot}2404$ m^3/kg and at 6 bar, $v_s = 0{\cdot}3157$ m^3/kg. Let x_2 be the final dryness fraction. Then, since the volume remains constant,

$$x_1 v_{s1} = x_2 v_{s2} \quad \therefore\ x_2 = \frac{x_1 v_{s1}}{v_{s2}} = \frac{0{\cdot}95 \times 0{\cdot}2404}{0{\cdot}3157} = \mathbf{0{\cdot}7249}\ \text{(final condition)}$$

(iii) From steam tables, at 8 bar, h = 721·11 kJ/kg, L = 2,048 kJ/kg;

and at 3 bar, H_s = 2,725·3 kJ/kg, t_s = 133·55°C

Enthalpy at 8 bar, and 0·95 dry (at commencement),

$$H = h + xL = 721{\cdot}11 + 0{\cdot}95 \times 2{,}048 = 2{,}666{\cdot}71 \text{ kJ/kg.}$$

84 kJ/kg is supplied to this steam at constant pressure. The enthalpy will be increased to 2,666·71 + 84 = 2,750·71 kJ/kg and by reference to steam tables, the steam is still wet. This wet steam (at 8 bar) is then throttled down to 3 bar. During throttling process enthalpy before throttling and enthalpy after throttling is same i.e. $H_1 = H_2$.

$\therefore$ Enthalpy of steam (H_2) at 3 bar must also be 2,750·71 kJ/kg, which is greater than that for dry saturated steam (the enthalpy of dry saturated steam at 3 bar from steam tables is, 2,725·3 kJ/kg). From this we conclude that the steam is *superheated* after throttling.

$$H_1 = H_2 = H_{s2} + k_p(t_{sup2} - t_{s2})$$

i.e. $2{,}750{\cdot}71 = 2{,}725{\cdot}3 + 2{\cdot}1\ (t_{sup2} - 133{\cdot}55)$

$\therefore\ t_{sup2} = 145{\cdot}64$°C

i.e. Steam is superheated by 145·64 − 133·55 = **12·09° C**

5.3 Steam Calorimeters

Many of the steam prime-movers are supplied with superheated steam. The enthalpy (total heat) of such a steam is readily determined when its pressure and temperature are known. However, there are many cases in which saturated steam or wet steam is supplied. The measurement of its temperature, when pressure is known, simply confirms the fact that the steam is saturated or wet. In no way it gives any information as to either the quality of steam or the enthalpy of steam.

To aid in the determination of the quality (dryness fraction) of wet steam, various types of steam calorimeters have been deviced. The types of colorimeters used for this purpose are :

- Barrel calorimeter,
- Throttling calorimeter,
- Separating calorimeter, and
- Combined Separating and Throttling calorimeter.

The combined Separating and Throttling calorimeter is the type commonly used for accurate determination of dryness fraction of steam.

5.3.1 Barrel Calorimeter : If the steam is fairly wet, its dryness fraction may be estimated very easily by blowing certain quantity of steam into cold water contained in a vessel of known water equivalent. A vessel (fig. 5-6) containing a measured quantity of water is placed upon the platform of a weighing machine and a supply of steam to be tested is brought into the vessel through the sampling tube placed in the main steam pipe. The steam, mixing in water, gives up some of its heat and raises the temperature of water. The mass of the apparatus is found before and after the experiment. The temperature of water is also taken before and after blowing in the steam.

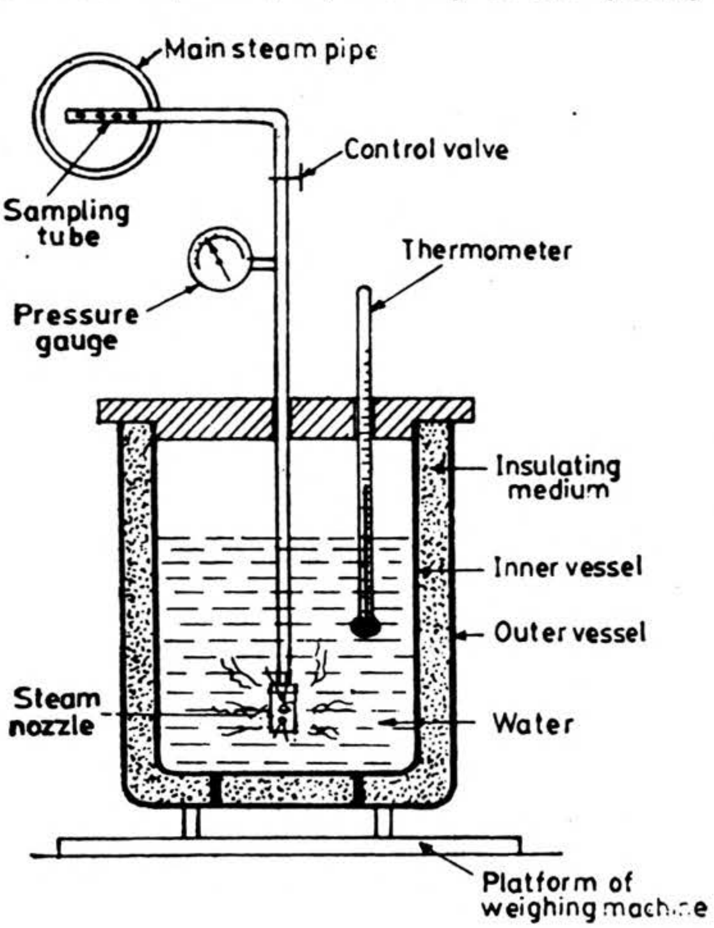


Fig. 5-6. Barrel calorimeter.

Let p = Pressure in main steam pipe.

L = Enthalpy of evaporation corresponding to pressure p,

t_s = Temperature of formation of steam corresponding to pressure p,

x = Dryness fraction of steam in the main pipe,

m = Mass of steam blown into the vessel,

M = Mass of cold water in the vessel at commencement *plus* the mass of water equivalent of the vessel,

t_1 = Initial temperature of water,

t_2 = Final temperature of water (mixture), and

k = Specific heat of water.

Then, Heat lost by wet steam = Heat gained by cold water

i.e. $m\,[\,xL + k\,(t_s - t_2)\,] = M \times k\,(t_2 - t_1)$

i.e. $xL + k\,(t_s - t_2) = (M/m)\,k\,(t_2 - t_1)$

$$\therefore \text{ Dryness fraction, } x = \frac{(M/m)\,k\,(t_2 - t_1) - k\,(t_s - t_2)}{L} \quad \text{...(5.17)}$$

The barrel calorimeter is only accurate when large quantities of steam and cold water are used. The measured final temperature of the water may be lower on account of losses due to heat conduction and radiation.

Problem-16 : *Two kg of steam at 7 bar was injected in a barrel type of calorimeter. The combined mass of water in the calorimeter and the water equivalent of the* *calorimeter was 56 kg at 15°C. The final temperature of the mixture resulting from the condensation of steam was 35°C. Calculate the dryness fraction of the sample of steam. Take specific heat of water as 4·187 kJ/kg K.*

From steam tables, at 7 bar, $L = 2{,}066{\cdot}3$ kJ/kg and $t_s = 164{\cdot}97$°C.

Using eqn. (5.17), Heat lost by steam = Heat gained by water

$$\text{i.e. } xL + k\,(t_s - t_2) = \frac{M}{m} \times k\,(t_2 - t_1)$$

$$\therefore\ x = \frac{\frac{M}{m} \times k\,(t_2 - t_1) - k\,(t_s - t_2)}{L}$$

$$= \frac{\frac{56}{2} \times 4{\cdot}187\,(35 - 15) - 4{\cdot}187 \times (164{\cdot}97 - 35)}{2{,}066{\cdot}3} = \mathbf{0{\cdot}87}$$

5.3.2 Throttling Calorimeter : This calorimeter works on the principle that moderate pressure steam has more total heat or enthalpy than that of low pressure steam. When steam passes through a small orifice, there is a drop in its pressure. As no external work is done by the steam when passing through the orifice in the throttling valve, the enthalpy of steam before throttling must be *equal* to the enthalpy of steam after throttling. Hence, if high quality steam at moderately high pressure is throttled, it tends to become superheated after throttling, i.e. if the steam is fairly dry initially, the pressure drop during throttling will release sufficient heat to superheat the steam.

The throttling calorimeter (fig. 5-7) consists of a chamber, which is shaped to provide a thermometer pocket in the centre. The chamber has a throttling valve (orifice) at the top through which steam enters the calorimeter. The steam expands to a pressure a little above atmospheric pressure after passing through the orifice. The steam leaves the calorimeter and passes off into the atmosphere through the exhaust pipe. The sampling tube placed in the main steam pipe is connected to the calorimeter chamber.

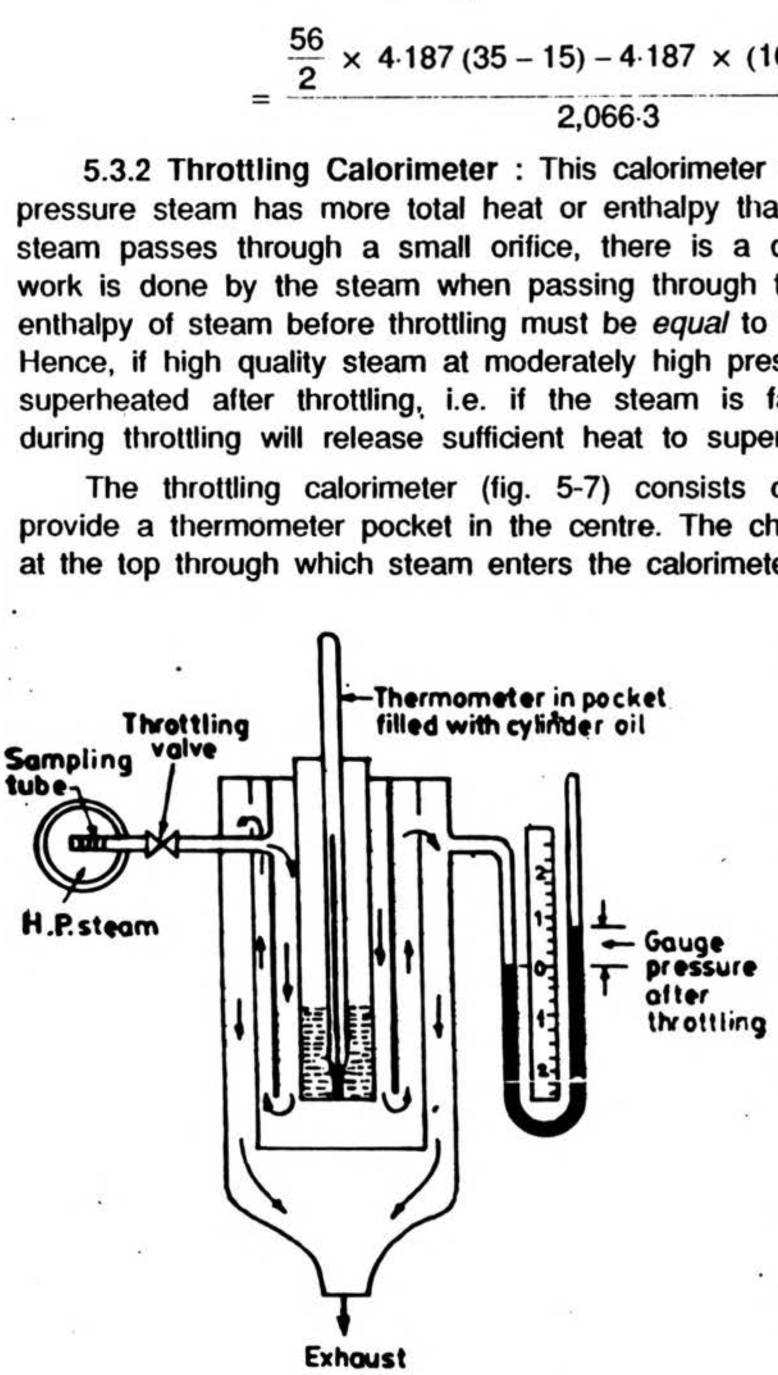


Fig. 5-7. Throttling calorimeter.

The pressure of steam before passing through the throttle valve is measured by a steam pressure gauge (not shown) attached to the main steam pipe. The pressure of steam inside the calorimeter chamber (after throttling) is measured by means of a manometer gauge, which is a glass U-tube containing mercury. The temperature of throttled steam is read by the thermometer placed in the pocket containing oil.

The throttling calorimeter is designed to throttle very nearly dry saturated steam, thus, producing superheated steam whose temperature and pressure can be measured. By reading the initial and final conditions (temperature and pressure) of the steam, its dryness fraction can be measured. The dryness fraction of steam (x_1) is calculated as under :

$$\begin{bmatrix} \text{Enthalpy of wet steam} \\ \text{before throttling (at pressure } p_1) \end{bmatrix} = \begin{bmatrix} \text{Enthalpy of superheated steam} \\ \text{after throttling (at pressure } p_2) \end{bmatrix}$$

i.e. $h_1 + x_1 L_1 = H_{s2} + k_p [\, t_{sup2} - t_{s2}]$

$$\therefore \; x_1 = \frac{H_{s2} + k_p\,(t_{sup2} - t_{s2}) - h_1}{L_1} \qquad \text{...(5.18)}$$

where t_{sup2} = temperature of superheated steam after throttling at pressure p_2, and

t_{s2} = saturation temperature of steam corresponding to pressure p_2 (from steam tables).

The value of k_p is generally taken as 2·1 kJ/kg K.

Throttling calorimeter will only be useful, if the steam after throttling is superheated. It is useless if the steam in the main steam pipe is so wet that it remains wet after throttling (because the enthalpy of the throttled steam or wet steam cannot be estimated). If the steam has an absolute pressure of 20 bar, its dryness fraction must be greater than 0·93 if it is to be measured with this instrument, and at 7 bar the dryness fraction must exceed 0·96.

Problem-17 : *In carrying out a test using a throttling calorimeter, to determine the quality of sample of steam from a steam main pipe at pressure of 11 bar, it was found that the pressure and temperature of steam after throttling was 1·15 bar and 130°C respectively. Calculate the dryness fraction of sample of steam. Take k_p of superheated steam as 2·1 kJ/kg K.*

From steam tables, at 11 bar, h = 781·34 kJ/kg, L = 2,000·4 kJ/kg.

At 1·15, H_s = 2,681·7 kJ/kg, t_s = 103·58°C.

Enthalpy of wet steam before throttling = Enthalpy of superheated steam after throttling

i.e. $h_1 + x_1 L_1 = H_{s2} + k_p(t_{sup2} - t_{s2})$

i.e. $781{\cdot}34 + x_1 \times 2{,}000{\cdot}4 = 2{,}681{\cdot}7 + 2{\cdot}1\ (130 - 103{\cdot}58)$

$$\therefore \text{ Dryness fraction, } x_1 = \frac{1{,}955{\cdot}84}{2{,}000{\cdot}4} = 0{\cdot}978 \text{ or } 97{\cdot}8\%$$

5.3.3 Separating Calorimeter : In case steam cannot be throttled sufficiently in a throttling calorimeter to obtain superheated steam (i.e. if the steam to be tested is not of high quality or fairly dry to become superheated after throttling), a separating calorimeter may be used.

The separating calorimeter is simple in action and it may be used for testing steam of practically any degree of wetness. It works on the principle of separating the water mechanically from the steam by abruptly (suddenly) changing the direction of steam flow and collecting it in a chamber where its mass can be accurately determined.

The separating calorimeter consists of a cast iron body so constructed that there is an inner chamber and outer chamber with space between them as shown in fig. 5-8.

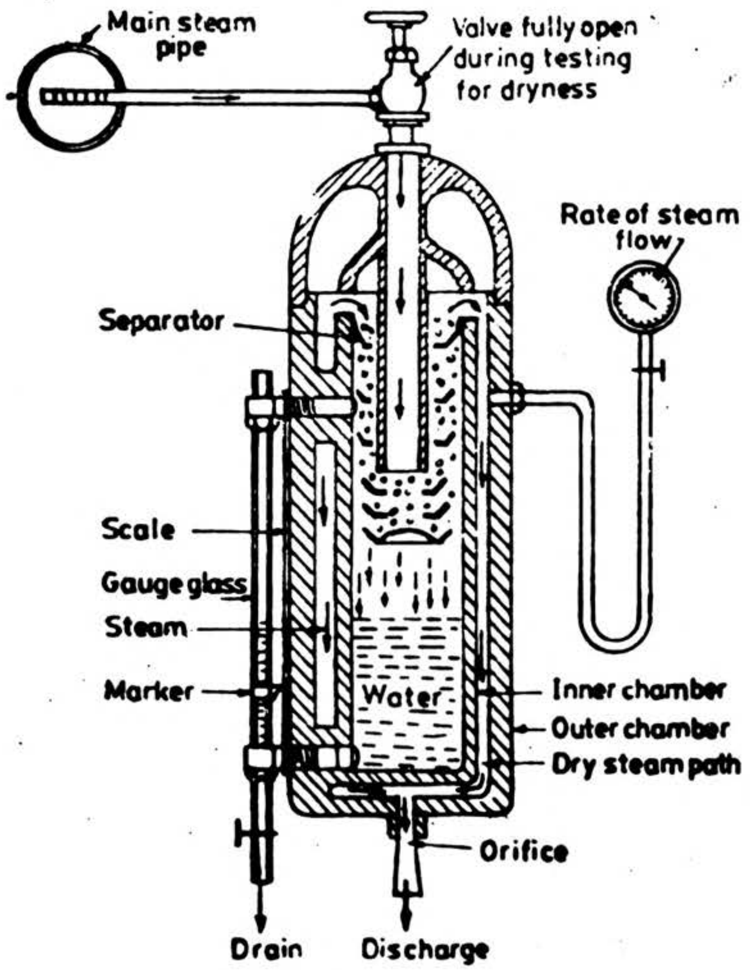


Fig. 5-8. Separating calorimeter.

The space between the chambers forms a jacket. The inner chamber has no direct connection with the outer jacket space, but it is connected at one side to a glass tube water gauge having an attached scale graduated to read in hundredths of a kg of water separated from steam. The outer chamber is connected to steam gauge which indicates the pressure in the outer jacket space. The guage is also graduated, by trial, to read the mass of steam passing through the instrument in ten minutes. At the bottom of the outer chamber is a small orifice of known size, through which the steam passes from the calorimeter.

When in operation, the sample of steam to be tested enters the calorimeter and passes down through the central passage into the perforated cup or separator, where its direction is reversed as shown by the arrows. The moisture being heavier than the steam is left in the cup and falls in the inner chamber where its amount may be read on the graduated scale attached to the gauge. The steam then passes upwards, enters the outer chamber, and flows through the orifice at the bottom as shown by arrows to the exhaust pipe. The gauge connected to the outer chamber indicates the mass of steam leaving the calorimeter in ten minutes.

If m is the mass of water collected in the inner chamber in a certain time, and M is the mass of steam measured in the same interval of time, then the dryness fraction of steam by definition, is

$$x = \frac{M}{M+m} \qquad \text{... (5.19)}$$

It may be noted that it is very difficult to separate out all the water from the wet steam and thus, the separating calorimeter is not very accurate. Under these conditions it would be necessary to use a combined separating and throttling calorimeter (fig. 5-9).

Problem-18 : *In carrying out a test to determine the dryness fraction of a sample of steam, a separating calorimeter was used. The amount of moisture collected over a given period of time was one kilogram and the mass of steam on leaving the calorimeter during the same period was 10 kg. Determine the dryness fraction of the steam.*

Using eqn. (5.19), Dryness fraction of steam, $x = \frac{M}{M+m} = \frac{10}{10+1} = 0{\cdot}909$ or **90·9%**

5.3.4 Combined separating and throttling calorimeter : Since mechanical separation of suspended water particles from wet steam cannot be perfect, a separating calorimeter is not so accurate as a throttling calorimeter. The operation of the throttling calorimeter depends on the steam being superheated after throttling and it will fail in its purpose, if the steam is so wet before throttling that it remains wet after throttling.

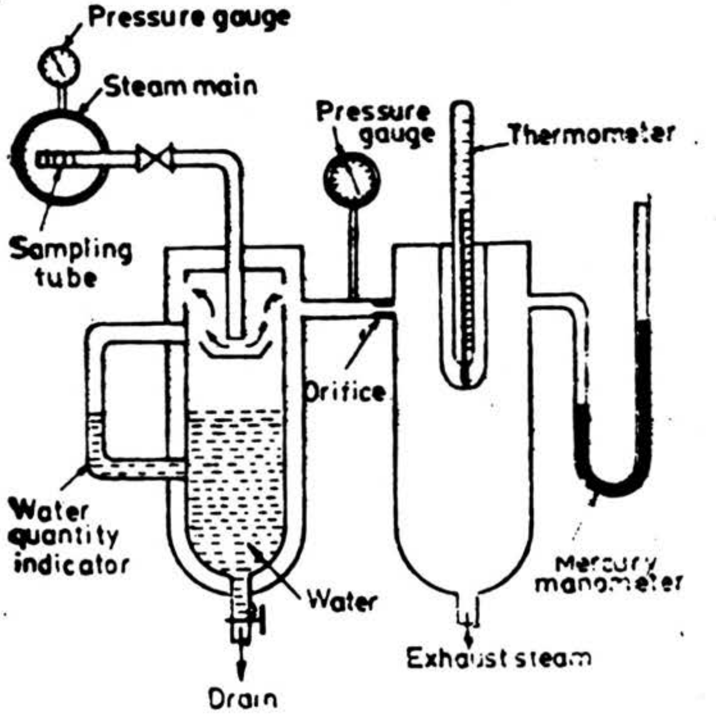


Fig. 5-9. Diagrammatic sketch of a combined separating and throttling calorimeter.

Therefore, a very successful method of measuring the dryness fraction of very wet steam is by a combined separating and throttling calorimeter shown in fig. 5-9. In this instrument the sample of wet steam is first passed through the separating calorimeter where most of the moisture content (water) of the steam is separated and collected. The more drier sample of steam then passes to the throttling calorimeter where it is superheated after throttling. In estimating the dryness fraction of steam supplied to the combined separating and throttling calorimeter, first the dryness fraction of the steam entering the throttling calorimeter is determined by using eqn. (5.18).

For the duration of experiment,

Let M = mass of steam passed through the throttling calorimeter after separating some water in the separating calorimeter,

m = mass of water collected in the separating calorimeter,

x_2 = dryness fraction of steam entering the throttling calorimeter, and

x = dryness fraction of the sample of steam (to be determined).

Then, total mass of dry steam present in the steam supplied = $M \times x_2$ and the total mass of steam supplied = $M + m$

$$\therefore\ x = \frac{\text{Mass of dry steam present in the mixture}}{\text{Total mass of the mixture}}$$

$$= \frac{M \times x_2}{M + m} = \frac{M}{M + m} \times x_2$$

$$\therefore\ x = x_1 \times x_2 \qquad \text{... (5.20)}$$

where $x_1 = \dfrac{M}{M + m}$ from eqn. (5.19)

Problem-19 : *With a combined throttling and separating calorimeter, arranged in series, the following data was obtained for the determination of the dryness fraction of sample of steam taken from the steam pipe :*

Water separated in separating calorimeter .. 0·5 kg
steam discharged from the throttling calorimeter .. 5 kg
Steam pressure in main steam pipe .. 9 bar
Manometer reading .. 166·8 mm of Hg
Barometer reading .. 733·6 mm of Hg
Temperature of steam after throttling .. 110·3°C

Calculate the dryness fraction of the sample of steam, taking specific heat of superheated steam at constant pressure as 2·1 kJ/kg K.

Absolute pressure of steam after throttling = 166·8 + 733·6 = 900·4 mm of Hg. As 760 mm of Hg = 1·01325 bar, 900·4 mm of Hg = 1·2 bar.

From steam tables, at 9 bar, $h = 742{\cdot}83$ kJ/kg, $L = 2{,}031{\cdot}1$ kJ/kg; and at 1·2 bar, $H_s = 2{,}683{\cdot}5$ kJ/kg, $t_s = 104{\cdot}8$°C.

Considering throttling calorimeter,

Enthalpy before throttling = Enthalpy after throttling

i.e. $h_2 + x_2 L_2 = H_{s3} + k_p\,(t_{sup3} - t_{s3})$

i.e. $742{\cdot}83 + x_2 \times 2{,}031{\cdot}1 = 2{,}683{\cdot}5 + 2{\cdot}1\,(110{\cdot}3 - 104{\cdot}8)$

$$\therefore\ x_2 = \frac{1{,}952{\cdot}72}{2{,}031{\cdot}1} = 0{\cdot}962$$

where x_2 is dryness fraction of steam entering the throttling calorimeter.

Mass of steam discharged from throttling calorimeter, $M = 5$ kg.

Mass of water collected in the separating calorimeter, $m = 0{\cdot}5$ kg.

Dryness fraction of steam on entry to the separating calorimeter,

$$x = \frac{\text{Mass of dry steam present in the mixture}}{\text{Total mass of the mixture}} = \frac{M \times x_2}{M + m} = \frac{5 \times 0{\cdot}962}{5 + 0{\cdot}5} = \mathbf{0{\cdot}874}$$

Alternative method by using eqn. (5.20) :

$$x_1 = \frac{M}{M + m} = \frac{5}{5 + 0{\cdot}5} = 0{\cdot}909;\ x_2 = 0{\cdot}962$$

∴ Using eqn.(5.20), Dryness fraction of sample of steam taken from main steam pipe,

$$x = x_1 \times x_2 = 0{\cdot}909 \times 0{\cdot}962 = \mathbf{0{\cdot}874}\ \textbf{(same as before)}$$

Problem-20 : *A combined separating and wire-drawing calorimeter was used to measure the dryness fraction of steam supplied to a steam engine. The following observations were made :*

Pressure of steam in steam pipe .. 7 bar
Temperature of steam inside the wire-drawing calorimeter .. 110°C
Pressure of steam inside the wire-drawing calorimeter .. 136 mm of water
Barometer reading .. 750 mm of mercury
Steam used by the steam engine .. 55 kg
Water collected in the separating calorimeter .. 1 kg
Steam condensed on leaving the wire-drawing calorimeter .. 6·5 kg

The entire steam was passed through the separating calorimeter. The last three readings were recorded simultaneously. Calculate the dryness fraction of the steam entering the calorimeter. Take k_p of superheated steam as 2·1 kJ/kg K.

Pressure inside the wire-drawing calorimeter after throttling

$$= \frac{136}{13{\cdot}6} + 750 = 760 \text{ mm of Hg, or } 1{\cdot}01325 \text{ bar.}$$

From steam tables :

p bar	t_s °C	h kJ/kg,	L kJ/kg	H_s kJ/kg
1·01325	100	–	–	2,676·1
7	164·97	697·22	2,066·3	–

Considering the throttling (wire-drawing) calorimeter,

Enthalpy before throttling = Enthalpy after throttling

i.e. $h_2 + x_2 L_2 = H_{s3} + k_p\ (t_{sup3} - t_{s3})$

i.e. $697{\cdot}22 + x_2 \times 2{,}006{\cdot}3 = 2{,}676{\cdot}1 + 2{\cdot}1\ (110 - 100)$

$\therefore$ $x_2 = 0{\cdot}9674$ where x_2 is the dryness fraction of the steam leaving the separator and enterting the throttling calorimeter.

$\therefore$ Mass of dry steam present in the steam supplied = $0{\cdot}9674\ (6{\cdot}5 + 55) = 59{\cdot}5$ kg.

$\therefore$ Total mass of wet steam (mixture) = $1 + 6{\cdot}5 + 55 = 62{\cdot}5$ kg.

$\therefore$ Dryness fraction of steam before it enters the separator, i.e., dryness fraction of steam in steam pipe line,

$$x = \frac{\text{Mass of dry steam present in the mixture}}{\text{Total mass of the mixture}} = \frac{59{\cdot}5}{62{\cdot}5} = \mathbf{0{\cdot}952}$$

Alternative method by using eqn. (5.20) :

$$x_1 = \frac{M}{M+m} = \frac{6{\cdot}5 + 55}{(6{\cdot}5 + 55) + 1} = \frac{61{\cdot}5}{62{\cdot}5} = 0{\cdot}984; \quad x_2 = 0{\cdot}9674$$

Using eqn. (5.20), Dryness fraction of steam before it enters the separator,

$x = x_1 \times x_2 = 0{\cdot}984 \times 0{\cdot}9674 = \mathbf{0{\cdot}952}$ (same as before)

Problem-21 : *A combined separating and throttling calorimeter was used to measure the dryness fraction of steam supplied to a steam engine. The following observations were made :*

Absolute pressure of steam in steam pipe .. 7 bar (700 kPa)

Pressure of steam inside the throttling calorimeter as measured by mercury manometer .. 8 kPa above atmospheric pressure

Barometer reading .. 97 kPa

Temperature of steam inside the throttling calorimeter .. 111°C

Steam used by the steam engine .. 56·1 kg

Water collected in the separating calorimeter .. 0·9 kg

Steam condensed on leaving the throttling calorimeter .. 6·5 kg

The entire steam was passed through the separating calorimeter. The last three readings were recorded simultaneously. Calculate the dryness fraction of steam entering the calorimeter. Take k_p of superheated steam as 2·1 kJ/kg K.

Absolute pressure inside the throttling calorimeter

= 8 + 97 = 105 kPa (1·05 bar)

From steam tables :

p bar	t_s °C	h kJ/kg,	L kJ/kg	H_s kJ/kg
1·05	101	–	–	2,677·6
7	164·97	697·22	2,066·3	–

Enthalpy before throttling = Enthalpy after throttling

i.e. $h_2 + x_2 L_2 = H_{s3} + k_p\ (t_{sup} - t_{s3})$

i.e. $697{\cdot}22 + x_2 \times 2{,}066{\cdot}3 = 2{,}677{\cdot}6 + 2{\cdot}1\ (111 - 101)$

$\therefore\ x_2 = 0{\cdot}968$

where x_2 is the dryness fraction of steam leaving the separator and enterting the throttling calorimeter.

Mass of dry steam present in the steam supplied = 0·968 (6·5 + 56·1) = 60·6 kg.

Thus, total mass of wet steam (mixture) = 0·9 + 6·5 + 56·1 = 63·5 kg.

∴ Dryness fraction of steam before it enters the separator i.e., dryness fraction of steam in steam pipe line,

$$x = \frac{\text{mass of dry steam present in the mixture}}{\text{total mass of the mixture (wet steam)}} = \frac{60{\cdot}6}{63{\cdot}5} = \mathbf{0{\cdot}954}$$

5.4 Summary of Results

Expressions derived in this chapter may be summarised as under :

Process	Remarks	Equation for final condition of steam	Work done on piston W, kJ/kg	Heat Transferred Q kJ/kg
Constant volume	–	$x_1 v_{s1} = x_2 v_{s2}$	0	$u_2 - u_1$
Constant pressure-Constant temperature (isothermal)	Represents evaporation of wet steam	–	$\frac{10^5\ p \times v_s\,(x_2 - x_1)}{10^3}$	$H_2 - H_1$
Hyperbolic	$pv = C$	$p_1\,(x_1 v_{s1}) = p_2\,(x_2 v_{s2})$	$\frac{10^5\ p_1 (x_1 v_{s1}) \log_e(r)}{10^3}$	$W + (u_2 - u_1)$
Polytropic	$pv^n = C$	$p_1\,(x_1 v_{s1})^n = p_2\,(x_2 v_{s2})^n$	$\frac{10^5 (p_1 x_1 v_{s1} - p_2 x_2 v_{s2})}{10^3\,(n-1)}$	$W + (u_2 - u_1)$
Isentropic (Frictionless adiabatic)	Constant entropy	$\Phi_1 = \Phi_2$	$u_1 - u_2$	0
Throttling	Constant enthalpy	$H_1 = H_2$	0	0

Note : p_1 and p_2 are pressures in bar

Tutorial – 5

1. (a) Differentiate between a vapour and a gas ?
   (b) State the important characteristics of the process of vapour formation from liquid state.

2. (a) State important thermodynamic vapour processes.
   (b) State equations expressed in terms of state parameters for different vapour processes.
3. (a) Explain how the quality of vapour at final state is determined in case of the following thermodynamic processes :
   - (a) Constant volume process,
   - (b) Isothermal process, and
   - (c) Throttling process.

   (b) Explain how the work done is determined for the following thermodynamic vapour processes :
   - (a) Constant pressure process,
   - (b) Isentropic process, and
   - (c) Polytropic process.
4. (a) Isothermal process is also a constant pressure process for a wet vapour, while it is a hyperbolic process for a superheated vapour. Explain this statement.
   (b) One kg of steam initially at 1 bar (100 kPa) and 0·65 dry is heated at constant volume until the pressure becomes 2·5 bar. Determine final temperature (state) and the heat supplied during the process. Take k_p of superheated steam as 2·1 kJ/kg K.

   [340·5°c (superheated by 213·06°C); 1,114·82 kJ/kg]
5. One kg of steam at 7 bar and 250°C is cooled at constant volume until its pressure is 1·5 bar. Find : (a) its final quality, and (b) the change of internal energy during the process. Take k_p for superheated steam as 2·1 kJ/kg K.

   [(a) 0·281 dry; (b) – 1,707·4 kJ]
6. One kg of steam at an initial condition of 19 bar and 325°C passes through constant pressure process until its quality (dryness fraction) becomes 0·85. Find : (i) the work done, (ii) the change in enthalpy, and (iii) the change of internal energy. Take k_p for superheated steam as 2·1 kJ/kg K.

   [(i) – 77·57 kJ; (ii) – 527·45 kJ; (iii) – 449·88 kJ]
7. One kg of steam at a pressure of 17 bar (1·7 MPa) and 0·85 dry expands according to the law pv = constant until its pressure is 0·7 bar. Calculate : (a) the final condition of steam, (b) the change of enthalpy, (c) the change in internal energy, (d) the work done during the process, and (e) the interchange of heat between the steam and the cylinder walls, stating its direction. Take k_p of superheated steam as 2·1 kJ/kg K.

   [(a) superheated by 96·6°C; (b) 166·85 kJ/kg; (c) 166·85 kJ/kg; (d) 537·9 kJ/kg; (e) + 704·75 kJ/kg, heat is received by steam from the cylinder walls]
8. One kg of dry saturated steam expands from 10 bar to 1·5 bar polytropically with the law $pv^{1.2}$ = constant. Calculate : (i) the final condition of steam, (ii) the change of internal energy, (iii) the work done during the process, and (iv) the heat transferred.

   [(i) 0·813 dry; (ii) – 447·85 kJ; (iii) 265·4 kJ; (iv) – 182·45 kJ]
9. One kg of steam at a pressure of 20 bar (2 MPa) and temperature 300°C expands isentropically to a pressure of 0·5 bar. Determine : (a) the condition of steam at the end of expansion, (b) the change in internal energy, (c) the work done per kg of steam during expansion. Take k_p of superheated steam as 2·3 kJ/kg K.

   [(a) 0·8206 dry; (b) 255·93 kJ/kg; (c) 255·93 kJ/kg]
10. One kg of steam at 18 bar (1·8 MPa) and 250°C is expanded isentropically to 4 bar. Calculate, using *steam tables* only, the condition and volume of steam at the end of isentropic expansion. Take k_p of superheated steam as 2·1 kJ/kg K.

    [0·934 dry, 0·432 m^3/kg]
11. Steam originally at 4 bar (400 kPa) and 0·9 dry expands polytropically in the cylinder with the law $pv^{1.1}$ = constant until the pressure is 1 bar. Determine : (i) the change in internal energy per kg, (ii) the work done during the process per kg, and (iii) the amount of heat supplied to or extracted from the steam per kg of steam during the process, stating the direction of the transfer.

    [(i) – 132·47 kJ/kg; (ii) 198 kJ/kg; (iii) + 65·53 kJ/kg, heat is supplied to steam]
12. At a particular instant during the expansion stroke in a steam engine, the volume of steam in the cylinder was 0·015 m^3, the pressure and dryness fraction being 8·5 bar and 0·98 respectively. At a later point during the expansion stroke the volume and pressure had become 0·05 m^3 and 2 bar respectively. Determine : (a) the index of polytropic expansion, (b) the work done per kg of steam between these points, and (c) the change in internal energy per kg.

    [(a) 1·2; (b) 201·65 kJ; (c) – 336·2 kJ]

13. Two kg of steam at 16 bar and 250°C undergoes frictionless adiabatic (isentropic) process until its pressure is 0·15 bar. Find : (a) the final quality of steam, (b) the change in enthalpy, and (c) the change in internal energy. Take k_p of superheated steam as 2·1 kJ/kg K.

[(a) 0·8106 dry; (b) – 1,492·2 kJ; (c) – 1,300·5 kJ]

14. One kg of steam at 8·5 bar and 175°C goes through frictionless adiabatic process until its pressure is 1·1 bar. Determine :

(a) the final quality of steam, and

(b) the work done. Take k_p for superheated steam as 2·1 kJ/kg K.

[(a) 0·8874 dry; (b) 305·19 kJ]

15. (a) One kg of steam is throttled from a pressure of 1·1 bar to a pressure of 11 bar (110 kPa) and 125°C. Using the *steam tables only*, find the quality of steam before throttling. Take k_p of superheated steam as 2·1 kJ/kg K. (b) Steam at 12 bar and 0·95 dryness is throttled to a pressure of 1 bar. Using steam tables only, evaluate the final quality of the steam. Also estimate the change of entropy per kg during this process. Represent the throttling process on T–Φ and H–Φ diagrams. State whether this process is reversible or not. Justify your statement giving reasons. Take k_p of superheated steam as 2·1 kJ/kg K.

[(a) 0·9728 dry; (b) 104·46°C (superheated by 4·83°C); 1·0802 kJ/kg K]

16. One kg of steam at 18 bar and 250°C is expanded isentropically to 4 bar. Calculate using the *steam tables only* : (a) the condition and volume of steam at the end of isentropic expansion, and (b) the final temperature, if the isentropic expansion is followed by a throttling process to a pressure of 1·1 bar. Assume k_p of superheated steam as 2·1 kJ/kg K. Show the operations on H–Φ chart.

[(a) 0·9315 dry, 0·4307 m^3;
(b) 102·31°C]

17. One kg of steam at pressure of 9 bar and 0·96 dry expands isentropically to 1 bar and then cooled at constant volume to a pressure of 0·15 bar. Determine, using the *steam tables only*, the dryness fraction of the steam in its final state.

[0·1434 dry]

18. One kg of steam at a pressure of 15 bar and with 100°C of superheat, is throttled to 9 bar and then expanded isentropically to 3·5 bar. Calculate with the help of *steam tables* :

(a) the temperature of steam after throttling, and

(b) the temperature of steam after isentropic expansion.
Take k_p of superheated steam as 2·1 kJ/kg K.

[(a) 284·4°C (superheated by 109·05°C);
(b) 166·8°C (superheated by 27·92°C)]

19. Steam at 8·5 bar and 182·6°C is throttled to 4·5 bar and then expanded isentropically to 0·3 bar. Determine using the *steam tables only* :
(a) the temperature of steam after throttling, (b) the gain of entropy per kg during throttling, (c) the final quality of steam, (d) the change in enthalpy per kg during isentropic expansion, and (e) the change in internal energy per kg during isentropic expansion. Is the change a gain or loss of internal energy ? Draw the operations on the T–Φ and H–Φ diagrams. Take k_p for superheated steam as 2·1 kJ/kg K.

[(a) 170·7°C (superheated by 22·77°C); (b) 0·2814 kJ/kg K; (c) 0·8824 dry;
(d) 440·61 kJ; (e) – 382·81 kJ i.e. loss of I.E.]

20. One kg of steam at a pressure of 10 bar and superheated to a temperature of 250°C, is expanded isentropically to a pressure of 3 bar and then throttled to 1 bar. Determine, using the steam tables only, the remaining quantities and fill in the blanks in the table given below :

Serial no.	Pressure bar	Quality of steam	Volume of steam m^3/kg	Total entropy of steam kJ/kg K	Enthalpy of steam kJ/kg
1	10	Superheated to temp. of 250°C	–	–	–
2	3	–	–	–	–
3	1	–	–	–	–

Also determine the pressure at which the steam becomes dry saturated during the expansions. Take k_p of superheated steam as 2·1 kJ/kg K. Show the operations on H–Φ diagram.

Serial no.	Pressure bar	Quality of steam	Volume of steam m^3/kg	Total entropy of steam kJ/kg K	Enthalpy of steam kJ/kg
1	10	Superheated to temp. of 250°C	0·225	6·8929	2,925·29
2	3	0·9815 dry	0·5947	6·8929	2,685
3	1	Superheated to temp. of 104·1°C	1·716	7·3818	2,685

Pressure at which steam becomes dry saturated, (i) during isentropic expansion, 4 bar, and (ii) during throttling, 1·25 bar]

21. Describe carefully any method of determining the dryness fraction of steam supplied to a steam engine. Discuss fully the precautions which have to be taken and state what degree of accuracy you expect to obtain.

22. (a) Explain the working of a wire-drawing calorimeter. Make suitable sketches to show its construction. What are the limits of use of such an instrument ?

(b) Two kg of wet steam at 7 bar are injected into a barrel type calorimeter containing 72 kg of water at 16°C. The water equivalent of the tank is 9 kg. If the final temperature resulting from condensing the steam is 28°C, determine the quality of the steam entering the calorimeter. Take specific heat of water as 4·187 kJ/kg K.

[0·7134 dry]

23. Three kg of steam at 1·5 bar was blown into a barrel type calorimeter which contained 36 kg of water at an initial temperature of 16°C. The mass of copper calorimeter is 11 kg and steam was blown through a pipe having water equivalent of 0·125 kg. If the temperature of the resulting mixture was 55°C and the specific heat of copper is 0·3852 kJ/kg K, find the dryness fraction of the steam entering the calorimeter. Take specific heat of water as 4·187 kJ/kg K.

[0·807]

24. Make a neat sketch of a throttling calorimeter and describe carefully how you would use it to determine the quality of steam supplied to an engine.

A boiler supplies superheated steam at a pressure of 7 bar and 30°C of superheat. The steam on its way to the engine loses some heat in the steam main pipe leading to the engine as a result of which the condition of the steam changes. The condition of steam at the engine stop valve is determined by means of a throttling calorimeter. The pressure of the steam after throttling in the calorimeter becomes 1·1 bar and its temperature becomes 112°C. With the help of the steam tables, calculate the dryness fraction of steam at the engine stop valve. Take k_p of superheated steam as 2 kJ/kg K.

[0·9687]

25. Steam on its way to an engine passes through a steam separator in which greater part of the moisture (water) in the steam is separated out. The mass of water, thus, collected in one hour was 5 kg, and the engine uses in the same time 96·5 kg of steam. The test by a throttling calorimeter showed that the steam leaving the separator has a dryness fraction of 0·99. Calculate the dryness fraction of steam entering the separator.

[0·9412]

26. Give a diagrammatic sketch of a throttling calorimeter. What essential conditions must be fulfilled when using a throttling calorimeter ?

Steam at a pressure of 10 bar (1 MPa) was passed through a throttling calorimeter and reduced in pressure to 1·25 bar. The temperature of steam after throttling was observed to be 122°C. What is the quality of steam entering the throttling calorimeter ? Take k_p of superheated steam as 2·01 kJ/kg K.

[0·97 dry]

27. In a test from a combined throttling and separating calorimeter, the following results were obtained :

Initial absolute pressure of steam, 13 bar (1·3 MPa); pressure of steam inside the throttling calorimeter after throttling as measured by mercury manometer, 17 kPa above atmospheric pressure; barometer reading, 98 kPa; steam discharged from the throttling calorimeter, 22 kg at 150°C; and water separated from separating calorimeter, 2·3 kg. Estimate the dryness fraction of the steam entering the calorimeter. Take k_p of superheated steam as 2·1 kJ/kg K.

[0·9021]

28. Sketch diagrammatically a combined separating and throttling calorimeter.

The following data were obtained from a test using a combined separating and throttling calorimeter :

Steam pressure in main steam pipe .. 4·5 bar
Water collected in separating calorimeter .. 0·65 kg

Steam passing out of throttling calorimeter	4·75 kg
Manometer reading after throttling	86·6 mm of Hg
Temperature of steam after throttling	109·3°C
Barometer reading	738·6 mm of Hg

Assuming k_p for superheated steam to be 2·1 kJ/kg K, determine the quality of steam entering the calorimeter from the boiler.

[0·8592 dry]

29. (a) State the principle of throttling calorimeter. Why is it necessary under certain conditions to incorporate a separator with a throttling calorimeter ?

(b) The following data were obtained from a test with combined separating and throttling calorimeter :

Water collected in separating calorimeter 2 kg, steam condensed after leaving throttling calorimeter 20·5 kg, steam pressure in the main steam pipe 12 bar, barometer reading 749·8 mm of mercury, temperature of steam in throttling calorimeter 145°C, and reading of manometer 113 mm of Hg.

Estimate the dryness fraction of the steam in the main steam pipe. Take k_p for superheated steam as 2·1 kJ/kg K.

[0·9036]

30. What are the methods of determining the dryness fraction of steam ? Give limitations of each method.

During a test to find the dryness fraction of steam, with separating and throttling calorimeters in series, the following readings were obtained :

Pressure of steam in the main steam pipe	10 bar
Manometer reading after throttling	956 mm of water
Barometer reading	754·91 mm of Hg
Temperature of steam after throttling	111°C
Condensate from throttling calorimeter	0·5 kg
Drainage from separating calorimeter	28·5 mg

Calculate the dryness fraction of the steam entering the calorimeter from the boiler assuming k_p of superheated steam as 2·1 kJ/kg K.

[0·9097]

31. Sketch and describe a combined separating and throttling calorimeter for determining the quality of steam.

In a test with a combined throttling and separating calorimeter, the following results were obtained :

Steam discharged from throttling calorimeter	22 kg at 150°C
Water separated in separating calorimeter	2·3 kg
Initial steam pressure	13 bar
Final pressure of steam after throttling	114 mm of Hg
Barometer reading	748·8 mm of Hg

Estimate the dryness fraction of the steam entering the calorimeter. Take k_p of superheated steam as 2·1 kJ/kg K.

[0·9012]

6
FUELS

6.1 Introduction

Fuel is defined as a substance composed mainly of carbon and hydrogen because these elements will combine with oxygen under suitable conditions and give out appreciable quantities of heat. Fuels are used in industry for production of heat and in turn this heat produced is converted into mechanical work. Before the invention of *nuclear energy* and *nuclear fuels,* chemical energy of fuels, which may be in solid, liquid or gaseous forms was only the natural source of energy mainly.

Coal, crude oil and natural gas are *fossil fuels.* Even today, major portion of energy demand is met by fossil fuels. At present, the major share of energy demand is met by crude oil which supplies nearly 42% of the total energy demand. Coal supplies nearly 32% and natural gas about 21% of the energy demand. Remaining 5% of energy demand is supplied by nuclear power and hydraulic power.

All human, industrial, and agricultural wastes constitute potential significant source of energy. Normally a large portion of such waste is combustible matter e.g. *wood, paper, tar, textile, plastic, plant, leaves* etc. *Bio-gas* is obtained by fermentation in the sewage disposal system, or by the fermentation of cattle waste, farm waste, etc. If the fermentation is carried out in the absense of air, the gas produced may have the concentration of *methane* as high as 75%. The remaining part is mostly carbon dioxide and hydrogen.

With the increasing gap between the demand and supply of crude oil, and with the fast reducing fossil fuel reserves, it has new become essential to find out alternatives to gasoline for automobiles, heavy vehicles, and aircrafts. Scientists are working on various *synthetic fuels* that can replace gasoline. There are three main alternative synthetic fuels which can replace gasoline :

— Synthetic fuels such as *ethanol* and *butanol* derived from carbohydrates,

— Synthetic fuels such as *methanol* derived from coal, and

— *Hydrogen* from water.

Since the discovery of nuclear energy in 1939 using *fussion reaction, nuclear fuels* have become a good source of energy. The main nuclear fuel for reactors is U^{235}. Though U^{233} and Pu^{239} are also *fissionable* but these are not available in nature. U^{233} can be produced from Th^{232} and Pu^{239} from U^{238}. U^{235} and U^{238} are available in nature in the form of uranium oxide U_3O_8 in pitch-blende. The world reserves of uranium are estimated to be about 10,00,000 tons. The uranium reserves in India are estimated to be around 1,00,000 tons. Besides uranium, India also has large deposits of thorium but it will take sometime before suitable breeder reactors are commissioned for the production of power from U^{238} produced from thorium. At the present time about 2% of the total energy demand is met by nuclear sources.

The cost of power generated from nuclear reactors is already in competition with the cost of thermal power generation, and it is expected that with the increase in the cost of production and transportation of fossil fuels, cost of nuclear power generation will be less than the cost of thermal power generation. However, the *disposal of waste from the nuclear reactors* is still a problem.

6.2 Solid Fuels

Solid fuels may be classified into two classes :

- Natural solid fuels, and
- Artificial or Prepared solid fuels.

6.2.1 Natural Solid Fuels : The fuels belonging to this group are :

- Wood,
- Peat,
- Lignite or brown coal,
- Bituminous coal, and
- Anthracite coal.

Wood consists largely of carbon and hydrogen chemically formed by the action of sunlight, and was at one time used extensively as a fuel. The calorific value of wood varies with kind of wood and with water content. Other solid fuels are the products of wood and vegetable matter which have, during the course of ages been decomposed and solidified under great pressure. The character of the solid fuel depends upon the length of time which has been occupied in its formation and on the amount of pressure and heat to which it has been subjected in the strata of the earth.

Peat represents the first stage at which the fuel derived from wood and vegetable matter is recovered from earth. It contains from 20 to 30 per cent of water and, therefore, needs to be dried off before it can be burnt satisfactorily. The calorific value of air dried peat is about 14,500 kJ/kg.

Lignite or Brown coal represents the intermediate stage, and is a very soft and an inferior type of coal containing not less than 60% carbon. It contains from 15 to 20% of moisture after drying in the air. It has been largely used as a low grade fuel. The average heating value of this type of coal after drying is 21,000 kJ/kg. It burns with large smoky flame and is generally non-caking (i.e. its parts do not stick together while burning).

Bituminous coal is the next intermediate stage in the development. It is soft and shiny black in appearance, often showing signs of its vegetable origin and consists of about 70% carbon and about 20 to 30% volatile (easier to ignite) matter. The average calorific value of bituminous coal is about 31,500 kJ/kg. It burns with a long yellow and smoky flame. Bituminous coal is of two varieties, namely, *non-caking bituminous coal* and *caking bituminous coal.*

Bituminous non-caking burns with little or no smoke and its parts do not stick together while burning. Thus, it is known as free buring carbon. Caking bituminous coal softens and swells and the parts stick together, forming a pasty mass. It burns with a fairly long smoky flame.

Anthracite is regarded as the final stage in the development. It is hard, brittle and consists of about 90% carbon and about 8 to 10 per cent volatile matter. It is difficult to ignite and burns without flame or smoke. It is non-caking and does not give smell when burning. It has very low ash content and is comparatively free from sulphur. Its calorific value is about 36,000 kJ/kg or 36 MJ/Kg.

6.2.2 Artificial or Prepared Solid Fuels : The fuels belonging to the group are : *wood charcoal, coke, briquetted coal* and *pulverised coal.*

Wood charcoal is made by heating wood out of contact with the atmosphere or with limited supply of air, to a temperature not lower than 310°C. The higher the temperature, the blacker and harder is the charcoal produced. Dry charcoal contains 80 to 90% carbon. The calorific value of dry charcoal is about 28,000 kJ/kg or 28 MJ/kg.

Coke is made from bituminous coal by driving out its volatile elements. It is hard, brittle and porous. It is not much used for steam raising purposes. It is especially used in gas producers. It contains 85 to 95% of carbon and its calorific value is about 32,500 kJ/kg.

Briquetted coal consists of finely ground coal mixed with a suitable binder and pressed together into briquettes (blocks). This treatment decreases the fuel loss through the grate and increase the heating value of the fuel. Briquetting of lignite coal will allow its handling and burning without disintegration.

Pulverised coal is the coal that has been reduced to a powder of such a fineness that it will float in air. For complete combustion, the fuel should be mixed intimately with the proper amount of air at sufficiently high temperature. It is easier to achieve this intimate mixing of the coal with air if the coal is finely divided (or atomised). The coal is reduced to its finely divided state by crushing, and many important plants are now working on pulverised coal. Thus, flexibility of control, complete combustion with less excess air and high temperature of flame, are advantages of pulverised coal.

6.3 Burning of Coal

When coal is heated it breaks up into two distinct fuels : Volatile matter and fixed carbon.

—**Volatile matter** consists of carbon combined with hydrogen together with the other gas forming constituents that are driven off by heat. Volatile matter burns with a long yellow flame.

— **Fixed carbon** burns with hardly any flame and is difficult to ignite. Fixed carbon is the carbon which is not in combustion with any other element of the fuel.

Bituminous coal when heated gives off 20 to 30% volatile matter, whereas anthracite contains very small quantity of volatile matter, and therfore bituminous coal is much easier to ignite than anthracite coal. Coke is the substance, containing almost entirely of fixed carbon and ash, which remains after the greater part of volatile matter and moisture has been distilled from the coal, as in the production of coal gas.

6.4 Coal Analysis

Coal, as found in nature, is neither a pure substance nor of uniform composition. A definite chemical formula cannot, therefore, be written for a coal found in coal mines. Consequently, *two* different methods of analysis are employed to know the composition of coal. They are known as *Ultimate analysis* and *Proximate analysis.*

In **Ultimate analysis** a complete chemical break down of coal into its chemical constituents is carried out by chemical process. This analysis is required when important large scale trials are being performed (i.e. a boiler trial), and is carried out by a chemist. The analysis serves the basis for calculation of the amount of air required for complete combustion of one kilogram of coal. The analysis gives the percentage content on mass basis of carbon, hydrogen, nitrogen, oxygen, sulphur and ash and their sum is taken as equal to 100 per cent. Moisture is expressed as a separate item. The analysis also enables us to determine the calorific value or heating value of the coal.

Proximate analysis is the separation of coal into its physical components and can be made without the knowledge of analytical chemistry. The analysis is made by means of a chemical balance and a temperature controlled furnace. The sample of fuel is heated in the furnace or oven. The components in the analysis are *fixed carbon, volatile matter* (combination of carbon, hydrogen and other gases which distil easily), *moisture* and *ash*. These components are expressed in percentage on mass basis and their sum is taken as 100 per cent. Sulphur is determined separately. This analysis also enables us to determine the heating value of the coal.

Both the above analysis may be expressed in terms of the following ;

.. Coal 'as received' or 'as fired',

.. Coal 'moisture free' or 'dry', or

.. Coal 'moisture-and-ash free' or Combustible.

6.5 Calorific Value

The calorific value or heating value of a solid fuel or liquid fuel may be defined as the *number of heat units (kJ* or *MJ) liberated by the complete combustion (burning) of one kilogram of fuel.* For solid or liquid fuels, the calorific value is generally expressed as kJ per kilogram. For gaseous fuels, the calorific value is generally expressed in kJ per cubic metre of gas measured at temperature of 15°C and a pressure of 760 mm of mercury, i.e., at S.T.P.

Fuels which contain hydrogen have two heating values, the higher and the lower calorific values. The **higher or gross or total calorific value** (H.C.V.) of a fuel is the total heat liberated by one kilogram or one cubic metre of fuel when completely burned, including the heat of steam formed by the combustion of hydrogen in the fuel which is recovered by condensing the steam from the products of combustion.

The **lower or net calorific value** (L.C.V.) of a fuel is the heat liberated per kilogram or per cubic metre of fuel after the enthalpy necessary to vaporise the steam formed by combustion of hydrogen in the fuel is deducted. The enthalpy referred to is the enthalpy of evaporation of dry saturated steam at the standard atmospheric pressure of 760 mm Hg.

The calorific value of a fuel may be found :

— Theoretically by calculations, when the composition of fuel on mass basis is known, and

— Experimentally by a fuel calorimeter.

6.5.1 Carbon Value : The evaporative power of fuels of different compositions can be compared by referring them to a common standard. This standard is the kilograms of carbon required to produce the same amount of heat as one kilogram of fuel. This is also known as the *carbon equivalent of the fuel* and is given by the ratio

$$\frac{\text{Calorific value of fuel}}{\text{Calorific value of carbon}}$$

6.5.2 Evaporative Power : Evaporative power of fuel is the number of kilograms of water which can be converted into dry saturated steam at atmospheric pressure by one kilogram of fuel, the initial temperature of the water being taken as 100°C.

$$\therefore \text{ Evaporation power of a fuel} = \frac{\text{calorific value of fuel in kJ per kg}}{2,257 \text{ (enthalpy of evaporation of steam at } 1{\cdot}01325\text{bar)}} \quad ..(6.1)$$

6.5.3 Theoretical Value : The theoretical calorific value (C.V.) of a fuel may be calculated by using a formula known as Dulong's formula. If the ultimate analysis of the fuel is available and if the calorific value of the elementary combustibles (carbon, hydrogen and sulphur) are known, then the calorific value of the fuel is *approximately* the sum of the heat liberated by these combustibles (carbon, hydrogen and sulphur).

According to Dulong's formula,

$$\text{Theoretical C.V. of the fuel} = 33{,}800C + 1{,}44{,}500\left(H_2 - \frac{O_2}{8}\right) + 9{,}300S \text{ kJ/kg} \quad ..(6.2)$$

The symbols C, H_2, O_2 and S represent the mass of carbon, hydrogen, oxygen and sulphur in kg per kg of fuel, and the numericals indicate their respective calorific values in kJ/kg. The expression $\left(H_2 - \frac{O_2}{8}\right)$ represents the available hydrogen for heat release, that is, the amount left after subtracting the amount of hydrogen that will unite with oxygen present in the fuel to form water. Mass of H_2 in water is 1/8th of the mass of O_2.

6.5.4. Experimental Value : The calorific value of a fuel is determined very accurately by the use of a fuel calorimeter in which a known quantity of fuel is burnt and heat liberated from its combustion is absorbed by a definite mass of water whose temperature rise is then measured.

Bomb Calorimeter (fig. 6-1) enables an accurate determination of the calorific value of a solid fuel, and also of a liquid fuel when burnt at constant volume in an atmosphere of oxygen.

The name of the apparatus arises from the shape of the vessel in which the fuel is burnt. The bomb is made of stainless steel and consists of a base, which supports the platinum crucible and is screwed to the body of the bomb. The top of the bomb carries an oxygen supply connection through which the bomb is charged with the oxygen, and a product

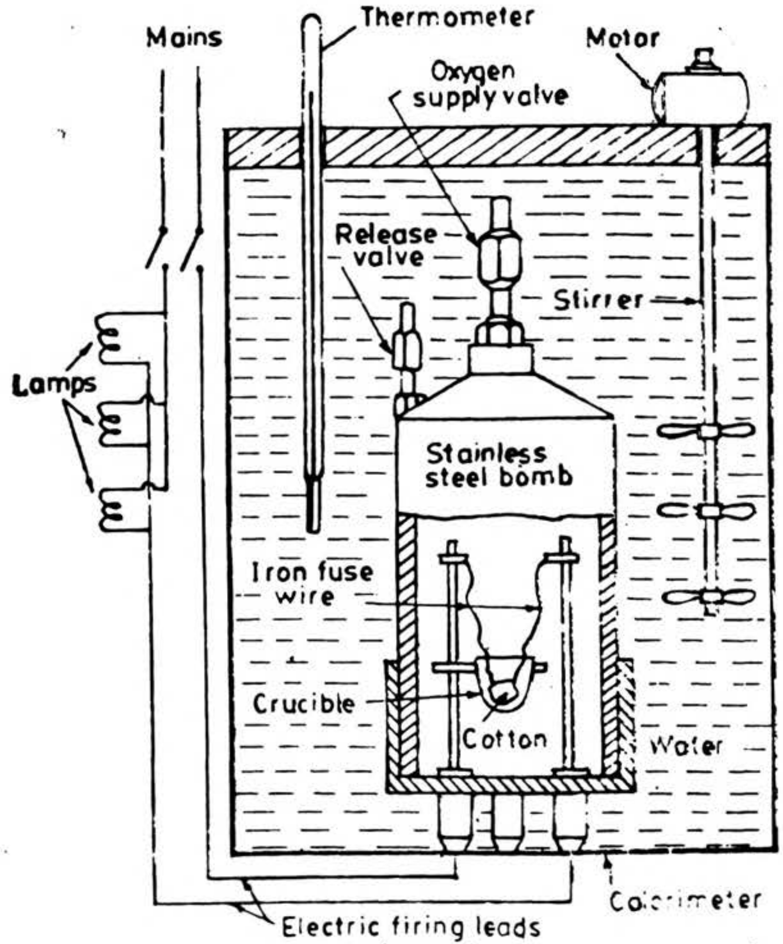


Fig. 6 1. Bomb calorimeter.

release valve. The selected sample of coal to be tested is reduced to powder and them compressed into briquette with the aid of a briquette mould apparatus. The mass of powdered sample of coal taken for the test is usually one gram and the calorimeter is filled with 2,000 c.c. of water.

The sample of coal (briquette) is then placed in the platinum crucible (fig. 6-1). The iron fuse wire, which surrounds the coal sample, is connected to the lower end of the two electrodes. The electrodes extend through the base of the bomb and connect the fuse wire to an electric circuit. The coal can be ignited by closing the electric circuit.

The bomb is placed inside a copper vessel known as the calorimeter which contains a definite weighed amount of water (2,000 c.c.). A stirring device for agitating (or stirring) the water within the calorimeter is attached to the apparatus and it is worked by an electric motor.

The calorimeter filled with water and containing the bomb, is surrounded by an outer vessel (not shown in fig. 6-1) which forms an air space around the calorimeter and thereby minimises the radiation losses to the surroundings.

The temperature of the water in the calorimeter is measured by an accurate thermometer graduated to read upto one hundredth degree centigrade. An oxygen cylinder (not shown in fig. 6-1) supplies oxygen to the bomb to burn the coal. The oxygen cylinder is coupled to the bomb and the oxygen is admitted to the bomb through the oxygen supply valve until the pressure gauge on the cylinder indicates a pressure of about 25 atmospheres.

Temperature readings of the water in the calorimeter are taken at one minute interval for a period of 10 to 15 minutes. When either the temperature is constant or the change of temperature is regular, the coal sample is ignited by closing the electric circuit. The stirrer is kept in motion all the time and temperature readings are taken at 1/2 minute interval for about 5 minutes, until the maximum temperature is reached. The temperature readings are taken at one minute interval for 10 minutes after maximum temperature is reached. These temperature readings are required for calculating the heat exchange with jacket.

The heat generated by combustion of the sample of the coal is partly absorbed by the weighed quantity of water in the calorimeter and partly by the metal of the bomb and the calorimeter. The water equivalent of the bomb and the calorimeter is usually furnished by manufacturers or it can be determined experimentally.

Let M_1 = mass of sample of coal burnt in the bomb in gram,

M = mass of water filled in the calorimeter in gram,

m = water equivalent of the appartus (bomb and calorimeter) in gram ($M + m$ gives equivalent mass of water in gram),

t_1 = temperature of appartus and water before starting the experiment in degree centigrade,

t_2 = maximum temperature of apparatus and water during experiment, i.e. temperature after combustion in degree centigrade.

k = specific heat of water, and

$C.V.$ = calorific value of coal sample in Joules per gram (J/gm) or kJ/kg.

Then, heat liberated by combustion of the coal sample

= heat absorbed by the apparatus and water

i.e. $C.V. \times M_1 = (M + m) \times k\ (t_2 - t_1)$

$$\therefore\ C.V. = \frac{(M+m) \times k\,(t_2 - t_1)}{M_1} \text{ J/gm or kJ/kg} \qquad ..(6.3)$$

The heating value, thus, determined is the *higher calorific value* at constant volume.

For the very accurate determination of calorific value, corrections are made for the following :

— Heat may be gained from the atmosphere or heat may be lost to the atmosphere by the apparatus during the experiment.

— Combustion taking place in oxygen, results in the formation of nitric acid and sulphuric acid.

— The fuse wire liberates some heat energy.

— The products of combustion are cooled to the original temperature.

The correction for radiation (*or cooling correction*) has to be made in the observed temperature rise during the period of burning which will eliminate the error due to first reason. The cooling correction is expressed in degree centigrade and should be added to the observed temperature rise of water to get the corrected or true temperature rise. The heat resulting from the formation of nitric acid and sulphuric acid and that due to heating of the fuse wire, when subtracted from the heat absorbed by apparatus and the water, will eliminate the error due to second and third reasons as stated above. The error due to the fourth reason is only a small fraction of about one per cent.

Calorific value of a liquid fuel (oil) is measured in the same manner as with solid fuels by a bomb calorimeter. The sample of liquid fuel is filled in a gelatine capsule and the capsule is placed in the platinum crucible. Refer illustrative problem-4.

Problem-1: *A sample of coal has the following composition on mass basis : Carbon, 82%; Hydrogen, 8%; Sulphur, 2%; Oxygen 4%; and Ash, 4%.*

Calculate, using Dulong's formula, the higher calorific value of the coal.

Considering 1 kg of coal and using Dulong's formula {eqn. (6.2)},

$$\text{H.C.V. of coal} = 33{,}800C + 1{,}44{,}500\left(H_2 - \frac{O_2}{8}\right) + 9{,}300S \text{ kJ/kg}$$

$$= 33{,}800 \times 0{\cdot}82 + 1{,}44{,}500\left(0{\cdot}08 - \frac{0{\cdot}04}{8}\right) + 9{,}300 \times 0{\cdot}02$$

$$= \mathbf{38{,}739{\cdot}5\ kJ/kg \text{ or } 38{\cdot}7395\ MJ/kg}$$

Problem-2 : *Calculate the theoretical higher and lower calorific values of a fuel having the following composition on mass basis : Carbon, 85·5%; Hydrogen, 12·5%; and Oxygen, 2·0%. Take C.V. of hydrogen = 1,44,500 kJ/kg and that of carbon = 33,800 kJ/kg. Take enthalpy of evaporation of steam as 2,257 kJ/kg at atmospheric pressure.*

We first assume that any oxygen if present in the fuel is already combined with some of the hydrogen, so that the heat released from this part of the hydrogen is not available.

1 kg of H_2 combines with 8 kg of O_2 to produce 9 kg of H_2O.

∴ 0·02 kg of O_2 will have combined with 0·02/8 = 0·0025 kg of H_2

∴ Hydrogen available for heat release = 0·125 − 0·0025 = 0·1225 kg

Calorific value of hydrogen = 1,44,500 kJ/kg

∴ Heat released from the combustion of 0·1225 kg of H_2 = 1,44,500 × 0·1225
= 18,701 kJ

Calorific value of carbon = 33,800kJ/kg

∴ Heat released from the combustion of 0·855 kg of C = 33,800 × 0·855
= 28,899 kJ

∴ H.C.V. of fuel = 28,899 + 18,701 = **47,600 kJ/kg** or **47·6 MJ/kg**

Alternatively, considering 1 kg of fuel and using Dulong's formula (eqn. 6.2)

$$\text{H.C.V. of fuel} = 33{,}800\ C + 1{,}44{,}500\left(H_2 - \frac{O_2}{8}\right) + 9{,}300S\ \text{kJ/kg}$$

$$= 33{,}800 \times 0{\cdot}855 + 1{,}44{,}500\left(0{\cdot}125 - \frac{0{\cdot}02}{8}\right) + 0 = \mathbf{47{,}600\ kJ/kg\ (same\ as\ before)}$$

Now, 1 kg of Hydrogen combines with 8 kg of oxygen to form 9 kg of steam.

∴ Amount of water vapour (steam) formed during combustion of one kg of fuel
$= 9H_2 = 9 \times 0{\cdot}125 = 1{\cdot}125$ kg.

Considering steam pressure as atmospheric, enthalpy of evaporation of 1·125 kg of steam at 100°C = 1·125 × 2,257 = 2,539 kJ.

∴ L.C.V. of fuel = H.C.V. − Enthalpy of evaporation of steam formed
= 47,600 − 2,539 = **45,061 kJ/kg** or **45·061 MJ/kg**

Problem-3: *The following data refers to experimental determination of the calorific value of a sample of coal containing 6·86% of hydrogen :*

Mass of the sample of coal 0·862 gm;

Mass of the fuse wire 0·0325 gm of calorific value of 6,800 J/gm;

Mass of water in calorimeter 2,000 gm;

Water equivalent of calorimeter 350 gm;

Observed temperature rise of water from 16·235°C to 19·28°C;

Cooling correction + 0·017°C

Take specific heat of water as 4·187 kJ/kg K (4·187 × 10^3 J/kg K). Find the higher and lower calorific values of the coal.

Equivalent mass of water = 2,000 + 350 = 2,350 gm

Observed temperature rise of water = 19·28 − 16·235 = 3·045°C

Cooling correction = + 0·017°C

Corrected temperature rise of water = 3·045 + 0·017 = 3·062°C

Heat absorbed by the equivalent mass of water
= {Equivalent mass of water} × {Specific heat of water} ×
{Corrected temperature rise of water}

$= 2{,}350 \times 4{\cdot}187 \times 3{\cdot}062 = 30{,}130$ J.

Heat received by this mass of water from coal = 30,130 − heat from fuse wire

$= 30{,}130 - (0{\cdot}0325 \times 6{,}800) = 29{,}909$ J.

Mass of the sample of coal = 0·862 gm.

Higher C.V. of Coal $= \dfrac{29{,}909}{0{\cdot}862} = 34{,}697$ J/gm or **34,697 kJ/kg**

Now, 1 kg of hydrogen combines with 8 kg of oxygen to form 9 kg of steam. Therefore, steam formed during combustion of one kg of fuel $= 9H_2 = 9 \times 0{\cdot}0686 = 0{\cdot}6174$ kg.

Considering steam pressure as atmospheric, enthalpy of evaporation of 0·6174 kg of steam at 100°C = 0·6174 × 2,257 = 1,393 kJ

∴ Lower C.V. of coal = Higher C.V. − Enthalpy of evaporation of steam formed

= 34,697 − 1,393 = **33,304 kJ/kg.**

6.6 Liquid Fuels

Liquid Fuels are hydrocarbons. Depending on the number of carbon atoms, they can be classified as light oils and heavy oils. Light oils are volatile. The general chemical formula for all hydrocarbons is C_mH_n. They are commonly used for internal combustion engines and for steam raising purposes.

Most of the liquid fuels used in industry are prepared from one or the other of the following substances :

— Crude petroleum,

— By-products from the manufacture of coal gas, and

— Vegetable matter.

Crude Petroleum is obtained from natural reservoirs in the earth's crust through wells which are drilled in likely places until oil is struck (found). Crude petroleum as it comes from the wells is of complex nature. It has to be to processed in order to break it down into many grades of oils required for numerous applications. The separation of crude petroleum into numerous oils used in industry is achieved by the process of distillation (i.e. by heating the crude petroleum and condensing the vapour thus formed) at various temperatures and pressures. When crude petroleum is refined by a process of distillation, a number of different substances come off, leaving finally a thick residue. Petrol (or gasoline), paraffin oil or kerosene, heavy fuel oils used in the furnaces of oil fired boilers and diesel engines, lubricating oils, greases, paraffin wax, bitumen, etc., are thus produced by the processes of distillation.

Petrol or **Gasoline** comes off at temperatures from 65°C to 220°C. It is a source of power for light petrol engines used for road traction, aviation and small industrial installations. Its calorific value is about 44,250 kJ/kg.

Kerosene or **Paraffin** oil comes off at temperatures from 220°C to 345°C. It is used as a heating and lighting fuel. It is heavier and less volatile than petrol. It requires to be heated before it becomes volatile. Its calorific value is 44,000 kJ/kg. It is used as a fuel for heavy road traction, tractors and internal combustion engines.

Fuel oils used in Diesel engines and in the furnaces of oil fired boilors, are distilled at temperature from 345°C to 470°C. These oils are heavy and non-volatile. The calorific value of these oils is about 44,250 kJ/kg.

Tar is an important by-product from the manufacture of coal gas and it may be redistilled to produce valuable fuel like the *Benzene* (C_6H_6). The important feature of such oil is that it is much less liable to detonation than standard petrol, and forms a good alternative to petrol for use as a fuel in internal combustion engines.

Alcohol is formed by fermentation of vegetable matter, and has been widely used as a commercial fuel. It can be used as a fuel in internal combustion engines in a similar manner as petrol but its cost prohibits its use as a fuel. Its calorific value is about 26,800 kJ/kg.

It is necessary to examine the different properties and characteristics of fuel oils, owing to the variable nature of the numerous oils now available. The properties of a fuel oil that affect its suitability and which should be examined are : viscosity, specific gravity, flash point, ignition point, cold test, sulphur, water and ash contents, mechanical impurities, mean specific heat and heat value. These characteristics are determined by laboratory tests carried out according to some specific standards.

The *advantages* of liquid fuel over solid fuel (coal), when used as a fuel in boilers are:

- Higher calorific value,
- Reduction in the cost of handling,
- Saving in storage space required,
- Easy control of combustion and thus evaporation rate,
- No loss in heat value during storage,
- Cleanliness and freedom from dust,
- Higher combustion efficiencies, and
- Ease of starting and shutting of boilers

The *disadvantages* of liquid fuel when used as a fuel in boilers are :

- Danger of explosion,
- Necessity of heating the oil in the storage tanks in very cold climates to lower the viscosity, and
- Higher cost and its non-availability in the areas of a country where it is not produced.

Problem-4 : *The following data refers to an experimental determination of the calorific value of a sample of fuel oil containing 14% of hydrogen :*

Specific gravity of fuel oil	.. *0·845*
Mass of water in calorimeter	.. *2 kg*
Water equivalent of calorimeter	.. *600 gm*
Mass of sample of fuel oil	.. *1 gm*
Temperature of water in calorimeter before combustion	.. *19·5°C*
Temperature of water in calorimeter after combustion	.. *23·75°C*

Find the higher and lower calorific values of fuel oil. Take specific heat of water as 4·187 kJ/kg K and enthalpy of evaporation of steam as 2,275 kJ/kg at atmospheric pressure.

Equivalent mass of water = 2 + 0·6 = 2·6 kg

Temperature rise of water = 23·75 − 19·5 = 4·25°C.

Specific heat of water = 4·187 kJ/kg K.

Heat absorbed by equivalent mass of water

= Equivalent mass of water × temperature rise of water × specific heat of water

= 2·6 × 4·25 × 4·187 = 46·266 kJ

Mass of sample of fuel oil = 1 gm = 0·001 kg.

$$\therefore \text{ Higher calorific value of fuel oil (H.C.V.)} = \frac{46{\cdot}266}{0{\cdot}001} = \mathbf{46{,}266 \text{ kJ/kg}}$$

This figure, i.e. 46,266 kJ/kg refers to what is known as the *higher calorific value of the fuel.* In all hydrocarbon fuels, one of the products of combustion is water, which at combustion temperature is in the form of steam. In the bomb calorimeter experiment, this steam on condensing gives up its enthalpy of evaporation, which assists in raising the temperature of water in the calorimeter. In an oil engine, however, the temperature of the expanding gases never falls below that of condensation, and therefore, the enthalpy of evaporation of steam cannot appear as sensible heat and cannot be made use of by the engine to develop mechanical work. It is therefore, customary to *subtract* the enthalpy of evaporation of steam from the H.V.C. in order to give *lower calorific value of the fuel,* which is the only heat value, the engine makes use of.

Now, 1 kg of hydrogen combines with 8 kg of oxygen to form 9 kg of steam.

Steam formed during combustion of 1 kg of fuel oil = $9H_2$ = 9 × 0·14 = 1·26 kg

Considering steam pressure as atmospheric, enthalpy of evaporation of 1·26 kg of steam formed at 100°C = 1·26 × 2,257 = 2,844 kJ

L.C.V. of fuel of = H.C.V. − Enthalpy of evaporation of steam formed

= 46,266 − 2,844 = **43,422 kJ/kg**

6.7 Gaseous Fuels

Gaseous fuels may be divided into two classes :

— *Natural Gas,* and

— *Prepared Gases.*

Natural gas is given off near the oil bearing areas. *Natural Gas* consists mainly of methane (marsh gas) and ethylene and also oxygen, CO and CO_2. Its calorific value varies from 35,500 to 37,200 kJ/m^3.

Prepared gases used as fuels are : coals gas, producer gas, water gas, mond gas, blast furnace gas, coke-oven gas, marsh gas and oil gas.

Coal gas is obtained by distilling coal in retorts. It consists mainly of hydrogen, CO, CO_2, methane (CH_4), Olefines ($C_n H_{2n}$) and nitrogen. Its calorific value varies from 21,000 to 25,000 kJ/m^3.

Producer gas consists of the products derived from the passage of air and steam through a bed of incandescent fuel in gas producers. The details of production and theoretical analysis of this gas are given fully in Chapter-7. Producer gas is a mixture of carbon monoxide, hydrogen, nitrogen and carbon dioxide. The essential component of producer gas is carbon monoxide. It has a low calorific value but it is cheap. Its calorific value varies from 4,200 to 6,600 kJ/m^3.

Water gas is a mixture of hydrogen and carbon monoxide and is obtained by the contact of steam with incandescent carbon in the form of coal or coke. Its calorific value ranges from 11,500 to 23,000 kJ/m^3. Because of use of steam (water) in its manufacture, it is known as water gas. As it burns with a blue flame it is also known as *blue gas*.

Mond gas is obtained by injecting large quantity of steam in a producer, and is suitable for use in gas engines. Its calorific value is about 5,800 kJ/m^3.

Blast furnace gas is obtained as a by-product in smelting of pig iron. It has a low heating value of about 3,800 kJ/m^3. This gas is a mixture of carbon monoxide, hydrogen, carbon dioxide, nitrogen and some traces of methane. Actual composition depends upon the fuel and method adopted for operation of blast furnace. Blast furnace gas contains considerable amount of dust and it must be cleaned before its use.

Cock-oven gas is produced by high temperature carbonisation of bituminous coal. It is a by-product from coke ovens in the manufacture of coke. Its calorific value ranges from 14,500 to 19,000 kJ/m^3. It is essentially a mixture of methane and hydrogen, but also contains ethylene, carbon monoxide, carbon dioxide and nitrogen.

Marsh gas is a simple hydrocarbon gas (methane, CH_4), produced in nature by the decay of vegetable matter under water. Its calorific value is about 23,000 kJ/m^3.

Oil gas is produced by vaporisation and thermal cracking of oils with air or steam. It is a combination of cracked oil vapours and blue (water) gas. The oil gas predominates and the blue gas is in much smaller quantity. Its calorific value varies from 17,000 to 25,000 kJ/m^3.

6.7.1 Calorific Value : Calorific value of gaseous fuel is generally expressed in kJ per cubic metre of the gas at S.T.P. or N.T.P. and may be obtained :

.. Theoretically by calculations, based on the chemical analysis of gas, or

.. Experimentally by means of a gas calorimeter.

Theoretically the calorific value in kJ/m^3 of a gaseous fuel can be calculated by making chemical analysis and then summing up the heat evolved by the combustible constituents when burnt separately. The heating value of some typical gases are given in table 6-1.

Table 6-1. Calorific (heating) value of gases

Gas	Symbol	Heating value in kJ per m^3 at N.T.P.
Hydrogen	H_2	13,000
Carbon monoxide	CO	12,900
Methane	CH_4	39,500
Ethane	C_2H_6	69,400
Ethylene	C_2H_4	62,400
Acetylene	C_2H_2	58,000

The calorific value of gaseous fuel can also be determined experimentally (very accurately) by a gas calorimeter, known as Junker's gas calorimeter.

Junker's Gas Calorimeter (shown in fig. 6-2) consists of a vertical cylindrical chamber containing vertical tubes. The gas is burned in a Bunsen burner placed underneath the water chamber. The products of combustion pass upward through the

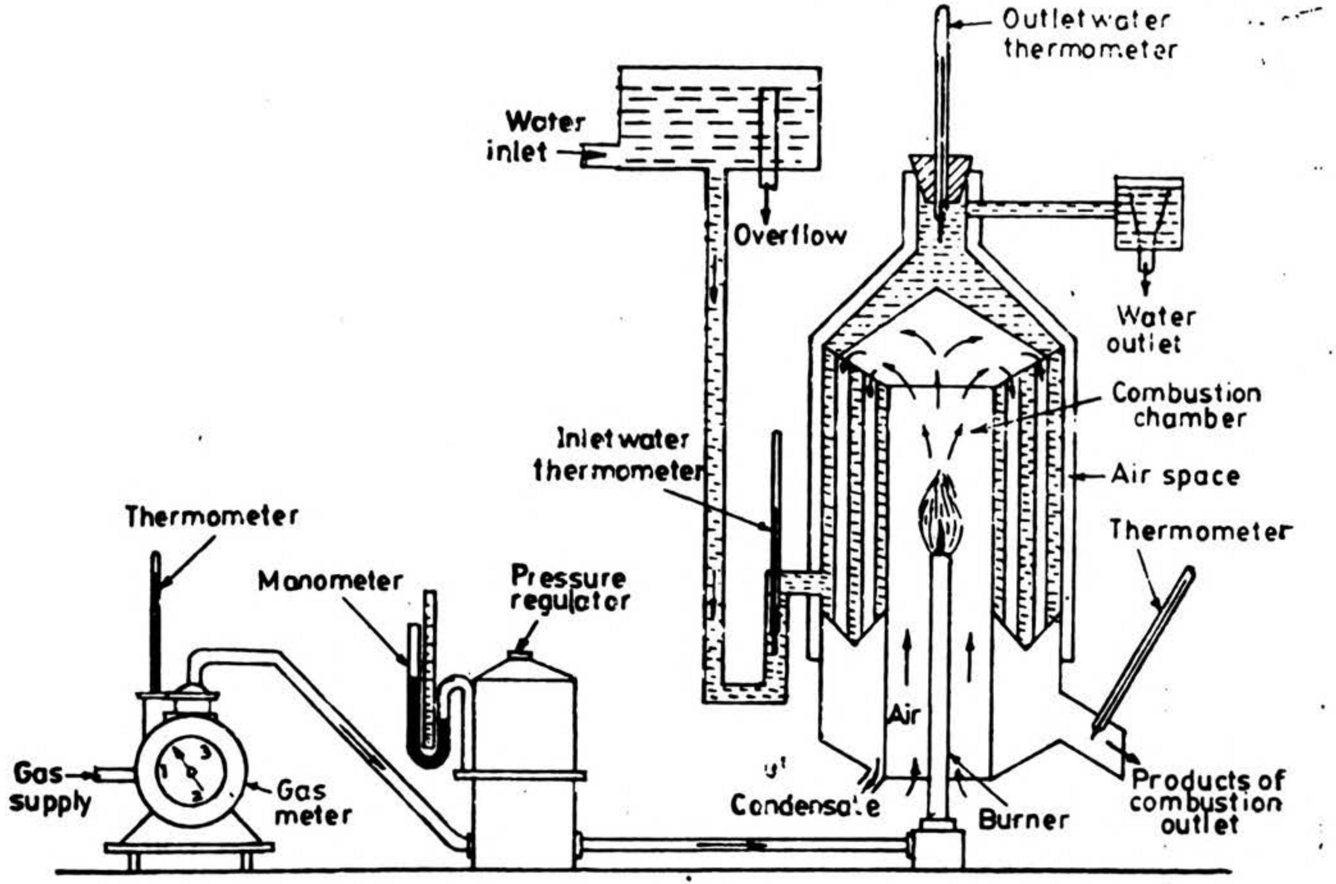


Fig. 6-2. Junker's gas calorimeter.

combustion chamber and then downward through the tubes. The tubes are surrounded by continuously flowing water. The water enters at the bottom and leaves at the top. The water is supplied under a constant head to ensure uniform rate of circulation.

The gas flow to the calorimeter is measured by a gas meter. The pressure of the gas is controlled by a gas pressure regulator. This is necessary to damp out pulsations in the gas supply which would affect the results adversely. The pressure of the gas is measured by a water manometer. The gas temperature is measured by a theremometer at the gas meter and the temperature of the products of combustion is measured by another thermometer placed in the products of combustion outlet pipe. The temperature of the water entering and leaving the calorimeter are also measured by two thermometers provided in the inlet and outlet water pipes. The water which flows through the calorimeter is collected and weighed. The amount of condensate formed due to the combustion of hydrogen in the fuel gas is determined by collecting it in a small graduated glass vessel (not shown). Certain specific standards are observed regarding the rate of flow of the gas and water. The rate of gas flow is specified to be about 6 litres per minute for a test of 4 minutes duration, during which time about 2,000 c.c. of water should be circulated and the temperature rise should be about 20°C.

Readings of the inlet and outlet water temperatures should be taken after every 30 seconds during the test. Other readings to be taken during the test are : the amount of gas burned and its pressure and temperature at the meter, the amount of water passing through the calorimeter, the amount of condensate collected, and the barometric pressure.

The calorific value of the gas in kJ per cubic metre is determined by multiplying the mass of water passed in kg by the rise of temperature of water in °C, and the specific heat of water in kJ/kg K and dividing the product by the volume of gas used in cubic metres reduced to S.T.P. (15°C and 760 mm Hg) or to N.T.P. (0°C and 760 mm Hg).

The reduction of volume from measured temperature and pressure to standard temperature and pressure (S.T.P.) is done as follows :

Let p_1, v_1 and T_1 be absolute pressure, volume and absolute temperature of gas at the meter, and p_2, v_2 and T_2 be absolute pressure, volume and absolute temperature of the gas at 15°C and 760 mm Hg (at S.T.P.).

Here, $p_1 = \dfrac{\text{pressure shown by manometer in mm of water}}{13{\cdot}6} + \begin{bmatrix}\text{observed barometric}\\ \text{pressure in mm of Hg}\end{bmatrix}$

v_1 = volume of gas used (burnt) as shown by the meter in m^3,

T_1 = observed temperature of gas at meter in K (°C + 273),

p_2 = 760 mm Hg,

v_2 = volume of gas used, reduced to S.T.P. in m^3, and

T_2 = 15 + 273 = 288 K.

$$\frac{p_1 v_1}{T_1} \text{ (at supply conditions)} = \frac{p_2 v_2}{T_2} \text{ (at S.T.P. conditions)}$$

From which the volume of gas used at S.T.P., $v_2 = \dfrac{p_1 v_1 T_2}{p_2 T_1}$ m^3

The higher calorific value in kJ per m^3 at S.T.P. should be calculated from the heat balance equation,

i.e. heat lost by gas = heat gained by water.

Hence,

$$\begin{bmatrix}\text{H.C.V.}\\ \text{in kJ/m}^3\end{bmatrix} = \frac{\text{mass of water collected in kg} \times \text{specific heat of water in kJ/kg K} \times \text{temp. rise in °C}}{\text{volume of gas burnt in m}^3 \text{ at S.T.P.}}$$

$$\begin{bmatrix}\text{L.C.V.}\\ \text{in kJ/m}^3\end{bmatrix} = \text{H.C.V.} - \frac{\begin{bmatrix}\text{mass of condensate}\\ \text{in kg}\end{bmatrix} \times \begin{bmatrix}\text{total heat recovered from}\\ \text{condensate in kJ per kg}\end{bmatrix}}{\text{volume of gas burnt in m}^3 \text{ at S.T.P.}} \quad ..\ (6.6)$$

Problem-5 : *The following observations were made during a test of a sample of gas in Junker's gas calorimeter :*

Gas burnt, 0·03 m^3; pressure of gas supplied, 68 mm of water above atmospheric pressure; temperature of gas, 25°C; barometric pressure, 750 mm of Hg; mass of water passing through the calorimeter, 10 kg; temperature of circulating water at inlet, 26°C and at outlet, 38·5°C; steam condensed during the test, 0·025 kg; temperature of steam condensed during the test, 32°C.

Determine the higher and lower calorific values of the gas per m^3 at a temperature of 15°C and pressure of 760 mm Hg (S.T.P.).

Take the enthalpy of evaporation of steam as 2,257 kJ/kg at atmospheric pressure, specific gravity of mercury as 13·6 and specific heat of water as 4·187 kJ/kg K.

The volume of gas burned is measured at a pressure of 68 mm of water above atmospheric pressure and at a temperature of 25°C. This volume (0·03 m^3) should be reduced to standard pressure of 760 mm Hg and standard temperature of 15°C.

Let p_1, v_1, T_1 be the absolute pressure, volume and temperature of gas at the meter, and p_2 v_2, T_2 be absolute pressure, volume and absolute temperature at 15°C and 760 mm Hg (or at S.T.P.).

Applying the general law,

$$\frac{p_1 v_1}{T_1} \text{ (at supply conditions)} = \frac{p_2 v_2}{T_2} \text{ (at S.T.P. conditions)}$$

From which the volume of gas at S.T.P., $v_2 = \frac{p_1 v_1 T_2}{p_2 T_1}$

Here, $v_1 = 0{\cdot}03\ m^3$, $p_1 = \frac{68}{13{\cdot}6} + 750 = 755$ mm Hg, $T_1 = 25 + 273 = 298$ K.

p_2 = 760 mm Hg, T_2 = 15 + 273 = 288 K, v_2 is to be determined.

Substituting the values,

Volume of gas burned at S.T.P., $v_2 = \frac{755 \times 0{\cdot}03 \times 288}{760 \times 298} = 0{\cdot}0293\ \text{m}^3$

Heat absorbed to raise the temperature of 10 kg of water from 26°C to 38·5°C

$= 10 \times 4{\cdot}187 \times (38{\cdot}5 - 26) = 523{\cdot}4$ kJ

$\therefore$ Higher C.V. of the gas $= \frac{523{\cdot}4}{0{\cdot}0293} =$ **17,863 kJ/m³ at S.T.P.**

Heat given out by 1 kg of steam on condensing and cooling down to 32°C is equal to enthalpy of evaporation of steam, 2,257 kJ at atmospheric pressure, *plus* cooling of condensed water from 100°C to 32°C = 68 × 4·187 = 284·72 kJ or a total of 2,257 + 284·72 = 2,541·72 kJ/kg.

Hence, 0·025 kg of steam on condensing and cooling down to 32°C, gives out 0·025 × 2,541·72 = 63·54 kJ.

Therefore, the actual (net) heat absorbed by water passing through the calorimeter from combustion of gas = 523·4 − 63·54 = 459·86 kJ.

$\therefore$ Lower C.V. of the gas at S.T.P. $= \frac{459{\cdot}86}{0{\cdot}0293} =$ **15,695 kJ/m³ at S.T.P.**

Problem-6 : *During an engine trial, the gas as used was tested in a calorimeter and gave the following results :*

20 litres (0·02 m³) of gas at pressure 50 mm Hg above atmospheric pressure of 740 mm Hg and temperature 18°C was burned, which raised the temperature of 10·2 kg of water from 16°C to 24·5°C and the steam formed was condensed to 20 grams of water. Calculate the higher and lower calorific values per cubic metre of the gas at a temperature of 0°C and pressure of 760 mm of Hg (N.T.P.). Take the heat liberated in condensing the water vapour (steam) and cooling the condensate as 2,520 kJ/kg and specific heat of water as 4·187 kJ/kg K.

Let p_1, v_1, T_1 and p_2, v_2, T_2, be the observed and normal absolute pressure, volume and absolute temperature respectively.

Applying the general law,

$$\frac{p_1 v_1}{T_1} \text{ (at supply conditions)} = \frac{p_2 v_2}{T_2} \text{ (at N.T.P., conditions)}$$

From which the volume of gas at N.T.P., $v_2 = \frac{p_1 v_1 T_2}{p_2 T_1}$

Given, $v_1 = \frac{20}{1,000} = 0{\cdot}02\ \text{m}^3$; $p_1 = 50 + 740 = 790$ mm Hg; $T_1 = 18 + 273 = 291$ K.

v_2 is to be determined; $p_2 = 760$ mm Hg; $T_2 = 0 + 273 = 273$ K.

Substituting these values,

Volume of gas burnt at N.T.P., $v_2 = \frac{790 \times 0{\cdot}02 \times 273}{760 \times 291} = 0{\cdot}0195\ \text{m}^3$.

Heat absorbed by water in the calorimeter $= 10{\cdot}2 \times 4{\cdot}187 \times (24{\cdot}5 - 16) = 363$ kJ.

$\therefore$ H.C.V. of gas $= \frac{363}{0{\cdot}0195} =$ **18,616 kJ/m³ at N.T.P.**

Heat given up by 0·02 kg of steam on condensing and cooling

$= 0{\cdot}02 \times 2,520 = 50{\cdot}4$ kJ

$\therefore$ Actual heat absorbed by water from combustion of gas $= 363 - 50{\cdot}4 = 312{\cdot}6$ kJ

$\therefore$ L.C.V. of gas $= \frac{312{\cdot}6}{0{\cdot}0195} =$ **16,000 kJ/m³ at N.T.P.**

Tutorial - 6

1. Define the term 'fuel'. What are the principal constituents of a fuel ? Classify the various fuels.
2. Explain the procedures for the following analysis for determining the composition of coal :
   (i) Ultimate analysis, and
   (ii) Proximate analysis.
3. State and explain Dulong's formula for theoretical determination of calorific value of a fuel
   (i) when there is no oxygen present in the fuel, and
   (ii) when oxygen is present in the fuel.
4. Differentiate between natural solid fuels and artificial or prepared solid fuels. Name some important natural solid fuels and some important artificial solid fuels.
5. Name the important liquid fuels used in the industry and give their important properties. Give the advantages of liquid fuels over solid fuels.
6. Define the term calorific value of a fuel. Distinguish between the lower calorific value and higher calorific value of a fuel.
7. Describe with the help of sketches the working of a Bomb calorimeter.
8. Describe with neat sketches, an experiment for the determination of calorific value of a given sample of a coal. Prepare a list of observations you would make and give sample calculations for the determination of calorific value of coal.
9. You are required to determine, by means of the bomb calorimeter, the clarific value of the coal being used in a boiler trial. Describe :
   (a) How would you prepare the sample of coal ?
   (b) The procedure you would adopt in carrying out the experiment and the observations you would make.
   (c) How would you use your observations to calculate the calorific value of the coal ? Does the bomb calorimeter give a higher calorific value or lower calorific value ?
10. The analysis of a sample of coal gave : 89% carbon and 3·4% hydrogen. It was tested in a bomb calorimeter and the following data was obtained :

    Mass of coal, 0·86 gm; mass of water in the calorimeter, 2,000 gm; water equivalent of calorimeter, 350 gm; observed temperature rise of water from 25°C to 28·45°C; cooling correction + 0·016° C;mass of fuse wire

0·03 gm of C.V. 6,700 J/gm. Take specific heat of water as 4·187 kJ/kg K and enthalpy of evaporation of steam as 2,257 kJ/kg at atmospheric pressure.

Calculate the higher and lower calorific values of the coal.

[39,419 kJ/kg; 38,728 kJ/kg]

11. Describe with the help of a neat sketch an experiment for determing the calorific value of a sample of oil fuel, showing how the value is calculated from the readings obtained.

12. The sample of oil fuel gave the following composition on mass basis : carbon, 86·5% and hydrogen, 13·5%. Taking the calorific value of carbon = 33,800 kJ/kg and that of hydrogen = 1,44,500 kJ/kg when completely burnt, calculate the theoretical higher and lower calorific values of the oil fuel. Take enthalpy of evaporation of steam at atmospheric pressure as 2,257 kJ/kg.

[48,745 kJ/kg; 46,003 kJ/kg]

13. The following observations were made when a sample of fuel oil having 0·86 specific gravity and containing 86·5% carbon and 13·5% hydrogen was tested in the bomb calorimeter for its heating value :

Mass of fuel oil, 0·9 gm; mass of water in calorimeter, 2·1 kg; water equivalent of calorimeter, 750 gm; observed rise in temperature of water, 3·47°C; correction for loss by radiation +0·03°C. Determine the higher and lower calorific values of fuel oil. Take specific heat of water as 4·187 kJ/kg K and enthalpy of evaporation of steam as 2,257 kJ/kg at atmospheric pressure.

[46,405 kJ/kg; 43,662 kJ/kg]

14. (a) Differentiate between natural gas and prepared gas.
(b) Write short notes on producer gas and blast furnace gas.

15. (a) Name some of the important gaseous fuels commonly used in industry for heating and power producing purposes and give their physical properties ?
(b) What are the advantages of gaseous fuels over solid fuels ?

16. (a) Describe with the help of a neat sketch the method of determining the calorific value of a sample of gaseous fuel, showing how the result is calculated from the data obtained.
(b) Distinguish between the higher and lower calorific values of a gaseous fuel.

17. The following data refer to a calorific value test of a fuel gas by a gas calorimeter : Volume of gas used, 0·065 m^3 (reckoned at S.T.P.); mass of water heated from 11°C to 25°C, 22·86 kg; mass of steam condensed, 0·0355 kg. Find the higher and lower calorific values per cubic metre of the gas at S.T.P. Take the heat liberated in condensing the water vapour and cooling the condensate as 2,500 kJ/kg. Take specific heat of water as 4·187 kJ/kg K.

[20,615 kJ/kg; 19,250 kJ/kg]

18. Describe with the help of sketches the working of Junker's gas calorimeter.

A sample of gas was tested by Junker's gas calorimeter and the results obtained were :

Gas burned	.. 0·06 m^3
Temperature of gas at meter	.. 19°C
Pressure of gas at meter	.. 76·2 mm of water
Barometer reading	.. 750 mm of Hg
Mass of water circulated through the calorimeter	.. 25 kg
Temperature of water at inlet	.. 20·4°C
Temperature of water at outlet	.. 30·5°C
Steam condensed during the test	.. 50 gm
Temperature of condensate leaving calorimeter	.. 28°C

Calculate the higher and lower calorific values of the gas per cubic metre at a temperature of 15°C and pressure 760 mm Hg (S.T.P.). Take enthalpy of evaporation of steam as 2,257 kJ/kg at 100°C and specific heat of water as 4·187 kJ/kg K. (Specific gravity of mercury = 13·6).

[17,980 kJ/kg; 15,805 kJ/kg]

7
COMBUSTION

7.1 Introduction

Combustion or burning is a chemical process in which the inflammable matter in a substance combines with oxygen at a temperature above the ignition temperature of the substance and results in the evolution of heat and light. A fuel, at the time of burning, must be in gaseous forms so as to unite with oxygen, which is present in the air. A liquid fuel has to be vaporised or atomised before it is burnt. The process of gassifying a solid fuel and mixing it with the air at the proper temperature is more difficult and many means have been adopted in recent years to accomplish the proper burning of solid fuels.

The combustion process involves the oxidation of constituents in the fuel that are capable of being oxidized, and can, therefore, be represented by a chemical equation. These equations indicate how much oxygen is required to combine with the given amount of fuel and also the amount of resulting gases produced by the combustion. The knowledge of these tests enables an engineer to determine the correct amount of air to be supplied and the amount of the heat carried away by the products of combustion up the chimney.

To accomplish perfect combustion, there should be an ample supply of air, a thorough mixing of the air, and a sufficiently high temperature to maintain combustion.

7.2 Chemistry of Combustion

Fuels are made up of substances such as carbon and hydrogen with small amount of oxygen, nitrogen, sulphur, etc. These substances are basic elements which cannot be broken down into simpler parts having different properties. However, if temperature and pressure conditions are favourable, these elements may combine and produce an entirely different substance. This substance is known as *chemical compound*. For example, when carbon combines with oxygen, the compound formed is carbon dioxide. Again, if hydrogen combines with oxygen, the compound formed is water.

All substances are composed of minute particles of chemical elements. These minute particles are called atoms and these atoms are arranged in groups, called molecules. The elements can be represented, for convenience, by symbols. The symbols C, H, O, and S represent one atom each of carbon, hydrogen, oxygen and sulphur respectively.

Hydrogen is used as the standard gas in order to introduce *quantity* in combustion calculations. The volume of one kg of hydrogen at 0°C and 760 mm of Hg pressure is 11·2 m^3 and this is taken as the standard volume.

The mass of one standard volume of any element under the above condition of pressure and temperature and in the form of a gas, is the atomic weight of the element.

The molecular weight of a substance is the mass of two standard volumes (22·4 m^3) of that substance when in the form of a gas at normal temperature and pressure. Both elements and compounds have molecular weights. The atomic weights are purely relative and are arbitrarily established. Hydrogen which is the lightest known element has been given the atomic weight 1 (one) and the weights of other elements are fixed in relation to it.

7.3 Mol and Molar Volume

When dealing with mixture of gases and gas reactions, it is more convenient to use the molecular weight of the gases as unit of mass and not the kilogram. Although the values in table 7-1 is not the true mass of a molecule in kilogram, they may be considered, for convenience, as mass in kilogram. The mass of a substance in kilogram equal to its tabulated molecular weight is called a kg-mol or kilogram-mol. Thus, one kg-mol of hydrogen means 2 kg of hydrogen, and one kg-mol of oxygen means 32 kg of oxygen.

The corresponding volume of the gas of this unit (kg-mol) at 0°C and 760 mm Hg is known as the molar volume. For example, the molar volume of hydrogen is 22·4 m^3 which is the volume of 2 kg of hydrogen at 0°C and 760 mm Hg.

Table 7-1 Atomic and Molecular Weights

Substance	Atomic symbol	Atomic weight	Molecular symbol	Molecular weight
Hydrogen	H	1	H_2	2
Oxygen	O	16	O_2	32
Nitrogen	N	14	N_2	28
Carbon	C	12	C	12
Sulphur	S	32	S	32
Water or Steam	..	..	H_2O	18
Carbon dioxide	..	..	CO_2	44
Carbon monoxide	..	..	CO	28
Sulphuric dioxide	..	..	SO_2	64
Methane	..	..	CH_4	16
Ethylene or Olefiant gas	..	..	C_2H_4	28
Acetylene	..	..	C_2H_2	26

When the substance (compounds) are made up of different elements, their molecular weight can be calculated by adding the atomic weight of elements concerned. For example one molecule of steam (H_2O) consists of 2 atoms of hydrogen + 1 atom of oxygen.

$\therefore$ Molecular weight of $H_2O = (1 \times 2) + 16 = 18$

Similarly, 1 molecule of CO_2 consists of 1 atom of carbon + 2 atoms of oxygen.

$\therefore$ Molecular weight of $CO_2 = 12 + (2 \times 16) = 44$

The table 7-1 gives the atomic weights of elements, and molecular weights of gaseous elements and chemical compounds which are frequently used in combustion calculations.

The molar volume is constant for all gases when they are at the same pressure and temperature. This is according to *Avogadro's law* which states that the equal volumes of different gases at the same temperature and pressure contain equal number of molecules. The numerical value of the molar volume of all gases at a temperature of 0°C and 760 mm Hg is 22·4 cubic metres. In other words, under these specified conditions, 22·4 m^3 of hydrogen weighs 2 kg, whilst the same volume (22·4 m^3) of oxygen weighs 32 kg, and so on.

7.4 Chemical Reactions

The chemical union of substances may be expressed by an equation or reaction, as it is called, Chemical equations are very valuable to engineers because they also indicate the proportions with which the elements react when there is to be no surplus. The union of hydrogen with oxygen produces steam. Symbolically this reaction is written as

$$\begin{aligned} 2H_2 + O_2 &= 2H_2O \\ 2(1 \times 2) + (16 \times 2) &= 2(1 \times 2 + 16) \\ \text{i.e. } 4 + 32 &= 36 \\ \text{or } 1 + 8 &= 9 \end{aligned}$$

The positive sign indicates that a chemical reaction has taken place. The sign has no algebraic interpretation. The equation reads that one kg of hydrogen unites with 8 kg of oxygen to produce 9 kg of steam or 2 kg-moles of hydrogen combine with 1 kg-mol of oxygen to form 2 kg-moles of steam.

All the initial substances that undergo the combustion process are called the **reactants,** and the substances that result from the combustion process are called **products.** From this example, it should be understood that a chemical equation not only expresses the result of reactions but it also has a quantitative significance. The equation obeys mathematical laws as the total mass on either side of the equation is the same.

Since, it is difficult to measure the mass of gas directly, it is more usual to consider the volumes which unite instead of the masses. Take for example, the chemical equation,

$$2H_2 + O_2 = 2H_2O$$

This equation can be read as "two cubic metres of H_2 combine with one cubic metre of O_2 to produce two cubic metres of steam at the same state as that of H_2 and O_2", provided the steam produced obeys the laws of perfect gases.

At high temperatures steam obeys the laws of perfect gases but at low temperatures steam would condense and its volume then would be negligible in comparison with that of the gases. A similar situation arises with the solid carbon in the equation $C + O_2 = CO_2$, which should not be read as one cubic metre of carbon combines with one cubic metre of oxygen because the volume of oxygen uniting with carbon is so great that the volume of carbon becomes negligible in comparison. It may, therefore, be said that one cubic metre of O_2 after reacting with C, produces one cubic metre of CO_2 at the same pressure and temperature as that of O_2.

The following chemical equations represent the combustion of different constituents of a fuel and they are most frequently used in combustion calculations :

Burning of *Carbon to Carbon dioxide* (complete combustion)

$$\begin{aligned} C + O_2 &= CO_2 \\ 12 + 16 \times 2 &= 12 + 16 \times 2 \\ \text{i.e. } 12 + 32 &= 44 \end{aligned}$$

or $1 + 2{\cdot}67 = 3{\cdot}67$

i.e. 1 kg of Carbon needs 2·67 kg of Oxygen and produces 3·67 kg of Carbon dioxide.

Burning of *Carbon to Carbon monoxide* (incomplete combustion)

$$2\,C + O_2 = 2\,CO$$

$$2 \times 12 + 16 \times 2 = 2\,(12 + 16)$$

i.e. $24 + 32 = 56$

or $1 + 1{\cdot}33 = 2{\cdot}33$

i.e. 1 kg of C takes 1·33 kg of O_2 and produces 2·33 kg of CO.

Burning of *Sulphur to Sulphur dioxide*

$$S + O_2 = SO_2$$

$$32 + 16 \times 2 = 32 + 16 \times 2$$

i.e. $1 + 1 = 2$

i.e. 1 kg of S needs 1 kg of O_2 and produces 2 kg of SO_2.

Burning of *Hydrogen to Steam*

$$2\,H + O_2 = 2\,H_2O$$

$$2(1 \times 2) + 16 \times 2 = 2\,(1 \times 2 + 16)$$

i.e. $1 + 8 = 9$

i.e. 1 kg of Hydrogen needs 8 kg of O_2 and produces 9 kg of steam.

Burning of *Carbon monoxide to Carbon dioxide*

$$2\,CO + O_2 = 2\,CO_2$$

i.e. $2\,(12 + 16) + 2 \times 16 = 2(12 + 16 \times 2)$

i.e. $1 + \frac{4}{7} = 1\frac{4}{7}$

i.e. 1 kg of CO needs 4/7 kg of O_2 and produces 11/7 kg of CO_2.

Methane :

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

i.e. $(12 + 1 \times 4) + 2(16 \times 2) = (12 + 16 \times 2) + 2(1 \times 2 + 16)$

or $1 + 4 = \frac{11}{4} + \frac{9}{4}$

Ethylene :

$$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$$

i.e. $(12 \times 2 + 1 \times 4) + 3(16 \times 2) = 2(12 + 16 \times 2) + 2(1 \times 2 + 16)$

or $1 + 3{\cdot}43 = 3{\cdot}14 + 1{\cdot}29$

Acetylene

$$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$$

i.e. $2(12 \times 2 + 1 \times 2) + 5\,(16 \times 2) = 4(12 + 2 \times 16) + 2\,(1 \times 2 + 16)$

or $1 + 3{\cdot}08 = 3{\cdot}39 + 0{\cdot}69$

From the point of view of combustion it is usual to assume that air is a mixture of oxygen and nitrogen, their proportions being :

On mass basis, oxygen 23 per cent, nitrogen 77 per cent, and

By volume, oxygen 21 per cent, nitrogen 79 per cent.

This means that one kg of oxygen is associated with 77/23 = 3·35 kg of nitrogen in (1 + 3·35) = 4·35 kg of air. Also, one cubic metre of oxygen is associated with 79/21 = 3·76 m^3 of nitrogen in (1 + 3·76) = 4·76 m^3 of air.

The nitrogen is an inert gas and takes no part in the chemical reaction occuring. Its presence in the air lowers the temperature of combustion, retards the intimate mixing of the fuel with oxygen, and also carries away heat up the chimney of the boiler.

The knowledge of the chemical composition of the fuel to be burnt and the application of chemistry to combustion problems enables the determination of the theoretical or minimum amount of the air required to burn completely a solid fuel or a liquid fuel or a gaseous fuel.

Once the mass of theoretical air required is determined, the actual air supplied can be adjusted to avoid undue heat losses. In practice, the quantity of air in excess of the theoretical air must be supplied to ensure complete combustion as some of the air may pass off without any chemical reaction. This extra amount of air supplied, is known as *excess air* and varies with the type and quantity of the fuel used, and the method of firing. In practice, excess air to the extent of 30% to 50% of the theoretical or minimum air is supplied to burn the fuel completely.

It should be noted that too great an excess air results in :

— lowering furnace temperature,

— increased heat loss in the chimney gases,

— increased smoke, and

— more power required to handle air and flue gases to produce artificial draught.

Too small an excess air results in :

— increasing furnace temperature, which may have bad effects on the structures and components of furnace, and

— Incomplete combustion.

7.5 Combustion Problems

In the problems of combustion of fuels, the values of the following items are usually required to be determined by calculations :

.. Mass and volume of minimum or theoretical air required for the complete combustion of 1 kg of fuel,

.. Conversion of volumetric composition of a gas to composition on mass basis.

.. Conversion of composition of a gas on mass basis to composition by volume,

.. Mass of carbon in one kg of flue gases,

.. Mass of flue gases produced per kg of fuel burned,

.. Mass of total or actual air, and excess air supplied per kg of fuel,

.. Flue gas analysis by volume and on mass basis,

.. Heat carried away by flue gases per kg of fuel, and

.. Volume of minimum air required for complete combustion of one m^3 of gaseous fuel.

The products of combustion that pass away through the combustion chamber to the chimney are called *flue gases.* The products of combustion from a fuel containing H,C, and S are : water vapour, CO_2, and/or CO and SO_2.These products form part of the flue gases from the combustion chamber. Other components of flue gases present are nitrogen and excess oxygen supplied with excess air. The nitrogen is from the air supplied for combustion purpose, and it may also originally be present in fuel.

The CO_2, CO, SO_2 and N_2 are called dry flue gases or products of combustion and the volumetric percentage of these gases can be determined by means of the Orsat apparatus described later.

7.6 Theoretical or Minimum Air Required for Combustion

It will be seen from the chemical equations in Art. 7.4 that for combustion,

Carbon requires 2·67 times its own mass of oxygen,

Hydrogen requires 8 times its own mass of oxygen, and

Sulphur requires its own mass of oxygen.

Considering 1 kg of a fuel, whose ultimate analysis gives Carbon, C kg; Hydrogen, H kg; Oxygen, O kg; and Sulphur, S kg (remainer being incombustible), then

Oxygen required to burn C kg of carbon = 2·67 C kg.

Oxygen required to burn H kg of hydrogen = 8 H kg.

Oxygen required to burn S kg of sulphur = S kg.

∴ Total oxygen required for complete combustion of 1 kg of fuel = $(2{\cdot}67C + 8H + S)$ kg.

As the fuel already contains O kg of oxygen which we assume can be used for combustion,

Theoretical oxygen required for complete combustion of 1 kg of fuel

$= (2{\cdot}67C + 8H + S - O)$ kg.

Since, the air contains 23% of oxygen on mass basis (the remaining 77% being considered as nitrogen),

1 kg of oxygen is contained in 100/23 kg of air.

Hence, minimum or theoretical mass of air required for complete combustion of

$$1 \text{ kg of fuel} = 100/23\ (2{\cdot}67C + 8H + S - O) \text{ kg} \quad ..(7.1)$$

Problem-1 : *Percentage composition of fuel on mass basis is C, 87·1; H_2 4·4; O_2, 1·2; and ash, 7·3. Find the minimum or theoretical mass and volume of air required to burn 1 kg of the fuel.*

The oxygen required to burn the two combustibles (carbon and hydrogen) may first be found. The chemical reactions are :

(i) $C + O_2 = CO_2$ (ii) $2H_2 + O_2 = 2H_2O$

$12 + 32 = 44$ $4 + 32 = 36$

or $1 + 2{\cdot}67 = 3{\cdot}67$ or $1 + 8 = 9$

The equations show that 1 kg of C requires 2·67 of O_2 to burn it, and

1 kg of H_2 requires 8 kg of O_2 to burn it.

Mass of C in 1 kg of fuel = 0·871 kg (by given composition).

∴ O_2 required to burn 0·871 kg of C = 0·871 × 2·67 = 2·323 kg.

Mass of H_2 in 1 kg of fuel = 0·044 (by given composition).

∴ O_2 required to burn 0·044 kg of H_2 = 0·044 × 8 = 0·352 kg.

∴ Total O_2 required = 2·323 + 0·352 = 2·675 kg/kg of fuel.

But, 0·012 of O_2 is already present in 1 kg of fuel (by given composition).

∴ Net O_2 required to burn 1 kg of fuel = 2·675 − 0·012 = 2·663 kg

As oxygen present in 100 kg of air is 23 kg,

Minimum mass of air required to burn 1 kg of fuel completely

= 2·663 x 100/23 = **11·66 kg.**

Alternatively, using eqn. (7.1), minimum mass of air required for complete combustion of one kg of fuel

$$= \frac{100}{23}\,[\,2{\cdot}67C + 8H + S - O\,]$$

$$= \frac{100}{23}\,[(2{\cdot}67 \times 0{\cdot}871) + (8 \times 0{\cdot}044) - 0{\cdot}012] = \mathbf{11{\cdot}66\ kg\ per\ kg\ of\ fuel}\ \text{(same as before).}$$

To find theoretical or minimum volume of air required :

The density of gas is proportional to its molecular weight and inversely to its molar volume. The molar volume of a gas at 0°C and 760 mm Hg (N.T.P.) is 22·4 m^3.

$$\therefore \text{ Density of } O_2 \text{ at 0°C and 760 mm Hg} = \frac{\text{molecular weight}}{\text{molar volume}} = \frac{32}{22{\cdot}4} = 1{\cdot}429 \text{ kg/m}^3$$

$$\text{Hence, volume of } O_2 \text{ required} = \frac{\text{mass of } O_2}{\text{density}} = \frac{2{\cdot}663}{1{\cdot}429} = 1{\cdot}843 \text{ m}^3$$

As O_2 is 21 per cent by volume in air, minimum volume of air required for complete combustion of 1 kg of fuel = $1{\cdot}843 \times \frac{100}{21}$ = **8·776 m^3 per kg of fuel.**

Problem-2 : *A sample of fuel was found to have the following percentage composition on mass basis : C, 80; H_2, 16; and ash etc., 4. Determine the minimum or theoretical mass and volume of air required to burn 1 kg of this fuel.*

Chemical reactions are :

(1) $C + O_2 = CO_2$ (ii) $2H_2 + O_2 = 2H_2O$

$12 + 32 = 44$ $4 + 32 = 36$

$1 + 2{\cdot}67 = 3{\cdot}67$ $1 + 8 = 9$

∴ O_2 required to burn 0·8 kg of carbon = 0·8 × 2·67 = 2·13 kg

∴ O_2 required to burn o·16 kg of hydrogen = 0·16 × 8 = 1·28 kg

∴ Total O_2 required = 2·13 + 1·28 = 3·41 kg/kg ,of fuel

∴ Minimum mass of air required to burn one kg of fuel.

= 3·41 × 100/23 = **14·83 kg/kg of fuel.**

Alternatively, using eqn. (7.1), minimum mass of air required for complete combustion of one kg of fuel = $\frac{100}{23}$ $[2{\cdot}67C + 8H + S - O]$ = $\frac{100}{23}$ $[(2{\cdot}67 \times 0{\cdot}8) + (8 \times 0{\cdot}16)] = 14{\cdot}83$ kg/kg of fuel (same as before)

Density of O_2 at N.T.P. $\frac{\text{molecular weight}}{\text{molar volume}} = \frac{32}{22{\cdot}4} = 1{\cdot}429$ kg/m^3

Hence, volume O_2 required to burn 1 kg of fuel = $\frac{3{\cdot}41}{1{\cdot}429} = 2{\cdot}387$ m^3.

$\therefore$ Minimum volume of air required = 2.387 × 100/21 = 11·366 m^3/kg of fuel

7·7 Conversion of Volumetric Composition on Mass Basis

Following example illustrates the procedure to convert the given volumetric composition of gas on mass basis.

Problem-3 : *The volumetric analysis of a fuel gas is :* CO_2*, 15%;* CO*, 2·2%;* O_2*, 1·6%;* N_2*, 81·2%. Convert this volumetric composition to percentage composition on mass basis.*

To convert volume of CO_2 to mass of CO_2 :

Let v be the volume of CO_2 in m^3 at the existing pressure and temperature of the flue gas whose analysis is given above. The volumes of the other constituents CO, O_2 and N_2 will also be v, since they are all at the same temperature and pressure. It is not necessary to know the exact value of this volume v, since it will ultimately be cancelled on division. The volume of CO_2 per cubic metre of flue gas is 0.15 m^3. Since, the mass of v m^3 volume of CO_2 is proportional to 44 (its molecular weight),

The proportional mass of 0·15 m^3 of CO_2 = $\frac{44}{v} \times 0{\cdot}15 = \frac{6{\cdot}6}{v}$ kg.

The proportional mass of CO, O_2 and N_2 may similarly be calculated and tabulated as shown in the table. The sum of all these proportional masses may then be found and it is $\frac{30{\cdot}477}{v}$ in this case. In $\frac{30{\cdot}477}{v}$ kg of flue gas, the mass of CO_2 is $\frac{6{\cdot}6}{v}$ kg.

$\therefore$ Mass of CO_2 in 1 kg of flue gas = $\frac{6{\cdot}6}{v} \times \frac{v}{30{\cdot}477} = 0{\cdot}216$ kg i.e. 21·6%.

The mass of other products per kg of flue gas and their percentages on mass basis can be calculated in the same manner and tabulated as shown below. The process of conversion of volumetric analysis to analysis on mass basis is simplified by constructing a table as under :

Constituent	Volume in one m^3 of flue gas *(a)*	Molecular weight *(b)*	Proportional mass in kg *(c)* = *(a)* × *(b)*	Mass per kg of flue gas *(d)* = $\frac{(c)}{\Sigma\,(c)}$	Percentage mass *(e)*
CO_2	0·150	44	$\frac{44}{v} \times 0{\cdot}15 = \frac{6{\cdot}6}{v}$	$\frac{6{\cdot}6}{30{\cdot}477} = 0{\cdot}216$	21·6
CO	0·022	28	$\frac{28}{v} \times 0{\cdot}022 = \frac{0{\cdot}615}{v}$	$\frac{0{\cdot}615}{30{\cdot}477} = 0{\cdot}022$	2·2
O_2	0·016	32	$\frac{32}{v} \times 0{\cdot}016 = \frac{0{\cdot}512}{v}$	$\frac{0{\cdot}512}{30{\cdot}477} = 0{\cdot}017$	1·7
N_2	0·812	28	$\frac{28}{v} \times 0{\cdot}812 = \frac{22{\cdot}75}{v}$	$\frac{22{\cdot}75}{30{\cdot}477} = 0{\cdot}745$	74·5
Total	= 1·000	–	$= \frac{30{\cdot}477}{v}$	= 1·000	= 100·00

- In short, the above procedure can be summarised as follows :
- Multiply volume of each constituents in one m^3 of flue gas, by its molecular weight.
- Find the sum of the products and divide each constituent by this sum.
- The result is the mass of each constituent in 1 kg of the flue gas.
- To have its percentage mass, multiply this mass by 100.

7.8 Conversion of Composition on Mass Basis to Composition by Volume

In order to convert analysis on mass basis to volume basis,

- Divide mass of each constituent in one kg of flue gas by its molecular weight.
- Find the sum of the quotients, and divide each quotient by this sum.
- The result is the volume of each constituent in one m^3 of flue gas.
- For percentage volume multiply this volume by 100.

This is reverse of the procedure just described for the conversion of volumetric analysis to analysis on mass basis.

Problem-4 : *The percentage composition of the flue gas on mass basis : CO_2, 16; CO, 1; O_2, 7; and N_2, 76. Convert this analysis on mass basis to percentage volumetric analysis.*

The process of conversion of analysis on mass basis to volumetric analysis is simplified by constructing the table as under :

Constituent	% mass analysis in kg (*a*)	Molecular weight (*b*)	Relative volume $(c) = \frac{(a)}{(b)}$	Percentage volumetric analysis $(d) = \frac{(c)}{\Sigma(c)} \times 100$
CO_2	16	44	$\frac{16}{44} = 0.3640$	$\frac{0.3640}{3.3382} \times 100 = 10.90$
CO	1	28	$\frac{1}{28} = 0.0357$	$\frac{0.0357}{3.3382} \times 100 = 1.05$
O_2	7	32	$\frac{7}{32} = 0.2185$	$\frac{0.2185}{3.3382} \times 100 = 6.55$
N_2	76	28	$\frac{76}{28} = 2.7200$	$\frac{2.7200}{3.3382} \times 100 = 81.50$
Total	= 100	–	= 3.3382	= 100.00

7.9 Mass of Carbon in Unit Mass of Flue Gases

Following example illustrates the method of estimation of mass of carbon in one kg of flue gases, if the volumetric analysis of flue gases is available.

Problem-5 : *Volumetric analysis of flue gas is CO_2, 15%; CO, 2.2%; O_2, 1.6%; and N_2, 81.2%. Find the mass of carbon in one kg of this flue gas.*

The data is the same as that considered in problem 3. The conversion of the volumetric analysis of CO_2 and CO into the mass has been done in problem 3 and the mass of CO_2 and CO were found to be 0.216 kg and 0.022 kg respectively. The carbon is contained only in CO_2 and CO. Equations for reactions are :

(i) $C + CO_2 = CO_2$ (ii) $2C + O_2 = 2CO$

$12 + 32 = 44$ $24 + 32 = 56$

Mass of carbon in 0.216 kg of CO_2 = $12/44 \times 0.216 = 0.059$ kg

Mass of carbon in 0·022 kg of CO = 24/56 × 0·022 = 0·00943 kg

∴ Total mass of carbon = 0·059 + 0·00943 = 0·06843 kg/kg of flue gas.

7.10 Mass of Flue Gases produced per Unit Mass of Fuel

Problem-6 : *The percentage composition of a certain fuel on mass basis is : C, 87·1; H_2, 4·4; O_2, 1·2; and ash, 7·3. The volumetric composition of dry flue gas is : CO_2, 15%; CO, 2·2%; O_2, 1·6%; and N_2, 81·2%. Find the mass of flue gases produced per kg of fuel burned.*

It may be noted that there is no loss of carbon during the process of combustion. The mass of carbon in the products must be the same if the combustion is complete.

The volumetric composition of dry flue gas is same as considered in problem 5. The total mass of carbon in 1 kg of the flue gas is found to be 0·06843 kg. The mass of carbon in 1 kg of fuel in this problem is 0·871 kg (87·1% as per composition of the fuel.)

Since, 0·06843 kg of carbon is contained in 1 kg of flue gas, 0·871 kg of carbon will be contained in $\frac{0{\cdot}871}{0{\cdot}06843} = 12{\cdot}8$ kg of flue gas.

Thus, mass of flue gas produced per kg of fuel burnt is **12·8 kg.**

7.11 Mass of Actual and Excess Air Supplied per Unit Mass of Fuel

The procedure of estimation of actual air and excess air supplied per kg of fuel when fuel and flue gas analysis is available, is outlined in the following problem.

Problem-7 : *The following is a percentage analysis by volume of dry flue gas :*

$CO_2 = 11{\cdot}9$; $CO = 0{\cdot}1$; $O_2 = 6{\cdot}2$; and $N_2 = 81{\cdot}8$. Calculate the actual air drawn through the furnace per kg of coal fired and the excess air supplied per kg of coal fired assuming that the coal contains 86·8% of carbon.

Relative composition on mass basis of the flue gas

CO_2, 11·9 x 44; CO, 0·1 × 28; O_2, 6·2 × 32; and N_2, 81·8 × 28.

∴ 11·9 × 44 kg CO_2 requires $11{\cdot}9 \times 44 \times \frac{12}{44}$ kg carbon for formation,

and 0·1 × 28 kg CO requires $0{\cdot}1 \times 28 \times \frac{12}{28}$ kg carbon for formation.

Total amount of carbon in flue gases = 11·9 × 12 + 0·1 × 12 = 144 kg.

∴ Ratio of relative mass of nitrogen to carbon in flue gases $= \frac{81{\cdot}8 \times 28}{144}$

But, as each kg of coal contains 0·868 kg of carbon,

Mass of nitrogen per kg of coal $= \frac{81{\cdot}8 \times 28}{144} \times 0{\cdot}868 = 13{\cdot}775$ kg.

Air contains 77% nitrogen by weight.

∴ Actual air required (supplied) per kg of coal fired $= 13{\cdot}775 \times \frac{100}{77} =$ **17·9 kg.**

Alternative method of determining actual air supplied (approximate) :

Actual air required (supplied) per kg of coal $= \frac{N \times C}{33(C_1 + C_2)}$

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 fuel on mass basis.

Actual air supplied per kg of coal fired $= \dfrac{81{\cdot}8 \times 86{\cdot}8}{33\,(11{\cdot}9 + 0{\cdot}1)} = 17{\cdot}9$ kg (same as before)

Excess air :

Oxygen in the flue gases is 6·2% by volume. This oxygen is the oxygen of the excess air.

The amount of nitrogen associated with this oxygen $= 6{\cdot}2 \times \dfrac{79}{21} = 23{\cdot}32$

and the ratio of the excess air to the total air supplied $= \dfrac{23{\cdot}32}{81{\cdot}8}$

$\therefore$ Mass of excess air supplied per kg of coal fired $= \dfrac{23{\cdot}32}{81{\cdot}8} \times 17{\cdot}9 = \mathbf{5{\cdot}1\ kg.}$

Alternative method of determining excess air (approximate) :

Mass of excess air supplied per kg of coal $= \dfrac{79 \times O \times C}{21 \times 33\,(C_1 + C_2)}$

where C_1, C_2 and O are percentages of carbon dioxide, carbon monoxide and oxygen by volume in flue gases, and C is the percentage of carbon in fuel on mass basis.

$\therefore$ Mass of excess air supplied per kg of coal fired.

$$= \frac{79 \times 6{\cdot}2 \times 86{\cdot}8}{21 \times 33\,(11{\cdot}9 + 0{\cdot}1)} = \mathbf{5{\cdot}1\ kg}\ \text{(same as before)}$$

Problem-8 : *During a boiler trial, the dry flue gas analysis by volume was reported as : CO_2, 13%; CO, 0·3%; O_2, 6%; and N_2, 80·7%. The coal analysis on mass basis was reported as : C, 62·4%; H_2 4·2%; O_2, 4·5%; moisture, 15%; and ash, 13·9%.*

Calculate : (i) the theoretical air or minimum air required to burn 1 kg of coal (ii) the mass of air actually supplied per kg of coal, and (iii) the amount of excess air supplied per kg of coal burnt. (Air contains 23% oxygen on mass basis).

(i) Oxygen required to burn carbon and hydrogen may first be found.

The chemical reactions are :

(i)	$C + O_2 = CO_2$		(ii)	$2H_2 + O_2 = 2H_2O$
	$12 + 16 \times 2 = 12 + 16 \times 2$			$2 \times 1 \times 2 + 16 \times 2 = 2(1 \times 2 + 16)$
i.e.	$12 + 32 = 44$		i.e.	$4 + 32 = 36$
	or 1 kg + 2·67 kg = 3·67 kg			or 1 kg + 8 kg = 9 kg

The two equations show that 1 kg of carbon requires 2·67 kg of oxygen and 1 kg of hydrogen requires 8 kg of oxygen for complete combustion.

$\therefore$ O_2 required to burn 0·624 kg of carbon $= 0{\cdot}624 \times 2{\cdot}67 = 1{\cdot}66$ kg, and

O_2 required to burn 0·042 kg of hydrogen $= 0{\cdot}042 \times 8 = 0{\cdot}336$ kg.

$\therefore$ Total O_2 required $= 1{\cdot}66 + 0{\cdot}336 = 1{\cdot}996$ kg/kg of coal.

Oxygen already present in coal is 0·045 kg per kg of coal. This must be subtracted from the total oxygen required.

∴ Net O_2 required = 1·996 − 0·045 = 1·951 kg per kg of coal.

As oxygen present in 100 kg of air is 23 kg,

Theoretical or minimum air required $= 1{\cdot}951 \times \frac{100}{23}$ = **8·5 kg per kg of coal.**

Alternatively, using eqn. (7.1),

$$\text{Minimum air required} = \frac{100}{23}\,[2{\cdot}67C + 8H + S - O]$$

$$= \frac{100}{23}\,[(2{\cdot}67 \times 0{\cdot}0624) + (8 \times 0{\cdot}042) - 0{\cdot}045]$$

= 8·5 kg per kg of coal burnt (same as before)

(ii) Before the actual air or total air supplied is calculated, the dry flue gas composition by volume has to be converted to analysis on mass basis as shown in the table below.

Dry flue gas constituents	Volume in one m^3 of flue gas (a)	Molecular weight (b)	Proportional mass in kg (c) = (a) × (b)	Actual mass in kg per kg of flue gas $(d) = \frac{(c)}{\Sigma(c)}$	Mass of carbon in kg per kg of flue gas (e)
CO_2	0·13	44	5·72	$\frac{5{\cdot}72}{30{\cdot}314} = 0{\cdot}1887$	$0{\cdot}1887 \times \frac{12}{44} = 0{\cdot}05146$
CO	0·003	28	0·084	$\frac{0{\cdot}084}{30{\cdot}314} = 0{\cdot}0027$	$0{\cdot}0027 \times \frac{12}{28} = 0{\cdot}00118$
C_2	0·06	32	1·92	$\frac{1{\cdot}92}{30{\cdot}314} = 0{\cdot}0634$	
N_2	0·807	28	22·59	$\frac{22{\cdot}59}{30{\cdot}314} = 0{\cdot}7452$	
Total	= 1·000	. –	= 30·314	= 1·0000	= 0·05264

$$\text{Mass of flue gas produced} = \frac{\text{Mass of carbon in 1 kg of coal}}{\text{Mass of carbon in 1 kg of flue gas}}$$

$$= \frac{0{\cdot}624}{0{\cdot}05264} = 11{\cdot}85 \text{ kg/kg of coal.}$$

Mass of N_2 in the flue gas per kg of coal = 0·7452 × 11·85 = 8·85 kg

As nitrogen present in 100 kg of air is 77 kg,

Mass of air actually supplied $= 8{\cdot}85 \times \frac{100}{77} =$ **11·5 kg/kg of coal.**

(iii) Amount of excess air supplied per kg of coal burnt

= actual air supplied − minimum air required = 11·5 − 8·5 = 3 kg

Problem-9 : *The following is the percentage composition of a sample of coal on mass basis :*

C, 82; H_2, 6; O_2, 9; and ash, 3.

Find : (a) the minimum mass of air required for complete combustion of 1 kg of *coal, and (b) the volumetric analysis of the products of combustion, if 10% excess air is supplied. Assume that air contains 23% oxygen on mass basis.*

(a) Oxygen required to burn carbon and hydrogen may first be found.

The chemical reactions are :

(i) $C + O_2 = CO_2$ (ii) $2H + O_2 = 2H_2O$

$12 + 32 = 44$ $4 + 32 = 36$

or 1 kg + 2·67 kg = 3·67 kg or 1 kg + 8 kg = 9 kg

The two equations show that 1 kg of carbon requires 2·67 kg of oxygen and 1 kg of hydrogen requires 8 kg of oxygen for complete combustion.

∴ O_2 required to burn 0·82 kg of C = 0·82 × 2·67 kg, and

O_2 required to burn 0·06 kg of H_2 = 0·06 × 8 kg

∴ Total oxygen required = (0·82 × 2·67) + (0·06 × 8) = 2·67 kg/kg of coal.

The oxygen already present in 1 kg of coal must be subtracted from the total oxygen required.

∴ Net theoretical O_2 required to burn 1 kg of coal = 2·67 − 0·09 = 2·58 kg per kg of coal.

As oxygen present in 100 kg of air is 23 kg,

Theoretical or minimum air required to burn 1 kg of coal completely

$$= \frac{100}{23} \times 2{\cdot}58 = \mathbf{11{\cdot}21\ kg/kg\ of\ coal.}$$

(b) As 10% excess of air is supplied,

Excess air supplied per kg of coal = 0·1 × 11·21 = **1·121 kg**

∴ Actual air supplied, per kg of coal

= Minimum air required + Excess air supplied = 11·21 + 1·121 = 12·331 kg.

The products of combustion (wet) contain carbon dioxide, water vapour, oxygen and nitrogen. We must now determine the products of combustion as follows :

— Mass of CO_2 formed per kg of coal = 3·67 C = 3·67 x 0·82 = 3·01 kg.

— Mass of H_2O formed per kg of coal = $9H_2$ = 9 × 0.06 = 0·54 kg.

— Since, excess air has been supplied, excess oxygen will appear in the products of combustion. Hence, mass of excess O_2 per kg of coal

$$= \frac{23}{100} \times \text{excess air supplied} = 0{\cdot}23 \times 1{\cdot}121 = 0{\cdot}258\ \text{kg.}$$

— Nitrogen will also appear in the products of combustion as it does not take part in the combustion process and passes directly into the flue gases.

Hence, the mass of nitrogen per kg of coal

$$= \frac{77}{100} \times \text{actual air supplied} = 0{\cdot}77 \times 12{\cdot}331 = 9{\cdot}5\ \text{kg.}$$

Hence, the products of combustion per kg of coal, will consists of the following :

CO_2 = 3·01 kg; Water vapour (H_2O) = 0·54 kg; O_2 = 0·258 kg; N_2 = 9·5 kg.

If the mass of each constituent is divided by its own molecular weight; the proportional volume of each constituent is obtained. The proportional volume of each constituent so obtained is then divided by the sum of proportionate volumes of all the constituents to obtain the volumetric analysis.

For obtaining percentage volumetric analysis of products of combustion, table is constructed as shown below :

Constituent	Mass in kg (a)	Molecular weight (b)	Proportional volume $(c) = (a) + (b)$	% Volumetic analysis $(d) = \frac{(c)}{\Sigma(c)} \times 100$
CO_2	3·01	44	0·0684	$\frac{0.0684}{0.4460} \times 100 = 15.32$
H_2O	0·54	18	0·0300	$\frac{0.0300}{0.4460} \times 100 = 6.73$
O_2	0·258	32	0·0081	$\frac{0.0081}{0.4460} \times 100 = 1.82$
N_2	9·5	28	0·3395	$\frac{0.3395}{0.4460} \times 100 = 76.13$
Total	–	–	= 0·4460	= 100·00

Volumetric analysis of flue gases is,

CO_2, 15·32%; H_2O, 6·73%; O_2, 1·82%; and N_2, 76·13%.

Problem-10 : *The following is the percentage composition of coal on mass basis :*

C, 90; H_2, 3·3; O_2, 3; S, 0·9; and remainder ash.

Calculate : (a) the theoretical air required to burn 1 kg of coal completely, and (b) the percentage composition of dry flue gases on mass and volume basis, if 50% excess air is supplied. (Assume that air contains 23% O_2 on mass basis).

(a) The chemical reactions are :

(i) $C + O_2 = CO_2$ (ii) $2H_2 + O_2 = 2H_2O$ (iii) $S + O_2 = SO_2$

$12 + 32 = 44$ $4 + 32 = 36$ $32 + 32 = 64$

$1 + 2.67 = 3.67$ or $1 + 8 = 9$ or $1 + 1 = 2$

Equation (i) shows that 1 kg of carbon requires 2·67 kg of oxygen and produces 3·67 kg of CO_2.

∴ 0·9 kg of carbon requires 0·9 × 2·67 = 2·4 kg of oxygen, and produces

0·9 × 3·67 = 3·303 kg of CO_2.

Equation (ii) shows that 1 kg of hydrogen requires 8 kg of oxygen and produces 9 kg of water vapour (H_2O).

∴ 0·033 kg of hydrogen requires 0·033 × 8 = 0·264 kg of oxygen and produces

0·033 × 9 = 0·297 kg of H_2O.

Equation (iii) shows that 1 kg of sulphur requires 1 kg of oxygen and produces 2 kg of sulphur dioxide.

∴ 0·009 kg of sulphur requires 0·009 kg of oxygen and produces 0·009 × 2 = 0·018 kg of sulphur dioxide.

Hence, total oxygen required per kg of coal

$= 2{\cdot}4 + 0{\cdot}264 + 0{\cdot}009 = 2{\cdot}673$ kg.

But, 0·03 kg of oxygen is already present in 1 kg of coal.

∴ Net oxygen required for complete combustion of 1 kg of coal

$= 2{\cdot}673 - 0{\cdot}03 = 2{\cdot}643$ kg/kg of coal.

As 23 kg of oxygen is present in 100 kg of air, theoretical or minimum air required

$$= 2{\cdot}643 \times \frac{100}{23} = \mathbf{11{\cdot}5\ kg/kg\ of\ coal.}$$

i.e. minimum mass of air necessary for complete combustion of 1 kg of coal is **11·5 kg.**

(b) As 50% excess of air is supplied, air actually supplied

$= 11{\cdot}5 \times 1{\cdot}5 = 17{\cdot}25$ kg/kg of coal.

∴ Excess air supplied $= 17{\cdot}25 - 11{\cdot}5 = 5{\cdot}75$ kg/kg of coal.

Since, excess air has been supplied, excess oxygen will appear in the products of combustion.

Hence, oxygen present in the products of combustion will be given by

$0{\cdot}23 \times$ excess air supplied $= 0{\cdot}23 \times 5{\cdot}75$ kg per kg of coal.

Nitrogen will also appear in the products of combustion.

Hence, nitrogen in the products of combustion will be given by

$0{\cdot}77 \times$ actual air supplied $= 0{\cdot}77 \times 17{\cdot}25$ kg/kg of coal.

The products of combustion, per kg of coal, will consists of

$CO_2 = C \times 3{\cdot}67 = 0{\cdot}9 \times 3{\cdot}67 = 3{\cdot}303$ kg

$H_2O = H_2 \times 9 = 0{\cdot}033 \times 9 = 0{\cdot}297$ kg

$SO_2 = S \times 2 = 0{\cdot}009 \times 2 = 0{\cdot}018$ kg

Oxygen in excess air $= 0{\cdot}23 \times 5{\cdot}75 = 1{\cdot}322$ kg

Nitrogen in air actually supplied $= 0{\cdot}77 \times 17{\cdot}25 = 13{\cdot}282$ kg

As water vapour (H_2O) is wet gas, *dry flue gases* contained in the products of combustion are :

$CO_2 = 3{\cdot}303$ kg; $SO_2 = 0{\cdot}018$ kg; $O_2 = 1{\cdot}322$ kg; and $N_2 = 13{\cdot}282$ kg,

∴ Total mass of dry flue gases $= 3{\cdot}303 + 0{\cdot}018 + 1{\cdot}322 + 13{\cdot}282$

$= 17{\cdot}925$ kg/kg of coal.

Hence, percentage composition of *dry flue gases* on mass basis is,

$$CO_2 = \frac{3{\cdot}303}{17{\cdot}925} \times 100 = \mathbf{18{\cdot}42\%};\ SO_2 = \frac{0{\cdot}018}{17{\cdot}925} \times 100 = \mathbf{0{\cdot}1\%};$$

$$O_2 = \frac{1{\cdot}322}{17{\cdot}925} \times 100 = \mathbf{7{\cdot}38\%};\ N_2 = \frac{13{\cdot}282}{17{\cdot}925} \times 100 = \mathbf{74{\cdot}1\%}$$

Using tabular method (shown below) of coverting percentage analysis on mass basis to volumatric analysis of *dry flue gases,* we have,

Constituent	% composition on mass basis (a)	Molecular weight (b)	Relative volume $(c) = \frac{(a)}{(b)}$	% composition by volume $(d) = \frac{(c)}{\Sigma (c)} \times 100$
CO_2	18·42	44	$\frac{18{\cdot}42}{44} = 0{\cdot}419$	$\frac{0{\cdot}419}{3{\cdot}3} \times 100 = 12{\cdot}70$
SO_2	0·1	64	$\frac{0{\cdot}1}{64} = 0{\cdot}0016$	$\frac{0{\cdot}0016}{3{\cdot}3} = 100 = 0{\cdot}05$
O_2	7·38	32	$\frac{7{\cdot}38}{32} = 0{\cdot}231$	$\frac{0{\cdot}231}{3{\cdot}3} \times 100 = 7{\cdot}00$
N_2	74·1	28	$\frac{74{\cdot}1}{28} = 2{\cdot}649$	$\frac{2{\cdot}649}{3{\cdot}3} \times 100 = 80{\cdot}25$
Total	= 100·00	–	= 3·3	= 100·00

Volumetric analysis of dry flue gases is,

$$CO_2 = 12{\cdot}7\%;\ SO_2 = 0{\cdot}05\%;\ O_2 = 7\%;\ \text{and}\ N_2 = 80{\cdot}25\%.$$

Problem-11 : *A sample of oil fuel was found to be composed of 80% carbon and 20% hydrogen. If 6.25% of the carbon contained in one kg of fuel is burned to carbon monoxide due to incomplete combustion and remainder to carbon dioxide and if 40 per cent excess air was supplied per kg of fuel burnt, determine : (a) the theoretical air required for complete combustion of 1 kg of oil fuel, (b) the excess oxygen in the flue gases per kg of oil fuel burnt, (c) the percentage analysis on mass basis of the total products of combustion, and (d) the percentage volumetric analysis of the total products of combustion.*

(a) The chemical reactions are :

$$\begin{array}{llll} \text{(i)} & C + O_2 = CO_2 & \text{(ii)}\ 2C + O_2 = CO & \text{(iii)}\ 2H_2 + O_2 = H_2O \\ & 12 + 32 = 44 & 24 + 32 = 56 & 4 + 32 = 36 \\ \text{or} & 1 + 2{\cdot}67 = 3{\cdot}67 & \text{or}\ 1 + 1{\cdot}33 = 2{\cdot}33 & \text{or}\ 1 + 8 = 9 \end{array}$$

From eqns. (i) and (iii),

Oxygen required to burn completely carbon and hydrogen

$$= (0{\cdot}8 \times 2{\cdot}67) + (0{\cdot}2 \times 8) = 3{\cdot}725 \text{ kg per of oil fuel.}$$

$\therefore$ Theoretical air required for complete combustion of 1 kg oil fuel

$$= 3{\cdot}725 \times \frac{100}{23} = 16{\cdot}25 \text{ kg.}$$

(b) As 40% excess air is supplied,

Actual air supplied = 16·25 × 1·4 = 22·75 kg, of this air 77% is N_2 and 23% is O_2.

$\therefore$ N_2 in the total air supplied = 22·75 × 0·77 = 17·52 kg, and

O_2 in the total air supplied = 22·75 × 0·23 = 5·23 kg.

Per kg of oil fuel burnt, 0·8 × 0·0625 = 0·05 kg of C was burnt to CO,

0·8 (1 – 0·0625) = 0·75 kg of C was burnt to CO_2 and

0·2 kg of H_2 was burnt to H_2O.

$\therefore$ O_2 actually used to burn 1 kg of oil fuel

$$= (0{\cdot}05 \times 1{\cdot}33) + (0{\cdot}75 \times 2{\cdot}67) + (0{\cdot}2 \times 8) = 3{\cdot}67 \text{ kg.}$$

∴ Excess oxygen in the flue gases = O_2 in the total air supplied $-O_2$ actually used

$$= 5{\cdot}23 - 3{\cdot}67 = \mathbf{1{\cdot}56 \text{ kg per kg of oil fuel.}}$$

(c) The wet products of combustion are : CO_2, CO, H_2O, N_2 and excess O_2

Their masses can be calculated as under :

CO_2 = C × 3·67 = 0·75 × 3·67 = 2·753 kg

CO = C × 2·33 = 0·05 × 2·33 = 0·117 kg

H_2O = H_2 × 9 = 0·2 × 9 = 1·8 kg

Excess O_2 = — — = 1·56 kg

N_2 = — — = 17·52 kg

Total mass of wet products of combustion = 23·75 kg per kg of oil fuel

Hence, percentage analysis on mass basis of wet products of combustion is,

$$CO_2 = \frac{2{\cdot}753}{23{\cdot}75} \times 100 = 11{\cdot}37;\ CO = \frac{0{\cdot}117}{23{\cdot}75} \times 100 = 0{\cdot}49;\ H_2O = \frac{1{\cdot}8}{23{\cdot}75} \times 100 = \mathbf{7{\cdot}58};$$

$$\text{Excess } O_2 = \frac{1{\cdot}56}{23{\cdot}75} \times 100 = \mathbf{6{\cdot}56};\ N_2 = \frac{17{\cdot}52}{23{\cdot}75} \times 100 = \mathbf{73{\cdot}8}.$$

(d) The mass of each constituent when divided by its molecular weight, proportional volume of constituent is obtained. The proportional volume so obtained is then divided by the sum of proportionate volumes to obtain the analysis by volume (see table).

Constituent	Mass in kg (a)	Molecular weight (b)	Proportional volume (c) = (a) ÷ (b)	Percentage volumetric analysis $(d) = \frac{(c)}{\Sigma(c)} \times 100$
CO_2	2·753	44	0·06248	$\frac{0{\cdot}06248}{0{\cdot}841} \times 100 = 7{\cdot}426$
CO	0·117	28	0·00418	$\frac{0{\cdot}00418}{0{\cdot}841} \times 100 = 0{\cdot}497$
H_2O	1·8	18	0·1000	$\frac{0{\cdot}1}{0{\cdot}841} \times 100 = 11{\cdot}90$
O_2	1·56	32	0·04875	$\frac{0{\cdot}04875}{0{\cdot}841} \times 100 = 5{\cdot}79$
N_2	17·52	28	0·6256	$\frac{0{\cdot}6256}{0{\cdot}841} \times 100 = 74{\cdot}387$
Total	= 23·75	–	= 0·841	= 100·00

Volumetric analysis of the products of combustion is,

CO_2, 7·426%; CO, 0·497%; H_2O, 11·9%; O_2, 5·79%; N_2, 74·387%

7·12 Determination of Volumetric Analysis of Flue Gases

The Orsat apparatus is very convenient for analysing flue gases from the boiler or engine on the spot. In its simplest form the apparatus is arranged for the absorption of CO_2, CO and O_2. The nitrogen content of the gas is obtained by difference.

The schematic diagram of Orsat apparatus is shown in fig 7-1. The essential parts of this apparatus are :

.. A measuring burrette, called *eudiometer*, surrounded by a water filled jacket to maintain constant temperature during analysing operations.

.. Three *absorption double flasks* for the absorption of CO_2, O_2 and CO. For CO_2, the flask at the right (near eudiometer) contains 1 part of caustic soda (NaOH) or caustic potash (KOH) in 2 parts of water by mass. For O_2, the flask in the middle contains one part of pyrogallic acid in three parts of water, and 24 parts of either caustic soda or potash in 16 parts of water by mass. For CO, the flask at the left contains a solution of cuprous chloride in hydrochloric acid, which is prepared by dissolving copper oxide (CuO) in about twenty times its mass of strong hydrochloric acid and allowing it to stand in a corked flask containing copper wire until the solution is colourless. The reagent flask contains a number of lengths of glass tubes in order to increase the surface of the reagent exposed to the gases. At the entrance, each flask is provided with a cock. A three-way cock is provided for giving straight-through connection and also connection to the atmosphere.

.. An *aspirator bottle* connected to the lower end of the eudiometer is provided for drawing (taking in) exhaust or flue gases in the apparatus.

7.12.1 Procedure for operating the Orsat Apparatus : In order to obtain the sample of exhaust or flue gases, the reagents in the flasks are adjusted to some known level and cocks shut. The three-way cock is then opened to the atmosphere and the aspirator bottle raised until water is discharged to atmosphere. Three-way cock is then turned to the 'through' position and aspirator lowered, thus drawing in the flue gases from the boiler or engine through a suitable connection near the base of the chimney in the case of a boiler, and near the exhaust valve in the case of I.C. engine. The operation should be repeated several times, the aspirated gas being discharged to atmosphere in order to make sure that all air has been cleared from the connecting tubing. Having drawn in 100 c c. of the representative sample of gases, the three-way cock is closed.

The percentage of CO_2 is first obtained by opening the cock on the flask at the

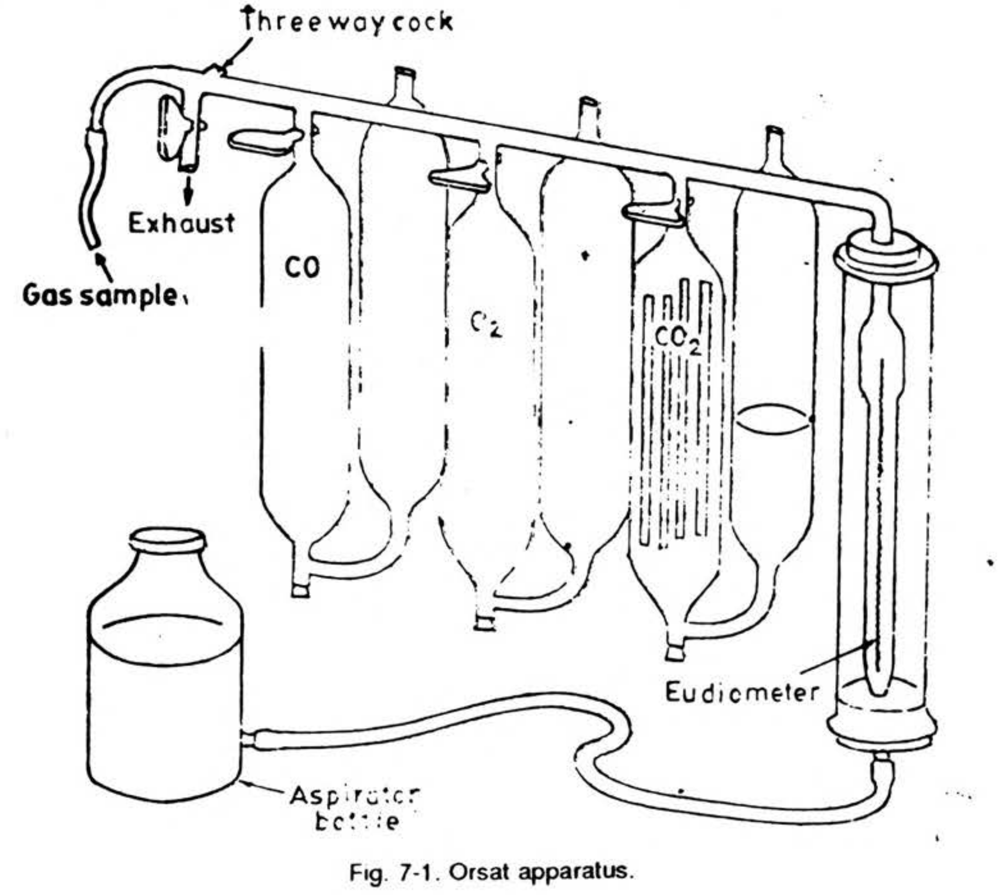


Fig. 7-1. Orsat apparatus.

right and forcing gases from the eudiometer into the flask by raising the aspirator bottle. When the gas is withdrawn by again lowering the aspirator, the cock on the flask is closed when the reagent reaches its original level. This is repeated several times, the reduction in volume as indicated by eudiometer being noted. When no further decrease is noted, all the CO_2 is absorbed and the total decrease in volume gives the percentage of CO_2 present in the flue gases.

Oxygen (O_2) is next absorbed in a similar manner in the flask, and finally carbon monoxide (CO) is absorbed in the flask at the left. After the absorption of CO_2, O_2 and CO, the gas remaining is assumed to be nitrogen.

When taking eudiometer reading, care should be taken to see that the water level is same in both the eudiometer and the aspirator bottle.

7.13 Heat Carried away by Flue Gases

With chimney draught in a boiler plant, it is necessary for the flue gases to leave the boiler at a considerably higher temperature above that of the steam produced. This high temperature as well as the excess air causes loss of heat amounting to about 25% of the total heat produced by the fuel. This loss is due to :

.. The heat carried away by the dry products of combustion, and

.. Heat carried away by the steam in the flue gases.

The formation of steam in the flue gases may be due to :

— Combustion of the free hydrogen in the fuel,

— Vaporisation of moisture in the fuel,

— Moisture contained in the air supplied for combustion, and

— Moisture contained in excess air supplied.

(i) *Heat carried away by dry flue gases or products of combustion :*

If m_g = mass of dry flue gases in kg per kg of fuel burnt,

k_p = mean specific heat of dry flue gases in kJ/kg K,

t_1 = temperature of flue gases leaving the boiler in °C, and

t_o = temperature of the boiler room in °C.

Then, heat carried away by dry flue gases produced per kg of fuel burnt

$$= m_g \times k_p\ (t_1 - t_o)\ \text{kJ} \qquad ..(7.2)$$

(ii) *Heat carried away by steam in flue gases :*

The enthalpy (total heat) of steam in flue gases will correspond to its partial pressure because it is now one of the constituents of a mixture of steam and other gases.

If the method of solution by partial pressures is applied to products of combustion of a fuel, it will be found that the partial pressure of the steam is very low (usually less than 0.1 bar) and the steam is in a highly superheated state.

The exact value of partial pressure of steam (p_s) can be determined as follows :

For 1 kg of flue gases, $10^5\ p_1 v_1 = RT_1$

where T_1 = absolute temperature of flue gases.

Knowing T_1 and taking pressure p_1 as atmospheric (1·01325 bar), and R as

287 J/kg K, volume v_1 can be calculated for 1 kg of flue gases.

Now, according to Dalton's Law, steam also occupies volume v_1 m^3 at its partial pressure p_s and temperature t_1°C. (For Dalton's law refer volume II).

If m_s is the mass of steam contained in one kg of flue gases,

then volume of 1 kg of superheated steam, $v_{sup} = \dfrac{v_1}{m_s}$ m^3.

Now, by using steam tables for superheated steam and solving by trial, the partial pressure of steam p_s can be found. Knowing p_s and t_1 of steam, its enthalpy (total heat) can be evaluated.

If m_s = mass of steam formed in kg per kg of fuel burnt,

H_{sup} = enthalpy (total heat) of superheated steam in kJ/kg at the temperature of flue gases t_1°C and its partial pressure p_s, and

h_o = enthalpy (sensible heat) of water in kJ/kg at the boiler room temperature, t_o°C,

Then, the heat carried away by the steam per kg of fuel burnt = $m_s\ (H_{sup} - h_o)$ kJ .. (7.3)

Heat carried away by water vapour (steam) can be estimated with sufficient accuracy by assuming partial pressure of steam (p_s) to be atmospheric (see illustrative problem-14).

∴ Total heat carried away by products of combustion per kg of fuel burnt

= heat carried away by dry flue gases + heat carried away by steam in flue gases

= $m_g \times k_p\ (t_1 - t_o) + m_s\ (H_{sup} - h_o)$ kJ per kg of fuel ..(7.4)

Problem-12 : *The percentage composition of a sample of coal on mass basis is as follows : C, 85; H, 12; and* O_2, *3.*

Determine the theoretical or minimum air required for complete combustion of 1 kg of coal. It the total air supplied is 1·5 times the minimum air required, calculate the mass of dry products of combustion per kg of coal burned and the heat carried away by the dry products of combustion leaving the boiler per kg of coal burned, if the average specific heat of dry flue gases is 1·005 kJ/kg K and the temperature of flue gases is 375°C and the boiler house temperature is 25°C.

Using eqn.. (7.1), minimum air required per kg of coal burned

$$= \frac{100}{23}[2{\cdot}67C + 8H + S - O] = \frac{100}{23}\ [(2{\cdot}67 \times 0{\cdot}85) + (8 \times 0{\cdot}12) - 0{\cdot}03] = \mathbf{13{\cdot}91\ kg}$$

∴ Actual air supplied per kg of coal = 13·91 × 1·5 = 20·865 kg and

Excess air supplied per kg of coal = 13·91 × 0·5 = 6·955 kg.

The dry products of combustion per kg of coal will consist of ;

CO_2 produced = C × 3·67 = 0·85 × 3·67 = 3·2 kg

Excess O_2 = Excess air × 0·23 = 6·955 × 0·23 = 1·6 kg

N_2 = Actual air × 0·77 = 20·865 × 0·77 = 16·07 kg

Total mass of dry products of combustion = 20·87 kg per kg of coal.

Using eqn. (7.2), heat carried away by dry flue gases per kg of coal burned

= $m_g \times k_p \times (t_1 - t_o)$ = 20·87 × 1·005 × (375 − 25) = **7,341 kJ per kg of coal.**

Problem-13 : *The fuel supplied to a boiler has the following composition on mass basis :*

85%, carbon; 13%, hydrogen; 2%, oxygen.

The air supplied is 60% in excess of that theoretically required for complete combustion. Estimate the volumetric composition of the dry flue gases. If the temperature of the boiler house is 15°C and the temperature of the flue gases is 330°C, estimate the heat carried away by the flue gases leaving the boiler plant. Assume k_p of dry flue gases to be 1·005 kJ/kg K and enthalpy of water vapour in the flue gases to be 2,680 kJ/kg. (Assume that air contains 23% O_2 on mass basis).

Using eqn. (7.1), theoretical or minimum air required per kg of fuel

$$= \frac{100}{23}(2{\cdot}67C + 8H + S - O) = \frac{100}{23}[(2{\cdot}67 \times 0{\cdot}85) + (8 \times 0{\cdot}13) - 0{\cdot}02] = 14{\cdot}3 \text{ kg}$$

As 60% excess air is supplied,

Excess air supplied per kg of fuel = 0·6 × 14·3 = 8·58 kg, and

actual air supplied per kg of fuel = 1·6 × 14·3 = 22·88 kg.

Dry products of combustion per kg of fuel will consist of :

(i) Carbon dioxide = 3·67C = 3·67 × 0·85 = 3·12 kg.

(ii) Excess O_2 = 0·23 × excess air supplied per kg of fuel = 0·23 × 8·58 = 1·97 kg.

(iii) Nitrogen = 0·77 × actual air supplied per kg of fuel = 0·77 × 22·88 = 17·61 kg.

∴ Total mass of dry flue gases = mass of CO_2 + mass of excess O_2 + mass of N_2

= 3·12 + 1·97 + 17·61 = 22·7 kg per kg of fuel.

Hence, percentage composition of dry flue gases on mass basis is,

$$CO_2 = \frac{3{\cdot}12}{22{\cdot}7} \times 100 = 13{\cdot}75\%$$

$$O_2 = \frac{1{\cdot}97}{22{\cdot}7} \times 100 = 8{\cdot}67\%$$

$$N_2 = \frac{17{\cdot}61}{22{\cdot}7} \times 100 = \underline{77{\cdot}58\%}$$

$$\text{Total} = 100{\cdot}00\%$$

Using tabular method (shown below) of converting analysis on mass basis into volumetric analysis *of dry flue gases,* we have

Constituent	% Composition on mass basis (a)	Molecular weight (b)	Relative volume $(c) = \frac{(a)}{(b)}$	Percentage composition by vol. $(d) = \frac{(c)}{\Sigma(c)}$
CO_2	13·75	44	0·312	$\frac{0{\cdot}312}{3{\cdot}351} = 0{\cdot}093 \times 100 \doteq 9{\cdot}3$
O_2	8·67	32	0·271	$\frac{0{\cdot}271}{3{\cdot}351} = 0{\cdot}081 \times 100 = 8{\cdot}1$
N_2	77·58	28	2·768	$\frac{2{\cdot}768}{3{\cdot}351} = 0{\cdot}826 \times 100 = 82{\cdot}6$
Total	= 100·00	—	= 3·351	= 100·00

Volumetric analysis of the dry flue gases is,

CO_2 = 9·3%; O_2 = 8·1%; and N_2 = 82·6%.

Total mass of dry flue gases per kg of fuel, m_g = 22·7 kg.

Using eqn. (7.2), Heat carried away by dry flue gases per kg of fuel

= $m_g \times k_p \times (t_1 - t_o)$ = 22·7 × 1·005 × (330 − 15) = 7,185 kJ/kg of fuel.

Mass of water vapour (H_2O) formed in the combustion process

= 9H = 9 × 0·13 = 1·17 kg/kg of fuel

∴ Heat carried away by water vapour = 1·17 × 2,680 = 3,135·6 kJ/kg of fuel

∴ Total heat carried away by the flue gases leaving the boiler

= 7,185 + 3,135·6 = **10,320·6 kJ per kg of fuel.**

Problem-14 : *The composition of a sample of coal shows 0·872 kg of carbon and 0·042 kg of hydrogen per kg of coal. If the air supplied is 1·52 time the minimum air, calculate the heat carried away by the products of combustion per kg of coal, if they escape to the chimney at a temperature of 350°C. Take specific heat of* CO_2, O_2, N_2, *and steam as 0·917, 1·047, 1·022, and 1·863 kJ/kg K respectively. Assume steam to be at atmospheric pressure. The boiler room temperature is 15°C. Air contains 23%* O_2 *on mass basis.*

To find minimum mass of air required per kg of coal :

O_2 required to burn C = C × 2·67 = 0·872 × 2·67 = 2·33 kg

O_2 required to burn H_2 = H_2 × 8 = 0·042 × 8 = 0·34 kg

Total O_2 required for complete combustion of 1 kg of coal = 2·67 kg

∴ Mass of minimum air required = 2·67 × 100/23 = 11·61 kg

As the total air supplied is 1·52 times the minimum required,

Total air supplied per kg of coal = 11·61 × 1·52 = 17·65 kg

∴ Excess air supplied = 17·65 − 11·61 = 6·04 kg

To find the mass of products of combustion per kg of coal :

CO_2 = 0·872 × 3·67 = 3·2 kg; N_2 = 17·65 × 0·77 = 13·58 kg.

O_2 from excess air = 6·04 × 0·23 = 1·39 kg; and steam = 0·042 × 9 = 0·38 kg.

Heat carried away by dry products of combustion per degree centigrade temperature difference = 3·2 × 0·971 + 1·39 × 1·047 + 13·58 × 1·022 = 18·267 kJ.

Total heat carried away by *dry products* of combustion

= 18·267 (350 − 15) = 6,119·5 kJ/kg of coal.

Using eqn. (7.3)

Heat carried away by *steam* at atmospheric pressure and 350°C

= $m_s\ (H_{sup} - h_o)$

= 0·38 [{2,676·1 + 1·863 (350 − 100)} − 15 × 4·187] = 1,170 kJ/kg of coal

∴ Total heat carried away by products of combustion per kg of coal

= 6,119·5 + 1,170 = **7,289·5 kJ/kg of coal.**

7.14 Volume of Minimum Air required for Complete Combustion of Gaseous Fuel

Problem-15 : *Calculate the volume of minimum air required to burn completely 1 m^3 of a gaseous fuel having the following composition by volume : H_2, 14; CH_4, 1%; CO, 24%; CO_2, 5%; O_2, 1%; and N_2, 55%. If 40% excess air is supplied, determine the actual volume of air supplied per m^3 of gas. (Assume that air contains 21% O_2 by volume).*

Reactions of combustion can be written as :

(i) $2H_2 + O_2 = 2H_2O$

2 vol. + 1 vol. = 2 vol.

1 m^3 + 1/2 m^3 = 1 m^3

(ii) $CH_4 + 2O_2 = CO_2 + 2H_2O$

1 vol. + 2 vol. = 1 vol. + 2 vol.

1 m^3 + 2 m^3 = 1 m^3 + 2 m^3

(iii) $2CO + O_2 = 2CO_2$

2 vol. + 1 vol. = 2 vol.

1 m^3 + 1/2 m^3 = 1 m^3

Equation (i) shows that 1 m^3 of hydrogen requires 1/2 m^3 of oxygen for complete combustion and the product of combustion will occupy 1 m^3.

Equation (ii) shows that 1 m^3 of CH_4 (marsh gas) requires 2 m^3 of oxygen for complete combustion.

Equation (iii) shows that 1 m^3 of carbon monoxide (*CO*) requires ½ m^3 of oxygen for complete combustion.

Volume of O_2 required for complete combustion of H_2

$= H_2 \times \frac{1}{2} = 0{\cdot}14 \times \frac{1}{2} = 0{\cdot}07$ m^3 from eqn. (i)

Volume of O_2 required for complete combustion of CH_4

$= CH_4 \times 2 = 0.01 \times 2 = 0{\cdot}02$ m^3 from eqn. (ii)

Volume of O_2 required for complete combustion of *CO*

$= CO \times \frac{1}{2} = 0.24 \times \frac{1}{2} = 0.12$ m^2 from eqn. (iii)

Total volume of O_2 required $= 0{\cdot}07 + 0{\cdot}02 + 0{\cdot}12 = 0{\cdot}21$ m^3.

As O_2 already present in gaseous fuel is 0·01 m^3,

Net O_2 required $= 0{\cdot}21 - 0{\cdot}01 = 0{\cdot}2$ m^3 per m^3 of gas

As air contains 21% oxygen by volume, minimum volume of air required

$$= 0{\cdot}2 \times \frac{100}{21} = \mathbf{0{\cdot}953\ m^3\ per\ m^3\ of\ gas.}$$

As 40% excess air is supplied, actual volume of air supplied

$= 0{\cdot}953 \times 1{\cdot}4 =$ **1·3342 m^3 per m^3 of gas.**

Problem-16 : *The following is the percentage composition of a gaseous fuel by volume : H_2, 46; C_2H_4, 2·5; CH_4, 37·5; N_2, 4·5; CO, 7·5; and moisture, 2.*

Determine the volume of minimum air required and the percentage volumetric composition of wet products of combustion. (Assume that air contains 21% O_2 by volume).

Reactions of combustion can be written as :

(i) $2H_2 + O_2 = 2H_2O$
$2\text{ vol.} + 1\text{ vol.} = 2\text{ vol.}$
$1\text{ m}^3 + 1/2\text{ m}^3 = 1\text{ m}^3$

(ii) $C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$
$1\text{ vol.} + 3\text{ vol.} = 2\text{ vol.} + 2\text{ vol.}$
$1\text{ m}^3 + 3\text{ m}^3 = 2\text{ m}^3 + 2\text{ m}^3$

(iii) $CH_4 + 2O_2 = CO_2 + 2H_2O$
$1\text{ vol.} + 2\text{ vol.} = 1\text{ vol.} + 2\text{ vol.}$
$1\text{ m}^3 + 2\text{ m}^3 = 1\text{ m}^3 + 2\text{ m}^3$

(iv) $2CO + O_2 = 2CO_2$
$2\text{ vol.} + 1\text{ vol.} = 2\text{ vol.}$
$1\text{ m}^3 + 1/2\text{ m}^3 = 1\text{ m}^3$

Volume of O_2 required to burn $H_2 = H_2 \times \frac{1}{2} = 0{\cdot}46 \times \frac{1}{2} = 0{\cdot}23\text{ m}^3$ from eqn. (i)
Volume of O_2 required to burn $C_2H_4 = C_2H_4 \times 3 = 0{\cdot}025 \times 3 = 0{\cdot}075\text{ m}^3$ from eqn. (ii)
Volume of O_2 required to burn $CH_4 = CH_4 \times 2 = 0{\cdot}375 \times 2 = 0{\cdot}75\text{ m}^3$ from eqn. (iii)
Volume of O_2 required to burn $CO = CO \times \frac{1}{2} = 0{\cdot}075 \times \frac{1}{2} = 0{\cdot}0375\text{ m}^3$ from eqn. (iv)
Total volume of O_2 required per m^3 of gas $= 1{\cdot}0925\text{ m}^3$

$\therefore$ Minimum volume of air required $= 1{\cdot}0925 \times \frac{100}{21} =$ **$5{\cdot}202\text{ m}^3$ per m^3 of gas.**

The *dry* products of combustion will consist of :

— CO_2 by combustion of C_2H_4, CH_4 and CO, and

— N_2 from fuel gas, and from air supplied.

Volume of *dry* product of combustion (CO_2) :
CO_2 from $C_2H_4 = C_2H_4 \times 2 = 0{\cdot}025 \times 2 = 0{\cdot}05\text{ m}^3$ from eqn. (ii)
CO_2 from $CH_4 = CH_4 \times 1 = 0{\cdot}375 \times 1 = 0{\cdot}375\text{ m}^3$ from eqn. (iii)
CO_2 from $CO = CO \times 1 = 0{\cdot}075 \times 1 = 0{\cdot}075\text{ m}^3$. from eqn. (iv)
Total volume of CO_2 formed $= 0{\cdot}5\text{ m}^3$

Volume of *dry* product of combustion (N_2) :
N_2 in gaseous fuel (by given composition) $= 0{\cdot}045\text{ m}^3$
N_2 in air supplied $= 5{\cdot}202 \times \frac{79}{100} = 4{\cdot}11\text{ m}^3$
Total volume of $N_2 = 4{\cdot}155\text{ m}^3$

Volume of *wet* products of combustion (H_2O) :
Steam (H_2O) from $H_2 = H_2 \times 1 = 0{\cdot}46 \times 1 = 0{\cdot}46\text{ m}^3$ from eqn. (i)
Steam (H_2O) from $C_2H_4 = C_2H_4 \times 2 = 0{\cdot}025 \times 2 = 0{\cdot}05\text{ m}^3$ from eqn. (ii)
Steam (H_2O) from $CH_4 = CH_4 \times 2 = 0{\cdot}375 \times 2 = 0{\cdot}75\text{ m}^3$ from eqn. (iii)
Steam (H_2O) from moisture
in the gaseous fuel (by given composition) $= 0{\cdot}02\text{ m}^3$
Total volume of steam formed $= 1{\cdot}28\text{ m}^3$

Thus, the volume of *wet* products of combustion is,
$CO_2 = 0{\cdot}5\text{ m}^3$; $N_2 = 4{\cdot}155\text{ m}^3$; $H_2O = 1{\cdot}28\text{ m}^3$,
Total volume of wet products of combustion $= 0{\cdot}5 + 4{\cdot}155 + 1{\cdot}28 = 5{\cdot}935\text{ m}^3$
Hence, *volumetric* analysis of wet products of combustion is,

$$CO_2 = \frac{0{\cdot}5}{5{\cdot}935} \times 100 = 8{\cdot}4\%; \; N_2 = \frac{4{\cdot}155}{5{\cdot}935} \times 100 = 70\%; \text{ and}$$

$$H_2O = \frac{1{\cdot}28}{5{\cdot}935} \times 100 = 21{\cdot}6\%$$

Volumetric analysis of wet products of combustion is,

$$CO_2 = 8{\cdot}4\%; \; N_2 = 70\%; \text{ and } H_2O = 21{\cdot}6\%$$

Problem-17: *The percentage composition of a producer gas by volume found from its analysis was reported as :*

H_2 = 14; CH_4 = 2; CO = 22; CO_2 = 5; O_2 = 2; and N_2 = 55.

Calculate : (a) the volume of minimum air required for the complete combustion of 1 m^3 of this gas, and (b) the volumetric analysis of dry products of combustion, if 40% excess air is supplied. (Assume that air contains 21% oxygen by volume).

(a) Reactions of combustion can be written as :

$$\text{(i)}\quad 2H_2 + O_2 = 2H_2O \qquad\qquad \text{(ii)}\quad 2CO + O_2 = 2\,CO_2$$

$$2 \text{ vol.} + 1 \text{ vol.} = 2 \text{ vol.} \qquad\qquad 2 \text{ vol.} + 1 \text{ vol.} = 2 \text{ vol.}$$

$$1 \text{ m}^3 + \tfrac{1}{2} \text{ m}^3 = 1 \text{ m}^3 \qquad\qquad 1 \text{ m}^3 + \tfrac{1}{2} \text{ m}^3 = 1 \text{ m}^3$$

$$\text{(iii)}\quad CH_4 + 2O_2 = 2H_2O + CO_2$$

$$1 \text{ vol.} + 2 \text{ vol.} = 2 \text{ vol.} + 1 \text{ vol.}$$

$$1 \text{ m}^3 + 2 \text{ m}^3 = 2 \text{ m}^3 + 1 \text{ m}^3$$

Equation (i) shows that 1 m^3 of hydrogen requires ½ m^3 of oxygen for complete combustion.

∴ Oxygen required for complete combustion of hydrogen = ½ H_2 = 0·5 × 0·14 = 0·07 m^3.

Equation (ii) shows that 1 m^3 of carbon monoxide requires ½ m^3 of oxygen for complete combustion.

∴ Oxygen required for complete combustion of carbon monoxide = 0·5 × 0·22 = 0·11m^3.

Equation (iii) shows that 1 m^3 of CH_4 requires 2 m^3 of oxygen for complete combustion.

∴ Oxygen required for complete combustion of CH_4 = 2 × CH_4 = 2 × 0·02 = 0·04 m^3.

∴ Total volume of oxygen required = 0·07 + 0·11 + 0·04 = 0·22 m^3.

The oxygen available in the fuel gas must be subtracted from the total oxygen required.

∴ Net oxygen to be supplied by air = 0·22 − 0·02 = 0·2 m^3

As air contains 21% oxygen by volume,

Volume of minimum air required for complete combustion of 1 m^3 of producer gas.

$$= 0{\cdot}2 \times \frac{100}{21} = \mathbf{0{\cdot}953 \text{ m}^3}.$$

(b) As 40% excess of air is supplied, actual volume of air supplied = 0·953 × 1·4 = 1·335 m^3.

The dry products of combustion will consist of :

— CO_2 from fuel gas, and by the combustion of CH_4 and CO,

— Oxygen due to excess air supplied, and

— Nitrogen from the actual air supplied, and from the fuel gas,

— Carbon dioxide from gaseous fuel (by given composition) = 0·05 m^3

CO_2 formed due to combustion of CH_4 = $CH_4 \times 1 = 0{\cdot}02 \times 1 = 0{\cdot}02\ m^3$ from eqn. (ii)

CO_2 formed due to combustion of CO = $CO \times 1 = 0{\cdot}22 \times 1 = 0{\cdot}22\ m^3$ from eqn. (iii)

∴ Total volume of CO_2 formed in the flue gas by the combustion of 1 m^3 of fuel gas
$= 0{\cdot}05 + 0{\cdot}02 + 0{\cdot}22 = 0{\cdot}29\ m^3$

— Volume of excess oxygen in flue gas

= (Actual air supplied − Minimum air required) × 0·21

$= (1{\cdot}335 - 0{\cdot}953) \times 0{\cdot}21 = 0{\cdot}0802\ m^3$

— Nitrogen from producer gas (by given composition) = 0·55 m^3

Nitrogen from the actual air supplied
= 0·79 × actual air supplied = $0{\cdot}79 \times 1{\cdot}335 = 1{\cdot}054\ m^3$

∴ Total volume of nitrogen in flue gas = $0{\cdot}55 + 1{\cdot}054 = 1{\cdot}604\ m^3$.

Hence, the volume of dry products of combustion per m^3 of producer gas is

Carbon dioxide = 0·29 m^3
Oxygen = 0·0802 m^3
Nitrogen = 1·604 m^3

Total volume of dry products of combustion = 1·9742 m^3

Hence, volumetric analysis of *dry* products of combustion is

$$\text{Carbon dioxide} = \frac{0{\cdot}29}{1{\cdot}9742} \times 100 = \mathbf{14{\cdot}7\%}$$

$$\text{Oxygen} = \frac{0{\cdot}0802}{1{\cdot}9742} \times 100 = \mathbf{4{\cdot}06\%}$$

$$\text{Nitrogen} = \frac{1{\cdot}604}{1{\cdot}9742} \times 100 = \underline{\mathbf{81{\cdot}24\%}}$$

$$\text{Total} = 100\%$$

Problem-18 : *A gas engine uses producer gas which has a percentage volumetric analysis as follows : $CO = 11$; $H_2 = 29$; $CH_4 = 1{\cdot}8$; $CO_2 = 16{\cdot}1$; and $N_2 = 42{\cdot}1$. Calculate the volume of minimum air required to completely burn one cubic metre of this gas.*

If the air supplied to the gas engine is 50% in excess of this quantity, what is the calorific value of one cubic metre of cylinder mixture ? Calorific values of CO, H_2 and CH_4 are 12,700, 10,800 and 35,000 kJ per cubic metre respectively. What is the gas consumption per kW-hour if the thermal efficiency of engine is 23% ?

Reactions for complete combustion are :

(i) .. $2CO + O_2 = 2CO_2$
2 vol. + 1 vol. = 2 vol.
$1\ m^3 + ½\ m^3 = 1\ m^3$

(ii) .. $2H_2 + O_2 = 2\ H_2O$
2 vol. + 1 vol. = 2 vol.
$1\ m^3 + ½\ m^3 = 1\ m^3$

(iii) .. $CH_4 + 2O_2 = 2H_2O + CO_2$

1 vol. + 2 vol. = 2 vol. + 1 vol.

$1\ m^3 + 2\ m^3 = 2\ m^3 + 1\ m^3$

With the help of above equations, oxygen required for complete combustion of 1 cubic metre of producer gas can be calculated as follows :

(i) Volume of O_2 required for CO $= 0{\cdot}11 \times ½ = 0{\cdot}055\ m^3$,

(ii) Volume of O_2 required for H_2 $= 0{\cdot}29 \times ½ = 0{\cdot}145\ m^3$, and

(iii) Volume of O_2 required for CH_4 $= 0{\cdot}018 \times 2 = \underline{0{\cdot}036\ m^3}$

Total oxygen required $= 0{\cdot}236\ m^3$

As air contains 21% oxygen by volume,

Volume of minimum air required for complete combustion of 1 m^3 of producer gas

$$= 0{\cdot}236 \times \frac{100}{21} = \mathbf{1{\cdot}129\ m^3}$$

As 50% excess air is supplied,

Total air supplied $= 1{\cdot}129 \times 1{\cdot}5 = 1{\cdot}694\ m^3$.

Heat given out by combustion of CO, H_2 and CH_4

$= 0{\cdot}11 \times 12{,}700 + 0{\cdot}29 \times 10{,}800 + 0{\cdot}018 \times 35{,}800$

$= 1{,}397 + 3{,}132 + 644{\cdot}4 = 5{,}173{\cdot}4$ kJ

$\therefore$ Heat supplied by gas per m^3 = **5,173·4 kJ**

Total mixture (gas and total air supplied) in the cylinder

$= 1 + 1{\cdot}694\ m^3 = 2{\cdot}694\ m^3$

Hence, calorific value of the cylinder mixture $= \dfrac{5{,}173{\cdot}4}{2{\cdot}694} = \mathbf{1{,}920{\cdot}3\ kJ/m^3}$

As 1 kW = 1 kJ/sec, then one kW-hour = 1 × 3,600 = 3,600 kJ.

Thermal efficiency of engine $= \dfrac{\text{Work done in kJ per hr.}}{\text{Heat supplied in kJ per hr.}}$

$$= \frac{\text{Work done in kJ equivalent to one kW–hour}}{V_g,\ \text{gas consumption per kW–hour in } m^3 \times \text{heat supplied in kJ per } m^3 \text{ of gas}}$$

i.e. $0{\cdot}23 = \dfrac{3{,}600}{V_g \times 5{,}173{\cdot}4}$

$$\therefore\ V_g = \frac{3{,}600}{0{\cdot}23 \times 5{,}173{\cdot}4} = 3{\cdot}025\ m^3$$

Hence, gas consumption per kW-hour = **3·025 m³**.

Problem-19 : *The fuel supplied to a petrol engine may be assumed to be gasoline having the formula* C_8H_{18}.

Calculate : (i) the theoretical or minimum air required for complete combustion of 1 kg of fuel, (ii) gross calorific value of fuel per kg, and (iii) the percentage of CO_2 *in the exhaust, if the fuel is burned with 50% more air than is theoretically required.*

In the formation of CO_2, 1 kg of carbon liberates 33,800 kJ and in the formation of H_2O, 1 kg of hydrogen liberates 1,44,500 kJ of heat.

(i) The products of combustion will be CO_2 and H_2O.

Combustion equation of petrol is

$$2C_8H_{18} + 25O_2 = 16CO_2 + 18H_2O$$

$$2 \times (12 \times 8 + 1 \times 18) + 25\,(2 \times 16) = 16\,(12 + 2 \times 16) + 18\,(2 \times 1 + 16)$$

$$228 + 800 = 704 + 324$$

Thus, 228 kg of C_8H_{18} require 800 kg of oxygen, for complete combustion.

$\therefore$ 1 kg of C_8H_{18} requires $\frac{800}{228} = 3{\cdot}5088$ kg of oxygen, and

1 kg of C_8H_{18} requires $3{\cdot}5088 \times \frac{100}{23} = 15{\cdot}26$ kg of minimum air for complete combustion.

(ii) Heat evolved due to combustion of carbon in 1 kg of petrol

$$= \frac{12 \times 8}{114} \times 33{,}800 = 28{,}460 \text{ kJ}.$$

Heat evolved due to combustion of hydrogen in 1 kg of petrol

$$= \frac{18}{114} \times 1{,}44{,}500 = 22{,}820 \text{ kJ}$$

$\therefore$ Total heat evolved per kg of petrol = 28,460 + 22,820 = 51,280 kJ

Thus, higher calorific value of petrol = **51,280 kJ/kg.**

(iii) The exhaust gases produced from 1 kg of petrol are :

CO_2 produced $= \frac{704}{228} = 3{\cdot}088$ kg,

H_2O produced $= \frac{324}{228} = 1{\cdot}421$ kg,

N_2 from the actual air supplied $= 0{\cdot}77 \times (15{\cdot}26 \times 1{\cdot}5) = 17{\cdot}625$ kg, and

O_2 from the excess air supplied $= 0{\cdot}23 \times (15{\cdot}26 \times 0{\cdot}5) = 1{\cdot}755$ kg

Total mass of exhaust gases = 23·889 kg

$\therefore$ % of CO_2 in exhaust gases with 50% excess air $= \frac{3{\cdot}088}{23{\cdot}889} \times 100 = 12{\cdot}93\%$

7.15 Gas Producer

A gas producer is an apparatus in which inflammable gaseous mixture is produced from solid or liquid fuels. Broadly speaking carburettors, vaporisers, coke oven plants, town's gas plants or blast furnace plants, etc. are all gas producers. However, the term gas producer is applied to an apparatus in which inflammable gaseous mixture is produced in large quantities for power production, heating and other domestic purposes.

Advantages of producing gas from solid fuel are as under :

.. When coal or other solid fuel is burnt in open fire, considerable waste of fuel and loss of energy occurs and the process of combustion is very inefficient. Gas can be burnt more efficiently and considerable saving of fuel can be effected. This is because the chances for fuel molecules to come in contact with oxygen are more in case of gaseous fuel than that in case of solid or liquid fuel.

.. Gaseous fuel can be distributed conveniently from a central station for use in number of places.

.. When gas is produced in a gas producer as a main product, number of by-products are obtained, such as coke, ammonia and tar. Also many tar derivatives such as carbonic acid, naphthalene, dyes, etc. are produced.

.. Any low quality of fuel containing an excessive amount of ash and moisture such as peat, etc. may be successfully burnt in gas producer to produce gas when their utilisation on steam boiler grate is impossible.

.. The control of supply of gaseous fuel is easier. This in turn makes temperature control possible and easier.

Producer gas is a name given to a cheaply manufactured type of gas for use in gas engines. It is made by the combustion of any grade of coal with limited supply of oxygen derived from air or mixture of air and steam. Incomplete combustion results in the formation of *carbon monoxide*. Other constituents of the gas are *hydrogen, nitrogen* and small percentage of *carbon dioxide* and *oxygen*. Producer gas is continuously obtained by passing a current of air mixed with steam in proper proportion through an incandescent (highly heated) bed of fuel. Once the gas producer has started working, no external heat is necessary, as the heat liberated during the process is sufficient for the purpose.

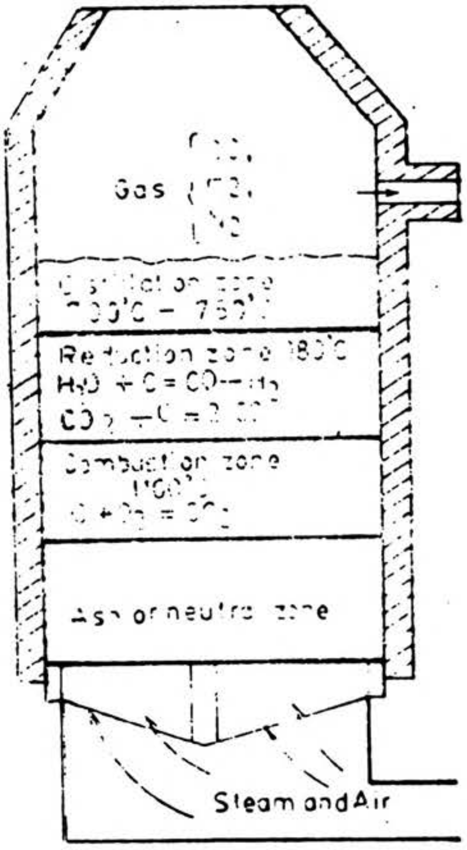


Fig. 7-2. Diagrammatic sketch of a gas generator showing various zones.

Gas is produced in a gas producer or generator (fig. 7-2). The generator consists of a long cylinder lined with fire bricks throughout and having a grate at the bottom, supporting a deep bed of fuel. The coal is fed from top through a closed hopper; air and steam is drawn through the grate. The hot gases produced pass through an opening near the top of the generator. The gas is then washed, purified and passed into a storage tank from which it is drawn for driving gas engines or for heating purposes. Producer gas has a heating value of 3,770 to 6,490 kJ/m^3.

7.15.1 Action of A Gas Producer : Figure 7-2 shows the approximate reactions which take place in the generator and the condition of fuel. The action of gas producer depends upon the depth and temperature of the fuel bed. The bed may be divided into four distinct zones (fig. 7-2) through which the flow of air and steam is maintaied by the suction of the engine or by means of a steam blower or injector.

When fresh coal drops on the top of the fuel bed it is heated by the hot gases passing upwards through it, and more volatile gases are immediately given off, leaving carbon (coke) behind. This zone is known as *distillation zone.* The temperature in this zone is between 370°C and 760°C.

As the hot carbon from distillation zone sinks lower in the generator, it meets the highly heated steam and CO_2 which was formed below in the combustion zone. The result is that CO_2 is reduced to CO by carbon according to reaction, $CO_2 + C = 2CO$, and steam is decomposed into oxygen and hydrogen. Oxygen combines with carbon forming CO and free hydrogen is added to the gas, according to reaction,

$$H_2O + C = CO + H_2.$$

Hydrogen, thus, formed may combine with carbon and form CH_4 or C_2H_4. This zone is known as *reduction zone.* The average temperature in this zone is about 980°C.

Lower down in the *combustion zone,* remaining carbon is burnt to CO_2 according to reaction, $C + O_2 = CO_2$ by the incoming air with the production of considerable heat which superheats the steam and raises its temperature above its decomposition temperature. The average temperature in this zone is about 1,100°C.

The lowest zone consists mainly of ash, about 75 cm thick in depth which is sufficient to superheat the steam. This zone is known as *ash* or neutral zone.

There is, of course, no sharp division line in the generator and the temperature may vary considerably from time to time. The relative amounts of CO and CO_2 ultimately leaving the generator depend largely on the temperature attained in various zones. Higher the temperature, the greater is the amount of CO produced and also a slow velocity of air favours the production of CO.

7.16 Producer Gas Theory

The chief reactants in a gas producer which take part in the chemical reaction are C, CO, CO_2, H_2O, H_2 and O_2. The amount of heat produced by the burning of one kilogram of carbon to CO is 10,130 kJ, and that when one kilogram of carbon is burnt to CO_2 is 33,830 kJ. One kilogram of hydrogen when burnt to H_2O gives out 1,21,420 kJ (lower value). If the operations are reversed, that is if CO, CO_2 and H_2O are decomposed, then the heat required for their decomposition is the same as the heat given out by their combustion. When carbon is burnt in full supply of air, CO_2 is formed, which cannot be used as a fuel. This will be the case when there is thin fire in the producer and large quantity of air is supplied. The reaction is represented by chemical equation,

$$C + O_2 = CO_2 + \text{Heat} \qquad ..(7.5)$$

This reaction occurs in actual producer at the bottom of the producer, i.e. in the combustion zone where air enters.

When considerable depth of fuel is used (several centimetres), CO_2 thus formed at the bottom of the producer will be reduced to CO according to the equation,

$$C + CO_2 = 2CO - \text{Heat} \qquad ..(7.6)$$

This reaction occurs in the gas producer in the reduction zone. Eqn. (7.5) represents burning of carbon to CO_2 and eqn.(7.6) represents subsequent reduction of CO_2 into CO.

CO may also be produced by the direct combustion of carbon with oxygen according to the equation,

$$2C + O_2 = 2CO + \text{Heat} \qquad ..(7.7)$$

Reations (7.6) and (7.7) may or probably do occur in an actual generator.

Calorific value of carbon burnt to CO per kg = 10,130 kJ. Calorific value of carbon burnt to CO_2 per kg = 33,830 kJ. Thus, the percentage heat available in CO gas per kg of carbon and evolved by burning to CO_2 is $\frac{33{,}830 - 10{,}130}{33{,}830} \times 100$ i.e. about 70% of the total heat obtained by completely burning the solid carbon. This means that if there are no heat losses, the efficiency of the producer will be about 70%. The remaining 30% of the total heat is wasted in preparing the combustible gas. This is in a way not permissible.

The 30% (i.e. 10,130 kJ per kg) of the total heat of carbon need not, however, be entirely lost. Out of this, about 8% is lost in radiation and conduction, and a large proportion will be carried away with gas as sensible heat, and the gas produced will be at a high temperature. The high temperature obtained in the producer may be excessive and may cause trouble in the working of the producer by the formation of clinker, which may cause serious trouble by blocking the air passages. To avoid the production of too high a temperature and to raise the theoretical efficiency of the producer to 100% (i.e. to reduce 30% loss), some of the sensible heat may be used to generate steam to be added to the air blast. *By the mixing of steam with air and passing it through the producer, heat is absorbed in the decomposition of the steam, the oxygen of steam combines with carbon and free hydrogen is added to the gas, while temperature of the producer is under control (heat absorbed to decompose steam is 1,21,420 kJ per kg of steam). The gas is enriched, both by the presence of hydrogen and reduction in the amount of nitrogen.*

Steam may react on the carbon in the following ways :

$$C + H_2O = CO + H_2 - \text{Heat} \qquad ..(7.8)$$

$$C + 2H_2O = CO_2 + 2H_2 - \text{Heat} \qquad ..(7.9)$$

Both these reactions cause a large absorption of heat. At temperature above 1,000°C, reaction represented by eqn. (7.8) is more likely to occur but at temperature of about 600°C and under, reaction represented by eqn. (7.9) takes place; while at temperature between 600°C and 1,000°C, the two reactions take place simultaneously. It is evident that since eqn. (7.8) gives rich gas and greater absorption of heat, the best results will be obtained in practice when working at the highest temperature consistent with practical considerations.

The proportion of water which should be used to obtain gas of the highest calorific value can be determined theoretically as shown below, but this in practice is largely

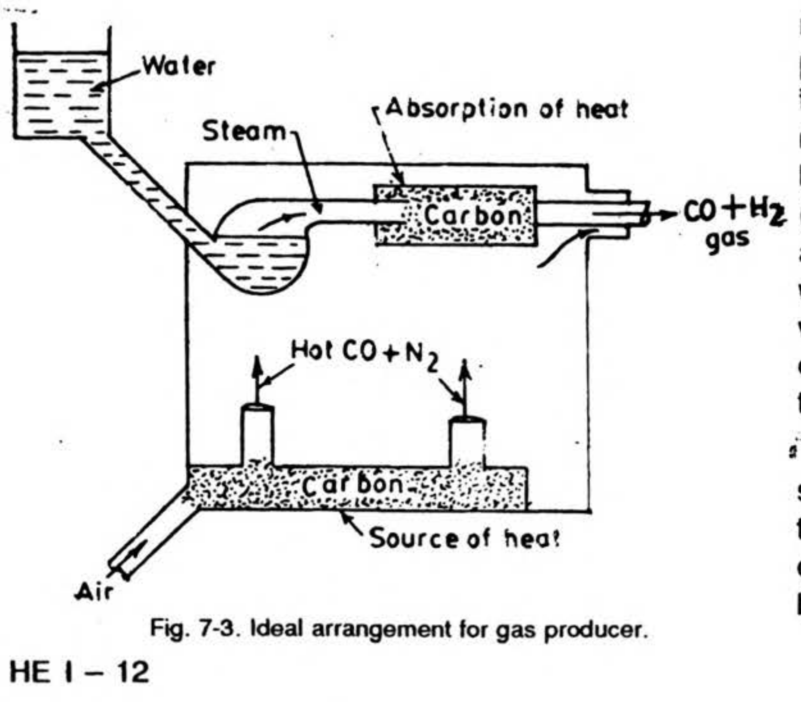


Fig. 7-3. Ideal arrangement for gas producer.

affected by the nature and composition of fuel used and the size of the producer (generator). The mass of water varies from 0·5 kg for large producers to 0·7 kg for small producer per kg of coal gassified.

The richest gas and maximum efficiency will be obtained by the ideal reactions given below;

$2C + O_2 = 2CO$ + Heat (primary reaction), and

$C + H_2O = CO + H_2$ − Heat (secondary reaction)

The arrangement is represented by diagrammatic sketch in fig. 7-3. Assume no heat loss by radiation and all sensible heat of gases leaving the producer is utilised in the reactions, so that all the heat energy of the fuel is available. Then, the amount of heat absorbed by the interaction of steam and carbon will be equal to heat evolved (developed) by the formation of CO from carbon with oxygen in the air.

Let the primary reaction be denoted by (a) and the secondary by (b). Then,

(a) $2C + O_2 = 2CO$ + Heat

i.e. 24 kg C + 32 kg O_2 = 56 kg CO + 24 × 10,130 kJ heat

= 56 kg CO + 2,43,120 kJ heat

(b) $C + H_2O = CO + H_2$ + 12 × 10,130 − 2 × 1,21,420 kJ heat

i.e. 12 kg C + 18 kg H_2O = 28 kg CO + 2 kg H_2 + (1,21,420 − 2,43,120) kJ heat

= 28 kg CO + 2 kg H_2 − 1,21,700 kJ heat

which shows that the addition of 18 kg of steam to 12 kg of carbon absorbs 2,43,120 kJ to decompose the steam and produces 1,21,420 kJ by the combustion of carbon to CO; the net result of the double reaction being the absorption of 2,43,120 − 1,21,420 = 1,21,700 kJ from the generator.

H_2O is supplied to the producer as water and not as steam. If the water enters the generator at 18°C and is evaporated into steam at 100°C, then the heat absorbed per kg of water = 4·187 (100 − 18) + 2,257 ≐ 2,590 kJ.

From reaction (b) it will be seen that, 1,21,700 kJ are absorbed when 18 kg of steam is decomposed by 12 kg of carbon. In addition to the heat absorbed directly from the furnace by 18 kg of steam during decomposition, there is, heat required to produce this steam from water before it enters the furnace. This heat is provided by the hot gases leaving the generator and therefore, comes from the heat available in the fuel. The amount of heat absorbed from the furnace in producing 18 kg of steam is 18 × 2,590 = 46,620 kJ. Therefore, the total heat absorbed from the furnace in producing and decomposing 18 kg of steam in the secondary reaction is 1,21,700 + 46,620 = 1,68,320 kJ heat,

i.e. 12 kg C + 18 kg H_2O = 28 kg CO + 2 kg H_2 − 1,68,320 kJ heat.

In order to obtain heat balance or thermal equilibrium between the primary and secondary reactions, reaction (b) must be multiplied by the ratio $\frac{2,43,120}{1,68,320}$ i.e. by 1·444 throughout.

The reaction (b) becomes 12 × 1·444 kg C + 18 × 1·444 kg H_2O

= 28 × 1·444 kg CO + 2 × 1·444 kg H_2 − 1,68,320 × 1·444 kJ heat i.e. 17·328 kg C + 25·992 kg H_2O absorb 2,43,120 kJ

and from reaction (a) 24 kg C + 32 kg O_2 liberate 2,43,120 kJ

Total carbon used = 17·328 + 24 = 41·328 kg

Total steam (H_2O) used = 25·992 kg

Total oxygen used = 32 kg.

$\therefore$ 1 kg of carbon needs $\frac{25\cdot992}{41\cdot328}$ = **0·629 kg of H_2O (water)**

and 1 kg of carbon needs $\frac{32}{41\cdot328}$ = 0·774 kg of O_2 (oxygen)

$= 0\cdot774 \times \frac{100}{23\cdot1}$ kg of air = **3·352 kg of air.**

7.17 Types of Gas Producers

There are two main types of gas producers. These are :

— Suction gas producer which works under a slight vaccum, and

— Pressure gas producer which works under a slight gauge pressure.

Both types may be designed to work with bituminous or non-bituminous coal.

In suction gas producer which is illustrated in fig. 7-4, mixture of air and steam is drawn through the fuel bed due to suction created by the piston of gas engine during its charging (suction) stroke. The interior of the producer is kept at a pressure less than atmospheric pressure. The producer consists of a cylindrical shell of mild steel or cast iron. It is lined throughout with the fire bricks. An annular space is provided between the fire bricks and the casing for the passage of steam and air which is thereby preheated, and the loss from radiation is reduced in consequences.

On the top of the generator, a coal charging hopper is provided to admit fresh coal to the fuel bed of the producer without interfering with the operation of the producer. A conical bell makes a gas tight seal between the hopper and generator. A number of holes are arranged around the top of the generator for the insertion of poker (a metal rod). Taper plugs are inserted in these holes to prevent air entering the generator.

A starting blower is used for starting the producer, and during the starting period the products of combustion are blown to waste by opening the starting cock.

The hot gases leaving the generator are utilised in evaporating the water in the steam boiler (fig. 7-4). No external boiler for generating steam is, therefore, require. At the base of the generator, a fire grate, a hand hole for removing ash, and a valve for controlling the supply of steam and air are provided.

The hot gases which are still hot for use in gas engine and contain dust and little tar, pass to the coke scrubber. The coke scrubber is vertical steel cylinder containing pieces of coke (about 8 cm in size), over which water is continuously falling from a spraying arrangement on the top of the scrubber. The gas is cooled and cleaned on passing through the scrubber.

In addition to coke scrubber, a saw dust scrubber (not shown in fig. 7-4) is provided through which the gas finally passes, so as to remove the last trace of the dust and moisture present in the gas. The arrows in fig. 7-4 show the path of the gas.

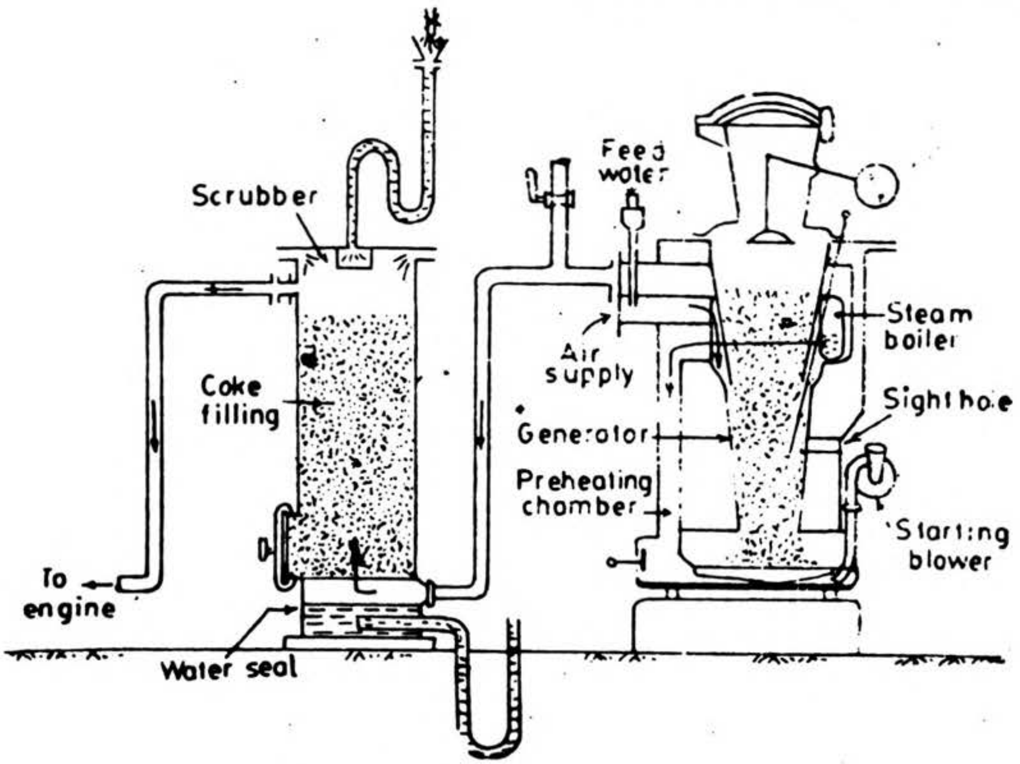


Fig. 7-4. Suction gas producer.

To operate the plant, small quantity of firewood and coal sprinkled with kerosene is dropped on the hearth (fuel bed) through the hopper. The starting cock (fig. 7-4) is opened and the steam boiler is filled with water. The starting blower is worked and after about 20 minutes the fire is sufficient to give off gas. The starting cock is then closed and the formation of gas becomes automatic, the suction of the engine furnishing the draught through the fire. The gas coming out is tested at a small test cock until it comes out with orange red flame. Water spray in the coke scrubber must be turned on as soon as the gas begins to pass through the scrubber. The spray of water should be sufficient to keep the coke scrubber vessel cool.

The suction gas producers is largely used where power only is required (i.e. for driving gas engines).

Pressure Gas Producer shown in Fig. 7-5 is a Downson pressure gas producer for use with *non-bituminous coal or coke.* The gas producer using non-bituminous coal is chiefly adopted for those plants which require gas for combined power and heating purposes or heating purpose only. For heating purposes the gas must be supplied at a steady pressure and should be of uniform quality to prevent the gas jets being extinguished. A gas holder is, therefore, valuable with pressure gas producer.

In this producer (fig. 7-5), there is small, separately fired, vertical steam boiler for producing steam. The steam produced is used in an injector for drawing in supply of air, and forcing mixture of steam and air through the grate. On the top of the generator, a coal charging hopper is provided to admit fresh coal to the fuel bed.

The products of combustion (gas) from the generator are conveyed through a pipe into a coke scrubber filled with coke, over which water is continuously falling through a spraying arrangement at the top of the coke scrubber. The function of coke scrubber is to cool and partly clean the hot smoky gas issuing from the producer. The tar and dust laden water from the coke scrubber runs away through a simple water seal as

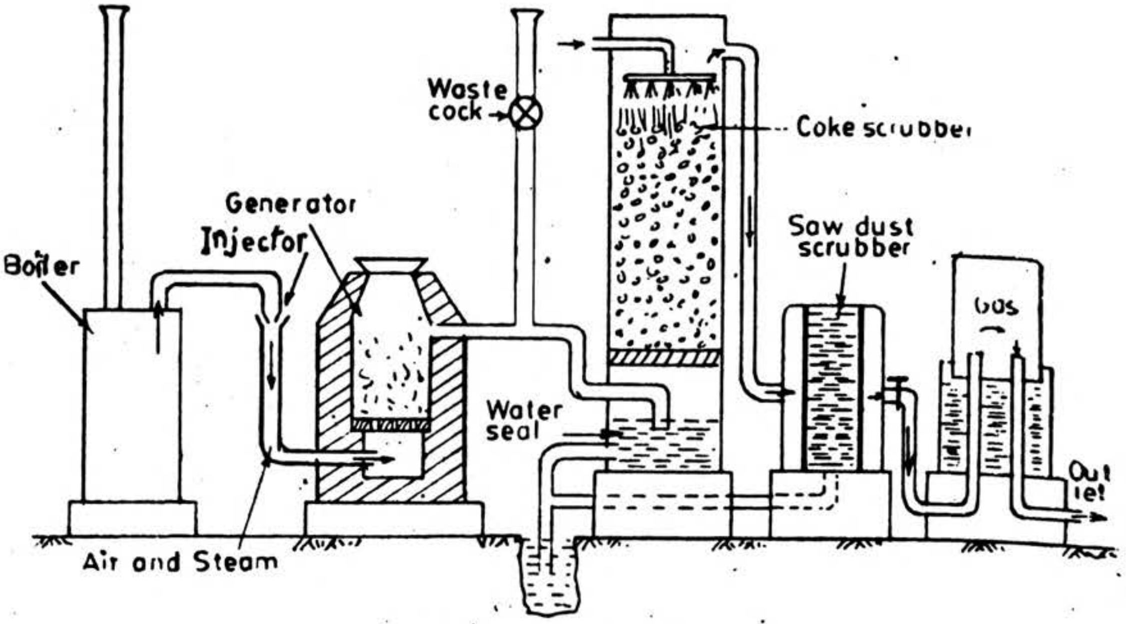


Fig. 7-5. Pressure gas producer.

shown in fig. 7-5.

As in the case of suction gas producer, in addition to the coke scrubber, a saw dust scrubber is provided through which the gas passes, so as to remove the last trace of the dust and moisture.

This clean and cooled gas is then taken through the gas valve into the gas holder from which it is drawn for use. The gas holder is used for supplying gas at steady pressure and of uniform quality, and for storing gas. The arrows in fig. 7-5 show the path of the gas. For the operation of plant refer suction gas producer.

Problem-20 : *A gas producer is supplied with fuel containing 93 per cent carbon and 7 per cent ash on mass basis. The reactions are such that gas given out from the producer consists solely of carbon monoxide, hydrogen and nitrogen. Determine the percentage volumetric analysis of the producer gas; also determine, the mass of water and mass of air needed per kilogram of fuel burnt.*

Assume that the calorific value of carbon burnt to carbon monoxide is 10,260 kJ/kg; that the L.C.V. of hydrogen is 1,23,520 kJ/kg; that the heat required to generate one kilogram of steam from water supplied is 2,610 kJ; that all heat losses are negligible. Air contains 23.1 per cent oxygen on mass basis.

(a) $2C + O_2 \rightarrow 2CO$ + heat (primary reaction)

i.e. 24 kg C + 32 kg $O_2 \rightarrow$ 56 kg CO + 24 × 10,260 kg

$\rightarrow$ 56 kg CO + 2,46,240 kJ ...(i)

(b) $C + H_2O \rightarrow CO + H_2$ − Heat (secondary reaction)

i.e. 12 kg C + 18 kg $H_2O \rightarrow$ 28 kg CO + 2 kg H_2 − Heat

The heat required for dissociation of steam to form 1 kg of H_2 is given as 1,23,520 kJ. 1 kg of steam requires 2,610 kJ for its generation.

18 kg of H_2O is evaporated and absorb 18 × 2,610 = 46,980 kJ.

18 kg of H_2O is dissociated into 2 kg of hydrogen and absorb 2 × 1,23,520 = 2,47,040 kJ.

12 kg of carbon combine with 16 kg of O_2 and liberate $12 \times 10{,}260 = 1{,}23{,}120$ kJ.

$\therefore$ Net absorption of heat in secondary reaction

$= 2{,}47{,}040 + 46{,}980 - 1{,}23{,}120 = 1{,}70{,}900$ kJ

i.e. 12 kg of C + 18 kg of $H_2O \rightarrow -1{,}70{,}900$ kJ ..(ii)

In order to obtain heat balance between the primary and secondary reactions, secondary reaction (ii) should be multiplied throughout by the ratio $\frac{2{,}46{,}240}{1{,}70{,}900}$, i.e. by 1·441 throughout

i.e. 17·29 kg of C + 25·94 kg of $H_2O \longrightarrow$ absorb 2,46,240 kJ

and from (a) 24 kg of C + 32 kg of $O_2 \rightarrow$ liberate 2,46,240 kJ

$\therefore$ 41·29 kg of C will need 25·94 kg of H_2O.

$\therefore$ 1 kg of C will need $\frac{25 \cdot 94}{41 \cdot 29} = 0 \cdot 628$ kg of H_2O.

and $0 \cdot 628 \times 0 \cdot 93 =$ **0·584 kg of H_2O per kg of fuel.**

Again 41·29 kg of C will need 32 kg of O_2.

$\therefore$ 1 kg of C will need $\frac{32}{41 \cdot 29} = 0 \cdot 775$ kg of O_2.

and $0 \cdot 775 \times 0 \cdot 93 \times \frac{100}{23 \cdot 1} =$ **3·12 kg of air per kg of fuel.**

The products are :

$$CO,\ 56 + 28\left(\frac{2{,}46{,}240}{1{,}70{,}900}\right) = 56 + 40 \cdot 35 = 96 \cdot 35 \text{ kg, i.e. } \frac{96 \cdot 35}{28} = \mathbf{3 \cdot 441 \text{ mols.}}$$

$$H_2,\ 2\left(\frac{2{,}46{,}240}{1{,}70{,}900}\right) = 2 \cdot 882 \text{ kg, i.e. } \frac{2 \cdot 882}{2} = 1 \cdot 441 \text{ mols.}$$

$$N_2,\ \frac{76 \cdot 9}{23 \cdot 1} \times 32 = 106 \cdot 53 \text{ kg, i.e. } \frac{106 \cdot 53}{28} = 3 \cdot 804 \text{ mols.}$$

Total products = 3·441 + 1·441 + 3·804 = 8·686 mols.

$$\left\{\begin{array}{l}\text{Volumetric analysis of}\\ \text{producer gas by percentage}\end{array}\right\} \quad CO,\ \frac{3 \cdot 441}{8 \cdot 686} \times 100 = 39 \cdot 62\%$$

$$H_2,\ \frac{1 \cdot 441}{8 \cdot 686} \times 100 = 16 \cdot 59\%$$

$$N_2,\ \frac{3 \cdot 804}{8 \cdot 686} \times 100 = 43 \cdot 79\%$$

$$\Sigma = 100 \cdot 00\%$$

Volumetric analysis of producer gas is

$CO = 39 \cdot 62\%$; $H_2 = 16 \cdot 59\%$; $N_2 = 43 \cdot 79\%$.

Problem-21 : *Calculate the mass of air and steam which should be supplied per kg of carbon in a suction gas plant assuming that no CO_2 is formed. Calorific value of hydrogen is 1,42,360 kJ/kg and one kilogram of carbon burned to CO gives 10,130 kJ. Also calculate the percentage volumetric analysis of the producer gas produced. Air contains 23% oxygen on mass basis.*

In the production of the gas following two reactions take place :

$$\begin{array}{llll} \text{(a)} \quad 2C & + \; O_2 & = 2CO & + \; 24 \times 10{,}130 \text{ kJ} \\ \quad 24 \text{ kg} & + \; 32 \text{ kg} & = 56 \text{ kg} & + \; 24 \times 10{,}130 \text{ kJ} \\ \text{i.e. } 1 \text{ kg} & + \; 1\tfrac{1}{3} \text{ kg} & = 2\tfrac{1}{3} \text{ kg} & + \; 10{,}130 \text{ kJ} \end{array}$$

$$1 \text{ kg of } C + \tfrac{4}{3} \text{ kg of } O_2 = \tfrac{7}{3} \text{ kg } CO + 10{,}130 \text{ kJ} \quad ..\text{(i)}$$

$$\begin{array}{llll} \text{(b)} \quad C + H_2O & = CO & + \; H_2 - \text{Heat} \\ \quad 12 \text{ kg} + 18 \text{ kg} & = 28 \text{ kg} & + \; 2 \text{ kg} - \text{Heat} \end{array}$$

$$\text{i.e. } 1 \text{ kg} + 1\tfrac{1}{2} \text{ kg} = 2\tfrac{1}{3} \text{ kg} + [10{,}130 - \tfrac{1}{6}\,(1{,}42{,}360) \text{ kJ}],$$

($\because$ 1 kg of C burning to CO produces 10,130 kJ, and 1 kg of H_2 liberated from steam requires 1,42,360 kJ).

Hence, we have,

$$1 \text{ kg of } C + \tfrac{3}{2} \text{ kg of } H_2O = \tfrac{7}{3} \text{ kg } CO + \tfrac{1}{6} \text{ kg } H_2 \text{ produces } 10{,}130 \text{ kJ and absorbs } \tfrac{1}{6} \times 1{,}42{,}360 \text{ kJ}$$

$$= \tfrac{7}{3} \text{ kg } CO + \tfrac{1}{6} \text{ kg } H_2 + 10{,}130 - \tfrac{1}{6} \times 1{,}42{,}360$$

$$= \tfrac{7}{3} \text{ kg } CO + \tfrac{1}{6} \text{ kg } H_2 - 13{,}596 \text{ kJ} \quad ..\text{(iia)}$$

It follows that for reaction (iia) to be brought about, sufficient carbon must be burnt in reaction (i) to supply 13,596 kJ. Hence to obtain a heat balance between two reactions, we must multiply reaction (iia) by the ratio $\frac{10{,}130}{13{,}596}$.

Thus, we get,

$$\frac{10{,}130}{13{,}596}\,[\,1 \text{ kg } C + \tfrac{3}{2} \text{ kg } H_2O\,]$$

$$= \frac{10{,}130}{13{,}596}\,[\,\tfrac{7}{3} \text{ kg } CO + \tfrac{1}{6} \text{ kg } H_2 - 13{,}596\,] \text{ kJ}$$

or $0{\cdot}745 \text{ kg } C + 1{\cdot}12 \text{ kg } H_2O = 1{\cdot}74 \text{ kg } CO + 0{\cdot}124 \text{ kg } H_2 - 10{,}130 \text{ kJ}$ ..(ii b)

The heat required for reaction (ii b) is now equal to the heat liberated in reaction (i).

Total carbon required per kg of C burnt to CO = 1 + 0·745 kg = 1·745 kg.

Total O_2 required per kg of C burnt in both reactions $= \frac{1{\cdot}333}{1{\cdot}745} = 0{\cdot}765$ kg.

Mass of air required per kg of C burnt $= 0{\cdot}765 \times \frac{100}{23} = 3{\cdot}32$ kg

Mass of steam required per kg of C burnt $= \frac{1{\cdot}12}{1{\cdot}745} = 0{\cdot}642$ **kg**

The two reactions yield, 1·74 + 2·33 = 4·07 kg of CO; 0·124 kg of H_2;

and $\frac{1{\cdot}333 \times 77}{23} = 4{\cdot}453$ kg of N_2.

From this % composition on mass basis = CO, 47·07; H_2, 1·434; and N_2, 51·496

Conversion to percentage analysis by volume of producer gas :

Constituent	% by mass ÷ molecular weight	% composition by volume
CO	$47{\cdot}07 \div 28 = 1{\cdot}681$	$\frac{1{\cdot}681}{4{\cdot}238} \times 100 = 39{\cdot}67$
H_2	$1{\cdot}434 \div 2 = 0{\cdot}717$	$\frac{0{\cdot}717}{4{\cdot}238} \times 100 = 16{\cdot}92$
N_2	$51{\cdot}496 \div 28 = 1{\cdot}84$	$\frac{1{\cdot}84}{4{\cdot}238} \times 100 = 43{\cdot}41$
Total	$= 4{\cdot}238$	$= 100{\cdot}00$

Volumetric analysis of producer gas is,

$CO = 39{\cdot}67\%$; $H_2 = 16{\cdot}92\%$; and $N_2 = 43{\cdot}41\%$

Tutorial–7

1. What do you understand by combustion ? Explain the terms "minimum mass of air" and "excess air" with reference to combustion of fuel.

2. Determine from first principles the minimum quantity of air required for complete combustion of 1 kg of (i) carbon (C), (ii) hydrogen (H), and (iii) methane gas (CH_4).

   [(i) 11·6 kg; (ii) 34·78 kg; (iii) 17·39 kg]

3. Prove that minimum air required for complete combustion of one kg fuel having C kg of carbon, H kg of hydrogen, S kg sulphur and O kg oxygen in one kg fuel is given by $\frac{100}{23}(2{\cdot}67C + 8H + S - O)$ kg

4. The percentage composition of a fuel oil on mass basis is as follows :

   C, 85·1; H_2, 13·7, and O_2 1·2·

   Calculate, the minimum air required to burn completely 1 kg of this fuel oil and hence the percentage composition of the dry products of combustion on mass and volume basis.

   [14·55 kg; percentage composition by mass - CO_2, 21·8; N_2, 78·2;
   percentage composition by volume - CO_2, 15·1; N_2, 84·9]

5. The percentage composition of coal on mass basis is given as :

   C, 90; H_2, 3·5; O_2, 1; S, 0·5; and ash, 5.

   Calculate : (a) the minimum air required for the complete combustion of 1 kg of this coal, and (b) the composition of dry flue gases on mass basis if 50% excess air is supplied.

   [(a) 11·6 kg; (b) composition by mass - CO_2; 18·253%; O_2, 7·4%; SO_2, 0·055%; N_2, 74·292%]

6. A certain coal has following percentage composition on mass basis : C, 0·8; H_2, 0·1; and S, 0·1. The analysis of the flue gases shows the following composition by volume :

   CO_2, 10%; CO, 1%; O_2, 10%; and N_2, 79%.

   Find, (a) the minimum air required per kg of coal, (b) the air actually supplied per kg of coal, and (c) the excess air supplied per kg of coal.

   [(a) 13·18 kg; (b) 17·36 kg; (c) 4·18 kg]

7. In a boiler trial, the percentage composition of coal on mass basis was reported as :

   C, 88; H_2, 3·6; O_2, 4·8; and ash, 3·6.

   The percentage composition of dry flue gases by volume was reported as :

   CO_2, 10.9; CO, 1; O_2, 7·1; and N_2, 81

   Find, (a) the mass of air actually supplied per kg of coal, and (b) the percentage of excess air supplied.

   [(a) 18·15 kg; (b) 38·3%]

8. In a boiler trial, the sample of coal fired, contained carbon 88% by mass and the ash in it was 3·6%. The volumetric analysis of the gases gave the following percentages :

   CO_2, 10·9; CO, 1; O_2; 7·1; and N_2, 81

   Find, (a) the proportion of carbon burned to CO, (b) the air required per kg of coal for the combustion as it actually occured, and (c) the mass of dry flue gases formed per kg of coal burned.

   [(a) 8·4%; (b) 18·15 kg; (c) 19·114 kg]

9. The percentage composition of a sample of coal on mass basis is as follows :

   C, 82; H_2, 6; O_2, 4; ash, 6; and moisture, 2.

   Determine the theoretical or minimum air required for complete combustion of 1 kg of coal.

   If the actual air supply is 18 kg per kg of coal, the hydrogen is completely burned, and 80% of carbon contained in 1 kg of coal is completely burned to CO_2 and the remainder is burned to CO, calculate the percentage composition by volume of the dry flue gases. (Air contains 23% O_2 on mass basis).

   [11·5 kg; percentage composition by volume CO_2, 8·84; CO, 2·25; O_2; 8·76; N_2, 80·15]

10. A fuel oil has the following analysis on mass basis : C, 85%; H_2, 12·5%; O_2, 2%; and residue, 0·5%. The dry exhaust gas has the following composition by volume :

    CO_2, 9%; CO, 1%; O_2, 7·77%; and N_2, 82·23%.

    Determine the air-fuel ratio.

    [21·25 : 1]

11. Describe the Orsat apparatus for the analysis of the dry flue gases from a boiler and briefly describe the procedure for carrying out the experiment.

12. The percentage dry flue gas analysis by volume in a boiler trial was : CO_2, 12; CO, 1; O_2, 7; and N_2, 80. The percentage analysis of the coal on mass basis was : C, 82; H_2, 4; O_2, 5; S, 1; moisture, 2; and ash 6.

    Determine : (a) the total mass of flue gases per kg of fuel burnt, (b) the mass of excess air supplied per kg of fuel burnt, and (c) the heat carried away by the products of combustion leaving the boiler per kg of fuel burnt, if the average specific heat of the dry flue gases is 1·005 kJ/kg K and the temperature of the flue gases is 325°C and the boiler house temperature is 25°C. Assume steam to be at atmospheric pressure and specific heat of superheated steam as 2·1 kJ/kg K.

    [(a) 16·26 kg; (b) 4·62 kg; (c) by dry flue gases, 4,793·85 kJ,
    by steam, 1,156·7, total heat, 5,950·55 kJ]

13. Estimate the minimum volume of air required for complete combustion of 1 m^3 of fuel gas having the following volumetric analysis :

    CH_4, 39·5%; H_2, 46%; N_2, 0·5%; CO, 7·5%; H_2O, 2%; and CO_2, 4·5%.

    Air contains 21% O_2 by volume. [5·04 m^3]

14. A fuel gas has the following percentage composition by volume :

    CO, 30; CH_4, 5; H_2, 10; and N_2, 55.

    Calculate : (a) the minimum volume of air required for the complete combustion of 1 m^3 of this gas, and (b) the percentage composition by volume of the dry products of combustion. (Air contains 21% O_2 by volume).

    [(a) 1·429 m^3; (b) percentage composition by volume - CO_2, 17·25; N_2, 82·75]

15. A fuel gas has the following composition by volume :

    H_2, 20%; CH_4, 2%; CO, 24%; CO_2, 6%; O_2, 3%; and N_2, 45%.

    If 25% excess air is supplied, calculate the mass of air at 0°C and 1·01325 bar (N.T.P) actually supplied for the combustion of 1 m^3 of this gas and also the percentage composition by volume of dry flue gases. Take the specific volume of air at 0°C and 1·01325 bar as 0·7734 m^3/kg. (% composition of air by volume : N_2, 79; O_2, 21)

    [1·765 kg; percentage composition by volume - CO_2, 16·74; N_2, 80·26; O_2, 3]

16. The fuel supplied to a petrol engine may be assumed to have the formula C_7H_{16}.

    Calculate : (i) the minimum air required for complete combustion of 1 kg of fuel, (ii) the gross calorific value of fuel per kg; and (iii) the percentage of CO_2 in the exhaust gases, if the fuel is burned with 30% excess air.

    Assume that 1 kg of carbon liberates 33,800 kJ forming CO_2 and 1 kg of hydrogen liberates 1,44,450 kJ forming H_2O.

    [(i) 15·3 kg; (ii) 51,504 kJ; (iii) 14·73%]

17. A producer gas used by a gas engine has the following percentage composition by volume :

$CO = 15$; $H_2 = 25$; $CH_4 = 2$; $CO_2 = 12$; and $N_2 = 46$.

Calculate the volume of minimum air required to completely burn one cubic meter of producer gas.

If the air supplied to the engine is 40% in excess, what is the calorific value of one cubic metre of cylinder mixture ? Calorific values of CO, H_2 and CH_4 are 12,560, 10,890 and 35,600 kJ per cubic metre respectively. What is the gas consumption per kW-hour, if the engine thermal efficiency is 24 percent ?

[1·143 m^3; 2,206 kJ/m^3, 2·615 m^3]

18. What is meant by the term "Gas producer". Explain in detail the advantages of producing gaseous fuel from solid and liquid fuels.

19. Differentiate between "Suction gas producer" and "Pressure gas Producer". Explain their working in brief.

20. Explain with the help of sketches, the construction and working of any one type of gas producer suitable for running a small gas engine.

A suction gas producer is supplied with fuel containing 94% carbon and 6% ash on mass basis. The reactions are such that the producer gas from the gas producer consists of hydrogen, carbon monoxide and nitrogen only.

Determine : (i) the mass of steam and air required per kilogram of fuel burnt, and (ii) the percentage volumetric analysis of the producer gas. Neglect all heat exchanges except those in the chemical reactions.

Assume that the calorific value of carbon burnt to carbon monoxide is 10,130 kJ/kg, the L.C.V. of hydrogen is 1,21,420 kJ/kg; heat given to steam is 2,610 kJ/kg and air contains 23·1% O_2 on mass basis and 21% O_2 on volume basis.

[(i) 0·5912 kg of steam/kg of fuel; 3·154 kg of air/kg of fuel;
(ii) CO, 39.62%; H_2 16·605%; N_2, 43·775%]

21. Give a short account of the reactions which may take place in a suction gas producer supplied with carbon, air and steam. What are the advantages of adding steam with air ?

Carbon in the form of coke with 12% ash, is used as a fuel in gas producer. Water is supplied at 15°C and the resulting reactions of both the water and the air with carbon produce CO, there being no CO_2 in the gas produced. The generation of the steam and its subsequent dissociation together absorb 94% of the heat liberated by the partial combustion of carbon with the air. The gross or higher calorific value of hydrogen is 1,44,450 kJ/kg and the heat produced by burning 1 kg of carbon to carbon monoxide is 10,360 kJ/kg. Air contains 23% oxygen on mass basis and temperature of the gas may be neglected. Estimate : (i) the mass of air and water required per kg of coke, and (iii) percentage volumetric analysis of the gas produced.

[(i) 2·96 kg of air and 0·5478 kg of water per kg of coke;
(ii) CO, 39·55%; H_2, 16·42; N_2, 44·03%]

22. Make a diagrammatic sketch of any one type of gas producer and explain its working. Give chemical reactions involved in the proces of manufacture of producer gas.

23. What is meant by the term "Producer gas" ?

Draw a layout plan of a suction gas producer gas plant incorporating therein a device to measure the quantity of gas produced. State the functions of various zones and chemical reactions involved in each one of these zones.

24. Sketch a typical form of gas producer using steam blast, and describe briefly the reactions that occur therein.

8

STEAM BOILERS

8.1 Introduction

A steam boiler is a closed vessel, strongly constructed of steel, in which steam is generated from water by the application of heat. The steam generated is used for producing power and for industrial work and heating work. The steam boiler is also known as *steam generator.* The function of a steam boiler or generator is to convert chemical energy of fuel by combustion into heat and to transfer this heat to water and thus to produce steam.

The following terms are commonly used in connection with various types of boilers and thus, their application and their meaning be clearly understood :

Boiler shell consists of one or more steel plates bent into cylindrical form and riveted or welded together. The ends of the shell are closed by means of *end plates or heads,* which are made flat, or concave. The shell together with closing heads is called the *drum.*

Setting forms the walls of the combustion chamber. It confines the heat to the boiler and forms a passage through which the gases pass. The passages so formed for the gases are called *flues.* The boiler setting also provides support for some types of boilers.

Grate in a coal or wood fired boiler is a platform in the furnace upon which the fuel is burned. The grate consists of cast iron bars which are spaced apart so that air for combustion can pass through them. The area of the grate surface on which the fire rests, in a coal or wood fired boiler, is usually expressed in square metres.

Furnace is also called a *firebox.* It is the space above the grate and below the boiler shell in which the fuel is burnt.

Volume of the shell that is occupied by water is termed **water space.** The **steam space** is the entire shell volume, less that occupied by water and tubes.

Water level is a level at which water stands in the boiler shell. The remaining space above the water level is called *steam space.*

Heating surface is the part of the boiler surface that is exposed to the fire and to the hot gases from the fire as they (hot gases) pass from the furnace to the chimney.

Mounting is the term usually referred to such items as safety valves, main stop valve, high-steam and low-water alarm, feed check valve, pressure gauge, water-level gauge, blow-off cock, etc. Special provision is always made on the boiler to mount them. A boiler can not function safely without the above mentioned mountings.

Accessories is the term applied to those items which form an integral part of the boiler but are not mounted on the boiler. Superheater, economiser, feed pump etc., are considered as accessories.

Blowing-off is the act of removing the floating impurities that float at the water level, and is termed surface blow-off. This is done by means of some kind of surface blow-off appliance.

The terms boiler and steam generator carry practically the same meaning or sense. However, **boiler** is an old name used for a unit which generates saturated steam. The name steam generator is a modern expression and is used for unit which includes in its integral construction, quite separate units, such as an economiser, a superheater, an air preheater, etc. The main boiler with these accessories is now-a-days called **steam generator.**

8.2. Classification of Boilers

Boilers may be classified according to relative position of the water and the hot furnace gases under the two main classes :

- *Fire-tube boilers and shell or tank boiler,* in which furnace gases pass through the tubes, these tubes being surrounded by water which is to be evaporated.
- *Water-tube boilers,* in which furnace gases pass over the external surface of the tubes through which water is circulated.

Boilers may be further classified according to the following :

.. Their form—*vertical* or *horizontal boilers.*

.. Their construction—*tank* or *tubular boilers.*

.. The service to which they are put—*land (stationary), portable, marine or locomotive boilers.*

The principal types of boilers belonging to the shell or tank and **fire-tube** class are :

- Cornish and Lancashire boilers,
- Simple vertical boilers,
- Cochran boilers,
- Locomotive boilers, and
- Scotch marine boilers.

The boilers belonging to **water-tube** class are :

* Babcock and Wilcox boilers,
* Stirling boilers, and
* Yarrow boilers.

8.3 Shell or Tank type Boilers

The shell or tank type of boilers are particularly suitable for stationary work where working pressure and power required are moderate. These boilers give reliability, ease of operation, and easy steaming even with impure feed water. Although these boilers are economical steam generators, they raise steam slowly on account of the large quantity of water they store, and because of their restricted fire space and slowness of the water circulation. The boilers belonging to this class are : Cornish and Lancashire boilers.

8.3.1 **Cornish Boiler** : The Cornish boiler was first introduced by Cornish engineer from whom it derives its name. It consists of a cylindrical shell with flat ends through which passes a single flue tube (or furnace tube) usually centred on the vertical centre

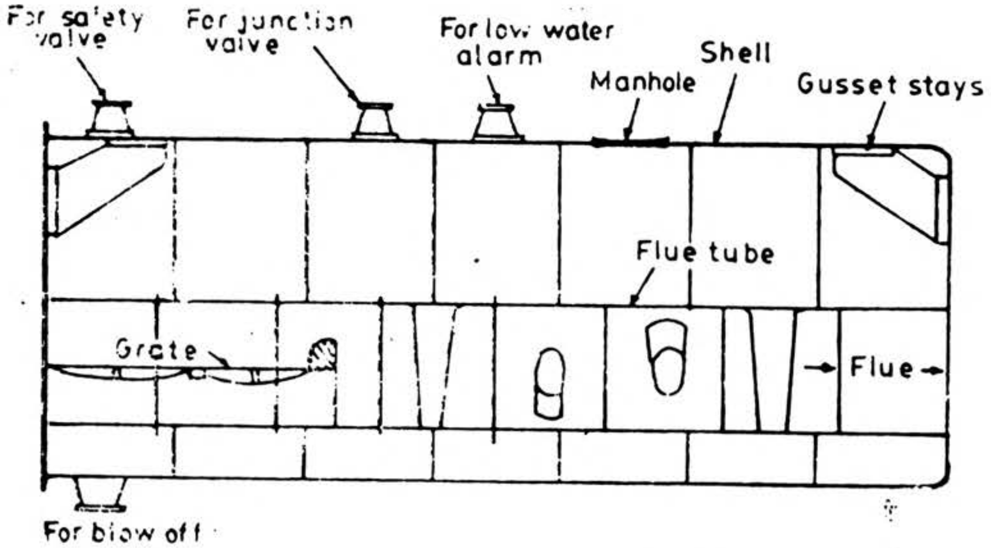


Fig. 8-1. Longitudinal section of the shell of a Cornish boiler.

line of the boiler. In some cases flue tube is arranged two or three centimetres off centre to make internal cleaning easy as shown in fig. 8-1. The claim has been made that this eccentricity encourages water circulation. The boiler is supported (set) on brickwork flues, forming two external *side flues* and one *bottom flue* (fig. 8-2). Thus, the part of the heating surface is on the external shell. The products of combustion pass from the fire grate to the end of the furnace, or flue tube. The gases then enter the side flues and return to the front end of the boiler. The gases now enter the bottom flue through which they pass to back end of the boiler and then to the chimney. The brickwork setting should be well maintained to reduce air in-leakage. The longitudinal section and other details of this boiler are similar to that of the Lancashire boiler (fig. 8-3) described hereafter.

Figure 8-1 shows Cornish boiler shell in section, and in fig. 8-2 is given cross-section through the boiler and brickwork setting. These two figures will be readily understood from what has been already said. The four cross-tubes inserted in the flue tube (fig. 8-2) must be specially noted. They are called *galloway tubes,* and their object is to improve the circulation of water in the boiler when heated by the furnace gases; its specific gravity being thereby lowered, and an ascending or rising current of water is set up in the galloway tubes. This draws the cold water from the bottom of the boiler to be heated in turn. Some of the galloway tubes are vertical and other diagonal; the object of this is to break up the current of flue gases passing along the furnace tube and to produce a scrubbing (rubbing) action of the gases on the outside walls of tubes. This arrangement increases the heating surfaces of the tubes, as every portion of the hot gases will, in turn, be brought into contact with plates having water to be heated on the other side.

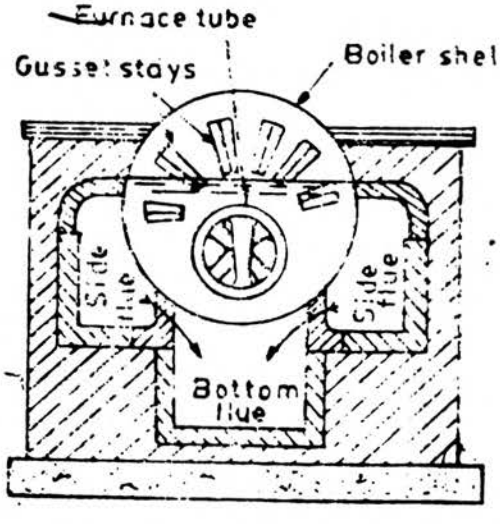


Fig. 8-2. Cross-section of a Cornish boiler and the brickwork setting and flues.

The Cornish boiler is sufficient for evaporation upto 2,000 kg of steam per hour and they are normally made for pressures upto 11 bar. Usually the Cornish boiler has cylindrical shell from 1·25 to 2 m in diameter and from 5 to 8 m long. The diameter of the furnace or flue tube may be about 0.8 times the shell diameter. But, Cornish boilers are now built with a shortened furnace tube and fitted with smoke tubes for pressures upto 14 bar and having evaporative capacity of 3,000 kg per hour. Cornish boilers fitted with smoke tubes are known as *multi-tubular* boilers. It may be said that Cornish boiler is simple and with due care it can be an economical steam generator.

8.3.2 Lancashire Boilers : This boiler is very widely used as a stationary boiler because of its good steaming quality and because it can also burn coal of inferior quality. This boiler is only a modification of the Cornish boiler. *It differs from the Cornish boiler in having two internal furnaces or flue tubes instead of one.* The Lancashire boiler has cylindrical shell usually from 2 to 3 metres in diameter and from 8 to 9.5 m long. The main features of the Lancashire boiler and its brickwork setting are shown in fig. 8-3. The boiler consists of a cylindrical shell with flat ends, and two furnace tubes pass right through this. There are two side flues and one bottom flue formed by brickwork setting. These flues provide part of the heating surface on the main external shell. The Lancashire boiler differs from the Cornish boiler in one or more ways. *The difference is in the manner or order of conducting the hot flue gases through the flues.* Here, the gases from the furnace pass to back end of the boiler where they dip (go down) and enter the bottom flue and travel through it to the front end of the boiler. The gases then divide themselves here and enter the side flues and travel through them to the back end of the boiler and then to the chimney.

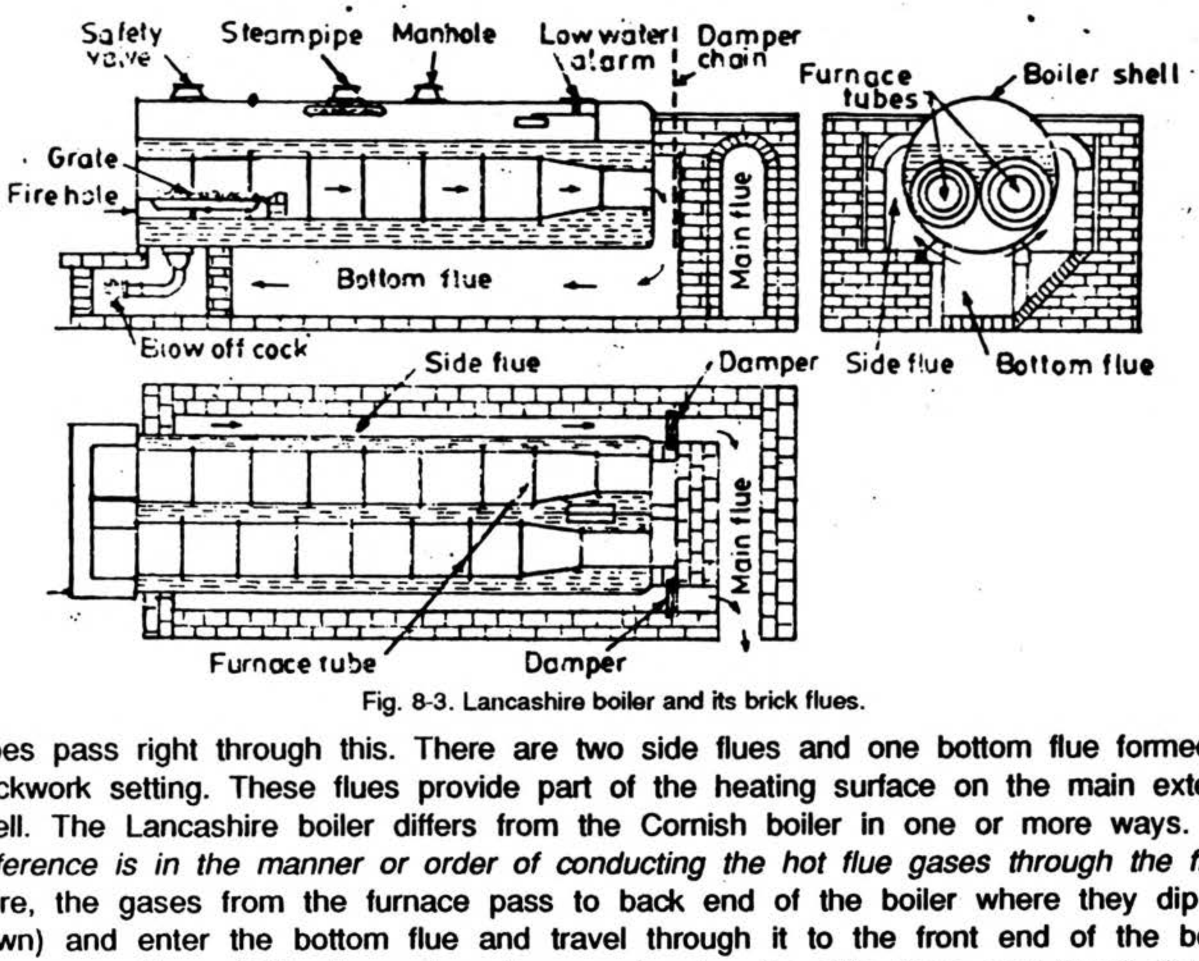


Fig. 8-3. Lancashire boiler and its brick flues.

This type of boiler is often fitted with cross-tubes called galloway tubes, as in the Cornish boiler (fig. 8-2). They are fitted across the furnace tubes. They provide an increased heating surface and improve the circulation of the water.

The boiler is provided with a blow-off cock at the bottom of the front end, and a feed check valve with a feed pipe on the front end plate. The mountings usually provided on the boiler are : pressure gauge, water level gauge (or water level indicator), stop valve, safety valve, high-steam and low-water safety valve, fusible plug, and anti-priming pipe. The accessories provided in large size boiler of this type are : superheater, economiser, and feed pump.

Some Lancashire boilers are fitted with shortened furnace and a number of smoke tubes. These smoke tubes increase the heating surface of the boiler. Lancashire boilers fitted with smoke tubes are known as *multi-tubular boilers.* They are also sometime called *economical* boilers.

The features which have made the Lancashire boiler so *popular* are as under :

— Simple in design – there is little to go wrong.

— Ease of operation – it will put up with rough treatment.

— The ratio of the volume of the boiler to its rated evaporative capacity is high, consequently it is able to meet heavy peak loads without very great variation in steam pressure.

The *disadvantages* of Lancashire boiler are :

— The shell construction restricts the maximum working pressure to about 17.5 bar.

— It occupies considerable floor space.

— There is so large a water capacity and so little encouragement to water circulation, especially between the furnace tubes and the bottom of the shell, that it is impossible in emergency to raise steam pressure rapidly from the cold water.

— Brickwork setting is expensive in the first cost and troublesome in maintenance.

— The grate area is restricted by the diameter of the internal furnace tubes or flue tubes.

Cornish and Lancashire boilers can be compared as shown in table 8-1.

Table 8-1 Comparison of Cornish and Lancashire Boilers

Feature	Cornish	Lancashire
Diameter of shell	1.25 – 2 m	2 – 4 m
Length of shell	5 – 8 m	8 – 10 m
Steam pressure	10 – 14 bar	15 – 20 bar
Steam capacity	2,000 – 4,000 kg/hr	8,000 – 10,000 kg/hr
No. of internal flue tubes	One	Two
No. of side flues	Two	Two
No. of bottom flues	One	One
Path of flue gases	Front to back through one internal flue (furnace) tube, back to front through two side flues, and front to back through one bottom flue.	Front to back through two internal flue tubes, back to front through one bottom flue and front to back through two side flues.
Initial cost	Less	More
Popularity	Less	More

8.4 Fire-tube Boiler

A fire-tube or smoke-tube boiler is one in which the hot products of combustion (hot gases) flow through the inside of tubes, known as smoke tubes, and water surrounding the tubes. The fire-tube boiler belongs to the old class of boilers, but they have still a place of usefulness where the steam pressure does not exceed about 10 bar and where a moderate quantity of steam is required. They have the advantage of low cost and compact design. Their evaporative capacities range from 200 to 3,000 kg of water per hour. Their thermal efficiencies vary from 65 to 68 per cent under normal conditions, and smaller size of these are easily portable. Boilers belonging to this class are : *Simple vertical boiler, Vertical boiler with horizontal smoke tubes (Cochran boiler), Locomotive boiler,* and *Scotch marine boiler.*

Fire-tube boilers are also termed as economical boilers. These boilers permit higher steam output evaporation capacity than that permitted by the Lancashire or Cornish boilers for a given floor space. These boilers have much less water capacity and so they require less time to raise steam from cold water.

8.4.1 Simple Vertical Boiler : The simplest form of a vertical boiler is illustrated in fig. 8-4. it consists of a cylindrical shell surrounding a nearly cylindrical fire box. At the bottom of the fire box is the grate. The uptake tube passes from the crown of the fire-box to the crown of the shell, and on the top of this uptake tube is fitted the chimney. The fire-box is fitted with two cross-tubes marked *F*. The cross-tubes are fitted slightly inclined to ensure efficient circulation of water. Hand holes are provided for cleaning the cross-tubes. A man-hole is also provided for cleaning and inspection of the boiler.

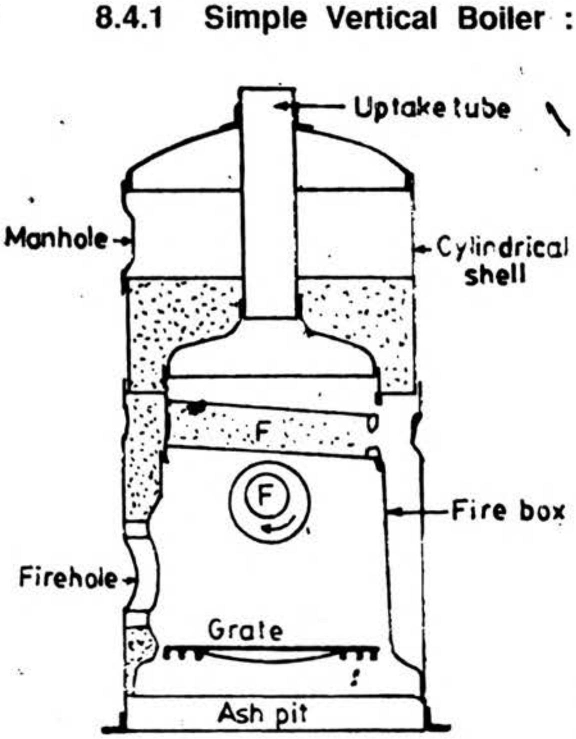


Fig. 8-4. Simple vertical boiler.

These boilers are used for small powers and where space is limited. The maximum working pressure is about 10 bar.

8.4.2 Cochran Boiler : This is one of the best type of vertical multi-tubular boilers. Figure 8-5 illustrates its design. It is made in numerous designs and sizes of evaporative capacities ranging from 150 to 3,000 kg of water per hour and for working pressures upto 20 bar and is suitable for different types of fuels. This boiler gives thermal efficiency of about 70 per cent with coal firing, and about 75 per cent with oil firing.

The boiler consists of a cylindrical shell with its crown having a hemispherical shape. Such a shape of the crown plate gives enough strength to withstand the bulging effect of the inside steam pressure.

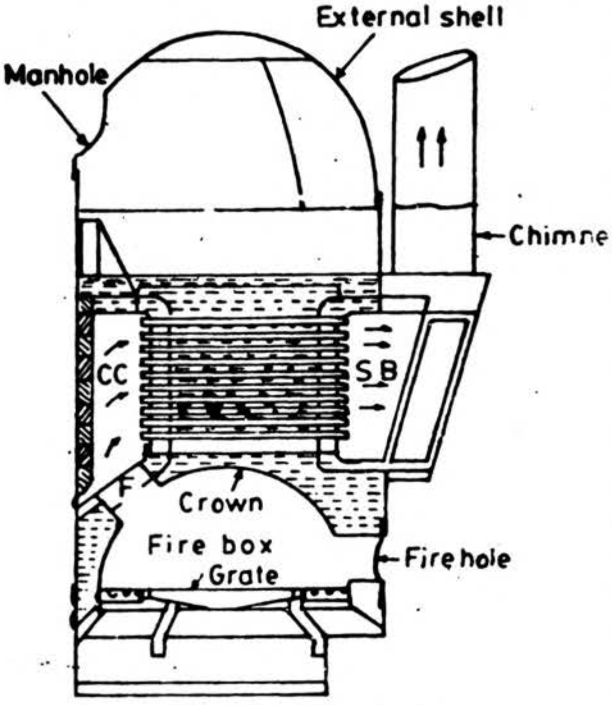


Fig. 8-5. Cochran boiler.

The fire-box is constructed in one piece and has no joints. The fire-box too has a crown of hemispherical shape. The shape is particularly advantageous for the absorption of the radiant heat from the furnace. The convection heat surfaces are provided by a large number of horizontal smoke tubes (about 150). The hot products of combustion from the fire box enter through the small flue-pipe F into the combustion chamber *CC* and strike on the boiler shell plate which forms the back of the combustion chamber. The back plate of the combustion chamber is lined with fire-bricks and can be conveniently dismantled and removed for cleaning smoke tubes. The back plate directs the gases into the smoke tubes.

The gases after passing through the horizontal smoke tubes enter the smoke box SB and then to the uptake or chimney. Most of the smoke tubes are fixed in the vertical tube plates by being expanded in the holes but some of them are fixed by screwing into the holes. The screwed tubes form stays to the vertical tubes and prevent them from bulging out due to the inside steam pressure. A number of hand holes are provided around the outer shell for cleaning purposes. The flat top of the combustion chamber *CC* is strengthend by gusset stays as shown in fig. 8-5.

Vertical multi-tubular boilers (Cochran boilers) have the advantage of taking up a comparatively small floor space or area and are used where space is limited. It is self contained and stronger from design point of view. As there are no seams (joints) in the furnace, this source of trouble is eliminated.

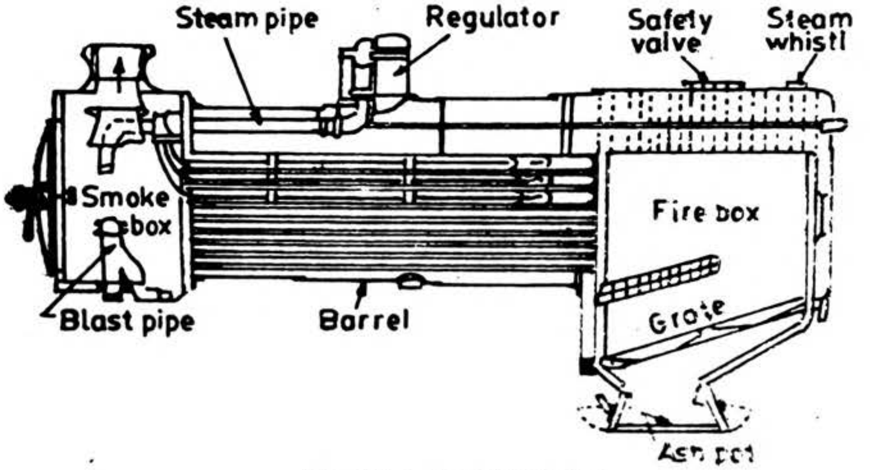


Fig. 8-6. Locomotive boiler.

8.4.3 Locomotive Boiler : The locomotive boiler is a horizontal fire-tube boiler with an internal fire box as shown in fig. 8-6. This boiler requires a large heating surface and a large grate area upon which coal can be burnt at a rapid rate. This is obtained by providing a large number of smoke tubes and by having strong induced draught (current of air) by means of steam jet.

The boiler consists of a cylinderical shell having a rectangular fire box at one end and a smoke-box at the other end. The fire-box is connected to the some-box by a number of horizontal smoke tubes. The hot gases from the furnace pass through these tubes into the smoke-box and are then discharged from the short chimney. The necessary draught is obtained by the steam exhausted from the engine cylinder. The exhaust steam is discharged in the form of a jet, with the help of a blast pipe and nozzle, placed at the base of the smoke-box shown in fig. 8-6. The jet of steam drives the gases upward from the smoke-box into the short chimney so as to create partial vacuum in the smoke box. This induces a strong draught (current of air) through the furnace box and tubes.

The fire-box is made of front plate, back plate, two side plates and a crown, riveted together to form a rectangular box. These inside plates are separated from the outside plates by a space which forms a *water space*. On the top of the shell and in front of the fire-box, an opening is provided over which is situated a dome shaped chamber known as the *steam dome* (fig. 8-7). Steam is taken to engine cylinders from the elevated dome so that it may contain as small amount of water particles as possible.

The last heating surface of the boiler is that which surrounds the fire-box. The part of the smoke tubes nearest to the fire-box is the most effective heating surface.

The side plates of the fire-box are stayed to the outer plates of the shell by means of screwed and riveted copper stays. The flat front tube plate is stayed to the flat smoke-box tube plate by means of longitudinal stays. The flat crown plate of the

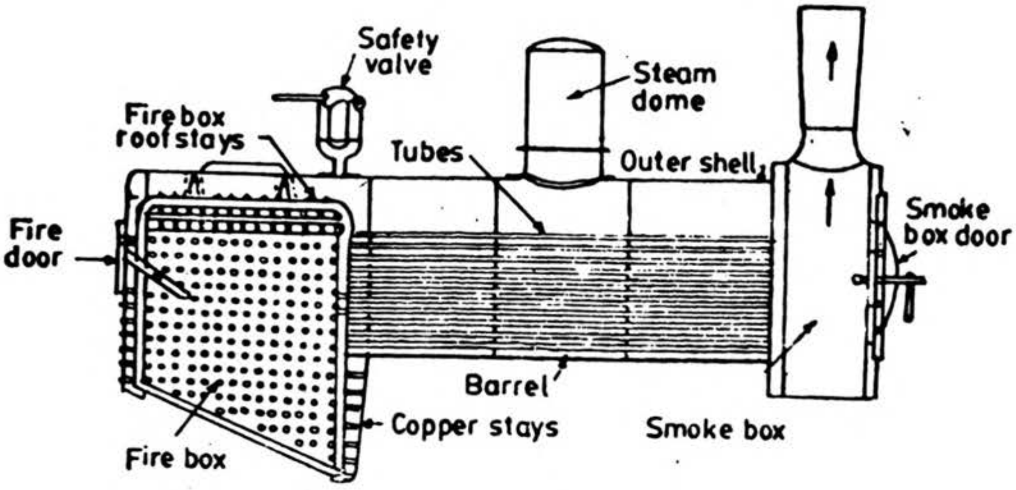


Fig. 8-7. Locomotive boiler.

fire-box requires to be well stayed to prevent it from collapsing under the pressure at its top. The staying is done by means of girder stays called fire-box roof stays (fig. 8-7).

The Locomotive boiler has the following *advantages* :

.. Compactness,

.. High steaming capacity,

.. Fair economy, and

.. Portability or Mobility.

It has also some *disadvantages* such as :

– Large flat surface needs sufficient bracing (supporting),

– There is corrosion in the water legs on account of sedimentary deposits, and

– The difficulty of reaching the inside for cleaning.

On ships, either fire-tube or water-tube boilers are used. The fire-tube type is used where lightness and high speed are not required as in heavy ships. The water-tube type is used where fast steaming and high pressure steam is required as in naval and fast passenger ships. The common types of marine boilers are : *single and double ended Scotch marine boilers, Yarrow marine boilers, and Babcock and Wilcox marine boilers.*

8.4.4 Single-ended Scotch Marine Boiler : Figure 8-8 illustrates a single-ended marine boiler commonly known as Scotch type. The cylindrical shell of this boiler contains from one to four cylindrical corrugated steel furnaces. The furnaces FB are internally fired and are surrounded by water. At the back end of the furnace is the combustion chamber CC which is also surrounded by water. Each furnace usually has its own combustion chamber but in some cases two or more furnaces open into common combustion chamber. A large number of fire tubes run from the front tube plate to the back tube plate. The hot gases in the furnaces pass forward, due to the draught, into the combustion chambers, and then through the tubes to smoke-box situated at the front end of the boiler, from where they move through uptake to the chimney.

The walls of the combustion chamber are the best heating surface of the boiler. The furnace tubes, the smoke tubes and the combustion chambers, all being surrounded by water, give a very large heating surface area in proportion to the cubical size of the boiler.

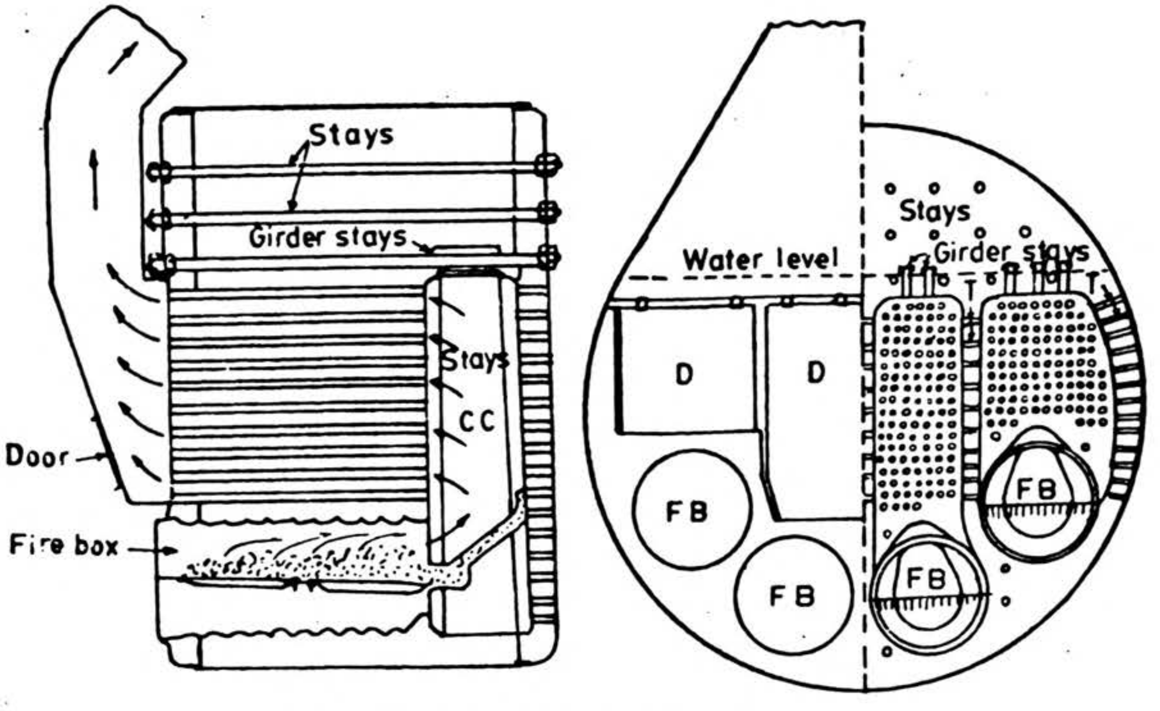


Fig. 8-8. Single-ended Scotch marine boiler.

The flat ends of the shell are stayed by longitudinal stays. Some of the smoke tubes are screwed into the tube plates and work as stays for the flat surfaces of tube plates. The flat plate of the combustion chambers are stayed by the screw stays, while their flat tops are strengthened by the girder stays as shown in fig. 8-8. Doors D are provided in the front end for cleaning the smoke tubes.

8.4.5 Double-ended Scotch Marine Boiler : They have furnaces at each end. They look like single-ended boiler placed back to back. The furnace tubes at each end open into a centrally placed combustion chamber from which the hot gases pass through the smoke tubes to the smoke boxes, one at each end of the boiler. A double-ended boiler has the advantage of being lighter, cheaper and occuping less space as compared with single-ended boiler for the same evaporation capacity.

8.5 Water-tube Boilers

Water-tube boilers have water inside the tubes and hot gases surrounding the tubes. These boilers are used extensively because they can be built for high pressures and large evaporative capacities. They are safe, quick steaming, and flexible in construction and operation. They consist of small drums in contrast to the shell or tank type boilers. The drum forms a small part of the total heating surface, the greater part of heating surface being provided by a number of water tubes fitted outside the drum in the furnace.

Units have been constructed in which steam is produced at the critical pressure of 220.9 bar, and evaporative capacity of half million kg of steam per hour have already been reached. In modern power stations, steam temperature of about 1,000 K at the stop valve is now available.

The water-tube boilers may be classified into four groups according to the following :

- *Service to which they are put* : Stationary or Marine,
- *Position of drum* : Vertical, Cross, or Longitudinal,

– *Type of tubes used* : Straight tube or Bent tube, and

– *Method of circulation* : Natural circulation or Forced circulation.

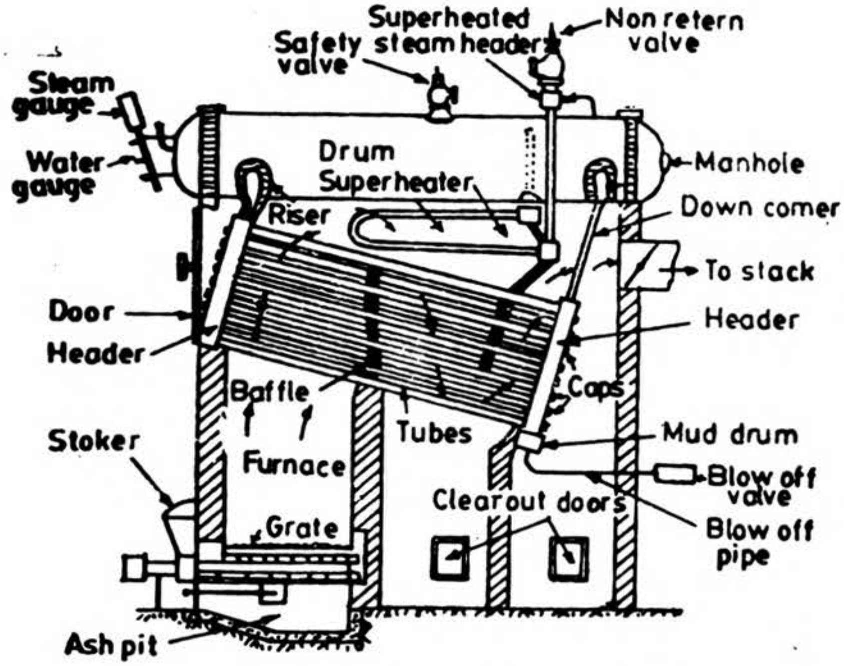


Fig. 8-9. Babcock and Wilcox water-tube boiler.

8.5.1 Babcock and Wilcox Water-tube Boiler : This is best known water-tube of boiler and is made of one or more horizontal, steam and water drums. The drum is connected to a series of front end and rear (back) end headers by short riser tubes. A series of inclined water tubes of solid drawn mild steel are connected to these *headers*. Each nest of tubes is made of several vertical rows. The tubes are expanded into headers which are provided with staggered or zigzag holes. Figure 8-9 illustrates the Babcock and Wilcox water-tube boiler of the land type.

The staggered arrangement of the water tubes will allow the surface of every tube to be exposed to the hot gases. A hand hole is provided in the header in front of each tube, which allows cleaning and inspection of tubes. Each hole is covered by a steel cap which is secured in its position by a steel clamp.

The hot gases from the furnace are forced to move upwards between the water tubes by fire brick baffles provided. They then move downwards between the tubes and then to chimney or stack. The movement of gases in this manner facilitates the heat transfer even to the highest part of the tubes. The feed water enters the front of the drum, passes to the back of the drum, and then descends through the down coming vartical tubes and enters the headers. The water then enters the water tubes, moves upwards through the inclined tubes and finally rises through the front riser tubes to the drum. The circulation of water is produced due to difference of density of water which in turn is due to difference of temperature in the front and rear (back) parts of the furnace. Thus, a thermo-siphon effect is created which results in continuous and rapid circulation of water.

The steam and water drum of the boiler is suspended from horizontal beams by means of metalic slings or straps. The horizontal beams in turn are supported on cast iron columns. This arrangement makes the boiler unit independent of the brickwork which eliminates troubles due to expansion. The brickwork around the boiler is only meant to enclose the furnace and the hot gases. A mud drum is provided at the lowest part of the inclined tubes. The sediment in water collects in the mud drum from where it is blown off by means of blow-off valve at regular intervals.

The boiler is provided with the usual mountings, as well as a superheater. The soot from the gases accumulating on the surface of the water tubes is removed at intervals, either by mechanical scrapers or it is blown off by high pressure steam blowers. This is necessary to keep the heat transfer by conduction effective.

8.6 Comparison between Water-tube Boilers and Fire-tube Boilers

These two types of boilers can best be compared by listing their advantages and disadvantages and compared with one another.

The *advantage* of water-tube boilers over the tank or shell and fire-tube boilers are as under :

.. Water-tube boilers generate steam of high pressure which has become a present day demand.

.. The water is divided into small portions, and therefore, water-tube boilers raise steam quickly.

.. The heating surface of water-tube boilers is much more effective than an equivalent area of surface in the ordinary tubular boilers.

.. The direction of water circulation in water-tube boilers is well defined. The circulation is rapid all over the boiler, keeping the boiler at a nearly constant temperature.

.. The arrangement of water-tube boilers is such that it forms a flexible construction. Every member of the boiler is free to expand without unduly expanding or compressing any other member. This feature gives prolonged life to the boiler.

.. Water-tube boilers are of sectional construction, and therefore, can be transported and erected more readily than the other types of boilers.

.. An accident to any one tube or fitting does not produce the destruction of the whole boiler. Hence, water-tube boilers are sometimes called *safety boilers.*

The *disadvantages* of water-tube boilers as compared with fire-tube boilers are as under :

- They are less suitable for use with impure and dirty water. If the water contains scale forming material, a small deposit of scale will lead to overheating and bursting of the tubes.
- They require more expert attention. The cost of their upkeep is relatively high.
- They are somewhat more difficult to inspect.

8.7 Factors for Boiler Selection

As there are many types of boilers, the factors to be considered for the selection of a boiler for particular purpose are as under :

- The pressure at which the boiler is to operate and quality of steam required i.e., whether wet, dry or superheated.
- Rate of steam generation i.e., quantity of steam per hour required to be produced.
- Availability of floor area.
- Efficiency of boiler in the same range i.e., amount of heat extracted per unit mass of fuel burnt.
- Easy accessibility for cleaning, repairs and inspection.
- The boiler must conform to the "Boiler Act".
- Comparative initial cost.
- Erection facility i.e., erection of the boiler is easy or complex.

In short, the boiler selected should possess as many good qualities as necessary for the purpose for which it is to be used.

The high pressure and large evaporative capacity boilers (generators) are described in volume II. These boilers are specially used in big thermal stations for power generation. They have special provisions for water and flue gas circulation, water and air heaters, superheaters, coal pulverisation, etc.

8.8 Boiler Mountings

All boilers are fitted with fittings or mountings for the *safety* of the boilers, and for complete *control* of the process of steam generation.

The fittings are :

Safety fittings

- Safety valves
- Water-level indicator
- Combined high-steam and low-water safety valve
- Fusible plug

Control fittings

- Steam pressure gauge
- Feed check valve
- Junction or stop valve
- Blow-off cock or valve

8.8.1 Safety Valves : The main function of a safety valve is to prevent the steam pressure in a boiler exceeding the pre-determined maximum working pressure by automatically opening the valve and discharging the steam as soon as this maximum pressure is reached. Moreover, besides operating at the set pressure, safety valves must be capable of discharging the full evaporative capacity of the boiler, otherwise the possibility of continued pressure build up will remain. The great damage to the boiler which may result from safety valve failure needs no detailed description.

Safety valves may be classified into two distinct groups according to the method of loading the valve, namely,

— Weight loaded safety valves, and

— Spring loaded safety valves.

The valves in the first group (weight loaded) may be again sudivided into :

— Dead-weight safety valve, and

— Lever and weight safety valve (lever safety valve).

Dead-weight safety valve is perhaps the most simple type of safety valve in which the valve is loaded by the direct application of weights above the valve. Such a valve is known as dead-weight safety valve.

Figure 8-10 illustrates dead-weight safety valve made by M/s. Hopkinson & Co. In this safety valve the valve rests (sits) on the valve seat which is fixed to the top of

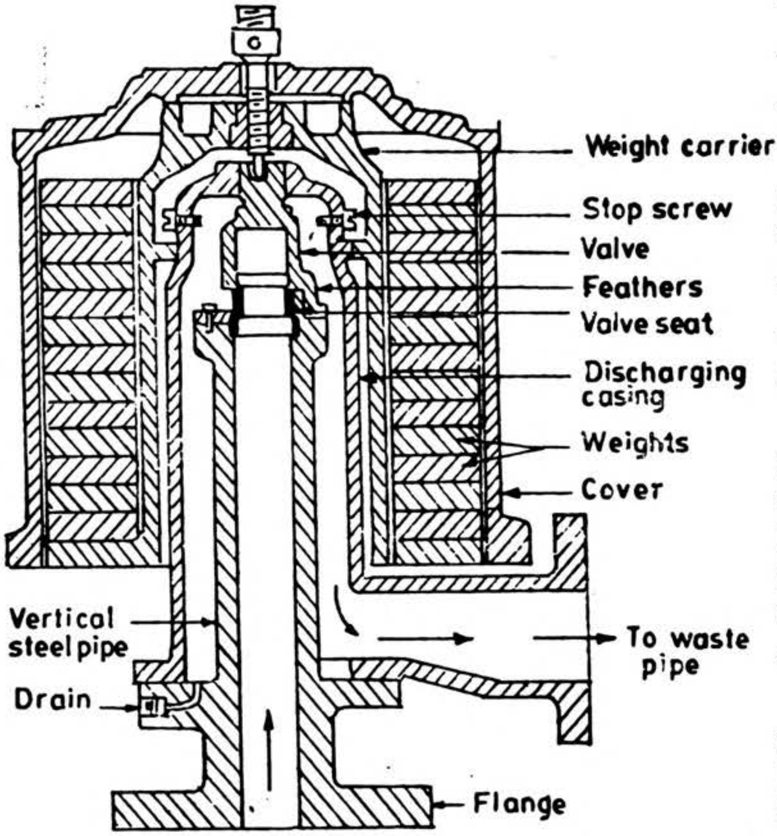


Fig. 8-10. Dead-weight safety valve.

the vertical steel pipe by a ring and screws as shown. The ring has a feather cast on it, which acts as a guide for the valve. Suspended from the top of the valve is a large cast iron casting which acts as a weight carrier. A cast iron cover is fitted over the weights and weight carrier. The load on the valve is made up of weights, the weight of the weight carrier and cover, and the weight of the valve itself, and this load balances the pressure of steam on the valve. When this pressure becomes too high, the valve and the weight carrier will lift and surplus steam will escape to the enclosed discharge casing from which it is carried to the waste pipe.

The valve and weights which it carries are prevented from blowing away by a ring cast on the inside of weight carrier and stop screw fitted on discharge casing as shown in fig. 8-10. The condensed steam in the discharge casing is drained by drain pipe connected at the bottom.

The vertical steel pipe has a flange at the bottom for bolting to a mounting block which is connected to the boiler shell by rivets.

To find the dead-weight required (including casting and weights) for a valve of given area : multiply the area of the valve in m^2 by the steam pressure in N/m^2 or Pa at which the valve is required to lift or open. Thus, a valve of 8 cm diameter to blow off at 10 bar, requires the following dead weights :

Dead-weight = valve area x pressure of steam

$$= \frac{\pi}{4}\left(\frac{8}{100}\right)^2 \times (10 \times 10^5) = 5{,}026{\cdot}55 \text{ N or } = 512{\cdot}4 \text{ kg}$$

The dead-weight safety valve is probably the most reliable type of safety valve and actually gives quite a satisfactory performance during operation, but is suffers from certain disadvantages which entirely prevents its use on many types of boilers. One great drawback is its unsuitability for use on boilers where extensive vibrations and movements are experienced, as for example, in locomotive and marine boilers. A second disadvantage arose as higher pressure was introduced in the steam industry. This increase in working pressures brought with it the need for heavier valve loadings, which in turn, necessitated heavier and bulkier safety valves. Moreover, together with the rise in working pressures, there has been a marked increase in the evaporative capacity of boilers particularly since the introduction of the water-tube boilers and this has brought the need for safety valves possessing large discharge area. Now, the necessary loading for a safety valve varies proportionately to the square of the valve diameter and so, any increase in valve diameter made in order to obtain a large discharge area, will require a very great increase in valve loading.

From these considerations it will be seen that *"the dead-weight safety valve has very limited range of application, being mainly for low pressure, low capacity, stationary boilers of the Cornish and Lancashire type.*

Lever safety valve is very close to the dead-weight safety-valve which is the lever and weight pattern safety valve. As will be seen from the diagrammatic sketch (fig. 8-11), the valve is loaded by means of a weighted lever which is pivoted about a fulcrum situated close to the valve. The actual value of the valve load L being dependent upon the lever ratio b/a viz.,

$$L = W \times \frac{b}{a} \qquad \text{.. (8.1)}$$

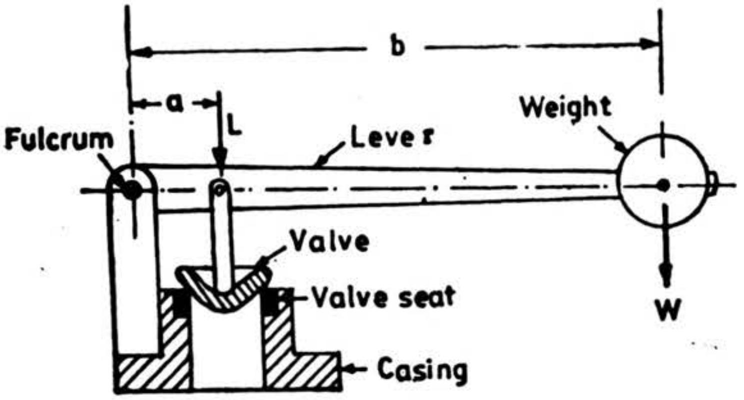


Fig. 8-11. Lever safety valve.

This compares with the expression $L = W$ for a dead-weight safety valve, and since the lever ratio b/a is usually about 8, it follows that the loading weight W is something like 1/8th the weight necessary for a correspnding dead-weight valve. Consequently, lever and weight safety valves do not suffer to such an extent from the pressure limitations experienced by dead-weight safety valves and are occasionally used for pressures upto 40 bar. *This type of valve is suitable for stationary boilers only as it is affecated by vibration.*

To find the weight W (load on outer end of lever in N) for a given steam pressure, lever ratio b/a, and area of the valve in m^2.

Let d = diameter of valve in m, and

p = pressure of steam in N/m^2 or in Pa at which valve is about to open or lift.

If the effect of weight of valve and lever be omitted, we have, when valve is about to lift,

Moment of downward pressure = Moment of upward pressure

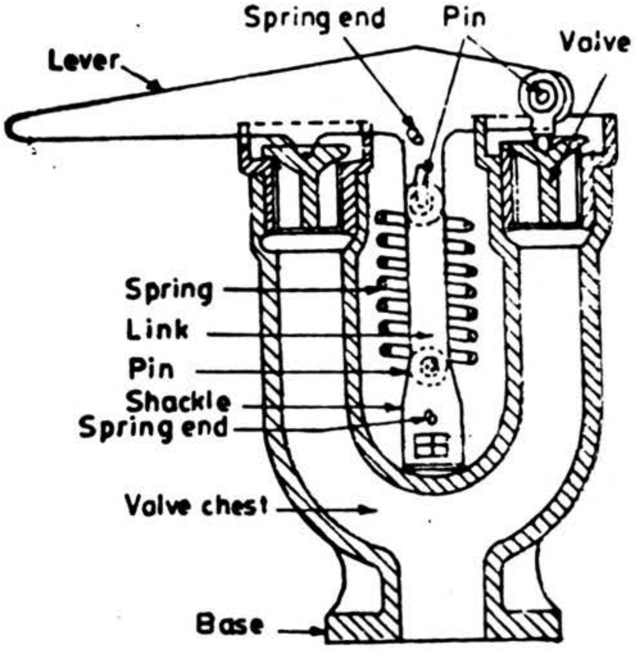
$$\text{i.e., } W \times b = \left(\frac{\pi d^2}{4} \times p\right) \times a \qquad \text{.. (8.2)}$$

An important disadvantage possessed by lever safety valve might be mentioned here. Due to the necessity of atleast two pivoting points (at the fulcrum and at the valve spindle), the valves are liable to suffer, to much greater extent than other types, from the effect of friction which in bad cases may prevent the valve from opening when the pressure exceeds the maximum working pressure.

In order to prevent unauthorised changes, the weight W is fitted on the lever by a pin and lock (now shown), or similar locking means.

For locomotive and marine boilers, both the dead-weight and lever safety valves are not suitable. **Spring loaded safety valve** is therefore mostly used on locomotives and marine boilers, as this type of valve is not affected by vibration.

Figure 8-12 illustrates a Ramsbottom spring loaded safety valve. It consists of two separate valves and seatings. The valves are held down on their seats by the helical spring and lever. The lever has two pivots, of which one is forged on the



ig 8-12. Ramsbottom spring loaded safety valve.

lever and the other is joined to it by a pin. The spring is hooked to an arm of the lever midway between the valves. The lower end of the spring is hooked to the shackle which is secured to the valve chest by studs and nuts.

To prevent the valves from being blown away in the event of spring breaking, there are two links, one behind the other, on either side of the lever connected by pins at their ends. The upper pin passes through a slot in the lever arm and the lower pin passes through the shackle.

By pressing down or raising the lever the engine driver or boiler attendant can relieve the pressure from either valve separately and find out that the valve is not sticking on the seating and is free to act properly.

One disadvantage of this lever is that the load on the valve increases as the valves lift, so that the pressure required to just lift the valve is less than that required to open it fully.

8.8.2 Combined High-Pressure Steam and Low-Water Safety Valves : Description of safety valves would be incomplete if no reference is made to the most important fitting of Lancashire boiler and other internally fired boilers. This is the high-steam and low-water safety valve, sometimes known as *compound safety valve.* It is suitable for stationary boilers.

This valve was introduced by M/s. Hopkinsons Ltd., U.K. in 1852. It is still regarded as a standard mounting on boilers. Although many improvements in design have been made since its introduction, the principle working remains basically the same (combining in one single fitting two separate and distinct features).

The first safety features consits of a high-pressure steam valve held down on its seat partly by the dead weight and partly by the lever and weight. The second safety feature consists of a float device which gives warning of the near approach of a dangerously low water level.

The two essential features will be seen from the sectional view shown in fig. 8-13. The float *A* usually made of firebrick, is suspended from the end *C* of the cast iron lever *BC,* and when fully immersed in the water is balanced by the weights *D* suspened from the other end of the lever. An alternative type of float, whose use is to be recommended where the feed water is likely to have harmful effect on a firebrick float, is the all metal float. Much longer life may be expected from a float of this type. Both types of floats are incapable of floatation on their own account and in order to cause

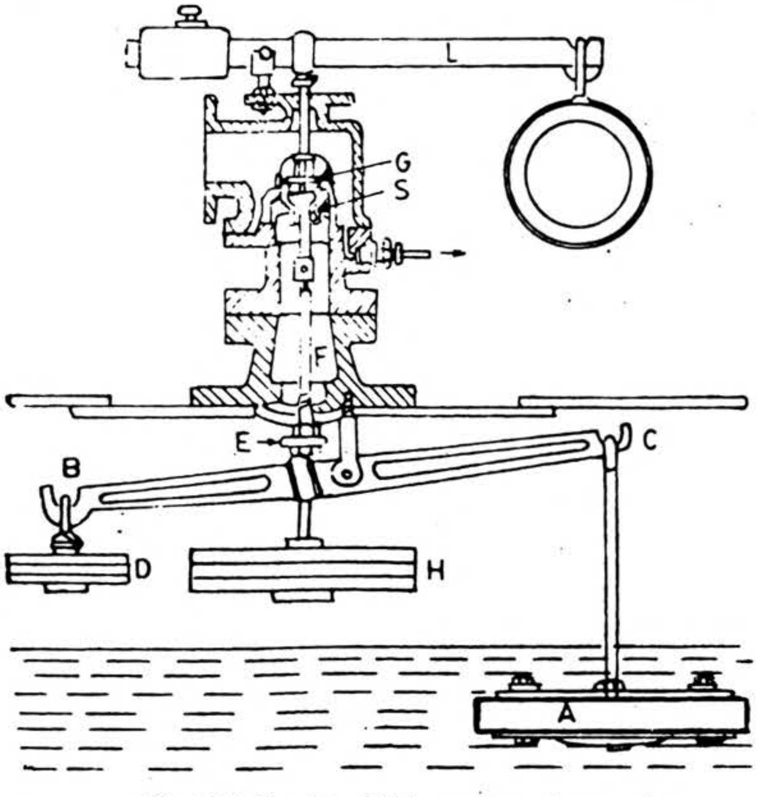


Fig. 8-13. Combined high pressure steam and low-water safety valve.

them to follow the variation in water level, balance weights *D* must be placed on the opposite end of the lever *BC*. The weight of the float *A* acting through the leverge provided, should be sufficient to lift the balance weight *D* and the hemispherical valve G. When the water level falls and the float *A* is sufficiently uncovered, the balanced weight *D* will not be sufficient to balance the float *A* and the float will descend (float being heavier than the balance weight), causing the lever *BC* to move on its pivot. On the lever there are two projections one on the front and the other on the back of a boss on the lever through which the rod *F* passes. The descent of the float causes these projection on the lever to come in contact with the collar *E* fixed to the rod *F*, and the hemispherical valve *G* is lifted and steam escapes, giving the boiler attendant due warning of fall in water level.

When the projections on the lever *BC* are clear of the collar *E* in the position shown in fig. 8-13 i.e., under normal working conditions, the high pressure steam valve *S* acts as an ordinary valve loaded partly by the dead weights *H* and partly by the loaded lever *L* (combined dead weight and lever safety valve). The low-water hemispherical valve *G* is held down against the steam pressure by dead weight *H* secured to a rod *F* connected to valve *G*. When the steam pressure rises too high, both valves will rises as one, permitting a free escape of steam outside of the boiler house so as to discharge surplus steam into the atmosphere.

8.8.3 Water level Indicator : The most satisfactory water-level indicator is the glass tube water gauge. It makes the water-level in the boiler visible from the boiler room floor. The gauge is a glass tube, the lower end of which communicates with the water space of the boiler and the upper end with the steam space. There are usually two gauges provided on each boiler, one placed at the left hand side of the boiler front, and the other at the right hand side. Where the boiler drum is situated at considerable height from the floor, the water gauge is often inclined in order to make the water level visible from any position.

A common form of glass tube-gauge is shown in fig. 8-14. This is Hopkinson's absolute water gauge. *AA* is the front end plate of the boiler and *WW* is the water level. *G* is a very hard glass tube indicating water level and is connected to the boiler plate through stuffing boxes in hollow gunmetal casting having flanges *F* for bolting to the plate. There are two cocks *C* for controlling the passages of water and steam from the boiler. When these cocks are open, the water stands in the glass tube at the same level as in the boiler. A third cock *B*, called a blow-through cock, is ordinarily closed and is for keeping the passages clear by frequent blowing through.

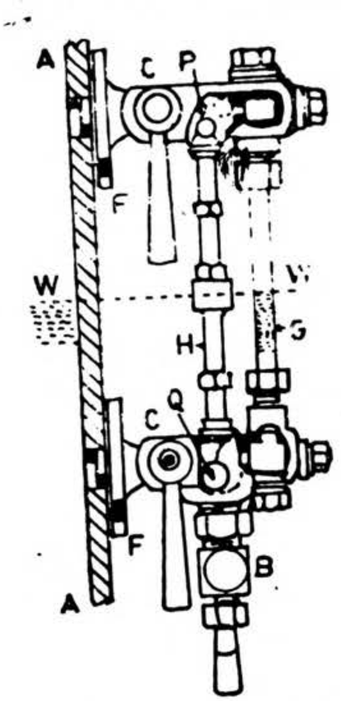


Fig. 8-14. Water level indicator.

In the gauge shown in fig. 8-14, provision is made for automatically shutting off the steam and water supply to the glass tube when the glass tube gets broken. Upper and lower stuffing boxes are connected by hollow column *H*. Balls *P* and *Q* are in positions shown in normal working condition. In case the glass tube gets broken, the rush of water from bottom passage and steam from top passage carries the balls *Q* and *P* in the positions shown dotted and shuts off the water and steam. Then the attendant can safely close the cock *C* and replace the broken glass tube.

In some cases the construction is simplified by removing the ball P and the hollow column *H*. In this gauge only water will be shut off when the glass tube is broken, but there is much less danger from the rush of steam from the top than from a rush of water from the bottom because the water as soon as it escapes into atmosphere flashes into steam, the volume of which is much greater than the volume of steam issuing from the top of gauge.

In some large modern boilers, hydraulic and electric-operated water-level indicators are used. They are located at the operating floor level from where the water level can be more easily observed. For high pressure boilers, the water gauge is made with flat prismatic glass. The inner surface of the glass is grooved to form prisms. When the grooves are filled with water the appearance is dark, but when they are above water level they present a silvery appearance. The contrast is marked as to make reading easier from a distance.

8.8.4 Fusible Plug : The crown of the furnace of some boilers is fitted with a plug held in position by fusible metal or alloy. This plug under normal conditions is covered with water in the boiler which keeps the temperature of the fusible metal below its melting point. But when the water level in the boiler falls low enough to uncover the top of the plug, the fusible metal quickly melts, the plug drops out, and the opening so made allows the steam to rush into the furnace. The steam, thus, puts out the fire or gives warning that the crown of furnace is in danger of being overheated.

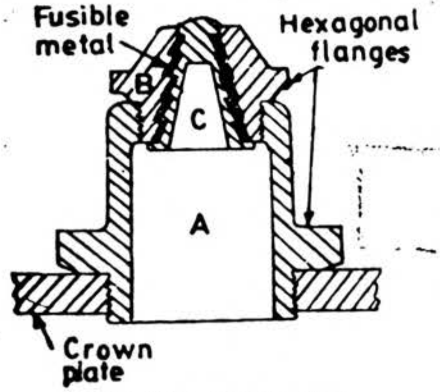


Fig. 8-15. Fusible plug.

Figure 8-15 illustrates a common form of fusible plug. *A* is a hollow gunmetal plug screwed into the crown plate. *B* is a second hollow gunmetal plug screwed into the plug *A*, and *C* is a third hollow gunmetal plug separated from plug *B* by fusible metal. The inner surface of *B* and the outer surface of *C* are grooved as shown so that when the fusible metal is poured in, the plugs *B* and *C* are locked together. Hexagonal flange is provided in the base of plug *A* so that it can be removed by using a spanner. There is another hexagonal flange on *B* for fixing or removing plug *B*.

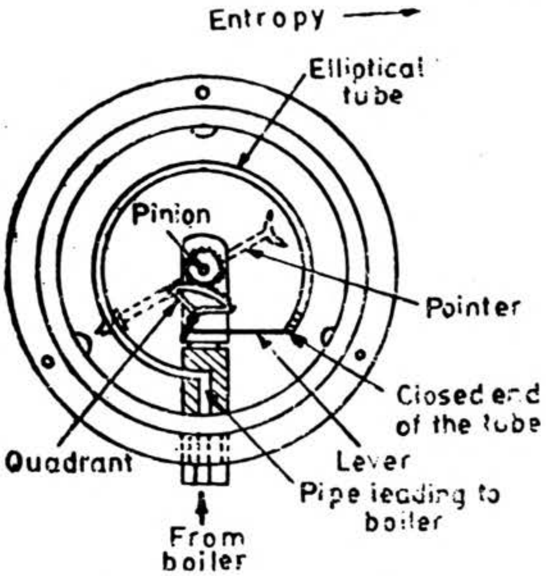


Fig. 8-16. Bourdon pressure gauge.

8.8.5 Steam Pressure Gauge : Each boiler must have a steam pressure gauge to show (or read) the pressure of steam in the boiler. The gauge is usually mounted on the front top of the shell or the drum. Its dial is graduated to read the pressure in kN/m^2 or KPa above atmospheric pressure. The most common type of pressure gauge used is the Bourdon pressure gauge.

Figure 8-16 is a single-tube Bourdon gauge with dial removed to show the interior mechanism. The essential feature of a Bourdon pressure gauge is the elliptical spring tube which is made of a special quality of bronze and is solid drawn. The one end of the tube is closed by a plug and the other end is connected to steam space of the boiler. The closed end of tube is attached by links and pins to a toothed quadrant, which in turn meshes with a small pinion fitted on the central spindle. When steam pressure is supplied to interior of the elliptical tube, it tends to assume a circular cross-section, but before the tube can do so it must straighten out. This tendency to straighten moves the free end (closed end), turning the spindle by lever and gearing (pinion and quadrant), and causing the pointer to move and register the pressure on a graduated dial (not shown in the figure).

The movement of the free end of the tube is proportional to the difference between external and internal pressures on the tube. Since the outside pressure on the tube is atmospheric, the movement of the free end is a measure of the boiler steam pressure above atmospheric pressure, i.e., gauge pressure. The steam pressure gauge should be graduated to read atleast 1½ times the set pressure of safety valve.

The gauge is connected to the boiler through U-tube siphon which is connected to the steam space of the boiler.

8.8.6 Feed Check Valve : The feed-water pipe carrying water from the feed pump usually enters the boiler in the water space of the boiler. A valve is placed in the feed pipe to control or regulate the flow of water into the boiler. The valve is attached directly to the boiler front. It is a non-return valve which permits flow of water in one direction only and automatically prevents the back flow of water from the boiler when the feed water pump is not working. The amount of water entering the boiler can be adjusted by controlling the lift of the valve. This valve is known as feed check valve or boiler feed valve.

A common design of feed check valve is shown in fig. 8-17. The lift of the check valve *C* is controlled by screwing down valve *V*. It is very important that check valve *C* is kept in perfect working condition. It can be cleaned and reground by closing the valve *V* even when the boiler is working. The flange is bolted to the front end of the boiler shell, at a point from which an internally perforated pipe leads (takes) the feed water to the boiler and distributes it near the working level of water in boiler.

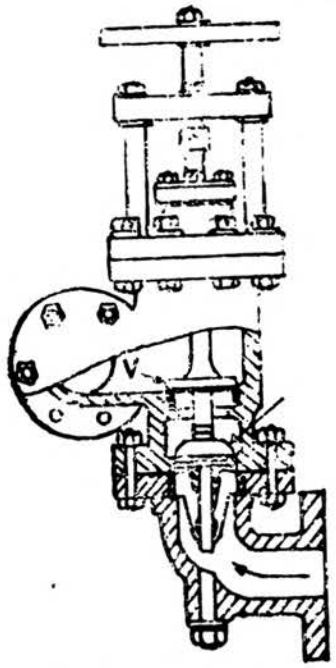


Fig. 8-17 Feed check valve.

8.8.7 Junction Valve or Stop Valve : A valve place directly on a boiler and connected to the steam pipe which carries steam to the engine is called a junction valve. The valve is necessary for purpose of shutting off steam when not required. A valve placed in the steam pipe, leading (taking) steam to the engine and placed near the engine is called a stop valve, but junction valves are also very frequently called stop valves.

Stop or junction valves are operated by hand, and their function is to regulate the amount of steam and to shut it off altogether if required. There is no essential difference between the construction of a junction valve and that of a stop valve.

The common type of stop valve is shown in fig. 8-18. When used as a junction valve, the lower flange is bolted to the boiler at the highest point of the steam space. The valve seat *S* is screwed into the valve body by the aid of lugs *L* cast on its interior. The valve disc *B* has a renewable disc seat *D*.

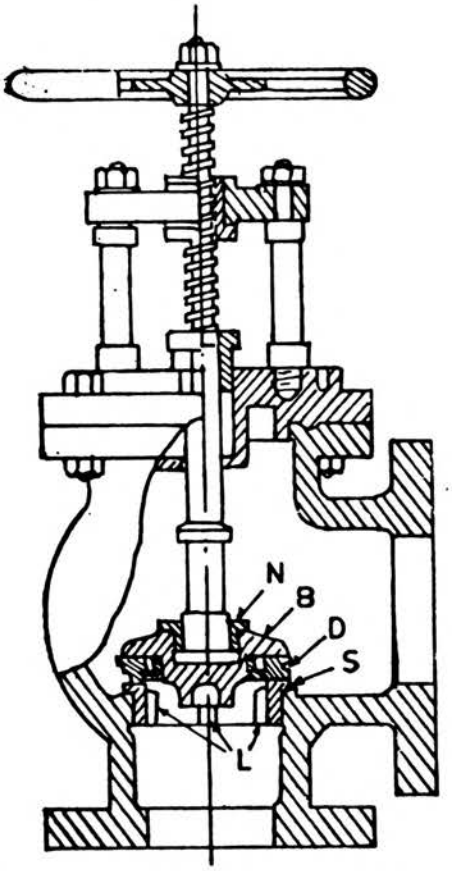


Fig. 8-18. Steam stop valve.

The valve disc *B* is connected with the spindle by the nut, *N*, the lower edge of which comes in contact with a collar on the end of the spindle. The spindle as it is raised or lowered, carries the disc with it but is free to rotate within the disc.

The spindle passes through a gland and stuffing box fixed in the cover of the valve body. The upper portion of the spindle is threaded and passes through a nut in a cross-head or yoke carried by two pillars, which are screwed into the cover of the valve body as shown. By turning the hand wheel fitted on the spindle, the valve spindle is raised or lowered.

8.8.8 Blow-off Cock : The blow-off pipe is attached at the lowest point of the boiler for the purpose of emptying the boiler when necessary, and for discharging the loose mud and sediment deposited from the feed water at the lowest point to which water circulates. A valve or a cock, known as blow-off cock, is placed on this pipe which can be opened to blow-off the dirt and sediment whenever necessary. Arrangements for automatic blow-off instead of manual are also available.

When several boilers are arranged to discharge into the same waste pipe, each blow-off cock should have with it an isolating valve which will prevent the discharge of one boiler from entering into another.

8.8.9 Manholes : These are openings on the boiler shell at suitable locations with covers. These openings allow a man to enter inside the boiler for inspection, cleaning and repairing.

Manholes are of oval shape, 40 cm x 30 cm is size. Due to oval shape, it is possible to fit the manhole cover or manhole door from inside of the manhole. This door is secured in position by bots and bridge bars.

8.9 Boiler Accessories

Most of the boilers are fitted with accessories. The major functions of boiler accessories are :

.. to increase the efficiency of the boiler plant, and

.. to help in the smooth working of the boiler plant.

The principal steam boiler accessories attached to modern boilers are :

- Feed water pump,
- Economiser,
- Air pre-heater.
- Injector,
- Superheater, and

8.9.1 Feed Water Pump and Injector : The **feed pump** is used to deliver feed water to the boiler, and it is required to supply a quantity of water atleast equal to that converted into steam and used by the engine. Feed Pumps may be either reciprocating or rotary pumps.

The feed pump is sometimes worked from the engine direct or from the shaft by an eccentric attached to the plunger. Pumps are also worked independently by using steam directly from the boiler. Such pumps are called direct-acting pumps.

A well-known form of a Duplex direct-acting reciprocating pump is shown in fig. 8-19. In this steam pump there are two simple steam engine cylinders placed side by side. Steam distribution in each cylinder is obtained by means of slide valves. The slide valve is each cylinder steam chest is operated by the cross head on the piston rod of the opposite cylinder, through an arrangement of rods and rocker arms. In Fig. 8-19 one water pump and its steam cylinder is shown in section.

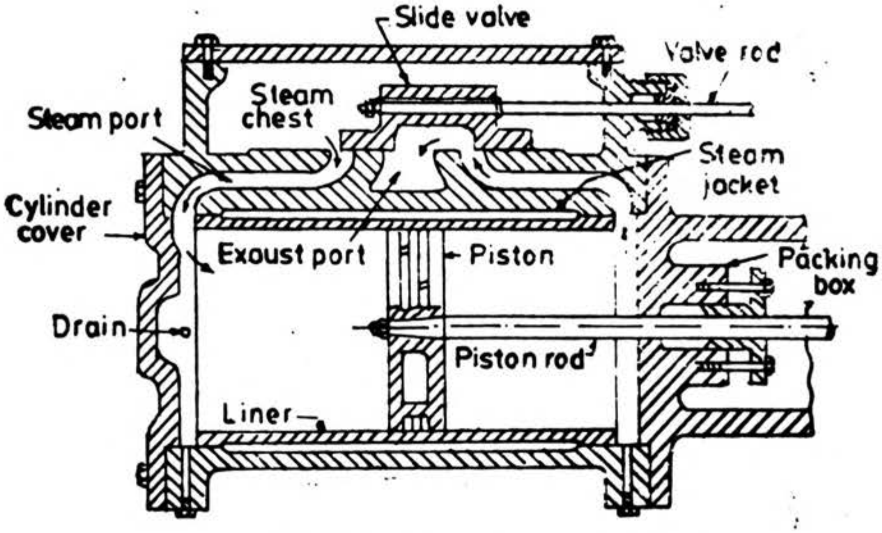


Fig. 8-19. Duplex direct-acting feed water pump.

The feed water pump is generally doube-acting, i.e., it delivers water on each forward and backward stroke. There are suction and discharge valves for each side of the pump plunger. The two pumps work alternatively, thus keeping up a practically continuous flow of water.

Rotary feed pumps are generally of the high speed centrifugal type, driven directly by a small steam turbine or by an electric motor. Rotary pumps may be single-stage or multi-stage according to whether one impeller or more than one impellers are used. In the single-stage pump, the full pressure of water is obtained in one chamber in which the impeller revolves. In a multi-stage pump a number of impellers are keyed on the same shaft, each impeller working in its own chamber or stage. These pumps are also known as turbine pumps.

The **injector** is a simple appliance used to deliver feed water into the boiler using live steam from the same boiler. In this appliance there are no moving parts or plunger, the water being forced into the boiler by the action of steam flowing through a tapering nozzle. Its use is generally limited to small boilers and locomotive boilers. It is sometime fitted as a standby or reserve feed water pump owing to its cheapness and simplicity.

Where an injector is **compared** with a feed pump, the injector is more economical because all the heat in the steam used in the injector is returned to the boiler. The injector therefore works as feed water heater as well as a feed water pump. The boilers in which large quantity of feed water is required to be pumped, the usual practice is to use feed water pumps. Feed water pumps are more reliable and require lesser attention than injectors. The injectors, though economical, are not very reliable.

The working principle of the injector may best be explained by reference to the sketch shown in fig. 8-20. The injector consits of a steam chamber with an outlet in the shape of a convergent nozzle. The position of this nozzle can be adjusted by means of the hand-wheel. By turning the hand-wheel the annular opening between the nozzle and mixing tube can be altered, thereby adjusting the amount of water supplied

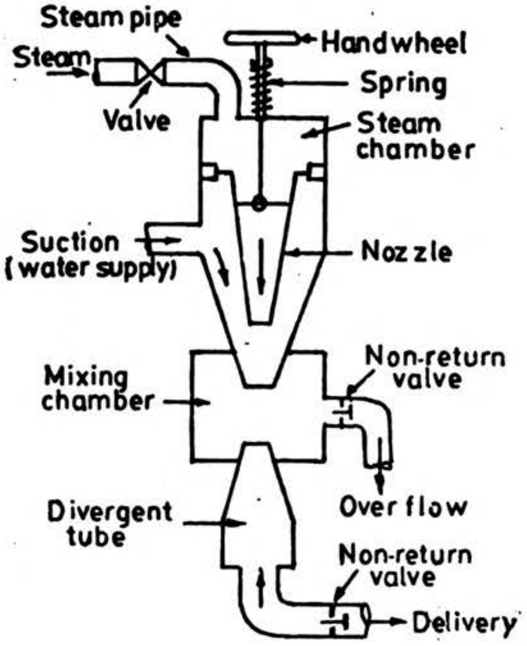


Fig. 8-20. Steam injector.

through the suction pipe. The steam admitted through the steam pipe and discharged from the nozzle, mixes with the water contained in mixing tube, where condensation of steam takes places. The jet then passes through the mixing chamber and enters the divergent tube, which reduces the velocity and increases the presssure of water, the increase of pressure being sufficient to open the non-return valve fitted in the delivery pipe and enters the boiler.

When the injector is started, the pressure in the mixing chamber is above the atmospheric pressure, and water and uncondensed steam pass out through the non-return valve and overflow pipe to atmosphere. As soon as the steam nozzle is brought into its correct position, jet action is established with the result that a pressure below atmospheric pressure (vacuum) is created in the mixing chamber, which causes the non-return valve to close and to stop the overflow to atmosphere. Regular operation of the injector is then established, but when the back pressure exerted upon the non-return valve in the delivery pipe becomes excessive so that it cannot be overcome by the injector, the water again fills the mixing chamber so that the vacuum is lost. Non-return valve in the overflow pipe, therefore, opens again and the water-steam mixture is discharged through the overflow pipe. As soon as normal back pressure is established again, the injector resumes its normal operation.

As a boiler feeding device, the injector has a limitation that hot water cannot be used for pumping because the operating steam must be condensed by mixing with cold water. Thus injector cannot be used where feed water is to be pre-heated or high proportion of condensate (condensed steam) is used again.

8.9.2 Feed Water Heater (Economiser) : A feed water heater (economiser) is an appliance in which the feed water is heated before it is supplied to the boiler. Feed water heaters may be of two classes :

- Those which take the required heat from steam, which is generally the exhaust steam from a non-condensing engine, or the steam used may be fresh steam direct from the boiler, and
- Those which take the heat from the waste furnace gases (flue gases).

Feed water heaters of the second class are called economisers.

One of the major heat losses in a boiler plant is the heat carried away by the flue gases. An **economiser** is a heat recovery appliance placed in the path of the flue gases (between the boiler and chimney) to pre-heat the feed water. An economiser consists of a number of horizontal or vertical tubes through which passes the feed water from the pump on its way to the boiler, whilst the hot flue gases pass over the external surface of the water tubes. The Green's vertical tube economiser (fig. 8-21), is fitted with scrapers which move up and down the water tubes by a mechanical drive, thereby keeping the exterior surface of the tubes free from soot deposits. The horizontal type (figure not given) needs some form of soot blower because of the fact that the

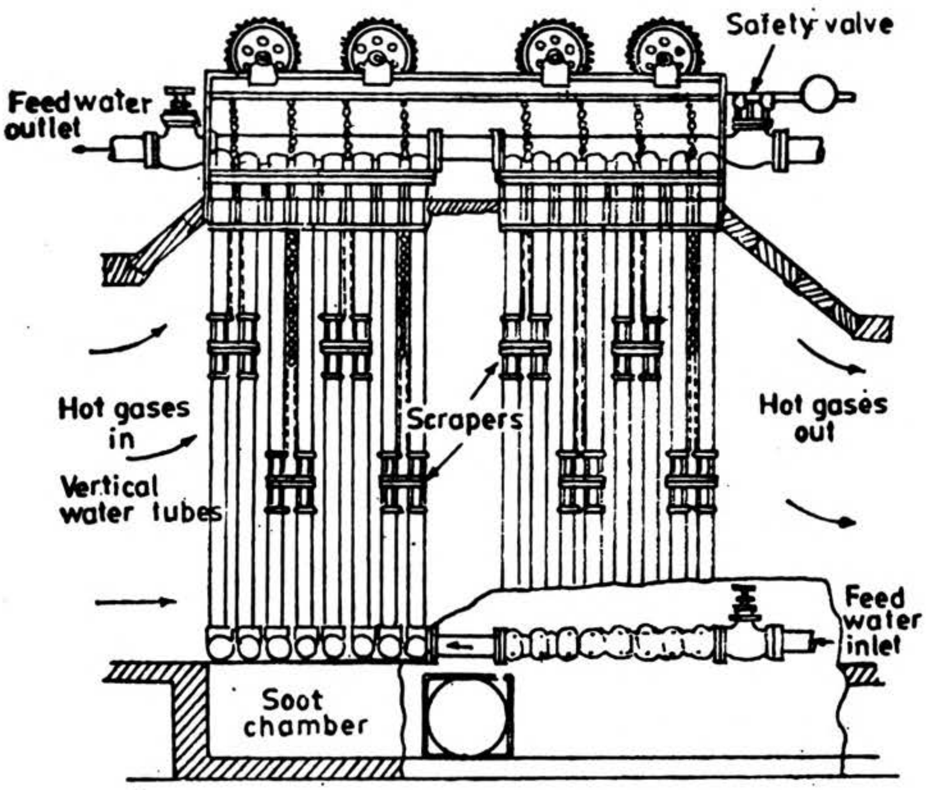


Fig. 8-21. Green's vertical tube economiser.

water tubes of such a type of economiser are fitted with finned casting to increase the area of heat-absorbing surface. Every economiser is fitted with a pressure gauge, a safety valve, a drain valve, an air release valve and two thermometers (for water temperature at inlet and outlet).

By-pass arrangements for the furnace gases and feed water must always be provided so that the economiser may be put out of action when necessary. Figure 8-22 shows an arrangement for diverting the hot gases to pass over the economiser tubes, i.e., when the economiser is in action. The economiser is put out of action by closing the dampers *A* and *B*, and opening damper *C*. In this position of the dampers, the hot furnace gases pass direct to the chimney without passing over the economiser water tubes.

The *advantages* derived by the boiler from the use of an economiser are :

- Feeding the boiler with cold water results in cooling of the boiler metal. By pre-heating the feed water in the economiser, the temperature difference between the different parts of the boiler is reduced, which results in the reduction of stresses due to unequal expansion. Therefore by installing an economiser, *the life of the boiler is increased.*
- The economiser increases the heating surface of the boiler and therefore the *evaporative capacity of the boiler is increased.* Evaporation also becomes rapid and more rapid evaporation results in quick circulation of the water, which makes the heating surface more effective.

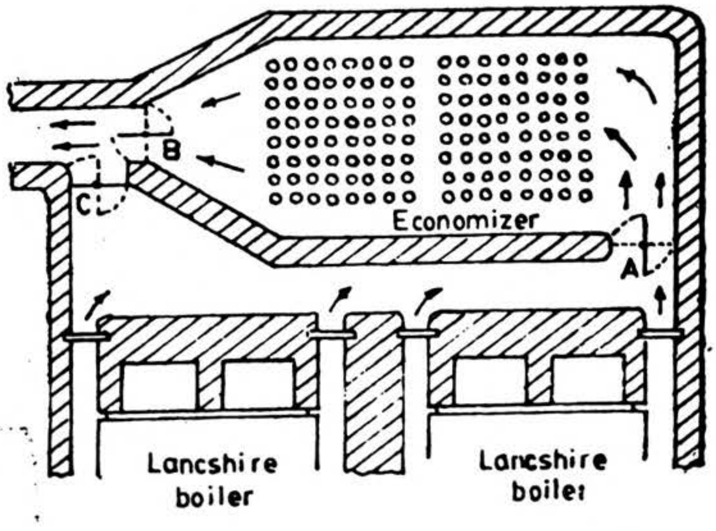


Fig. 8-22. By-pass arrangement of flue gases for two Lancashire boilers fitted with an economiser.

- The heat taken up by the economiser from the flue gases represents a saving of energy which may have been lost to the atmosphere. This saving of energy results in *saving in fuel* and an *increase of the overall efficiency of the boiler plant.*

Problem-1 : *A boiler generates dry saturated steam at a pressure of 11 bar. The feed water is heated by economiser before it is supplied to the boiler. If the feed water enters the economiser at 30°C and leaves at 90°C, find the percentage saving in heat by the use of economiser. Take specific heat of water as 4·187 kJ/kg K.*

Enthalpy of 1 kg of dry saturated steam at 11 bar, H_s = 2,781·7 kJ/kg (from steam tables)

Enthalpy of 1 kg of feed water at 30°C, $h = 4{\cdot}187 \times 30 = 125{\cdot}61$ kJ/kg

∴ Net heat required to produce 1 kg of dry saturated steam at 11 bar from feed water at 30°C = H_s, − h = 2,781·7 − 125·61 = 2,656·09 kJ/kg.

Net heat required to produce 1 kg of dry saturated steam at 11 bar from feed water at 90°C (instead of at 30°C) = 2,781·7 − 90 × 4·187 = 2,781·7 − 376·83 = 2,404·87kJ/kg.

∴ Saving in heat by using feed water at 90°C instead of at 30°C (i.e., by the use of economiser) = 2,656·09 − 2,404·87 = 251·22 kJ/kg.

$$\therefore \quad \text{Percentage saving in heat} = \frac{251{\cdot}22}{2{,}656{\cdot}09} \times 100 = \mathbf{9{\cdot}46\%}$$

8.9.3 Air Pre-heater : An air pre-heater is another appliance which enables to recover heat from the flue gases. It is installed between the economiser and the chimney. The air required for the purpose of combustion is drawn through the pre-heater where its temperature is raised. It is then passed through ducts (pipes) to the furnace. The air is passed through the tubes of the pre-heater internally whilst the hot flue gases are passed over the outside of the tubes.

The use of an air pre-heater in a boiler results in certain *advantages listed as under :*

.. Pre-heated air gives higher furance temperature, which results in more heat transfer to the water and steam per kg of fuel.

.. Better combustion conditions are achieved, as the hot air tends to accelerate the chemical reaction between the oxygen and the inflammable constituent of the fuel (increases the percentage of CO_2 in flue gases).

.. Pre-heated air also tends to result in a short and more stable flame which reduces smoke production.

.. It often enables a low grade coal to be burnt with less excess air.

.. Use of pre-heater increases the efficiency of the boiler plant.

8.9.4 Superheaters : Superheating is effected by passing the boiler steam through a nest of steel tubes bent to U-form and expanded into mild steel boxes called headers. The whole arrangement is known as superheater. The use of a superheater enables the wet steam from the boiler to be completely dried and raised in temperature at constant pressure.

Superheaters are generally located in the path of furnace gases so that heat is recovered by the superheater from the hot gases. When the steam is superheated in this manner by the flow of hot gases, the superheater is called a *convection superheater.* Superheaters are sometimes placed in one or more walls of the boiler furnace where superheater tubes receive heat by direct radiation from the fire. Such a superheater is called a *radiant superheater.* This type of superheater is generally used where high degree of superheat is desired.

In large boilers, superheater may be an independent unit having its own furnace independently fired. Superheaters with separate furnaces are known as *separately fired or portable superheaters.*

Superheaters for water-tube boilers can be further classified with reference to their position in relation to the water tubes. If the superheater is placed in the space over the water tubes, it is termed *over-deck* (fig. 8-9). If it is between the water tubes located near the furnace, it is termed *inter-deck* and when placed between the banks of water tubes, it is termed *inter-bank.*

A form of superheater used with stationary boilers, specially those of Lancashire type is shown in fig. 8-23. This superheater consists of two mild steel boxes or heaters from which hang groups of solid drawn steel tubes bent to U-form as shown in fig. 8-23. The superheater is placed at the back of boiler where the temperature of flue gases is generally not less than 560°C.

All superheaters should be designed and constructed so as to give rapid transfer of heat from flue gases to the steam, be easily cleaned, and be free from the danger of being burnt out. When the temperature of the furnace gases does not exceed 725 K there is no danger of burning of tubes, but for higher temperatures, arrangement must be made to protect the tubes when no steam is passing through them, as for instance, when the prime mover supplied with steam is stopped for a short time.

The methods adopted for the purpose of protecting superheater unit from overheating are :

— Flooding (filling) the superheater with water from the boiler, this water being drained out before the delivery of steam to the prime mover is again started.

— Diverting the hot gases, or stopping their flow over the superheater.

The second method, i.e., diverting the hot gases, is adopted in the superheater shown in fig. 8-23. The superheater is put out of action by turning the damper upward in the vertical position. In this position of the damper, the gases pass directly into the bottom flue without passing over the superheater tubes.

Figure 8-23 also shows how the steam pipes may be arranged so as to pass the steam through the superheater or direct to the main steam pipe. When the steam is taken from the boiler direct to the main steam pipe, the valves *A* and *B* are closed and valve *C* is opened. When the steam is passed through the superheater, i.e., when the superheater is in action the valves *A* and *B* are opened and valve *C* is closed.

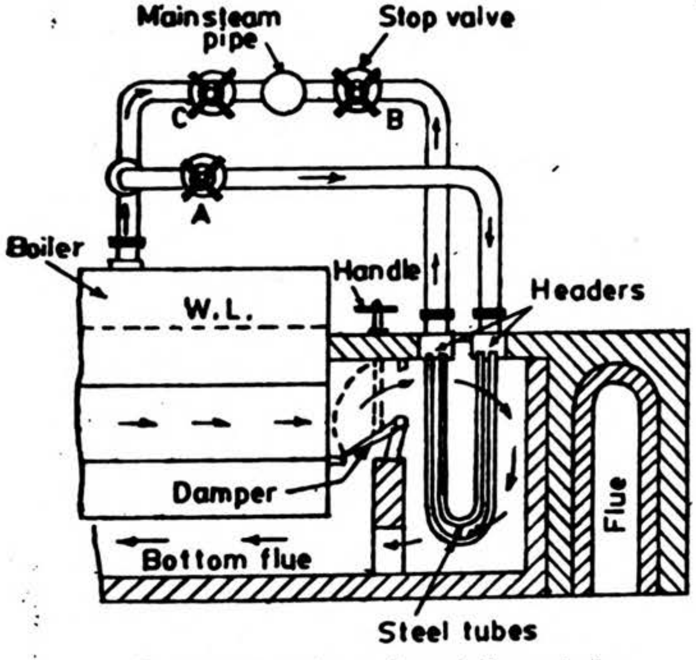


Fig 8 23. Superheater for a stationary boiler.

Problem-2 : *A boiler generates steam at a pressure of 8 bar and 0.8 dry. The steam after leaving the boiler stop valve enters the superheater where its temperature is raised* *to 200°C at constant pressure. Calculate the heat received by steam in the superheater per kg of steam. Take specific heat of superheated steam at constant pressure as 2·3 kJ/kg K.*

From steam tables, at 8 bar,

$$h = 721{\cdot}11 \text{ kJ/kg}, \ L = 2{,}048 \text{ kJ/kg and } t_s = 170{\cdot}43\text{°C}.$$

Enthalpy of *wet* steam *entering* the superheater

$$= h + xL = 721{\cdot}11 + 0{\cdot}8 \times 2{,}048 = 2{,}359{\cdot}51 \text{ kJ/kg}.$$

Enthalpy of superheated steam *leaving* the superheater

$$= h + L + k_p(t_{sup} - t_s) = 721{\cdot}11 + 2{,}048 + 2{\cdot}3(200 - 170{\cdot}43) = 2{,}837{\cdot}12 \text{ kJ/kg}.$$

∴ Heat received by steam in the superheater = 2,837·12 − 2,359·51 = **477·61 kJ/kg.**

8.10 Fittings for Separating Water Particles from Steam

Boilers are fitted with an anti-priming pipe, water separator and steam trap to separate the water particles going along with the steam before being supplied to the engine.

When a boiler is generating steam rapidly, particles of water are thrown up into the steam from where they are carried away by the steam to the engine. This is known as 'priming'. An anti-priming pipe is a device which prevents the carrying away of water particles with the steam. It is fitted to the boiler shell just above the steam space and underneath the steam stop valve. It is an iron box with closed ends. Its upper half is perforated with a number of slots (rectangular holes) through which the steam enters on the way to the main pipe. The steam current is broken up while it passes through the slots, causing the heavy water particles to separate out and fall back into the boiler through a small hole provided at bottom of the anti-priming pipe.

The object of a steam separator is to remove as far as possible fine particles of water carried along with steam on its way from the boiler to the engine. It is placed on the main steam pipe line leading from the boiler to the engine and as close as possible to the engine. A common type of separator contains baffle plates. The steam striking the baffle plates is suddenly deflected so that the direction of its flow is changed and the velocity of steam is reduced. The particles of water, due to their greater mass and inertia, strike the baffle plates and fall to the bottom of the separator.

Steam traps are devices used to collect and automatically discharge the water resulting from partial condensation of steam without allowing any steam to escape. The trap is so located that water from the condensation of the steam in the steam pipe flows by gravity to it. The rising level of water in the trap eventually causes a valve to open through a simple mechanism and the water is discharged through the opening. As soon as the water is discharged, the valve closes automatically so that the steam which follows the water can not escape with it. Steam traps are divided into two general classes :

- .. Traps which depend for their action on the expansion of metals under heat, called *expansion steam traps,* and
- .. Traps in which the discharge of condensed steam is controlled by floats or buckets called *bucket steam traps or float steam traps.*

8.11 Pressure Reducing Valves

When steam is required at a lower pressure than that supplied by a boiler, the steam is passed through a pressure reducing valve whose function is to maintain a constant reduced pressure on its delivery side. Low pressure steam is required in industry for process or heating purposes.

It is more difficult to keep pressure uniform with water-tube boilers, which hold a comparatively small quantity of water. Therefore, it is a practice to work such boilers at a pressure higher than that for which the engine is designed. The steam is then passed through a reducing valve on its way to the engine. The reducing valve maintains a constant reduced pressure on the engine side of the valve, while the higher pressure on the boiler side may be variable.

The principle upon which all reducing valves work is the principle of the throttle valve.

8.12 Scale Cleaners and Soot Blowers

A considerable amount of scale forming material may be removed from the feed water by using feed water heaters or by chemical treatment of feed water. However some scale will be deposited in boilers which requires some mechanical means for its removal. The scale can be removed from the accessible parts or parts of the boiler drum or shells by means of hammer and blunt chisel. However, scale formed in the water tubes of the boiler cannot be removed so easily and needs some mechanical tubes cleaners. The water tubes are freed from scale by a motor driven by pneumatic, electrical or water power. The loosened scale is carried away from the water tubes by the air or water used to drive the cutter.

Tubes and other surfaces exposed to the flue gases will have a layer of soot deposited on them. Such deposits reduce the rate of the heat transfer through the tubes. One method of cleaning the soot deposited on the surface of the tubes is by the use of *soot blowers*. The cleaning agent is steam issuing from nozzles.

8.13 Boiler Draught

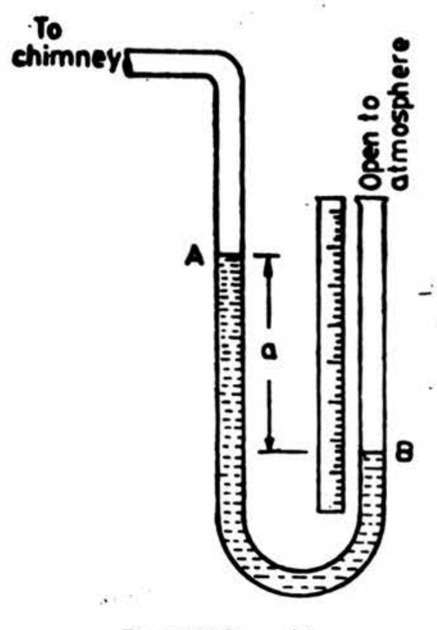


Fig. 8-24 Draught gauge.

The rate of steam generation in a boiler depends upon the rate at which the fuel is burnt. The rate of fuel burning depends upon the difference in the static pressure available to produce the flow of air through the bed of fuel on the furnace grate to the chimney. This difference in static pressure is known as *draught*. The object of producing the draught in a boiler is (i) to provide an adequate supply of air for combustion of the fuel, (ii) to draw (move) the resulting hot gases through the system, and finally (iii) to discharge these gases to atmosphere through the chimney.

Draught is ordinarily measured by draught gauge known as manometer, and its intensity is expressed in mm of water. It is usually measured taking difference in level between the surface of two columns of water in the two legs of U-tube, one leg being connected with the chimney and the other open to atmosphere. Thus, the difference *a* (fig. 8-24) between the water levels at A and B in the two legs of U-tube is the draught in mm of water head. In ordinary chimneys, the draught is about

12 mm of water in small chimneys and about 20 mm in high (tall) chimneys.

The main factors which determine the amount of draught to be provided for are as under :

- – Rate of burning the fuel,
- – Type and condition (size, moisture content, etc.) of the fuel,
- – Method of stoking or burning the fuel,
- – Depth of fuel bed,
- – Resistance in the flue gas circuit created by baffles, flues, smoke and water tubes, superheater, economiser, air pre-heater and dampers, etc.,
- – Design of the combustion chamber or furnace.

8.14 Methods of Producing Draught

Draught may be classified as under :

.. Natural draught produced by a chimney, and

.. Artificial draught produced by a fan or steam jet.

8.14.1 Natural Draught : Natural draught is produced by the boiler chimney and therefore it is also known as *chimney draught*. The draught is produced due to the difference in weight between the column of hot gases inside the chimney and the weight of equal column of cold air outside the chimney. Roughly speaking cold air outside weighs twice as much (volume by volume) as the hot gases inside the chimney, i.e., cold air outside is heavier than the hot gases inside the chimney. The pressure in the chimney is, therefore, less than the pressure of the outside air. As a result of this, the outside cold air will flow through the furnace into the chimney and the hot gases will pass up the chimney. The outside air rushing to the chimney will be utilised for combustion of the fuel in the furnace. The draught is measured by a draught gauge shown in fig. 8-24.

The amount of the draught produced in the boiler depends on the following factors :

- – *Climatic conditions* – draught decreases with the increase of outside air temperature.
- – *Temperature of furnace gases* – draught increases with the increase of furnace gas temperature.
- – *Height of chimney* – draught increases with the increase of height of chimney.

8.14.2 Artificial Draught : When the rate of fuel burning required is to be very high, natural draught is not sufficient and it becomes necessary to provide an artificial draught by some mechanical means. An artificial draught may be produced by fan, blower, or steam jet. If the draught is produced by a fan, it is known as *fan draught* or *mechanical draught* and if produced by steam jet, it is known as *steam jet draught.*

Artificial draught reduces the necessary height of chimney and provides a draught that is easily controlled. The artificial draught is of two kinds, namely *forced draught and induced draught.* In the *forced fan draught,* the air passes through the fan before entering the furnace. The air is forced by the fan into the furnace at a pressure higher than that of the atmosphere. In the *forced fan draught,* the fan is placed at or near the base of the chimney. The fan draws the air through the furnace by reducing the pressure in the furnace below that of the atmosphere. This draught is similar in action to the natural draught. It is usually employed when economiser and air pre-heater are provided with the boiler.

In the *forced steam jet draught,* the draught is produced by placing the steam jet in the ash pit which is situated under the fire grate of the furnace. In such a case the air will be forced into the furnace by the steam jet and the draught produced will be the *forced draught.* In steam locomotive the draught is produced by the exhaust steam blast in the smoke box (see fig. 8-6), which reduces the pressure of the escaping gases below the atmospheric pressure and greatly increases the air-flow through the grate, smoke tubes and up the chimney. The exhaust steam from the steam locomotive engine (non-condensing) is used for producing draught. The system is simple and cheap. In this system the draught is automatically adjusted to suit the requirements. In the *induced steam jet draught,* the draught is produced by steam jet issuing from a nozzle placed in chimney. This will drag with it the surrounding gas and produce a partial vacuum in its neighbourhood. The air is, thus, drawn through the furnace by the difference in pressure created by the steam jet. The draught produced by the steam jet in this manner is of the *induced* type.

Balanced draught is the combination of the forced and induced draught systems. The forced draught supplies air for burning the fuel. The induced draught removes the gases from the furnace. A balanced draught system minimises the losses due to inward and outward leakages and is the best compromise between induced and forced draughts.

Forced and induced draughts can be compared as under :

.. Forced draught system requires less fan power, since the fan has to handle only cold air.

.. Forced draught gives better control than induced draught. With forced draught the air penetrates (enters) into the fire-bed better than that with the induced draught and therefore, the rate of burning of fuel is more.

.. With forced draught all leakages are outward and therefore, there is a serious danger of blow out, if the fire doors are opened when the fan is operating. Care, therefore, has to be taken while opening the fire doors.

.. With induced draught all leakages of air are inward and, therefore, heavy air infiltration (leakage) will occur, reducing the available draught unless all brickwork, joints etc., in the flue system are kept in good condition. -

.. The fan power required with induced draught is greater than forced draught, since in induced draught fan has to handle a large amount of the hot flue gases.

.. With induced draught there will be an inrush of cold air into the furnace when the furnace doors are opened for firing or cleaning.

Artificial draught and natural draught : Induced draught artificially produced is better and more economical than the chimney draught. It is more widely used on large boiler plants. As the temperature of flue gases can be lowered with induced draught (artificial draught), the boiler efficiency is higher with induced draught than with natural or chimney draught. Induced draught created by a steam jet is not very economical from the point of view of steam consumption.

The artificial draught system has the following *advantages* over the natural or chimney draught :

— Higher evaporative power of the boiler,

— Burning of low grade fuel is possible,

— Proper control of combustion is possible,

— Smoke is greatly reduced,

— Chimney height required is less, and

— Over-all efficiency of the plant is higher.

The *disadvantages* of artificial draught are as under :

— Installation cost is higher,

— Running cost is higher, and

— Maintenance cost is more.

8.15 Chimney Height

The amount of natural draught produced in a boiler depends on the height of the chimney and the difference between temperature of the hot flue gases leaving the boiler and that of the outside cold air. A relationship between the height of a chimney and the draught it produces in terms of temperature of outside air and the average flue gas temperature, can be deduced in the following manner :

Let m = mass of air used in kg to burn 1 kg of fuel,

T = average absolute temperature of chimney gases in K, and

T_a = absolute temperature of the air outside the chimney in K.

Thus, mass of flue gases produced = $(m + 1)$ kg per kg of fuel burnt.

The volume of chimney gases produced may be taken as equal to the volume of air supplied. Since the volume of the solid or liquid fuel burnt is so small as compared with the volume of air supplied that it may be neglected.

Taking the volume of 1 kg of air at 0°C and 760 mm of Hg as 0·7734 m^3,

Volume of gases at 0°C = $0{\cdot}7734\ m$ m^3 per kg of fuel burnt.

Since volume of gas is proportional to its absolute temperature (by Charle's law),

Volume of chimney gases at TK = $\dfrac{0{\cdot}7734\ m \times T}{273}$ m^3/kg of fuel burnt,

and the density of chimney gases at temperature TK

$$= \frac{(m+1)}{0{\cdot}7734m \times \dfrac{T}{273}} = \frac{(m+1)}{0{\cdot}7734\ m} \times \frac{273}{T} \text{ kg/m}^3.$$

Similarly, the density of atmospheric air at T_aK

$$= \frac{m}{0{\cdot}7734\ m \times \dfrac{T_a}{273}} = \frac{1}{0{\cdot}7734} \times \frac{273}{T_a} \text{ kg/m}^3$$

Let H in fig. 8-25 be height of the chimney required in metres, measured from the level of furnace grate.

The pressure exerted per square metre at the furnace grate level by a column of hot gas one metre in height = density of the gas.

∴ Pressure exerted by a column of hot chimney gas of H metres height

$$= \frac{\rho A H g}{A} = \rho\, g\, H \text{ N/m}^2$$

$$= \text{density} \times 9{\cdot}81 H \text{ N/m}^2$$

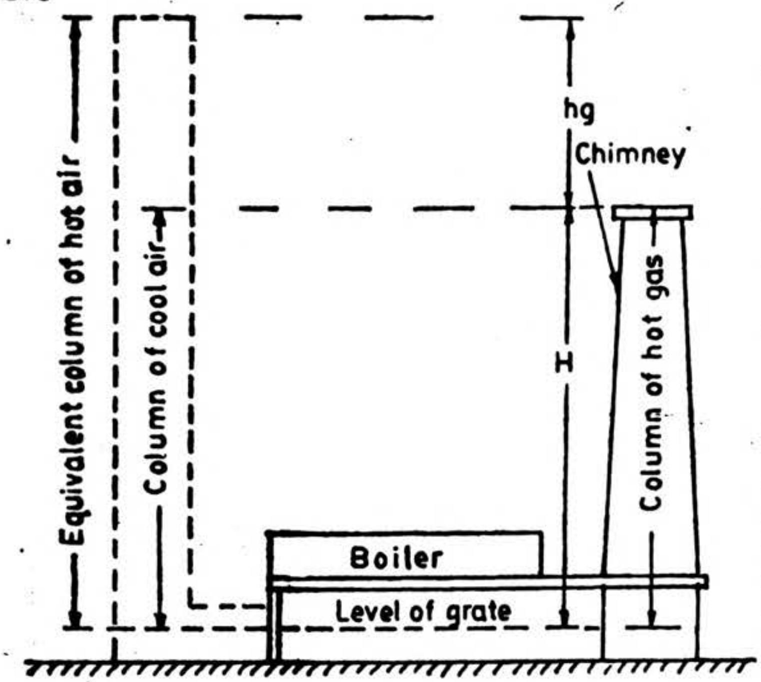


Fig. 8-25. Chimney draught.

$$= \frac{m+1}{0.7734\,m} \times \frac{273}{T} \times 9.81H \text{ N/m}^2$$

Similarly, pressure due to column of outside (cold) air of same area and H metres height

$$= \frac{1}{0.7734} \times \frac{273}{T_a} \times 9.81H \text{ N/m}^2.$$

Let p be the pressure causing the draught in N/m^2. Since the pressure causing the draught is due to the difference of pressure due to column of hot gases within the chimney and the pressure due to an equal column of outside (cold) air,

$$p = \left\{\frac{1}{0.7734} \times \frac{273}{T_a} - \frac{m+1}{0.7734\,m} \times \frac{273}{T}\right\} \times H\ 9.81$$

$$= \frac{273 \times 9.81H}{0.7734}\left\{\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right\} \text{ N/m}^2 \qquad ..(8.3)$$

The draught pressure is usually expressed in terms of mm of water column as indicated by the U-tube water pressure gauge called manometer.

Since the density of water is 1,000 kg/m^3, a water column of 1 metre or 1,000 mm height will exert a pressure of 1,000 × 9.81 N/m^2.

Let h be draught pressure in mm of water, then the draught pressure, p in N/m^2 is given as

$$p = \rho g\ h = 1{,}000 \times 9.81 \times \frac{h}{1{,}000} = 9.81h \text{ N/m}^2$$

$$\therefore\ h = \frac{p}{9.81} \text{ mm of water}$$

Substituting the value of p from the eqn. (8.3), we have

$$h = \frac{273\,H}{0.7734}\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right] \text{ mm of water}$$

$$= 353\,H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right] \text{ mm of water} \qquad ..(8.4)$$

The theoretical draught obtained by calculation from eqn. (8.4) is known as *static draught.* The actual or available draught is less in value than the static draught due to :

.. Frictional resistance to the flow of flue gases in the flue passages,

.. Energy required to impart velocity to flue gases, and

.. Losses in the bends and curves in the flue gas passages.

Problem-3 : *A chimney of 30 m height is full with hot gases at a temperature of 288°C. The outside air temperature is 21°C. If the available draught is 80 per cent of the theoretical draught, calculate the available draught. The air supplied for combustion is 18 kg per kg of fuel burnt. Take the density of air at 0°C and 760 mm of Hg to be 1·293 kg/m³.*

Using eqn. (8.4),

$$h = 353\,H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$$

where h = draught in mm of water,

H = minimum height of chimney in m,

m = quantity of air used in kg per kg of fuel burnt,

T_a = absolute temperature of the outside (cold) air in K, and

T = absolute temperature of chimney flue gases in K.

Here, H = 30 m, m = 18 kg of air per kg of fuel burnt,

$T_a = 21 + 273 = 294$ K, and $T = 288 + 273 = 561$ K.

Substituting the above values in the eqn. (8.4), we get,

$$h = 353 \times 30\left[\frac{1}{294} - \frac{18+1}{18} \times \frac{1}{561}\right] = 16{\cdot}1 \text{ mm of water column}$$

$\therefore$ Available draught = 0·8 × 16·1 = **12·88 mm of water column.**

Problem-4 : *A boiler uses 18 kg of fuel per minute and is supplied with 18 kg of air per kg of fuel burnt. Determine the minimum height of the chimney required to produce a draught of 25 mm of water. The mean temperature of the chimney gases is 315°C and that of the outside air is 27°C.*

Using eqn. (8.4), $h = 353\,H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$

Here, h = 25 mm of water, m = 18 kg of air per kg of fuel burnt,

$T_a = 27 + 273 = 300$ K, and $T = 315 + 273 = 588$ K

Substituting the above values in eqn. (8.4),

$$25 = 353 \times H\left[\frac{1}{300} - \frac{18+1}{18} \times \frac{1}{588}\right]$$

$$= 353 \times H\,(0{\cdot}0033 - 0{\cdot}0016)$$

$\therefore$ H = **41·6 metres** (minimum height of chimney)

Problem-5 : *How much air is used per kg of coal burnt in a boiler having a chimney of 30 m height to produce a draught of 16 mm of water when the temperature of the flue gases in the chimney is 317°C and that of the boiler house is 30°C ?*

Using eqn. (8.4), $h = 353\,H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$

Here, $H = 30m$, h = 16 mm of water,

$T_a = 30 + 273 = 303$ K, and $T = 317 + 273 = 590$ K.

Substituting the above values in the eqn. (8.4),

$$16 = 353 \times 30\left[\frac{1}{303} - \frac{m+1}{m} \times \frac{1}{590}\right]$$

$$\therefore \frac{16}{353 \times 30} = \frac{1}{303} - \frac{m+1}{590\, m}$$

$$\therefore \frac{m+1}{590\, m} = \frac{1}{303} - \frac{16}{353 \times 30}$$

$$\therefore m = \frac{1}{0{\cdot}0561} = \textbf{17·83 kg of air per kg of fuel burnt.}$$

Problem-6 : *Estimate the mean temperature of the flue gases leaving the chimney 30 m high to produce a draught of 16 mm of water column, if 18 kg of air is required per kg of fuel burnt on the grate. The temperature of atmospheric air is 27°C. Take the density of air al 0°C and 760 mm of Hg. as 1·293 kg/m³.*

Using eqn. (8.4), $h = 353\, H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$

Here, h = 16 mm of water, m = 18 kg of air per kg of fuel burnt,

H = 30 m, and T_a = 27 + 273 = 300 K.

Substituting the above values in eqn. (8.4),

$$16 = 353 \times 30\left[\frac{1}{300} - \frac{18+1}{18} \times \frac{1}{T}\right]$$

$$\therefore 16 = 10{,}590\left[0{\cdot}0033 - \frac{1{\cdot}056}{T}\right] = 35{\cdot}265 - \frac{11{,}183}{T}$$

$$\therefore T = \frac{11{,}183}{19{\cdot}265} = 580 \text{ K or } t = \textbf{307°C (temperature of flue gases)}$$

8.16 Maximum Discharge of Hot Flue Gases through the Chimney

The eqn. (8.4) may be modified to express the draught in terms of column of hot gases.

Let h_g (fig. 8-25) be the height of column of hot gases which would produce the pressure p.

Pressure exerted by this column of hot gases, p = density × h_g × 9·81

$$\therefore p = \frac{m+1}{0{\cdot}7734\, m} \times \frac{273}{T} \times h_g \times 9{\cdot}81$$

Substituting this value of p in eqn. (8.3), we get,

$$\frac{m+1}{0{\cdot}7734\, m} \times \frac{273}{T} \times h_g \times 9{\cdot}81 = \frac{273 \times 9{\cdot}81\, H}{0{\cdot}7734}\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$$

$$\therefore h_g = \frac{273\, H}{0{\cdot}7734}\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right] \times \frac{0{\cdot}7734\, m}{m+1} \times \frac{T}{273}$$

Simplifying,

$$h_g = H\left[\frac{m}{m+1} \times \frac{T}{T_a} - 1\right] \quad ..(8.5)$$

The chimney draught is more effective when the chimney will discharge a maximum mass of hot gases in a given time. This condition is produced when the absolute temperature of the chimney gases bears a certain ratio to the absolute temperature of the outside (cold) air.

The mass of gases discharged in a given-time is proportional to the product of its density and velocity of its discharge.

Since, $V^2 = 2gh_g$ i.e. $V = \sqrt{2\,g\,h_g}$

Substituting the value of h_g from eqn. (8.5), we have,

$$V = \sqrt{2\,gH\left[\frac{m}{m+1} \times \frac{T}{T_a} - 1\right]}$$

And since the density of the hot gases is proportional to $\frac{1}{T}$, the mass of the hot gases discharged,

$$M = \frac{A}{T}\sqrt{2\,gH\left[\frac{m}{m+1} \times \frac{T}{T_a} - 1\right]}$$

where A is a constant of proportionality.

Putting B as another constant, the equation may be written as

$$M = \frac{B}{T}\left[\sqrt{\frac{m}{m+1} \times \frac{1}{T_a} - 1}\right]$$

$$= B\left[\sqrt{\frac{m}{m+1} \times \frac{1}{T_a\,T} - \frac{1}{T^2}}\right]$$

Differentiating M with respect to T for maximum discharge,

$$\frac{dM}{dT} = B \times ½ \times \frac{-\frac{m}{m+1} \times \frac{1}{T_a} \times \frac{1}{T^2} + \frac{2}{T^3}}{\sqrt{\frac{m}{m+1} \times \frac{1}{T_a\,T} - \frac{1}{T^2}}}$$

For maximum discharge $\frac{dM}{dT} = 0$,

Therefore, the numerator of the above equation is zero, giving,

$$\frac{m}{m+1} \times \frac{1}{T_a\,T^2} = \frac{2}{T^3}$$

$$\therefore\ \frac{m}{m+1} \times \frac{1}{T_a} = \frac{2}{T}$$

$$\therefore\ T = 2 \times \frac{m+1}{m} \times T_a$$

This shows that maximum mass of hot gases are discharged when,

$$T = 2\left[\frac{m+1}{m} \times T_a\right] \quad ..\ (8.6)$$

If this value of T is substituted in eqn. (8.5),

$$h_g = H\left[\frac{m}{m+1} \times \frac{2 \times \frac{m+1}{m} \times T_a}{T_a} - 1\right]$$

$$= H\ (2 - 1) = H \text{ metres}$$

which means that when maximum discharge takes place, height of the column of hot gases expressing the draught pressure will be equal to the height of the chimney.

From eqn. (8.4), $h = 353\ H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$

Using eqn. (8.6), for maximum discharge, $T = 2\left[\frac{m+1}{m} \times T_a\right]$

Substituting the value of T in eqn. (8.4), the draught in mm of water under the condition of maximum discharge is

$$h = 353\ H\left[\frac{1}{T_a} - \frac{m+1}{m} + \frac{1}{2 \times \frac{m+1}{m} \times T_a}\right] = 353 \times \frac{H}{2\ T_a}$$

$$\text{or } h = \frac{176{\cdot}5\ H}{T_a} \qquad ..(8.7)$$

Problem-7 : *A chimney 30 m high deals with flue gases at 288°C, when the outside air temperature is 21°C. The air supplied for combustion is 18 kg per kg of coal burnt. Calculate :*

(i) the draught in mm of water column produced by the chimney,

(ii) the draught produced in terms of height of column of hot gases in metres,

(iii) the velocity of flue gases in the chimney, if 50% of the draught is lost in friction at the grate and passages, and

(iv) the draught produced in mm of water and the temperature of flue gases under the condition of maximum discharge.

(i) From eqn. (8.4),

$$h = 353\ H\left[\frac{1}{T_a} - \frac{m+1}{m} \times \frac{1}{T}\right]$$

Here, $T_a = 21 + 273 = 294$ K, $T = 288 + 273 = 561$ K

$H = 30$ metres, and $m = 18$ kg of air/kg of fuel burnt.

Substituting the above values in eqn. (8.4),

$$h = 353 \times 30\left[\frac{1}{294} - \frac{18+1}{18} \times \frac{1}{561}\right]$$

$$= \mathbf{16{\cdot}1 \text{ mm of water column}}$$

(ii) From eqn. (8.5), height of column of hot gases,

$$h_g = H\left[\frac{m}{m+1} \times \frac{T}{T_a} - 1\right]$$

Substituting the values in eqn. (8.5),

$$h_g = 30\left[\frac{18}{18+1} \times \frac{561}{294} - 1\right]$$

$= 30 \times 0{\cdot}81 =$ **24·3 metres of hot gases column.**

(iii) As 50% of the draught is lost in friction,

Available draught, $h = 24{\cdot}3 \times 0{\cdot}5 = 12{\cdot}15$ m.

If V is the velocity of gases in m/sec.

then, $V = \sqrt{2gh} = \sqrt{2 \times 9{\cdot}81 \times 12{\cdot}15} =$ **15·45 m/sec.**

(iv) Using eqn. (8.7), the draught in mm of water under the condition of maximum discharge is

$$h = \frac{176{\cdot}5\,H}{T_a}$$

Here, $T_a = 21 + 273 = 294$ K and $H = 30$ m.

$$\therefore\ h = \frac{176{\cdot}5 \times 30}{294} = 18 \text{ mm of water}$$

Using eqn. (8.6) for maximum discharge, temperature of the flue gases,

$$T = 2\left[\frac{m+1}{m} \times T_a\right]$$

$$= 2\left[\frac{19}{18} \times 294\right] = 620 \text{ K or } t = \mathbf{347^\circ C} \text{ (temperature of flue gases)}$$

8.17 Power Required to Drive the Draught Fan

The fan power or air power is the power required to move air or gas by the fan.

If p = draught pressure in bar (equivalent to draught in mm of water, h), and

v = volume of air or gas handled by the fan per second in m^3, then,

The work done by the fan = $10^5 \times p \times v$ Joules per second, and hence

$$\text{Air power} = \frac{10^5 \times p \times v}{1{,}000} = \text{kJ/sec or kW}$$

If η = efficiency of the fan,

$$\text{Then, power required to drive the fan} = \frac{10^5 \times p \times v}{1{,}000 \times \eta} \text{ kW} \qquad \text{...(8.8)}$$

Volume v of air handled by a *forced draught fan* can be calculated as under :

If m = mass of air supplied per kg of fuel burnt, and

M = mass of fuel burnt per second,

then the quantity of air supplied per second = $m \times M$ kg.

If T_a = absolute temperature of outside (cold) air,

T_o = absolute temperature of air at N.T.P. (273 K), and

V_o = volume of 1 kg of air at N.T.P. (0·7734 m^2),

$$\text{Then, } v = m \times M \times v_o \times \frac{T_a}{T_o} \text{ m}^3\text{/sec.}$$

Substituting the value of v in eqn. (8.8),

Power required to drive a forced draught fan, i.e., Brake

$$\text{Power of motor} = \frac{10^5 \times p \times m \times M \times v_o \times T_a}{1{,}000 \times \eta \times T_o} \text{ kW} \qquad ..(8.9)$$

Volume v of flue gases handled by an *induced draught fan* can be calculated as under : The mass of flue gases formed per kg of fuel = (m + 1) kg. Therefore, for M kg of fuel burnt per second, mass of flue gases handled by induced draught fan per second = $M\ (m + 1)$ kg

If T = absolute temperature of flue gases and assuming the density of flue gases to be that of air, then

$$v = (m + 1)\ M \times v_o \times \frac{T}{T_o} \text{ m}^3\text{/sec.}$$

Substituting the value of v in eqn. (8.8),

Power required to drive an induced draught fan, i.e. Brake

$$\text{Power of Motor} = \frac{10^5 \times p \times (m+1)\, M \times v_o \times T}{1{,}000 \times \eta \times T_o} \text{ kW} \qquad ..(8.10)$$

Comparison of fan power for induced and forced draughts can be made from the following expression when both the fans have the same efficiency and produce equal draught :

$$\frac{\text{Power required for induced draught fan}}{\text{Power required for forced draught fan}} = \frac{m+1}{m} \times \frac{T}{T_a} \qquad ..(8.11)$$

where T = absolute temperature of flue gases handled by induced draught fan,

T_a = absolute temperature of air handled by forced draught fan, and

m = quantity of air used in kg to burn 1 kg of fuel.

Problem-8 : *The following data was obtained during a test on two boilers working under similar conditions, except that the draught in the first boiler was produced by an induced draught fan and in the second boiler by a forced draught fan. Boiler house temperature was 20°C.*

Mean temperature of flue gases leaving the boilers	*.. 190°C*
Air supplied per kg of fuel burnt	*.. 19 kg*
Density of air under given conditions	*.. 1·205 kg/m^3*
Density of flue gases at the specified temperature	*.. 0·769 kg/m^3*
Combustion rate	*.. 150 kg of fuel per hour*
Fan draught produced in each case	*.. 75 mm of water*
Efficiency of fan in both cases	*.. 50%*

Calculate the brake power of fan in each case and make a comparison of fan power expended (used) in each case. Allow 20% leakage air in case of induced draught system and 10% in case of forced draught system.

Induced draught fan :

Mass of fuel burnt per second $= \frac{150}{3,600} = 0{\cdot}0417$ kg.

Mass of gases produced per kg of fuel = 19 + 1 = 20 kg.

$\therefore$ Mass of gases handled by the fan per second $= 20 \times 0{\cdot}0417 = 0{\cdot}834$ kg.

Volume of gases to be handled by the induced draught fan per second, including leakage air,

$$v = \frac{0{\cdot}834}{0{\cdot}769} \times 1{\cdot}2 = 1{\cdot}3 \text{ m}^3.$$

Draught pressure, $p = \frac{75}{13{\cdot}6} \times \frac{1}{750} = 0{\cdot}00753$ bar

Using eqn. (8.8), power of the motor to drive induced draught fan

$$= \frac{10^5 \times p \times v}{1,000 \times \eta} = \frac{10^5 \times 0{\cdot}00753 \times 1{\cdot}3}{1,000 \times 0{\cdot}5} = \mathbf{1{\cdot}9578 \text{ kW}}$$

Forced draught fan :

Volume of air to be handled by forced draught fan per second, including leakage air,

$$v = \frac{19 \times 0{\cdot}0417}{1{\cdot}205} \times 1{\cdot}1 = 0{\cdot}723 \text{ m}^3; \text{ Draught pressure, } p = 0{\cdot}00753 \text{ bar.}$$

Using eqn. (8.8) power of the motor required to drive forced draught fan.

$$= \frac{10^5 \times p \times v}{1,000 \times \eta} = \frac{10^5 \times 0{\cdot}00753 \times 0{\cdot}723}{1,000 \times 0{\cdot}5} = \mathbf{1{\cdot}0888 \text{ kW}}$$

$$\therefore \frac{\text{Power for induced draught fan}}{\text{Power for forced draught fan}} = \frac{1{\cdot}9578}{1{\cdot}0888} = \mathbf{1{\cdot}8} \text{ (ratio of power)}$$

Alternatively, using eqn. (8.11), the ratio of power

$$= \frac{m+1}{m} \times \frac{1{\cdot}2}{1{\cdot}1} \times \frac{T}{T_a}$$

$$= \frac{19+1}{19} \times \frac{1{\cdot}2}{1{\cdot}1} \times \frac{(190+273)}{(20+273)} = \mathbf{1{\cdot}8} \text{ (same as before)}$$

Problem-9 : *The mean temperature of flue gases in the chimney of boiler is 200°C, while the temperature of the air in the boiler house is 20°C. The boiler consumes 1,000 kg of fuel per hour and 18 kg of air is supplied to burn 1 kg of this fuel. Calculate the air power of the fan of this boiler plant to maintain a draught of 50 mm of water, when (a) the fan produces an induced draught, and (b) the fan produces forced draught. Volume of 1 kg of air at 0°C and 760 mm of Hg may be taken as 0·7734 m³.*

(a) Air power of induced draught fan :

Mass of *flue* gases handled by induced draught fan per second

$$= \frac{1,000}{3,600} \times (18+1) = 5{\cdot}278 \text{ kg}$$

Volume of *flue* gases at N.T.P.(0°C) $= 5{\cdot}278 \times 0{\cdot}7734 = 4{\cdot}082$ m^3/second

$\therefore$ Volume of flue gases handled by induced draught fan at 200°C,

$$v = 4{\cdot}082 \times \frac{473}{273} = 7{\cdot}0725 \text{ m}^3\text{/second}$$

Draught pressure, $p = \frac{50}{13{\cdot}6} \times \frac{1}{750} = 0{\cdot}0049$ bar

Air power of the induced draught fan $= \frac{10^5 \times p \times v}{1{,}000 \times \eta}$

$$= \frac{10^5 \times 0{\cdot}0049 \times 7{\cdot}0725}{1{,}000 \times 1} = \mathbf{3{\cdot}4655 \text{ kW}}$$

Alternatively, using eqn. (8.10), Air power $= \frac{10^5 \times p \times M(m+1) \times v_o \times T}{1{,}000 \times \eta \times T_o}$

$$= \frac{10^5 \times 0{\cdot}0049 \times \frac{1{,}000}{3{,}600}(18+1) \times 0{\cdot}7734 \times 473}{1{,}000 \times 1 \times 273} = 3{\cdot}4655 \text{ kW (same as before)}$$

(b) Air power of forced draught fan :

Mass of air supplied by the fan per second $= \frac{1{,}000 \times 18}{3{,}600} = 5{\cdot}0$ kg

Volume of this air supplied at 0°C and 760 mm of Hg $= 0{\cdot}7734 \times 5{\cdot}0 = 3{\cdot}867 \text{ m}^3$/second

$\therefore$ Volume of this air supplied at 20°C, $v = 3{\cdot}867 \times \frac{293}{273} = 4{\cdot}15 \text{ m}^3$/second.

Draught pressure, $p = 0{\cdot}0049$ bar.

Air power of the forced draught fan

$$= \frac{10^5 \times p \times v}{1{,}000 \times \eta} = \frac{10^5 \times 0{\cdot}0049 \times 4{\cdot}15}{1{,}000 \times 1} = \mathbf{2{\cdot}0335 \text{ kW}}$$

Alternatively, using eqn. (8.9), Air power $= \frac{10^5 \times p \times M \times m \times v_o \times T_a}{1{,}000 \times \eta \times T_o}$

$$= \frac{10^5 \times 0{\cdot}0049 \times \frac{1{,}000}{3{,}600} \times 18 \times 0{\cdot}7734 \times 293}{1{,}000 \times 1 \times 273} = 2{\cdot}0335 \text{ kW (same as before)}$$

8.18 Performance of Boilers

The performance of steam boiler is judged by calculating quantity of heat produced by the utilizing heat of combustion of fuel. The *evaporative capacity or output* of a boiler is frequently given as the kilograms of water evaporated per hour. Since, the steam produced at various pressures and temperatures (from feed water at various temperatures) contains varying amounts of heat, the number of kilograms of water evaporated per hour is exact measure of its performance.

Efforts have been made to provide common basis for comparing the evaporative capacity (output) of boilers. For comparison purpose, the output of a boiler is expressed in terms of :

— Equivalent evaporation from and at 100°C per kg of coal burnt, and

— Evaporation per square metre of heating surface.

The amount of steam generated by the boiler in kilograms per hour at the observed pressure and temperature, quality of steam and feed water temperature (i.e. evaporation in kg per hour under actual working conditions), is called *total evaporation.* The actual evaporation (m_a) is expressed in terms of kilograms of steam generated per kilogram of fuel burned,

i.e. actual evaporation, $m_a = \dfrac{\text{total evaporation per hour}}{\text{fuel used per hour}}$

Different boilers generate different quantities of steam at different pressures and different dryness fraction and degrees of superheat, from feed water at different temperatures. To illustrate, boiler 'A' generates 8·5 kg of dry saturated steam per kg of coal at 13 bar, from feed water at 15°C, boiler 'B' generates 9·5 kg of wet steam per kg of coal at 11 bar and dryness fraction 0·98, from feed water at 30°C, and boiler 'C' generates 9 kg of super-heated steam per kg of coal at 14 bar and with 50°C of superheat, from feed water at 20°C.

It will be seen from the above illustration that all the three boilers are generating steam under different working conditions. Therefore, to provide common basis for comparing the evaporative capacity of boilers working under different conditions, it is necessary that the water be supposed to be evaporated under some standard conditions. The standard conditions adopted are : feed water supplied to the boiler at 100°C and converted into dry saturated steam at 100°C. Under these conditions the evaporation of 1 kg of water at 100°C requires 2,257 kJ to be converted into dry saturated steam at 100°C, which is the enthalpy of evaporation of steam at 100°C (1·01325 bar pressure).

Equivalent evaporation may be defined as the evaporation which would be obtained if the feed water were supplied at 100°C and converted into dry saturated steam at 100°C (1·01325 bar pressure).

Under actual working conditions of the boiler, let

m_a = actual mass of water evaporated in kg per kg of coal burnt under actual working conditions,

H = Enthalpy of 1 kg of steam raised (produced) under actual working conditions in kJ,

h = Enthalpy of 1 kg of feed water entering the boiler in kJ,

L_s = Enthalpy of evaporation of 1 kg of steam at 100°C (2,257 kJ), and

m_e = equivalent evaporation in kg of water from and at 100°C per kg of fuel burnt.

Then, heat transferred to 1 kg of feed water in converting it to dry saturated steam or heat required to produce 1 kg of steam = $H - h$ kJ and

Heat required to produce m_a kg of steam under actual working conditions

$= m_a\ (H - h)$ kJ

Equivalent evaporation in kg of water from and at 100°C per kg of fuel burnt,

$$m_e = \frac{m_a\,(H-h)}{L_s} = \frac{m_a\,(H-h)}{2{,}257} \quad ..(8.12)$$

For wet steam, $m_e = \dfrac{m_a\,(H_{wet}-h)}{2{,}257}$

Factor of equivalent evaporation is the ratio of heat absorbed by 1 kg of feed water under observed conditions (actual working conditions), to that absorbed by 1 kg of feed water evaporated from and at 100°C (standard conditions).

Hence, factor of equivalent evaporation $= \dfrac{H-h}{L_s} = \dfrac{H-h}{2{,}257}$ ..(8.13)

The mass of water evaporated is also expressed in terms of **evaporation per hour per square metre** of heating surface of the boiler. This is obtained by dividing the total water evaporated per hour by the total area of heating surface in square metres.

i.e. evaporation per m^2 of heating surface $= \dfrac{m \text{ kg per hour}}{\text{Total area of heating surface in } m^2}$

where m is the actual mass of water evaporated in kg per hour.

Boiler efficiency or **thermal efficiency** of a boiler is defined as *the ratio of the heat utilized by feed water in converting it to steam, to the heat released by complete combustion of the fuel used in the same time,* i.e., output divided by the input to the boiler.

The output or the heat transferred to feed water is based on the mass of steam produced under the *actual working conditions.* The input to a boiler or heat released by complete combustion of fuel may be based on the *higher calorific value of the fuel.*

Boiler efficiency $= \dfrac{m_a (H-h)}{C.V.}$ ..(8.14)

where m_a = actual evaporation in kg per kg of fuel burnt,

H = enthalpy of 1 kg of steam produced under actual working conditions in kJ,

h = enthalpy of 1 kg of feed water entering the boiler in kJ, and

$C.V.$ = calorific value of fuel in kJ/kg.

If a boiler is provided with an economiser and a superheater, then each of these elements of a boiler will have its own efficiency. If the boiler, economiser and superheater are considered as single unit, the efficiency in that case is known as the *overall efficiency* of the boiler plant or efficiency of the combined boiler plant.

A good water-tube boiler should have a thermal efficiency of about 80%. A Lancashire boiler, with automatic stokers and well maintained brickwork should have an efficiency of about 75%. Where does the remainder of the heat (about 25% heat in the case of Lancashire boiler) go ? Since it is not utilised in the boiler in converting water to steam, it is taken as losses and is distributed as follows :

(1) Heat lost to chimney gases or flue gases :

The chimney gases are made up of

(a) dry flue gases, and (b) steam in flue gases formed from the combustion of hydrogen present in the fuel together with any moisture present in the fuel.

Heat lost to dry flue gases per kg of fuel burnt = mass of dry flue gases in kg per kg of fuel (m_g) × specific heat of dry flue gases (k_p) in kJ/kg K × rise in temperature of flue gases in °C (difference between temperature of flue gases leaving the boiler, t_1 and temperature of the boiler room, t_o),

i.e. Heat lost to dry flue gases per kg of fuel $= m_g \times k_p \, (t_1 - t_o)$ kJ

Heat lost to steam in flue gases per kg of fuel burnt

Now, mass of steam formed per kg of fuel burnt

$= 9\ H_2 +$ mass of moisture per kg of fuel (m)

Assuming that the steam in flue gases exists as superheated steam at atmospheric pressure and at flue gas temperature,

Heat lost to steam in the flue gases per kg of fuel burnt

$= [9H_2 + m] \times [H_{sup} - h]$

$= [9H_2 + m] \times [2{,}676{\cdot}1 + k_p\ (t_1 - 100) - h]$ kJ

where t_1 = the temperature of flue gases leaving the boiler,

H_{sup} = enthalpy of 1 kg of superheated steam at atmospheric pressure (1·01325 bar) and at flue gas temperature in kJ,

k_p = specific heat of superheated steam in kJ/kg K,

h = enthalpy of 1 kg of water at boiler house temperature in kJ,

m = mass of moisture present in 1 kg of fuel, and

H_2 = mass of hydrogen present in 1 kg of fuel.

(2) Heat lost due to incomplete combustion (burning of carbon to CO) :

Any CO present in flue gases is due to insufficient air supply. One kg of carbon burnt to CO releases only 10,130 kJ and one kg of carbon burnt to CO_2 releases 33,830 kJ. Thus the heat available in CO (formed due to incomplete combustion) per kg of carbon is 33,830 − 10,130 = 23,700 kJ. This means that the presence of CO in the flue gases is a loss due to incomplete combustion.

(3) Heat lost due to unburnt fuel falling through the grate bars :

When solid fuels are used, some of the fuel falls through the grate bars and is lost with ash. The heat loss is calculated my multiplying mass of unburnt fuel lost through grate bars by the calorific value of the fuel.

(4) Heat lost to external radiation :

Effective lagging (covering with asbestos) of the surface of boiler exposed to atmosphere is necessary to reduce this loss to a minimum.

Heat lost in items 2, 3 and 4 is found out as the *difference* of the gross heat supplied (released) per kg of fuel, and the heat transferred to feed water in converting it to steam per kg of fuel burnt and heat lost in item 1, i.e., heat carried by flue gases.

Problem-10 : *A boiler generates 800 kg of steam per hour at a pressure of 10 bar and with 50°C superheat, and burns 100 kg of coal per hour. If the calorific value of the coal is 30,000 kJ/kg and feed water temperature is 40°C, calculate :*

(i) the factor of equivalent evaporation, (ii) the actural evaporation per kg of coal, (iii) the equivalent evaporation from and at 100°C per kg of coal, and (iv) the boiler efficiency.

Take specific heat of superheated steam at constant pressure as 2·1 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

(i) At 10 bar, $H_s = 2{,}778{\cdot}1$ kJ/kg (from steam tables).

$H_{sup} = H_s + K_p(t_{sup} - ts) = 2{,}778{\cdot}1 + (2{\cdot}1 \times 50) = 2{,}883{\cdot}1$ kJ/kg.

$h = 4{\cdot}187 \times (40 - 0) = 167{\cdot}48$ kJ/kg.

$\therefore$ Heat required to produce 1 kg of superheated steam from feed water at 40°C

$$= H_{sup} - h = 2{,}883{\cdot}1 - 167{\cdot}48 = 2{,}715{\cdot}62 \text{ kJ/kg}$$

$$\text{Factor of equivalent evaporation} = \frac{H_{sup} - h}{2{,}257} = \frac{2{,}715{\cdot}62}{2{,}257} = \mathbf{1{\cdot}203}$$

$$\text{(ii) Actual evaporation} = m_a = \frac{800}{100} = \mathbf{8 \text{ kg/kg of coal}}$$

(iii) Equivalent evaporation in kg of water from and at 100°C per kg of coal,

$$m_e = \frac{m_a (H_{sup} - h)}{2{,}257} = \frac{8 \times 2{,}715{\cdot}62}{2{,}257} = \mathbf{9{\cdot}625 \text{ kg/kg of coal}}$$

$$\text{(iv) Boiler efficiency,} = \frac{m_a (H_{sup} - h)}{C.V.} = \frac{8 \times 2{,}715{\cdot}62}{30{,}000} = 0{\cdot}7243 \text{ or } \mathbf{72{\cdot}43\%}$$

Problem-11 : *In a boiler test the following quantities were obtained :*

Mean temperature of feed water, 15°C; mean boiler pressure, 12 bar; mean steam dryness fraction, 0·95; mass of coal burnt per hour, 250 kg; calorific value of coal, 32,400 kJ per kg; mass of water supplied to the boiler in 7 hours and 14 minutes, 16,500 kg; mass of water in the boiler at the end of the test was less than that at the commencement by 1,000 kg.

Calculate : (i) the actual evaporation per kg of coal, (ii) the equivalent evaporation from and at 100°C per kg of coal, and (iii) the thermal efficiency of the boiler.

(i) Total mass of steam raised in 7 hours and 14 minutes = 16,500 + 1,000 = 17,500 kg

$$\therefore \text{ Mass of steam raised per hour} = \frac{17{,}500}{7\frac{14}{60}} = 2{,}417 \text{ kg.}$$

$$\therefore \text{ Actual evaporation of water per kg of coal, } m_a = \frac{2{,}417}{250} = \mathbf{9{\cdot}672 \text{ kg}}$$

(ii) At 12 bar, $h = 798{\cdot}65$ kJ/kg and $L = 1{,}986{\cdot}2$ kJ/kg (from steam tables).

Enthalpy of 1 kg of wet steam at 12 bar and 0·95 dry,

$$H_{wet} = h + xL = 798{\cdot}65 + 0{\cdot}95 \times 1{,}986{\cdot}2 = 2{,}685{\cdot}54 \text{ kJ/kg}$$

Enthalpy of 1 kg of feed water, $h = (15 - 0) \times 4{\cdot}187 = 62{\cdot}8$ kJ/kg

$\therefore$ Heat given to 1 kg of feed water in converting it to wet steam

$$= H_{wet} - h = 2{,}685{\cdot}54 - 62{\cdot}8 = 2{,}622{\cdot}74 \text{ kJ/kg}$$

Using eqn. (8.12), equivalent evaporation from and at 100°C,

$$m_e = \frac{m_a (H_{wet} - h)}{2{,}257} = \frac{9{\cdot}672\,(2{,}622{\cdot}74)}{2{,}257} = \mathbf{11{\cdot}4 \text{ kg per kg of coal.}}$$

(iii) Using eqn. (8.14), thermal efficiency of the boiler

$$= \frac{m_a (H_{wet} - h)}{C.V.} = \frac{9{\cdot}672 \times 2{,}622{\cdot}74}{32{,}400} \times 100 = \mathbf{78{\cdot}29\%}$$

Problem-12 : *A boiler produces 2,400 kg of dry saturated steam per hour at a pressure of 12 bar (1·2 MPa) and the feed water is heated by an economiser to a temperature* *of 120°C. 240 kg of coal of calorific value of 33,500 kJ/kg are fired per hour and it is found that 10% of the coal is unburnt. Find thermal efficiency of the boiler, and also of the boiler and grate combined. Take specific heat of water as 4·187 kJ/kg K.*

Enthalpy of 1 kg of dry saturated steam at a pressure of 12 bar,

H_s = 2,784·8 kJ/kg (from steam tables).

Enthalpy of 1 kg of feed water at 120°C, $h = 4{\cdot}187 \times (120 - 0) = 502{\cdot}4$ kJ/kg

Heat given to 1 kg of feed water in converting it to dry saturated steam in boiler

$= H_s - h = 2{,}784{\cdot}8 - 502{\cdot}4 = 2{,}282{\cdot}4$ kJ/kg

Mass of coal actually burnt = 240 × 0·9 = 216 kg

Thermal efficiency of the *boiler* considering 10 per cent of coal unburnt on the grate

$$= \frac{m_a\,(H_s - h)}{C.V.} = \frac{\frac{2{,}400}{216}\,(2{,}282{\cdot}4)}{33{,}500} \times 100 = \mathbf{75{\cdot}69\%}$$

Thermal efficiency of the *boiler and grate* combined

$$= \frac{m_a\,(H_s - h)}{C.V.} = \frac{\frac{2{,}400}{240}\,(2{,}282{\cdot}4)}{33{,}500} \times 100 = \mathbf{68{\cdot}13\%}$$

Problem-13 : *In a boiler trial of 12 hours duration, 800 kg of coal were consumed and water evaporated was 6,400 kg and the mean steam pressure was 8 bar (800 kPa). The coal contained 2·5 per cent moisture and 3·5 per cent ash on mass basis. The feed water temperature was 30°C. Calorific value of coal is 31,000 kJ/kg. The steam produced is dry saturated. Take specific heat of water as 4·187 kJ/kg K. Determine :*

(i) the thermal efficiency of the boiler, (ii) the equivalent evaporation from and at 100°C per kg of dry coal, and (iii) the equivalent evaporation from and at 100°C per kg of combustible.

(i) At 8 bar, H_s = 2,769·1 kJ/kg (from steam tables).

Enthalpy of one kg of dry saturated steam at 8 bar, H_s = 2,769·1 kJ/kg.

Enthalpy of one kg of feed water at 30°C, $h = 4{\cdot}187 \times (30 - 0) = 125{\cdot}61$ kJ/kg.

∴ Heat transferred to 1 kg of feed water at 30°C in converting it to dry saturated steam at 8 bar $= H_s - h = 2{,}769{\cdot}1 - 125{\cdot}61 = 2{,}643{\cdot}49$ kJ/kg.

Actual evaporation per kg of coal, $m_a = \frac{6{,}400}{800} = 8$ kg

Thermal efficiency of the boiler $= \frac{m_a\,(H_s - h)}{C.V.} = \frac{8\,(2{,}643{\cdot}49)}{31{,}000} \times 100 = 68{\cdot}22\%$

(ii) Actual water evaporated per kg of dry coal,

$$m_a = \frac{6{,}400}{800 \times 0{\cdot}975} = 8{\cdot}206 \text{ kg/kg of dry coal}$$

Equivalent evaporation from and at 100°C,

$$m_e = \frac{m_a\,(H_s - h)}{2{,}257} = \frac{8{\cdot}206 \times 2{,}643{\cdot}49}{2{,}257} = \mathbf{9{\cdot}61\ kg/kg\ of\ dry\ coal}$$

(iii) Moisture and ash in coal = 2·5 + 3·5 = 6 per cent.

∴ mass of combustible only = 100 − 6 = 94 per cent of the total coal consumed.

Mass of water evaporated per kg of combustible, $m_a = \frac{6{,}400}{800 \times 0{\cdot}94} \doteq 8{\cdot}51$ kg

Equivalent evaporation from and at 100°C,

$$m_e = \frac{m_a\,(H_s - h)}{2{,}257} = \frac{8{\cdot}51 \times 2{,}643{\cdot}49}{2{,}257} = \mathbf{9{\cdot}965\ kg/kg\ of\ combustible.}$$

Problem-14 : *A boiler generates steam at 14 bar and 0·97 dry. The steam produced by the boiler then passes through the superheater where its temperature is raised to 305°C at constant pressure. If the boiler generates 1,200 kg of steam/hour, calculate the amount of heat received by the superheater per hour.*

If the effectiveness of the above superheater is 60%, and if 5,000 kg/hr of flue gases pass over the superheater at an initial temperature of 285°C, find the temperature of the flue gases leaving the superheater. Take specific heat of the flue gases as 1·005 kJ/kg K and k_p of superheated steam as 2·1 kJ/kg K.

At 14 bar, h = 830·3 kJ/kg, L = 1,957·7 kJ/kg, H = 2,790 kJ/kg

and t_s = 195·07°C (from steam tables).

Enthalpy of wet steam at 14 bar and 0·97 dry (before entering the superheater),

$$H_{wet} = h + xL = 830{\cdot}3 + 0{\cdot}97 \times 1{,}957{\cdot}7 = 2{,}731{\cdot}2 \text{ kJ/kg}$$

Enthalpy of superheated steam at 14 bar and at 305°C (after leaving the superheater),

$$H_{sup} = H_s + K_p\,(t_{sup} - t_s) = 2{,}790 + 2{\cdot}1\ (305 - 195{\cdot}07) = 3{,}020{\cdot}85 \text{ kJ/kg}$$

∴ Heat received by 1 kg of wet steam in the superheater

= 3,020·85 − 2,731·2 = 289·65 kJ/kg

∴ Heat received by steam per hour in the superheater

= 1,200 × 289·65 = **3,47,628 kJ/hr.**

For 60% effectiveness (efficiency) of the superheater, the heat supplied by the flue gases per hour

$$= \frac{3{,}47{,}628}{0{\cdot}6} = 5{,}79{,}380 \text{ kJ/hr.}$$

Heat lost by flue gases/hr = 5,000 × 1·005 × (285 − t)

Heat supplied by flue gases/hr = Heat lost by flue gases/hr.

i.e. 5,79,380 = 5,000 × 1·005 × (285 − t)

$$\therefore\ t = 285 - \frac{5{,}79{,}380}{5{,}000 \times 1{\cdot}005} = \mathbf{169{\cdot}7^{\circ}C}$$

i.e. Temperature of flue gases leaving the superheater = 169·7°C.

Problem-15 : *A boiler generates steam at the rate of 10 kg per kg of coal at a pressure of 12 bar, with a dryness fraction of 0·9. The boiler receives its feed water at a temperature of 110°C from an economiser which has received the feed water at 30°C. The steam raised by the boiler then passes through a superheater where its temperature is raised to 300°C at constant pressure. The coal has a calorific value of 34,000 kJ/kg.*

Determine : (a) the equivalent evaporation at standard conditions per kg of coal, (b) the efficiency of the combined boiler plant, and (c) the percentage of available heat in 1 kg of coal utilized in boiler, economiser and superheater separately.

Take k_p of superheated steam as 2·1 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

(a) At 12 bar (from steam tables), $H_s = 2{,}784{\cdot}8$ kJ/kg, $h = 798{\cdot}65$ kJ/kg,

$L = 1{,}986{\cdot}2$ kJ/kg and $t_s = 187{\cdot}99$°C

Enthalpy of 1 kg of superheated steam,

$$H_{sup} = H_s + k_p\,(t_{sup} - t_s) = 2{,}784{\cdot}8 + 2{\cdot}1\,(300 - 187{\cdot}99) = 3{,}020{\cdot}12 \text{ kJ/kg}$$

Enthalpy of 1 kg of feed water, $h = 4{\cdot}187 \times (30 - 0) = 125{\cdot}61$ kJ/kg

∴ Heat given to feed water in converting it to superheated steam,

$$H_{sup} - h = 3{,}020{\cdot}12 - 125{\cdot}61 = 2{,}894{\cdot}49 \text{ kJ/kg}$$

Equivalent evaporation under standard conditions (from and at 100°C),

$$m_e = \frac{m_a(H_{sup} - h)}{L_s} = \frac{10 \times 2{,}894{\cdot}49}{2{,}257} = \mathbf{12{\cdot}82 \text{ kg/kg of coal}}$$

(b) Efficiency of the *combined boiler plant*

$$= \frac{\text{Heat utilized in boiler, superheater and economiser per kg of coal}}{\text{Heat in 1 kg of coal}}$$

$$= \frac{m_a\,(H_{sup} - h)}{C.V.} = \frac{10 \times 2{,}894{\cdot}49}{34{,}000} = 0{\cdot}8513 \text{ or } \mathbf{85{\cdot}13\%}.$$

(c) Heat utilized in boiler alone per kg of coal

$$= 10[\,798{\cdot}65 + 0{\cdot}9 \times 1{,}986{\cdot}2 - 110 \times 4{\cdot}187\,] = 21{,}256{\cdot}6 \text{ kJ/kg of coal}$$

Available heat in coal is 34,000 kJ/kg.

∴ Percentage of heat available in 1 kg of coal utilized in the boiler

$$= \frac{\text{Heat utilized in boiler per kg of coal}}{\text{C.V.}} \times 100 = \frac{21{,}256{\cdot}6}{34{,}000} \times 100 = \mathbf{62{\cdot}52\%}$$

Heat utilized by economiser per kg of coal

$$= 10 \times 4{\cdot}187 \times (110 - 30) = 3{,}349{\cdot}6 \text{ kJ/kg of coal.}$$

Percentage of the heat available in one kg of coal utilized in the *economiser*

$$= \frac{3{,}349{\cdot}6}{34{,}000} \times 100 = \mathbf{9{\cdot}85\%}$$

Heat absorbed in the *superheater* per kg of steam

$$= [(1 - x)\,L + k_p\,(t_{sup} - t_s)] \text{ kJ}$$

∴ Heat absorbed by 10 kg of steam in the superheater

$$= 10\,[(1 - 0{\cdot}9) \times 1{,}986{\cdot}2 + 2{\cdot}1\,(300 - 187{\cdot}99)] = 4{,}339{\cdot}4 \text{ kJ/kg of coal}$$

∴ Percentage of heat available in 1 kg of coal utilized in the superheater.

$$= \frac{4{,}339{\cdot}4}{34{,}000} \times 100 = \mathbf{12{\cdot}76\%}$$

Problem-16 : *A boiler generates 1,000 kg of dry saturated steam per hour at a pressure of 10 bar. The feed water is heated by economiser before it is supplied to the boiler. If the feed water enters the economiser at 35°C and leaves at 95°C, find the percentage saving in heat by the use of the economiser.*

If 2,500 kg per hour of flue gases pass over the economiser at 330°C and leave the economiser at 190°C, calculate the effectiveness (efficiency) of the economiser. Take specific heat of flue gases as 1·005 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

Enthalpy of 1 kg of dry saturated steam at 10 bar, H_s = 2,778·1 kJ/kg (from steam tables).

Enthalpy of 1 kg of feed water at 35°C, $h = 4{\cdot}187 \times (35 - 0) = 146{\cdot}55$ kJ/kg.

∴ Heat required to produce 1 kg of dry saturated steam at 10 bar from water at 35°C

$$= H_s - h = 2{,}778{\cdot}1 - 146{\cdot}55 = 2{,}631{\cdot}55 \text{ kJ/kg}$$

Enthalpy of 1 kg of feed water at 95°C, $h = 4{\cdot}187 \times (95 - 0) = 397{\cdot}77$ kJ/kg

∴ Heat required to produce 1 kg of dry saturated steam at 10 bar from water at 95°C

$$= H_s - h = 2{,}778{\cdot}1 - 397{\cdot}77 = 2{,}380{\cdot}33 \text{ kJ/kg}$$

∴ Saving in heat by using feed water at 95°C instead of at 35°C (i.e. by heating water in the economiser)

$$= 2{,}631{\cdot}55 - 2{,}380{\cdot}33 = \mathbf{251{\cdot}22 \text{ kJ/kg}}$$

∴ Percentage saving in heat by the use of the economiser

$$= \frac{251{\cdot}22}{2{,}631{\cdot}55} \times 100 = \mathbf{9{\cdot}58\%}$$

Heat received (or gained) by feed water from the flue gases in the economizer

$$= 1{,}000 \times 4{\cdot}187 \times (95 - 35) = 2{,}51{,}220 \text{ kJ/hr.}$$

Heat rejected (or lost) by flue gases in the economiser

$$= 2{,}500 \times 1{\cdot}005 \times (330 - 190) = 3{,}51{,}750 \text{ kJ/hr.}$$

∴ Effectiveness (or efficiency) of the economiser $= \frac{2{,}51{,}220}{3{,}51{,}750} \times 100 = \mathbf{71{\cdot}42\%}$

Problem-17 : *A Lancashire boiler is supplied with coal of calorific value 31,400 kJ/kg and an analysis of C, 82%; H_2, 6%; O_2, 9%; and ash, 3%. Calculate the minimum mass of air required per kg of coal. If the total air supplied is 1·4 times the minimum air required, calculate the mass of dry products of combustion per kg of coal burnt and the heat carried away by dry flue gases per kg of coal burnt if the average specific heat of flue gases is 1·005 kJ/kg K and the temperature of flue gases is 350°C and the boiler house temperature is 20°C. Determine also the thermal efficiency and equivalent evaporation from and at 100°C of the boiler if 2,250 kg of dry saturated steam is generated per hour at a pressure of 11 bar from feed water at 30°C, and 250 kg of coal are burnt per hour.*

Using eqn. (7.1), minimum air required for combustion of 1 kg of coal

$$= \frac{100}{23}\,[2{\cdot}67C + 8H + S - O]$$

$$= \frac{100}{23}\,[(2{\cdot}67 \times 0{\cdot}82) + (8 \times 0{\cdot}06) - 0{\cdot}09] = \mathbf{11{\cdot}21\ kg/kg\ of\ coal.}$$

Total (actual air supplied = 1·4 × 11·21 = 15·69 kg/kg of coal.

∴ Excess air supplied = 15·69 − 11·21 = 4·48 kg/kg of coal.

The dry flue gases per kg of coal will consist of

(1) $CO_2 = C \times 3{\cdot}67 = 0{\cdot}82 \times 3{\cdot}67 \quad = 3{\cdot}01$ kg

(2) $O_2 = 0{\cdot}23 \times$ excess air $= 0{\cdot}23 \times 4{\cdot}48 = 1{\cdot}03$ kg

(3) $N_2 = 0{\cdot}77 \times$ actual air $= 0{\cdot}77 \times 15{\cdot}69 = \underline{12{\cdot}08}$ kg

Total mass of dry products per kg of coal (m_g) = **16·12 kg**

Using eqn. (7.2), heat carried away by dry flue gases per kg of coal

$= m_g \times k_p\,(t_1 - t_o) = 16{\cdot}12 \times 1{\cdot}005\,(350 - 20) =$ **5,346·2 kJ/kg of coal**

Enthalpy of 1 kg of dry saturated steam at 11 bar, H_s = 2,781·7 kJ/kg (from steam tables).

Enthalpy of 1 kg of feed water before entering the boiler,

$h = (30 - 0) \times 4{\cdot}187 = 125{\cdot}6$ kJ/kg

∴ Heat given to 1 kg of feed water at 30°C in converting it to dry saturated steam at 11 bar in the boiler = $H_s - h$ = 2,781·7 − 125·6 = 2,656·1 kJ/kg

Using eqn. (8.14), thermal efficiency of the boiler

$$= \frac{m_a\,(H_s - h)}{C.V.} = \frac{\frac{2{,}250}{250}\,(2{,}656{\cdot}1)}{31{,}400} = 0{\cdot}7613 \text{ or } \mathbf{76{\cdot}13\%}$$

Using eqn. (8.12), equivalent evaporation from and at 100°C,

$$m_e = \frac{m_a\,(H_s - h)}{L_s} = \frac{\frac{2{,}250}{250}\,(2{,}656{\cdot}1)}{2{,}257} = \mathbf{10{\cdot}59\ kg/kg\ of\ coal}$$

Problem-18 : *During a trial on a Lancashire boiler plant consisting of an economiser, 8 kg of steam was generated per kg of coal burnt at a pressure of 12 bar and 0·9 dry. The boiler received its feed water at a temperature of 50°C from an economiser. The air supplied per kg of coal was 18 kg, and the calorific value of coal used was 31,400 kJ/kg. If 70 per cent of the heat loss was carried away by the flue gases, calculate the temperature of flue gases leaving the boiler and entering the economiser. The boiler room temperature was 25°C and the specific heat of flue gases was 1·005 kJ/kg K.*

At 12 bar, h = 798·65 kJ/kg and L = 1,986·2 kJ/kg (from steam tables).

$H_{wet} = h + xL = 798{\cdot}65 + 0{\cdot}9 \times 1{,}986{\cdot}2 = 2{,}586{\cdot}23$ kJ/kg

Enthalpy of 1 kg of feed water, $h = (50 - 0) \times 4{\cdot}187 = 209{\cdot}35$ kJ/kg

∴ Heat given to 1 kg of feed water in converting it to wet steam

$= H_{wet} - h = 2{,}586{\cdot}23 - 209{\cdot}35 = 2{,}376{\cdot}88$ kJ/kg

Amount of steam generated per kg of coal = 8 kg.

∴ Heat utilized in boiler per kg of coal = 8 × 2,376·88 = 19,015 kJ

∴ Heat lost per kg of coal = 31,400 − 19,015 = 12,385 kJ.

70 per cent of this heat lost is carried away by the flue gases.

∴ Heat carried away by the flue gases = 0·7 × 12,385 = 8,669·5 kJ/kg of coal

Mass of flue gases produced per kg of coal burnt = 18 + 1 = 19 kg.

Let m_g = mass of flue gases per kg of coal,

t_1 = temperature of flue gases leaving the boiler,

t_o = boiler room temperature, and

k_p = specific heat of flue gases.

Then, heat carried away by flue gases per kg of coal burnt = $m_g \times k_p \times (t_l - t_o)$ kJ

i.e. $8{,}669{\cdot}5 = 19 \times 1{\cdot}005 \times (t_1 - 25)$

$\therefore\ t_1 =$ **479°C**

i.e. temperature of flue gases leaving the boiler and entering the economiser is 479°C.

Problem-19 : *In a boiler trial of 24 hours duration, 1,600 kg of coal were consumed and water evaporated was 12,800 kg and the mean steam pressure was 7·5 bar. The coal contained 3 per cent moisture and 3·9 per cent ash on mass basis. The feed water temperature was 35°C. Calorific value of coal is 30,300 kJ/kg. The steam produced is dry saturated.*

Determine : (i) the thermal efficiency of the boiler, (ii) the equivalent evaporation from and at 100°C per kg of dry coal, and (iii) the equivalent evaporation from and at 100°C per kg of combustible.

(i) Enthalpy of dry saturated steam at 7·5 bar,

H_s = 2,766·4 kJ/kg (from steam tables),

Enthalpy of 1 kg of feed water, $h = (35 - 0) \times 4{\cdot}187 = 146{\cdot}55$ kJ/kg

∴ Heat transferred to 1 kg of feed water in converting it to dry saturated steam

$= H_s - h = 2{,}766{\cdot}4 - 146{\cdot}55 = 2{,}619{\cdot}85$ kJ/kg.

Actual evaporation per kg of coal, $m_a = \dfrac{12{,}800}{1{,}600} = 8$ kg

Thermal efficiency of the boiler

$$= \frac{m_a(H_s - h)}{C.V.} = \frac{8\,(2{,}619{\cdot}85)}{30{,}300} \times 100 = 69{\cdot}17\%$$

(ii) Mass of water evaporated per kg of *dry* coal, $m_a = \dfrac{12{,}800}{1{,}600 \times 0{\cdot}97} = 8{\cdot}25$ kg

Equivalent evaporation from and at 100°C,

$$m_e = \frac{8{\cdot}25\,(2{,}619{\cdot}85)}{2{,}257} = \mathbf{9{\cdot}56\ kg/kg\ of\ dry\ coal.}$$

(iii) Moisture and ash in coal = 3 + 3·9 = 6·9 per cent.

Mass of combustible only = 100 − 6·9 = 93·1 per cent of the total coal consumed.

Mass of water evaporated per kg of combustible, $m_a = \frac{12{,}800}{1{,}600 \times 0{\cdot}931}$ = **8·593 kg**

Equivalent evaporation from and at 100°C,

$$m_e = \frac{8{\cdot}593\,(2{,}619{\cdot}85)}{2{,}257} = \textbf{9·974 kg/kg of combustible.}$$

Problem-20 : *The following data was obtained during the trial of a water-tube boiler; Steam pressure, 15 bar; degree of superheat, 71·1°C; temperature of feed water, 92·6°C; water evaporated, 3,223·8 kg per hour; coal fired, 417·3 kg per hour; ash, 43·3 kg per hour; percentage of combustible in ash, 9·68; moisture in coal, 4·42%; heat value of one kilogram of dry coal, 30,800 kJ/kg.*

Determine : (a) the efficiency of boiler plant including superheater, and (b) the efficiency of boiler and furnace combined. Take K_p of superheated steam = 2·1 kJ/kg K.

(a) At 15 bar (from steam tables), H_s = 2,792·2 kJ/kg.

Enthalpy of 1 kg of steam at 15 bar and with 71·7°C superheat,

$H_{sup} = H_s + k_p \times$ degree of superheat = 2,792·2 + 2·1 (71·7) = 2,942·8 kJ/kg

Enthalpy of 1 kg of feed water, h = (92·6 − 0) × 4·187 = 387·7 kJ/kg

∴ Heat given to 1 kg of feed water in converting it to superheated steam

$= H_{sup} - h$ = 2,942·8 − 387·7 = 2,555·1 kJ/kg

Moisture in coal = $417{\cdot}3 \times \frac{4{\cdot}42}{100}$ = 18·4 kg per hour.

∴ Dry coal fired per hour = 417·3 − 18·4 = 398·9 kg.

Actual evaporation per kg of dry coal, $m_a = \frac{3{,}223{\cdot}8}{398{\cdot}9}$ = 8·07 kg.

Thermal efficiency of the boiler plant including superheater

$$= \frac{m_a\,(H_{sup} - h)}{C.V.} = \frac{8{\cdot}07\,(2{,}555{\cdot}1)}{30{,}800} \times 100 = \textbf{67·03\%}$$

(b) Combustible in ash per hour = $43{\cdot}3 \times \frac{9{\cdot}68}{100}$ = 4·2 kg

This combustible in the ash is practically carbon whose calorific value may be taken as 34,000 kJ per kg.

Actual heat supplied per hour = heat of dry coal − heat of combustible in the ash

= (398·9 × 30,800) − (4·2 × 34,000) = 1,21,43,320 kJ/hr.

Heat usefully utilized in the boiler per hour

$= m\,(H_{sup} - h)$ = 3,223·8 (2,555·1) = 82,37,131 kJ/hr

∴ Efficiency of the boiler and furnace combined

$$= \frac{\text{Heat utilized per hour}}{\text{Heat supplied per hour}} = \frac{82{,}37{,}131}{1{,}21{,}43{,}320} \times 100 = \textbf{67·83\%}$$

Note: Efficiency of the boiler and furnace combined, is found by using the input based upon the heat given out by the coal actually burnt on the fire grate (not taking into account the heat value of the coal which falls through the grate in the ash pit). When solid fuels are used, some coal is always lost with the ash, usually because molten ash freezes around pieces of coal and so prevents combustion. The mass of coal lost is determined by heating samples of ash and measuring the reduction of mass.

Tutorial-8

1. What is a steam boiler and what is its function ? What are the uses of steam ?

2. Explain the following terms used in boiler practice :

(a) Boiler shell, (b) Fire grate, (c) Furnace,
(d) Setting, (e) Steam space, (f) Mountings,
(g) Blowing-off, (h) Gusset stays, and (i) Flues.

3. (a) Classify boilers according to various factors. Give the name of atleast one boiler of each type.

(b) Give a diagrammatic sketch and describe the arrangement of the brickwork flues of a Cornish boiler. Indicate on it the path of the furnace gases to the chimney.

4. Describe, giving neat sketches, the construction of the furnace tubes of a Lancashire boiler. Explain how the successive sections are connected and how the furnace tube is fitted to the end plates. Why are Galloway tubes fitted ?

5. (a) Describe, with neat sketches, the construction and working of a Lancashire boiler, showing therein the main fittings and the path of the flue gases to the chimney.

(b) What are the advantages of a Lancashire boiler ? What are its principal defects ?

6. Describe, with the aid of sketches, the construction and working of a Co-chran boiler. Indicate on it the path of flue gases. What are the advantages of Co-chran boilers ?

7. Sketch and describe the construction and working of a locomotive boiler. Describe the method of obtaining draught in this boiler. What are the characteristics of this boiler ?

8. (a) Make a neat sketch of a locomotive boiler and explain its working. Describe the method of obtaining draught in such a boiler in the absence of a stack (or chimney) of appreciable height.

(b) State the advantages and disadvantages of a locomotive boiler.

9. (a) Describe with the aid of sketches either a Lancashire boiler or a water-tube boiler.

(b) Give an outline sketch showing the arrangement of water tubes and furnace in Babcock and Wilcox water-tube boiler. Indicate on it the path of the furnace gases and the water circulation.

10. (a) Distinguish between Cornish and Lancashire boilers. State their fields of application.

(b) Distinguish between 'water-tube' and 'fire-tube' boilers and state under what circumstances each type would be used.

11. Mention the chief advantages and disadvantages of water-tube boilers.

12. State the advantages of water-tube boilers over fire-tube boilers and tank boilers.

13. What are the considerations which would guide you in determining the type of boiler to be employed for a specific purpose ?

14. Explain the following statements :

(a) Steam boiler is also known as steam generator.

(b) Shell or tank type boilers are more suitable for moderate power generation.

(c) Flat ends of Lancashire and Cornish boilers are stayed.

(d) Furnace tubes of Lancashire and Cornish boilers are not made in one piece.

(e) Fire-tube boilers have advantages of low cost and compact design.

15. (a) What are boiler mountings ? Give their names.

(b) What is a water-level indicator ? Give a sketch and description of any one type of water-level indicator.

16. (a) Draw a neat sketch of a pressure gauge used on a steam boiler, and explain its working.

(b) Explain the action and give sketches showing the construction of the steam pressure gauge.

17. What is the function of the safety valve ? Name the different types of safety valves, and name the boilers on which they are used.

18. Sketch and describe the construction and operation of a dead-weight safety valve and give its advantages and disadvantages.

19. Sketch and describe the construction of any one type of safety valve. In the valve you select explain the method to alter the setting of the valve so as to cause it to blow off at an increased pressure.

20. The diameter of a safety valve of the lever pattern is 9 cm. The weight of the lever is 50 N and centre of gravity is 38 cm from the fulcrum. The movable weight on the lever is 450 N. The weight of the valve is 15 N and distance from the fulcrum to the centre of the valve is 12 cm. Calculate the distance of the movable weight on the lever from the fulcrum in order that the valve may blow off at a steam pressure of 6 bar.

[97·16 cm]

21. (a) What is the purpose of fitting the fusible plug in a boiler ? What is its location in the boiler ? Name the mountings that give warning of low water in a boiler.

(b) Describe the construction and working of any one type of stop valve. What is its best location to carry out the function properly ?

22. (a) Describe the construction and working of any one type of feed check valve. What is the function of a check valve ?

(b) Sketch and describe any pattern of a blow-off valve or cock. What precautions must be observed in this form of valve ?

23. Explain the construction and working principle of a steam injector. State where and when it is used. What are the advantages and disadvantages of a steam injector ?

24. Draw a neat sketch of the Babcock and Wilcox water tube boiler. Name the mountings and accessories that are used on the boiler you have sketched and indicate their positions on the sketch you have drawn.

25. Name the different mountings and accessories with which the Babcock and Wilcox water-tube boiler is fitted. State their uses and give a neat sketch of any one of them.

26. Make a sketch of a water-tube boiler. Show the position of the superheater and economiser, and the path of furnace gases on the sketch you have drawn.

27. (a) What is a man hole ? Describe its shape and cover.

(b) A boiler, fitted with an economiser, generates dry saturated steam at 1·2 N/mm^2 ab (12 bar). If the feed water enters the economiser at 35°C and leaves at 90°C, find the percentage saving in heat by the use of the economiser. [8·73%]

28. (a) Mention the advantages of installing an air pre-heater with the boiler. Describe construction and working of an air pre-heater.

(b) Describe with the help of a neat sketch any one type of superheater used with a Lancashire boiler. What are the advantages of installing a superheater with the boiler ?

(c) Steam is raised at 15 bar in a Lancashire boiler fitted with a superheater. Boiler feed is at 30°C and the steam leaves the boiler and enters the superheater with the dryness fraction of 0·95. The temperature of steam leaving the superheater is 300°C. Calculate the heat received per kg of steam in the superheater. Take k_p for superheated steam as 2·3 kJ/kg K. [331·23 kJ/kg]

29. What are the accessories used for delivering feed water to a boiler ? Describe the working of a Duplex feed pump with the help of a neat sketch.

30. Write short notes on the following giving sketches wherever necessary :

(a) Steam trap, (b) Fusible plug, (c) Economiser, (d) Superheater, (e) Pressure reducing valve, (f) Feed check valve, (g) High-steam and low-water safety valve, (h) Steam drier or Separator, (i) Air pre-heater, and (j) Steam injector.

31. Describe with the help of sketches the following devices fitted on a boiler for removing water particles from steam :

(i) Anti-priming pipe,

(ii) Steam separator, and

(iii) Steam trap.

32. Draw a neat sketch of Lancashire boiler and indicates on it the positions of its principal mountings. Show also, on the sketch the path of the flue gases and the water level in the drum. Enumerate the accessories you would normally find on this type of boiler.

33. What do you understand by the term 'boiler draught'? What are the various types of draughts used in usual practice ?

Estimate the minimum height of a chimney required to produce a draught of 16 mm of water, if 19 kg of air are required per kg of fuel burnt on the grate, the mean temperature of flue gases inside the chimney is 330°C and that of atmospheric air is 30°C.

[29·16 m]

34. Derive an expression connecting the height of a chimney and the draught it produces, in terms of temperature of outside air and the mean flue gas temperature.

Find the draught in mm of water column produced by a chimney 40 m high when the mean temperature of flue gases is 300°C and the temperature of outside air is 27°C, and 19 kg of air are supplied per kg of fuel burnt on the grate.

[21·13 mm]

35. Distinguish between the following :

(i) Natural draught and artificial draught, and

(ii) Forced draught and induced draught.

Why an artificial draught is considered advantageous over a natural draught ?

36. Estimate the minimum height of chimney required to produce a draught of 19 mm of water. The atmospheric air temperature is 25°C and the average flue gas temperature is 310°C. Air supplied per kg of fuel burnt is equal to 24 kg. Derive the formula you may use for establishing the height of the chimney.

[34·31 m]

37. Deduce the expression showing the relation between the draught, the height of the chimney and the temperature of flue gases, air, etc.

How much air will be used per kg of coal burnt in a boiler having a chimney of 45 metres height to produce a draught of 25 mm of water when the temperature of the flue gases in the chimney is 327°C and that of the boiler house is 27°C ?

[17·92 kg]

38. What is the function of a boiler chimney ? Explain why there is no chimney in the case of a locomotive boiler ?

Estimate the mean temperature of the flue gases leaving the chimney 40 m high to produce a draught of 20 mm water column if 19 kg of air are required per kg of fuel burnt on the grate. The temperature of atmospheric air is 30°C.

[285·8°C]

39. (a) State briefly what do you understand by natural and artificial draughts. What are the advantages of artificial draught over natural draught ?.

(b) Explain the different system of producing draught in a boiler giving their advantages and disadvantages.

40. Establish a condition of maximum discharge of flue gases through a chimney of a given height.

A chimney has a height of 35 m. The temperature of outside air is 20°C. Find the draught in mm of water when the temperature of chimney gases is such as to cause the mass of the gases discharged in a given time to be maximum.

[21·08 mm]

41. A chimney 40 m high deals with flue gases at 300°C, when the outside air temperature is 27°C. The air supplied for combustion is 19 kg per kg of fuel burnt in the furnace. Calculate :

(i) the draught in mm of water column produced by the chimney,

(ii) the draught produced in terms of height of column of hot flue gases in metres,

(iii) the velocity of flue gases in the chimney, if 60% of the draught is lost in friction at the grate and passages, and

(iv) the draught produced in mm of water, and the temperature of flue gases under the condition of maximum discharge.

[(i) 21·13 mm; (ii) 32·58 m; (iii) 16 m/sec; (iv) 23·53 mm, 358·6°C]

42. (a) Explain the terms mechanical draught and balanced draught.

(b) Distinguish between fan draught and steam jet draught.

(c) Explain the working principle of the steam jet draught.

43. An induced draught fan produced a draught of 23 mm of water. The temperature of the chimney gases leaving the boiler is 195 C and 18 kg of air is supplied per kg of fuel. If the fuel burnt is 50 kg per minute and the efficiency of the fan is 75%, find the power required to drive the fan. Volume of 1 kg of air at 0°C and 760 mm of Hg may be taken as 0·7734 m^3.

[6·312 kW]

44. The following data refer to a boiler equipped with forced draught fan :

Draught produced, 38 mm of water; efficiency of the fan, 60%; fuel burnt, 3,000 kg per hour; temperature of chimney gases, 150°C; temperature of air in the boiler house, 35°C. If 19 kg of air is supplied per kg of fuel, calculate the power required to drive the fan.

What will be the power required if the forced draught fan is substituted by an induced draught fan ? Volume of 1 kg of air at 0°C and 760 mm Hg may be taken as 0·7734 m^3. Take the efficiency of the induced draught fan as 60%.

[8·576 kW; 12·4 kW]

45. Explain the following terms as applied to steam boilers :

(i) Equivalent evaporation from and at 100°C, (ii) Factor of equivalent evaporation, and (iii) Thermal efficiency.

46. The following particulars are taken from the reports of three steam boilers *A*, *B* and *C* :

A – 8·7 kg of dry saturated steam per kg of coal at 3·5 bar, from feed water at 15°C.

B – 9·5 kg of steam per kg of coal at 11 bar and 0·98 dry, from feed water at 60°C.

C – 9·3 kg of steam per kg of coal at 13 bar and with 50°C of superheat, from feed water at 20°C. Take specific heat of superheated steam at constant pressure as 2·3 kJ/kg K.

Calculate for each boiler the equivalent evaporation from and at 100°C per kg of coal.

[A – 10·29 kg; B - 10·48 kg; C - 11·62 kg]

47. What do you understand by "Equivalent evaporation from and at 100°C" as applied to a steam boiler ?

The following data were recorded during a test on a steam boiler :

The pressure of steam generated, 10 bar; Condition of steam at boiler stop valve, 0·9 dry; Temperature of feed water, 25°C; Average quantity of steam generated per hour, 1,000 kg; Average quantity of coal used per hour, 125 kg; Calorific value of coal, 26,000 kJ/kg.

Calculate : (a) the thermal efficiency of the boiler, and (b) the equivalent evaporation from and at 100°C per kg of coal.

[(a) 76·06%; (b) 8·762 kg]

48. What do you understand by "factor of equivalent evaporation" of a boiler ?

A steam boiler evaporates 5,625 kg of water per hour at a pressure of 11 bar and at a temperature of 250°C. It consumes 750 kg of coal per hour, having calorific value of 32,000 kJ/kg. The feed water temperature is 30°C. Find : (a) the actual evaporation per kg of coal, (b) the factor of equivalent evaporation, (c) the equivalent evaporation from and at 100°C per kg of coal, and (d) the thermal efficiency of the boiler.

Take specific heat of superheated steam at constant pressure as 2·3 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

[(a) 7·5 kg per kg of coal; (b) 1·247; (c) 9·328 kg per kg of coal; (d) 65·83%]

49. What steps would you take to improve the thermal efficiency of a boiler ?

A boiler generates dry saturated steam at a pressure of 11 bar. The feed water is heated by an economiser before it is supplied to the boiler. If the feed water enters the economiser at 30°C and leaves at 79°C, find the percentage saving in heat by the use of the economiser.

[6·02%]

50. What do you understand by "evaporative capacity" of a steam boiler ?

900 kg of feed water enter the economiser of a boiler per hour. The feed water enters the economiser at 35°C and leaves at 95°C. If 2,000 kg/hr of flue gases pass over the economiser at 320°C, determine the temperature of the flue gases which leave the economiser and pass to the chimney stack. Take the specific heat of the flue gases as 1·005 kJ/kg K and specific heat of water as 4·187 kJ/kg K. Assume that all the heat lost by the flue gases is taken up by the feed water.

[207·5°C]

51. 850 kg of feed water enter the economiser of a boiler per hour. The feed water enters the economiser at 30°C and leaves at 95°C. If 1,800 kg/hr of flue gases pass over the economiser at 300°C, determine the temperature of the flue gases which leave the economiser and pass to the chimney stack. Take specific heat of flue gases as 1·005 kJ/kg K and specific heat of water as 4·187 kJ/kg K. Assume that all the heat lost by the flue gases is taken up by the feed water.

[172·1°C]

52. A steam plant consists of an economiser, a boiler and a superheater. During a test lasting for one hour, the following observations were made :

Steam pressure, 13 bar; coal consumed, 700 kg; calorific value of coal, 28,000 kJ/kg; mass of water evaporated, 5,250 kg; temperatures of feed water entering the economiser, 35°C and leaving the economiser, 130°C; dryness fraction of steam leaving the boiler, 0·9; temperature of steam leaving the superheater, 322°C.

Determine : (a) the overall efficiency of the combined plant, and (b) the percentage of the available heat in coal utilized in the economiser, boiler and superheater separately.

Take specific heat of superheated steam at constant pressure as 2·1 kJ/kg K and specific heat of water as 4·187 kJ/kg K.

[(a) 78·06%; (b) 10·65%, 54·8%, 12·61%]

53. In a boiler test the following quantities were obtained :

Mean temperature of feed water, 20°C; mean steam pressure, 10 bar; mean steam dryness fraction, 0·92; mass of coal burnt per hour, 130 kg; calorific value of coal, 31,500 kJ/kg; mass of water supplied to the boiler in 5 hours and 24 minutes, 5,500 kg; mass of water in the boiler at the end of the test was less than at the commencement by 500 kg. Calculate : (a) the actual evaporation per kg of coal, (b) the equivalent evaporation from and at 100°C per kg of coal, and (c) the thermal efficiency of the boiler.

[(a) 8·54 kg; (b) 9·58 kg; (c) 68·68%]

54. A boiler produces 1,700 kg of dry saturated steam per hour at a pressure of 10 bar. The feed water is heated by an economiser to a temperature of 120°C. If 190 kg of coal of a calorific value 31,000 kJ/kg are fired per hour, and it is ascertained that 8 per cent of the coal is unburnt, calculate :

(a) the thermal efficiency of the boiler, and

(b) the thermal efficiency of the boiler and grate combined.

Take specific heat of water as 4·187 kJ/kg K.

[(a) 71·39%; (b) 65·8%]

55. In a boiler trial of 10 hours duration, 600 kg of coal was consumed and the water evaporated was 5,000 kg. The mean steam pressure was 8 bar and the steam produced was dry saturated. The coal contained 3 per cent of moisture and 4 per cent of ash on mass basis. The mean feed water temperature was 35°C. Calorific value of coal was 30,000 kJ/kg.

Determine : (a) the thermal efficiency of the boiler, and (b) the equivalent evaporation from and at 100°C per kg of combustible.

Take specific heat of water as 4·187 kJ/kg K.

[(a) 72·85%; (b) 10·56 kg/kg of combustible]

56. State what are the principal heat losses in a boiler plant. State what methods are adopted to minimise the losses.

9
STEAM ENGINES

9.1 Heat Engines

Heat engines may be divided into two main classes, according to where the combustion of fuel takes place. In one class, the combustion of fuel takes place outside the cylinder, and such an engine is called *external combustion engine.* The most common examples of this class are steam engines and steam turbines, where the working medium is steam. In an external combustion engine the power is produced in two stages. The energy released from the fuel in the furnace of the boiler 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 power. When the combustion of fuel takes place inside the engine cylinder so that the products of combustion directly act on the piston, the engine is known as *internal combustion engine.* Diesel engine, gas engine and petrol engine are the common examples of this class where the working medium is the products of combustion.

Steam engines were manufactured upto the year 1930 for use as stationary prime-movers, particularly in the textile industry. They are still used for locomotives for railways and now slowly they are being replaced by Diesel locomotives. In addition, they are used on ships where they are slowly being replaced by steam turbines and Diesel engines.

9.2 Steam Engine Plant

As steam engine plant consists essentially of three main units : **Boiler, Engine** *and* **Condenser**. In many cases, particularly in locomotive steam engines, a separate condenser is not provided and the engine exhausts into the atmosphere.

The steam from the boiler is admitted into a steam chest from where it enters the engine cylinder through a valve driven by an eccentric on the engine crankshaft. After expansion in the engine cylinder and doing work on the piston, the steam is exhausted into a condenser where it is condensed and returned as feed water to the boiler, thus, completing the cycle. Nearly all reciprocating steam engines are

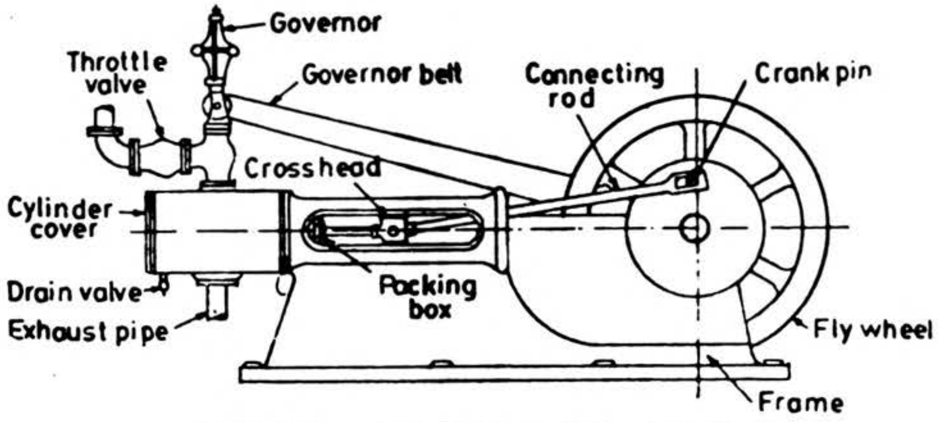


Fig. 9.1. Side view of a horizontal simple steam engine.

double-acting, i.e. steam is admitted in turn to each side of the piston and two working strokes are produced during each revolution of the crankshaft.

Figure 9-1 illustrates a simple form of a single-cylinder, horizontal, reciprocating steam engine. The figure shows major principal parts of the engine.

9.3 Classification (Types)

Steam engines may be classified in the following ways :

.. Position of the axis of the cylinder : *Vertical, Inclined* or *Horizontal engine.*

.. According to the action of steam upon the piston : *Single-acting* or *Double-acting engine.*

.. Number of cylinders used in which steam expands : *Single-expansion* or *Simple engine (total expansion of steam in one cylinder),* and *Multiple-expansion* or *compound engine (total expansion of steam in more than one cylinder).*

.. Method of removal of exhaust steam : *Condensing* or *Non-condensing engine.*

.. Magnitude of rotative speed : *Low, Medium* or *High speed engine.*

.. Type of valve used : *Slide valve, Corliss valve* or *Drop valve engine.*

.. Use or field of application : *Stationary, Portable (movable), Locomotive, Marine engine.*

9.4 Parts of Steam Engine

The parts of steam engine may be broadly divided into two groups, namely, *stationary parts and moving parts.*

- Stationary parts : *Engine frame, Cylinder, Steam chest, Stuffing box, Crosshead guides and Main bearings.*
- Moving parts : *Piston and piston rod, Crosshead, Connecting rod, Crankshaft, Flywheel, Slide valve and valve rod, Eccentric* and *eccentric rod* and *Governor.*

The function of the steam engine parts are as follows :

The **engine frame** is a heavy casting which supports all the stationary as well as moving parts of the engine and holds in proper alignment. It may rest directly on the engine foundation or upon the **engine bed plate** fixed on the engine foundation.

The **cylinder** shown in fig. 9-2 is a cast iron cylindrical hollow vessel in which the piston moves to and fro under the pressure of the steam. Both the ends of the cylinder are closed by covers and made steam-right.

The **steam chest** is a closed chamber integral with the cylinder. It supplies steam to the cylinder with the movement of the slide valve.

The **stuffing box** and gland are fitted on the crank end cover of the cylinder as shown in fig. 9-2 and

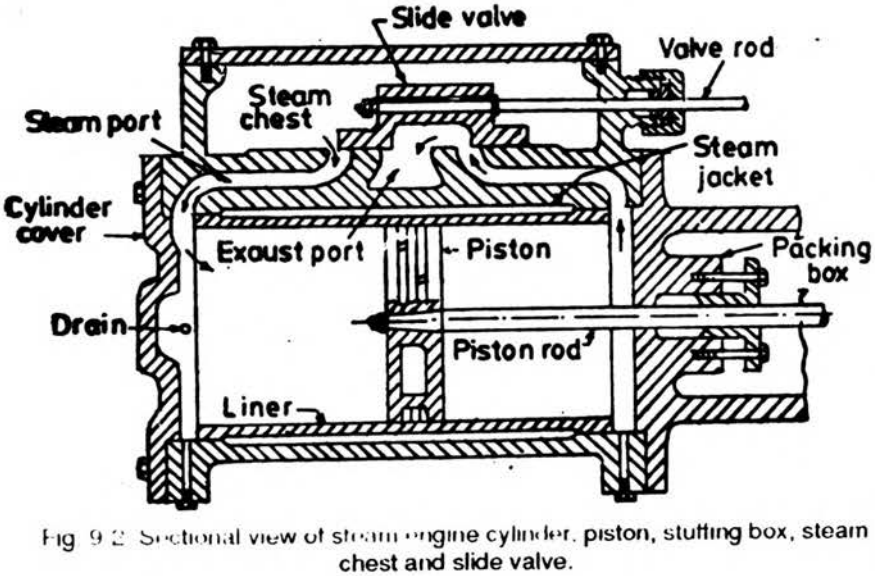


Fig. 9-2 Sectional view of steam engine cylinder, piston, stuffing box, steam chest and slide valve.

their function is to prevent the leakage of steam past the piston rod which moves to and fro.

The **piston** is a cast iron cylindrical disc moving to and fro in the cylinder under the action of the steam pressure. Its function is to convert the heat energy of the steam into mechanical work. Cast iron piston rings make the piston steam tight in the cylinder and thereby prevent the leakage of steam past the piston.

The **crosshead** is a link between piston rod and the connecting rod. It guides the motion of the piston rod and prevents it from bending.

The **connecting rod** helps in converting the reciprocating motion of the piston into rotary motion of the crank. Its one end is connected to the crosshead by means of *gudgeon pin* or *crosshead pin* and other end is connected to the crank.

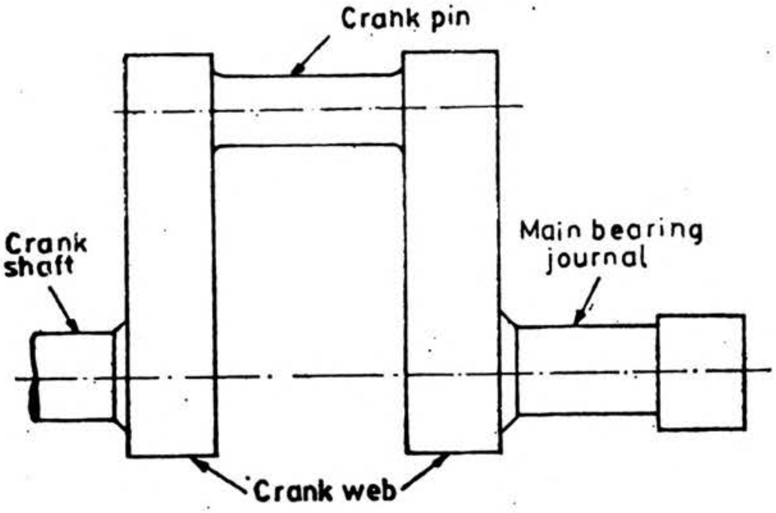


Fig. 9-3. Crank and Crankshaft.

The **crankshaft** (fig. 9-3) is the main shaft of the engine and carries on it the flywheel and the eccentric. It is supported on the main bearings of the engine and is free to rotate in them. It is made of mild steel. Crank formed on the crankshaft works on the lever principle and produces rotary motion of the crankshaft.

The slide valve (fig. 9-2) is situated in the steam chest and its function is to admit the steam from steam chest to the cylinder, and exhaust the steam from the cylinder at the proper moment. The valve gets to and fro motion from the eccentric fitted on the crankshaft.

The **eccentric** is fitted on the crankshaft. *The function of eccentric is to convert the rotary motion of the crankshaft into reciprocating motion of the slide valve.*

The **main bearings** support the engine crankshaft and are fitted on the engine frame. The part of the crankshaft which turns in the bearing is called a *main bearing journal* as shown in fig. 9-3.

The **flywheel** is a heavy cast iron or cast steel wheel mounted on the crankshaft to prevent the fluctuation of engine speed throughout the stroke and to carry the crank smoothly over the dead centres.

The steam **engine governor** is a device for keeping the speed of the engine more or less constant at all loads. For this it controls either the quantity or pressure of the steam supplied to the engine according to the load on the engine.

9.5 Working of a Simple, Double-acting, Condensing Steam Engine

The function of a steam engine is to convert the heat energy of steam into mechanical work. The pressure of the steam acts on the piston and moves it to and fro in the cylinder. It is necessary to have some method of converting this to and fro motion of the piston into a rotary motion, since the rotary motion can be conveniently transmitted from the engine to any other driven machine. This to and fro motion of the

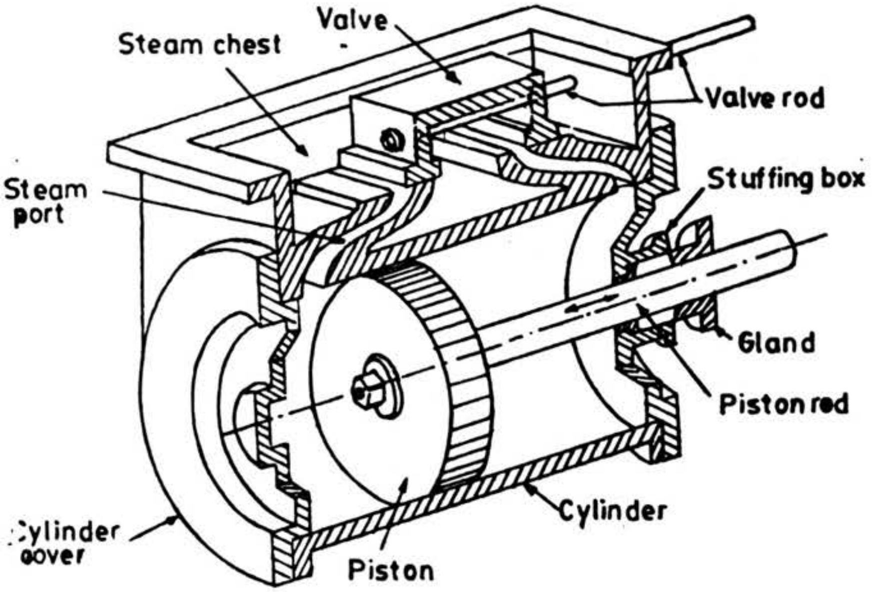


Fig. 9-4. Sectional pictorial view of horizontal steam engine cylinder, piston, stuffing box and gland, steam chest and D-slide valve.

piston is converted into rotary motion with the help of connecting rod and crank of the steam engine.

The steam (fig. 9-4) is first admitted to the cover end (left hand side) of the cylinder from the steam chest when steam admission port is uncovered (opened) by the D-slide valve, while the exhaust steam (which has done work on the piston) on the crank end (right hand side) of the cylinder passes at the same time into a vessel, called condenser (not shown in the figure) through the steam port and the exhaust port as shown in fig. 9-4. This steam admitted to the cover end exerts pressure on surface of the piston and pushes it to crank end (right hand side) of the cylinder.

At the end of this stroke, fresh steam from the steam chest is again admitted by the D-slide valve to the crank end of the cylinder (when admission steam port is opened), while the exhaust steam on the cover end of the cylinder passes at the same time into the condenser through steam port and exhaust port. Thus, the steam at the cover end exhausts while that at the crank end pushes the piston back to its original position.

The D-slide valve gets to and fro motion from the eccentric fitted on the crankshaft.

Thus, two working strokes are completed and the crankshaft turns by one revolution, i.e., the engine is *double-acting.* These operations are repeated.

When the exhaust steam is exhausted to atmosphere, the engine is known as *non-condensing engine.*

The motion of the piston and piston rod moves the crosshead, connecting rod, crank and crankshaft. The motion of the piston, piston rod and crosshead is to and fro. This to and fro motion is coverted into rotary motion with the help of the connecting rod, crank and the crank pin as shown in fig. 9-1. The end of the connecting rod which is attached to the crosshead can only move in a straight line, while the other end attached to the crank pin can move only in a circle, since, the crank carrying the crank pin is free to turn the crankshaft. The motion of the connecting rod is, thus, oscillating. Since, the crank is fixed on the crankshaft, the crankshaft will rotate in its bearing. The flywheel is mounted on the crankshaft.

The following **terms** are useful in understanding the working of the steam engine :

The *cylinder bore* is the inside diameter of the cylinder or the liner.

The *piston stroke* is the distance travelled (or moved) through by the piston from one end of the cylinder to the other end, while the crank is making half a revolution.

Two strokes of the piston are performed per revolution of the crankshaft.

The *forward stroke* is made by the piston while it moves from the head end to the crank end.

The *return stroke* is made while the piston travels from the crank end to the head end.

The *crank throw* or *crank radius* is the distance between the centre of the crankshaft and the centre of the crank pin. This distance is equal to half the travel of the piston, i.e., piston stroke.

The *piston displacement* or *swept volume* is the volume swept by the piston while moving from one end of the cylinder to the other end, while the crank is making half a revolution.

The *average linear piston speed* is the rate of motion of the piston expressed in metres/minute. It is equivalent to twice the product of the piston stroke (l) in metres and the number of revolutions of the engine per minute (N), i.e., piston speed = $2lN$ metres/minute.

The *dead centres* are the positions of the piston at the end of the stroke when the centre lines of the piston rod, the connecting rod and the crank are in the same straight line. There are two such dead centres, one for each end of the piston stroke. For horizontal engines, the two dead centres are known as *inner dead centre* and *outer dead centre*. For vertical engines, the two dead centres are known as *top dead centre* and *bottom dead centre.*

Clearance volume is the volume of the space between the piston and the cylinder cover, when the piston is at the end of the stroke, plus the volume of the steam port leading to this space.

9.6 Hypothetical Indicator Diagram

An indicator diagram is a plot of steam pressure in the cylinder on the basis of steam volume during the cycle of operations. The theoretical indicator diagram can be constructed geometrically or by calculating pressure of steam at different points of the stroke by applying the law, pv = constant, for the expansive working of steam and then raising the ordinates to get the points of the curve. Such a diagram (fig. 9-5) is known as theoretical or hypothetical indicator diagram because it is constructed after making certain assumptions as follows :

- The opening and closing of the ports is sudden.
- There is no pressure drop due to condensation.
- There is no pressure drop due to wire-drawing which is due to restricted port or valve opening.
- There is no compression to create the cushioning effect on the piston at the end of the stroke.
- The steam is admitted at boiler pressure and exhausted at atmospheric or condenser pressure.
- The expansion of steam is hyperbolic, following the law pv = constant.
- The clearance volume is neglected.

Referring to fig. 9-5, steam is admitted at point *a*. It is termed as the point of

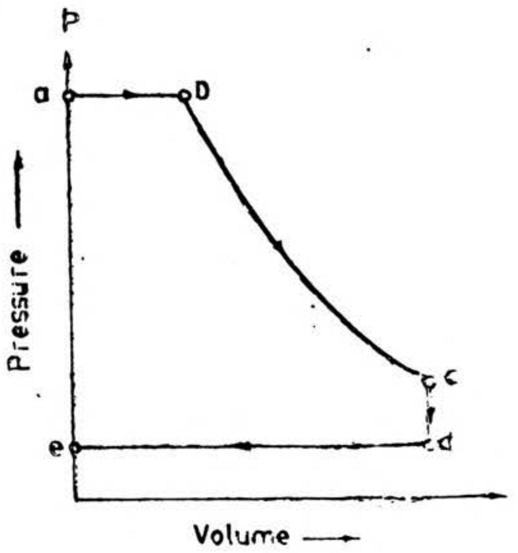


Fig. 9.5 Theoretical or hypothetical indicator diagram without clearance and compression.

admission. Steam is supplied to the cylinder at constant pressure, p_1 up to point of cut-off, *b*. Steam then expands in the cylinder as the piston moves further. This is accompanied by fall in pressure. The law of expansion *bc* is assumed as pv = constant. Expansion is carried to the end of the stroke. At point *c*, release of steam takes place. The pressure falls instantaneously to exhaust pressure or back pressure, p_b represented by line *cd*. Exhaust takes place at constant pressure p_b and conditions throughout the whole exhaust stroke are represented by line *de*.

From a thermodynamic stand point, it should be assumed that the expansion of an ideal diagram is adiabatic. However, such a condition never occurs in practice because some of the steam condenses before cut-off takes place and the steam is always more or less wet at the beginning of expansion, unless the steam is highly superheated at admission. Condensation also proceeds during part of the expansion and then a certain amount of re-evaporation occurs towards the end of expansion. All these factors take the expansion curve away from an adiabatic curve and very close to pv = constant curve or hyperbolic curve.

9.6.1 **Hypothetical Mean Effective Pressure** : As shown earlier, if the pressure and volume are plotted in kPa and m^3 respectively, the area of p - v diagram represents the work done in kJ.

(i) Hypothetical indicator diagram without considering clearance and compression :

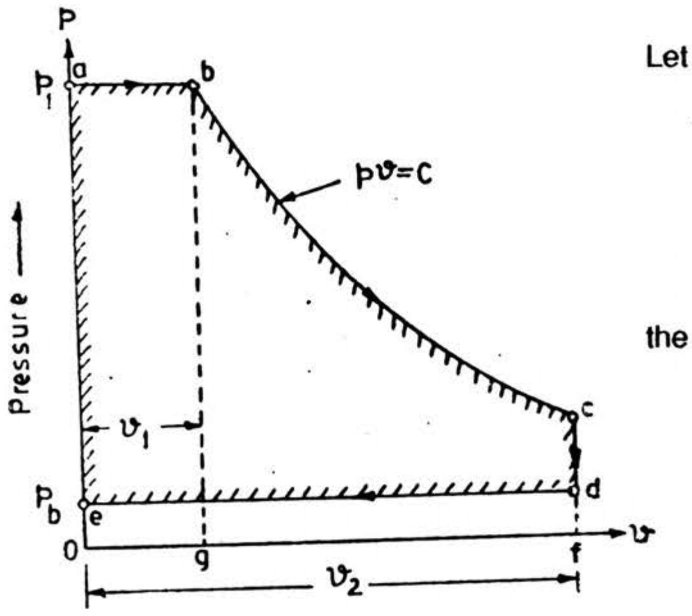


Fig. 9.6. Hypothetical indicator diagram without clearance and compression.

Referring to fig. 9-6.

Let p_1 = admission pressure kPa (or kN/m^2),

p_b = back pressure in kPa (or kN/M^2),

v_1 = volume of steam in the cylinder at cut-off in m^3, and

v_2 = swept volume in m^3.

Work done per cycle in kJ = area of the hypothetical indicator diagram

= area *a-b-c-d-e*

= area *a-b-g-o plus* area *b-c-f-g minus* area *d-e-o-f*

$$= p_1 v_1 + p_1 v_1 \log_e\left(\frac{V_2}{V_1}\right) - p_b v_2$$

$$= p_1 v_1 \left[1 + \log_e\left(\frac{v_2}{v_1}\right)\right] - p_b v_2 \qquad ..(9.1)$$

$$\text{Hypothetical m.e.p.} = \frac{\text{Area of hypothetical indicator diagram}}{\text{Length of the base of hypothetical indicator diagram}}$$

$$= \frac{\text{Work done per cycle in kJ}}{\text{Swept volume in m}^3} \text{ kPa (or kN/m}^2\text{)}$$

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

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

But, $\frac{v_2}{v_1} = r$ (expansion ratio)

$\therefore$ Hypothetical m.e.p. $= \frac{p_1}{r}\,[\,1 + \log_e(r)\,] - p_b$ kPa (or kN/m^2) .. (9.2)

where p_1 and p_b are pressures of steam in kPa (kN/m^2)

It may be noted that $\frac{v_1}{v_2} = \frac{1}{r}$ = cut-off ratio.

Problem-1 : *Steam is admitted to the cylinder of a steam engine at a gauge pressure of 1,050 kPa (1·05 MPa) and is cut-off at ⅓ stroke. The back pressure is 15 kPa. Calculate the hypothetical mean effective pressure in kPa and kN/m² on the piston during the stroke. The barometer reads 96 kPa. Neglect clearance.*

Using eqn. (9-2), Hypothetical m.e.p. $= \frac{p_1}{r}\,[1 + \log_e(r)] - p_b$ kPa

Here absolute pressure, p_1 = gauge pressure + atmospheric pressure

$= 1{,}050 + 96 = 1{,}146$ kPa

Expansion ratio, $r = \frac{1}{\text{cut-off ratio}} = \frac{1}{1/3} = 3$, and Back pressure, $p_b = 15$ kPa.

Substituting the values in eqn. (9.2), we have

$$\text{Hypothetical m.e.p.} = \frac{1{,}146}{3}\,[1 + \log_e(3)] - 15$$

$$= \frac{1{,}146}{3}\,[1 + 1{\cdot}0986] - 15$$

$$= \mathbf{786{\cdot}66 \text{ kPa} = 786{\cdot}66 \text{ kN/m}^2}$$

Note : 1 kPa = 1 kN/m^2 and 1 MPa = 10 bar = 1 MN/m^2.

Hypothetical indicator diagram considering clearance and compression :

Refer to fig. 9-7. The compression starts at *e* and finishes at *f* which is also the point of admission. The admission is continued up to *b*. The expansion curve is represented by *bc*. *c* is the point of release and *de* is exhaust line. The compression curve is represented by *ef*. The expansion and compression curves are assumed to be hyperbolic.

Let v = volume swept by the piston in one stroke in m^3,

c = fractional ratio of clearance volume to swept volume v,

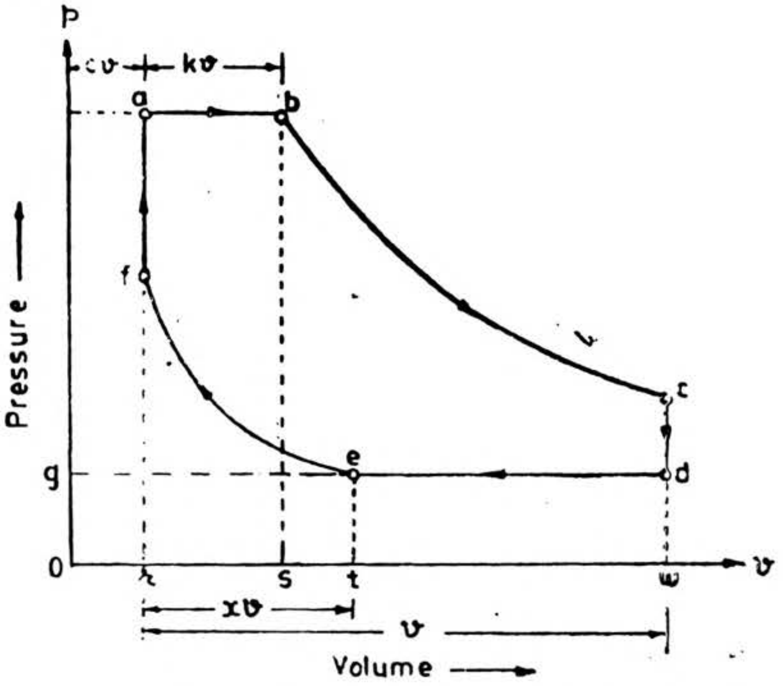


Fig 9.7 Hypothetical indicator diagram with clearance and compression.

x = fractional ratio of the volume between points of compression and admission to the swept volume v,

$k = 1/r$ = fractional ratio of the volume between the points of admission and cut-off to the swept volume v,

p_1 = initial pressure of steam or boiler pressure in kPa, and

p_b = back pressure of steam or condenser pressure in kPa.

Net work done per cycle = area *a–b–c–d–e–f*

= area *a–b–s–r plus* area *b–c–w–s minus* area *e–d–w–t minus* area *f–e–t–r*

$$= p_1 \times kv + p_1 (kv + cv) + \log_e \left[\frac{v + cv}{kv + cv} \right] - p_b\ (v - xv)$$

$$- p_b (xv + cv) \log_e \left[\frac{xv + cv}{cv} \right]$$

$$= v \left[p_1 \left\{ k + (k + c) \log_e \left(\frac{1 + c}{k + c} \right) \right\} - p_b \left\{ (1 - x) + (x + c) \log_e \left(\frac{x + c}{c} \right) \right\} \right] \quad ..(9.3)$$

Thus, hypothetical m.e.p. in kPa $= \dfrac{\text{Work done per cycle in kJ}}{\text{Swept volume in m}^3}$

$$= \frac{v \left[p_1 \left\{ k + (k + c) \log_e \left(\frac{1 + c}{k + c} \right) \right\} - p_b \left\{ (1 - x)\ (x + c) \log_e \left(\frac{x + c}{c} \right) \right\} \right]}{v}$$

Since $k = \dfrac{1}{r}$, Hypothetical m.e.p.

$$= p_1 \left[\frac{1}{r} + \left(\frac{1}{r} + c \right) \log_e \left(\frac{1 + c}{\frac{1}{r} + c} \right) \right] - p_b \left[(1 - x) + (x + c) \log_e \left(\frac{x + c}{c} \right) \right] \quad ..(9.4)$$

If the *compression effect* is neglected, i.e. $x = 0$, the eqn. (9.4) becomes,

$$\text{Hypothetical m.e.p.} = p_1 \left[\frac{1}{r} + \left(\frac{1}{r} + c \right) \log_e \left(\frac{1 + c}{\frac{1}{r} + c} \right) \right] - p_b \left[1 + c \log_e \left(\frac{c}{c} \right) \right]$$

$$= p_1 \left[\frac{1}{r} + \left(\frac{1}{r} + c \right) \log_e \left(\frac{1 + c}{\frac{1}{r} + c} \right) \right] - p_b \quad ..(9.5)$$

If the *effect of clearance volume* is neglected, i.e., $c = o$, the eqn. (9.5) becomes,

$$\text{Hypothetical m.e.p.} = \frac{p_1}{r}\,[1 + \log_e (r)] - p_b \qquad ..(9.6)$$

This is same as expression (9.2) directly derived earlier.

Point of cut-off is the point at which the supply of steam to the cylinder is closed. Cut-off ratio is expressed as the ratio of the volume of steam in the cylinder when the piston is at the point of cut-off, and the volume of steam in the cylinder when the piston is at the end of the stroke. If the clearance is not considered while taking the volume of steam at the point of cut-off and at the end of the stroke, the cut-off is known as *apparent cut-off.* If the clearance is taken into account, this ratio is known as *real cut-off.* Thus, referring to fig. 9-7,

$$\text{Apparent cut-off ratio} = \frac{kv}{v} = k = \frac{1}{r} \qquad ..(9.7)$$

$$\text{Real cut-off ratio} = \frac{kv + cv}{v + cv} = \frac{k + c}{1 + c} \qquad ..(9.8)$$

Ratio of expansion is the ratio of the volume of steam at the end of the stroke, and the volume of steam at the point of cut-off. It is the reciprocal of the cut-off ratio. If no account is taken of the clearance volume while considering this ratio, the ratio is called *apparent* or *normal ratio of expansion.* If the clearance volume is considered, the ratio is known as *real or actual ratio of expansion.*

$$\text{Apparent expansion ratio} = \frac{v}{kv} = \frac{1}{k} = r \qquad ..\ (9.9)$$

$$\text{and real expansion ratio} = \frac{v + cv}{kv + cv} = \frac{1 + c}{k + c} \qquad ..\ (9.10)$$

Problem–2 : *Steam is admitted to the cylinder of a single cylinder engine at a pressure of 1,400 kPa and it cut-off at 0·25 of the stroke. The back pressure is 120 kPa and the clearance volume is 10% of the swept volume. Calculate the hypothetical (theoretical) mean effective pressure on the piston during the stroke.*

Using eqn. (9.5),

$$\text{Hypothetical m.e.p.,} = p_1\left[\frac{1}{r} + \left(\frac{1}{r} + c\right)\log_e\left(\frac{1 + c}{\frac{1}{r} + c}\right)\right] - p_b$$

Here, $\frac{1}{r} = 0{\cdot}25$, $c = 0{\cdot}1$, $p_1 = 1{,}400$ kPa, and $p_b = 120$ kPa.

$$\therefore \text{ Hypothetical m.e.p.} = 1{,}400\left[0{\cdot}25 + (0{\cdot}25 + 0{\cdot}1)\log_e\left(\frac{1 + 0{\cdot}1}{0{\cdot}25 + 0{\cdot}1}\right)\right] - 120$$

$$= 1{,}400\ [0{\cdot}25 + 0{\cdot}35 \times 1{\cdot}1442] - 120 = \mathbf{790{\cdot}66\ kPa}$$

Problem–3 : *The cylinder of a non-condensing steam engine is supplied with steam at 1,220 kPa. The clearance volume is 1/10th of the stroke volume and the cut-off takes place at 1/4th of the stroke. If the pressure at the end of compression is 550 kPa, calculate the value of mean effective pressure of the steam on the piston. Assume that expansion and compression are hyperbolic. Take back pressure as 110 kPa.*

Let v = stroke volume, and cv = clearance volume,

then the volume of steam at the beginning of the compression stroke = $v\,(x + c)$

where, x = compression ratio without considering clearance volume, and the volume of steam at the end of compression = cv.

Applying hyperbolic law between the points of beginning and end of compression,

$$110\ (c + x)\ v = 550 \times cv$$

i.e. $110\,(0{\cdot}1 + x)\,v = 550 \times 0{\cdot}1\,v$ i.e., $0{\cdot}1 + x = 0{\cdot}5 \therefore x = 0{\cdot}4$

Here, $\frac{1}{r} = \frac{1}{4}$, $c = 0{\cdot}1$, $p_1 = 1{,}220$ kPa, $p_b = 110$ kPa, and $x = 0{\cdot}4$.

Now, using eqn. (9.4), Hypothetical m.e.p.

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

$$= 1{,}220\left[\frac{1}{4} + \left(\frac{1}{4} + 0{\cdot}1\right)\log_e\left(\frac{1 + 0{\cdot}1}{\frac{1}{4} + 0{\cdot}1}\right)\right] - 110\left[(1 - 0{\cdot}4) + (0{\cdot}4 + 0{\cdot}1)\log_e\left(\frac{0{\cdot}4 + 0{\cdot}1}{0{\cdot}1}\right)\right]$$

$$= 1{,}220\ [0{\cdot}25 + 0{\cdot}35\ \log_e 3{\cdot}14] - 110\ [0{\cdot}6 + 0{\cdot}5 \times \log_e 5]$$

$$= 1{,}220\ [0{\cdot}25 + 0{\cdot}35 \times 1{\cdot}1442] - 110\ [0{\cdot}6 + 0{\cdot}5 \times 1{\cdot}6094] = \mathbf{639{\cdot}1\ kPa}$$

Problem–4 : *In a steam engine, steam is admitted for 40% of the working stroke. Clearance volume is 10% of the swept volume. Calculate the apparent and real values of cut-off and expansion ratios.*

Assuming the swept volume = 100 units,

Apparent ratio of cut-off = 40/100 = 0·4 and

Apparent ratio of expansion = 100/40 = 2·5

$$\text{Real ratio of cut-off} = \frac{\text{total volume of steam at cut–off}}{\text{total cylinder volume}}$$

$$= \frac{\text{total volume of steam at cut–off}}{\text{swept volume + clearance volume}} = \frac{40 + 10}{100 + 10} = \mathbf{0{\cdot}454}$$

$$\text{Real ratio of expansion} = \frac{\text{total cylinder volume}}{\text{total volume of steam at cut–off}} = \frac{100 + 10}{40 + 10} = \mathbf{2{\cdot}2}$$

9.7 Cylinder Condensation

If saturated steam is brought in contact with a surface colder than temperature of the steam, condensation of steam begins immediately. During condensation the steam gives up part of its latent heat and becomes wetter.

During the period of exhaust, the cylinder walls are cooled by contact with the relatively cool low pressure exhaust steam. When the hot steam from boiler is admitted to the cylinder, a part of it condenses as it comes in contact with previously cooled cylinder walls during the exhaust stroke. The water thus formed by the process of condensation is deposited in the form of a thin film on the walls of the cylinder. The heat received by the cylinder walls from the fresh hot steam will be given back to the deposited film of water during the time the steam expands and consequently falls in temperature after the cut-off takes place. This transferred heat from the cylinder walls to the water film will re-evaporate some of the water into steam at the end of expansion stroke.

The condensation, thus, takes place during the early part of the admission stroke, re-evaporation occurs partly towards the end of the expansion stroke, and partly during exhaust stroke. The re-evaporation during expansion period behind the piston increases the total work done. But, the re-evaporated steam during the exhaust period in a single cylinder engine passes away to waste, without doing any useful work. However, in a compound steam engine this re-evaporated steam will be utilized in the next cylinder to which the steam is led from the first cylinder. Any degree of condensation results in loss of energy available for doing useful work, and this loss may vary from 10% to 40% of the available energy of the steam.

The **methods** adopted to *reduce the amount of condensation of steam in the cylinder* are :

- Obtaining the steam from the boiler in superheated state and lagging the steam pipe from boiler to engine by non-conducting material such as asbestos.
- Jacketing the cylinder with hot steam from the boiler.
- Compounding the cylinders, and thereby reducing the range of variation of temperature in each cylinder.

9.8 Actual Indicator Diagram

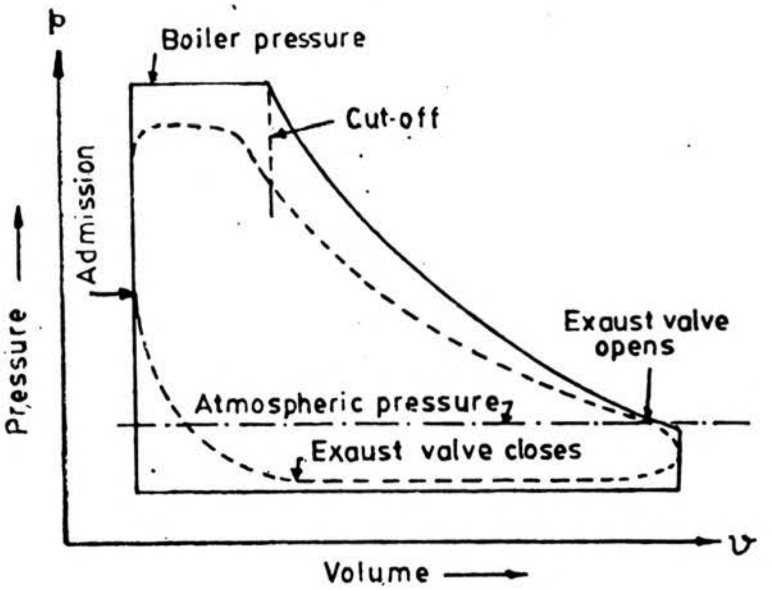


Fig. 9 8. Superimposed actual and hypothetical indicator diagrams.

An actual indicator diagram obtained on a steam engine is more likely to be of the form shown dotted in fig. 9-8. The hypothetical indicator diagram of this engine is shown in full lines.

The difference between two diagrams is on account of certain practical factors given below, which were not considered in arriving at the hypothetical indicator diagram :

- The big drop in the steam pressure between boiler and the engine cylinder is due to condensation caused by loss of heat in steam pipes, friction losses in the steam supply pipe and admission valve and to wire-drawing in the valves. This pressure difference will tend to increase toward the point of cut-off due to increasing velocity of the piston and consequent increased demand for steam and due to condensation of steam in the cylinder.
- The opening and closing of the ports is a gradual process as the valve moves over it. The pressure changes at the opening and closing of ports are not as sudden as shown on the hypothetical diagram, and there is rounding-off of the actual diagram at the beginning of admission, cut-off and release.
- The actual expansion curve is not a true hyperbola owing to the varying intercharge of heat through the cylinder walls. At the commencement of the expansion, the steam in

the cylinder is hotter than the cylinder wall; this causes condensation of steam and the volume of steam consequently decreases. Near the end of the expansion stroke, owing to the low pressure of steam, the steam is colder than the cylinder walls; this causes heat to flow from the cylinder walls to the steam which tends to re-dry the steam.

.. The release (opening of exhaust port) occurs before the end of the expansion stroke, and the time factor in the opening of the exhaust port will cause the rounding-off of the toe of the actual diagram.

.. Exhaust pressure in the cylinder will not be quite so low as the condenser pressure because the size of the exhaust port is limited and therefore, the exhaust from the port is not quick.

.. There is rounding-off of the heel of the actual diagram due to closing of the exhaust port before the end of the stroke. This results in compression of the steam in the cylinder, which serves as a *soft cushion* for bringing the piston to rest at the end of the stroke. Admission occurs just before the end of the compression stroke.

It will be seen from fig. 9-8 that the area of the actual indicator diagram is less than the area of the hypothetical indicator diagram. The ratio between the areas of these two diagrams is known as the *diagram factor.* Or

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

$$= \frac{\text{mean height of actual indicator diagram}}{\text{mean height of hypothetical indicator diagram}}$$

$$= \frac{\text{m.e.p. from actual indicator diagram}}{\text{m.e.p. from hypothetical indicator diagram}} \quad ..(9.11)$$

The diagram factor varies slightly with different types of engines, and its average value is about 0·75.

9.8.1 Indicator : Instrument used for drawing diagrams showing actual pressure – volume relations within the engine cylinder during one revolution of the engine crank is called *indicator.* The diagram is drawn on a piece of paper usually called an indicator card. The diagram showing actual pressure–volume relations within the engine cylinder is called an *indicator diagram* (fig. 9-8). The length of the indicator diagram represents the length of the stroke to a reduced scale and its height at any point represents the pressure on the piston at the corresponding point in the stroke. For double-acting steam engines a separate diagram is taken for each end of the cylinder.

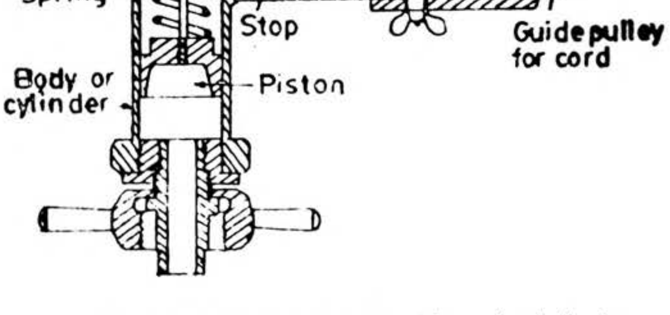


Fig. 9-9. Sectional view of inside spring indicator.

A common form of steam engine indicator, shown in fig. 9-9, consists of two main parts :

- a small cylinder, communicating at one end with the engine cylinder, and
- a drum for holding the indicator card.

The small cylinder contains a close-fitted piston to which is attached a strong helical spring. The piston is forced upward against the resistance of this spring by the steam pressure in the engine cylinder and forced downward by the spring when the steam pressure drops. The end of the piston rod of the indicator piston is attached through a series of links to a pencil lever that has a pencil point on its free end. The links are so designed that they magnify the motion of the indicator piston at the pencil point and yet cause it to move in a truly vertical line.

A coiled spring attached to the inside of the drum resists the pull of the cord on the outward stroke and keeps the cord tight on the return stroke.

In operation, the indicator card is wrapped around the indicator drum and held against it by spring clamps. One end of a fine cord is attached to and wrapped several times arround the groove at the base of the drum, and the other end is attached to some form of reducing motion, which in turn is connected to a reciprocating part of the engine, preferably to the engine crosshead pin. When indicator cylinder is put in communication with the engine cylinder, the lever carrying the pencil point will rise and fall according to the rise and fall of steam pressure. At the same time the drum will be rotated back and forth in some proportion to the engine stroke. Now, if the pencil point is pressed against the indicator card, it will trace an indicator diagram on the indicator card.

9.9. Power and Efficiencies

9.9.1 Indicated Power : The power developed in the cylinder of any engine is commonly known as *indicated power.* Thus, indicated power is the rate of doing work on the moving piston in the cylinder. It is called indicated power because an instrument known as the indicator, is used to measure it. The power developed in the cylinder of any engine can be determined, if the engine cylinder diameter, piston stroke and speed are known, and if an indicator diagram with its spring scale or number is available. The first step is to find the mean effective pressure. The mean effective pressure is the average effective pressure on the piston.

Mean height of indicator diagram in mm, $h = \dfrac{\text{area of indicator diagram in mm}^2}{\text{length of indicator diagram in mm}}$

Indicated mean effective pressure in kPa or kN/m^2 = mean height, h in mm × spring scale or spring number, kPa or kN/m^2 per 1 mm elongation.

The mean effective pressure is a constant pressure which will do the same work as the actual varying pressure. Hence, the actual varying pressure in the engine cylinder can be replaced by the mean effective pressure. To find the power, we must know the work done per second. This will be equal to the amount of work done per stroke multiplied by number of working strokes per second.

If p_m = actual mean effective pressure of steam in N/m^2 or Pa,

a = area of the piston in m^2,

l = length of piston stroke in metre, and

N = number of revolutions made by the engine per second (r.p.s),

Then, driving force on the piston during a stroke = $p_m \times a$ newtons

and Work done per stroke = $p_m \times a \times l$ N-m (newton-metre) or J (joules)

Since two working strokes are obtained in one revolution in a double-acting steam engine,

$$\text{Work done per second} = p_m \times a \times l \times 2N \text{ J/s or W (watts)}$$

$$\left.\begin{aligned} \therefore \text{ Indicated power} &= 2 \times p_m \times a \times l \times N \text{ W} \\ \text{or} &= \frac{2 \times p_m \times a \times l \times N}{1{,}000} \text{ kW} \end{aligned}\right] \quad ..(9.12)$$

A portion of the power developed in the engine cylinder is absorbed in overcoming the friction of the moving parts of the engine itself. The remainder is available for doing the required work. The power absorbed in overcoming the frictional resistance of the moving parts of the engine is termed *frictional power.*

9.9.2. **Brake Power** : The actual power available from the engine for doing useful work is termed the brake power or shaft power. The brake power of an engine can be determined by a brake of some kind applied to the brake pulley of the engine. The arrangement for the determination of brake power of the engine is known as *dynamometer.*

Dynamometers are broadly divided into two classes :

— *Absorption dynamometres,* and

— *Transmission dynamometers.*

In **absorption dynamometer,** the power available from the engine is absorbed in the form of friction at the brake. In transmission dynamometer, the power available from the engine is not wasted in friction. **Transmission dynamometer** transmits the power and measures it at the same time.

Absorption brake dynamometers are those that absorb the power to be measured by friction. This power absorbed in friction is finally dissipated in the form of heat energy. Common forms of absorption dynamometers are : the rope brake, the prony brake, the fan brake, the hydraulic brake and the electrical brake.

Rope brake dynamometer is a convenient type of brake and can be applied to the flywheel of a moderate size engine. The rope brake consists of a double rope passed round the brake wheel as shown in fig. 9-10, the upper end is connected to a spring balance suspended from overhead and the lower end carries the load *W* in newtons. The load *W* is usually called dead load. The ropes are generally held apart by the wooden blocks.

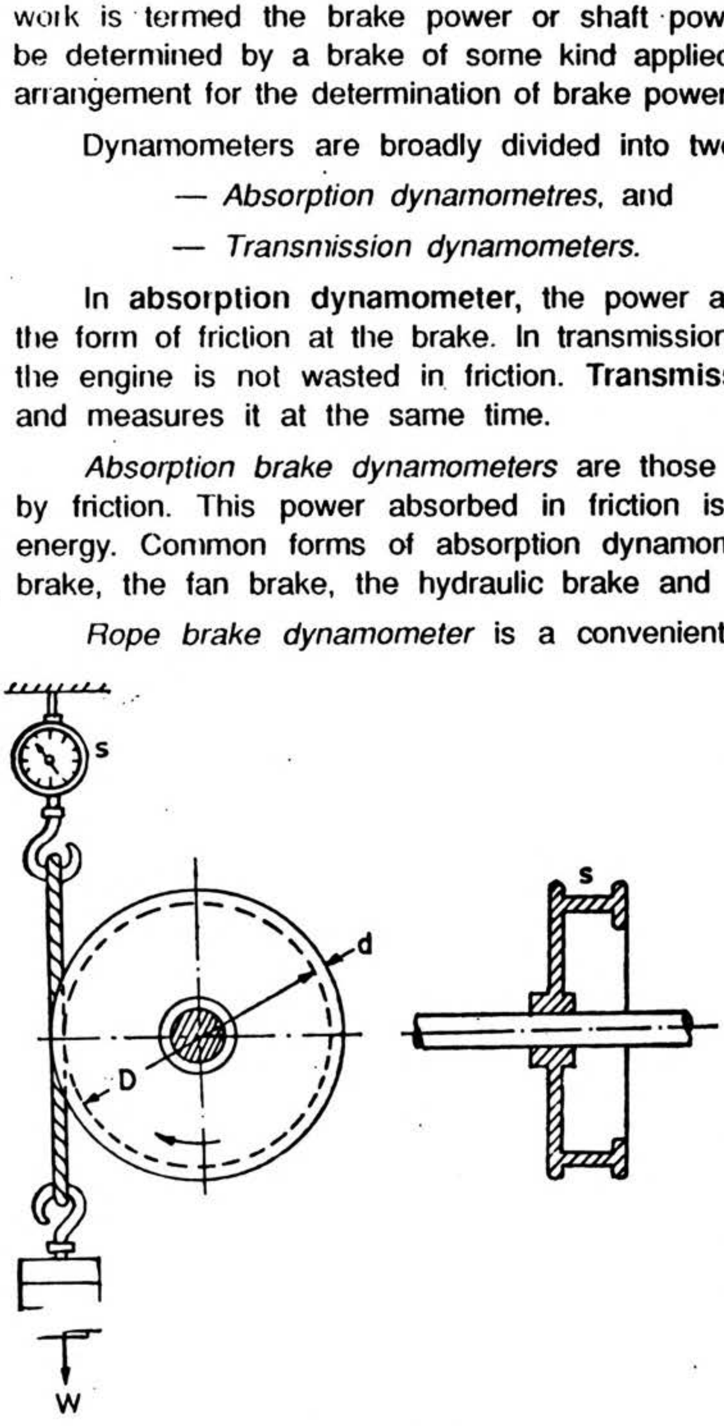


Fig. 9-10. Rope brake dynamometer.

The pull *S* in newtons indicated by the spring balance, is helping to turn the wheel while the load *W* is opposing its rotation. The direction of the rotation of the wheel is shown by the arrow. At the given load the whole power developed by the engine is absorbed by the friction produced at the rim of the wheel. Since, the whole power is converted into heat, the rim of the wheel must be water cooled.

The net resistance against the wheel or the frictional force or the net load = $(W - S)$ newtons. If R is the effective radius of the wheel in metres which works as the torque arm for frictional resistance, i.e. $R = \frac{D + d}{2}$ (wheel D = diameter of the brake wheel and d = diameter of the brake rope) then, the frictional torque = the resisting force × arm = $(W - S) \times R$ N-m.

If N = number of revolutions made by the engine per second (r.p.s), then the number of radians per second = $2\pi N$.

The work absorbed = $(W - S)\ R \times 2\pi N$ N.m/s or J/s or W (watts)

$$\left.\begin{aligned} \therefore \text{ Brake power} &= \frac{(W-S)\,R \times 2\,\pi\,N}{1{,}000}\ \text{kW} \\ &= \frac{\text{Torque} \times 2\,\pi\,N}{1{,}000}\ \text{kW} \end{aligned}\right] \qquad ..(9.13)$$

Power lost in overcoming the friction of rotating and sliding parts of the engine is the difference between the indicated power and brake power.

i.e. Friction power = Indicated power − Brake power .. (9.14)

9.9.3 Mechanical, Thermal and Overall Efficiencies : The performance of steam engines is usually stated in terms of the followings :

— Mechanical efficiency,

— Thermal efficiency, and

— Overall efficiency.

The mechanical efficiency of engine takes into account the energy loss due to the friction of the moving parts of the engine, and is calculated from the relation between the brake power and the indicated power. The power lost in friction is the difference between the indicated power and the brake power.

It is generally assumed that friction power is the same at all loads at constant speed. However, various tests indicate that the friction power generally increase with the load but not to any considerable extent.

The *mechanical efficiency* is defined as the ratio of the power output of the engine, as measured by the brake, to the power developed by the steam in the engine cylinder, as obtained from the indicator diagram,

$$\text{i.e. Mechanical efficiency, } \eta_m = \frac{\text{Brake power}}{\text{Indicated power}} \qquad .. (9.15)$$

The mechanical efficiency of high speed steam engines at full load is generally between 80 to 90 per cent. With forced lubrication, its value at full load is still higher.

The *thermal efficiency* is defined as the ratio of the heat converted into useful work, to the heat supplied. The heat supplied per kg of steam to the engine may be worked out as the difference between the enthalpy, H_1 of steam at conditions existing at engine stop valve and the enthalpy h_2 of water at the temperature of engine exhaust (heat of water or condensate returned to hot-well). So, heat (net) supplied in kJ per kg of steam = $H_1 - h_2$.

The useful work obtained may be worked out either from indicated power or brake power. If the heat equivalent of indicated power developed by the engine is considered as the output, the efficiency is termed *indicated thermal efficiency*,

$$\text{i.e. Indicated thermal efficiency, } \eta_i = \frac{\text{Heat equivalent of indicated power in kJ/sec}}{\text{Heat supplied in steam in kJ/sec.}}$$

$$= \frac{\text{Indicated power in kW}}{m_s (H_1 - h_2)} \qquad ..(9.16)$$

where, m_s = steam consumption in kg/sec,

H_1 = enthalpy of 1 kg of steam supplied in kJ, and

h_2 = enthalpy of 1 kg of water of exhaust steam in kJ.

$$\text{Brake thermal efficiency, } \eta_b = \frac{\text{Heat equivalent of brake power in kJ/sec.}}{\text{Heat supplied in steam in kJ/sec.}}$$

$$= \frac{\text{Brake power in kW}}{m_s (H_1 - h_2)} \qquad ..(9.17)$$

where, m_s = steam consumption in kg/sec.

H_1 = enthalpy of 1 kg of steam supplied in kJ, and

h_2 = enthalpy of 1 kg of water of exhaust steam in kJ.

Comparing eqns. (9.15), (9.16) and (9.17),

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{Brake thermal efficiency}}{\text{Indicated thermal efficiency}} \qquad ..(9.18)$$

The overall efficiency of *steam engine plant* is defined as the ratio of the power output of the engine as measured by the brake to the heat energy supplied by the fuel,

i.e. Overall efficiency of steam engine plant

$$= \frac{\text{Heat equivalent of brake power in kJ/sec.}}{\text{Heat supplied by fuel in kJ/sec.}}$$

$$= \frac{\text{brake power in kW}}{m_f \times C.V.} \qquad ..(9.19)$$

where, m_f = fuel supplied to the boiler in kg/sec, and

$C.V.$ = calorific value of fuel used in the boiler in kJ/kg.

Problem-5 : *A single-cylinder, double-acting steam engine of 20 cm diameter and 40 cm stroke is supplied with steam at 834 kPa and exhausts at 14·7 kPa. Cut-off takes place at ⅓rd stroke and the engine runs at 2 r.p.s. Using a diagram factor of 0·7, estimate the actual mean effective pressure and the indicated power of the engine.*

Here, p_1 = 834 kPa, Expansion ratio, $r = \frac{1}{\text{Cut-off}} = \frac{1}{1/3} = 3$, and p_b = 14·7 kPa.

Using eqn. (9.2), hypothetical m.e.p.

$$= \frac{p_1}{r}\left[1 + \log_e (r)\right] - p_b = \frac{834}{3}\left[1 + \log_e (3)\right] - 14{\cdot}7 = 568{\cdot}6 \text{ kPa}$$

$\therefore$ Actual m.e.p., p_m = diagram factor, f × theoretical m.e.p.

$= 0{\cdot}7 \times 568{\cdot}6 =$ **398·02 kPa.**

Using eqn. (9.12), Indicated power = $2 \times p_m \times a \times l \times N$ watts where, p_m = actual m.e.p. = $398{\cdot}02 \times 10^3$ Pa

a = area of the piston = $\frac{\pi}{4}\left(\frac{20}{100}\right)^2$ m^2,

l = length of piston stroke = $\frac{40}{100}$ m, and

N = number of revolutions made by the engine = 2 r.p.s.

Substituting the above values,

$$\text{Indicated power} = 2 \times 398{\cdot}92 \times 10^3 \times \frac{\pi}{4}\left(\frac{20}{100}\right)^2 \times \frac{40}{100} \times 2$$

$$= 20{,}000 \text{ W or } \mathbf{20\ kW.}$$

Problem-6 : *Find the dimensions of a simple steam engine cylinder to develop indicated power of 50 kW. The steam supply is at 800 kPa. The engine makes 2 r.p.s. and is double-acting. Cut-off is at ⅜ of the stroke, diagram factor is 0·65 and the back pressure is 120 kPa. The stroke is 1·5 times the diameter of the cylinder, and the clearance volume is 8% of the stroke volume.*

Using eqn. (9.5), Hypothetical m.e.p. = $p_1\left[\frac{1}{r} + \left(\frac{1}{r} + c\right)\log_e\left(\frac{1+c}{\frac{1}{r}+c}\right)\right] - p_b$

Here, $c = \frac{\text{clearance volume}}{\text{stroke volume}} = \frac{8}{100} = 0{\cdot}08$ (assuming stroke volume as 100 units),

Expansion, ratio, $r = \frac{1}{\text{cut–off}} = \frac{1}{3/8} = \frac{8}{3} = 2{\cdot}666$, p_1 = 800 kPa and p_b = 120 kPa.

$$\text{Hypothetical m.e.p.} = 800\left[\frac{1}{2{\cdot}666} + \left(\frac{1}{2{\cdot}666} + 0{\cdot}08\right)\log_e\left(\frac{1+0{\cdot}08}{\frac{1}{2{\cdot}666}+0{\cdot}08}\right)\right] - 120$$

$$= 800\ [0{\cdot}375 + 0{\cdot}455 \times 0{\cdot}8642] - 120 = 495 \text{ kPa}$$

∴ Actual m.e.p., p_m = hypothetical m.e.p. × f = 495 × 0·65 = 322 kPa

Let cylinder diameter = d metre, then piston stroke, l = 1·5d metre.

Indicated power = 2 × p_m × a × l × N watts

[where p_m is indicated m.e.p. in p_a (pascals)]

i.e. $50 \times 10^3 = 2 \times 322 \times 10^3 \times \frac{\pi}{4} d^2 \times 1{\cdot}5d \times 2$

$\therefore\ d^3 = 0{\cdot}03295$

∴ Diameter of the cylinder = $\sqrt[3]{0{\cdot}03295}$ = 0·32 m or = **32 cm.**

and Piston stroke, l = 1·5d = 1·5 × 32 = **48 cm.**

Problem-7 : *The diameter of the cylinder of a simple, double-acting steam engine is 30 cm. The stroke is 38 cm. The steam is admitted to the cylinder at a pressure of 700 kPa and is cut-off when the piston has advanced 9 cm from the dead centre position. Assume a diagram factor of 0·7, calculate the indicated power of the engine running at 150 r.p.m. Assume back pressure of 110 kPa. Neglect clearance effect.*

Here, Cut-off = 9/38 = 0·237 and expansion ratio, r = 1/0·237 = 4·22, p_1 = 700 kPa, and p_b = 110 kPa.

$$\text{Hypothetical m.e.p.} = \frac{p_1}{r}\left[1 + \log_e(r)\right] - p_b$$

$$= \frac{700}{4{\cdot}22}\left[1 + \log_e(4{\cdot}22)\right] - 110 = \mathbf{294\ kPa}$$

∴ Actual m.e.p., p_m = hypothetical m.e.p. × f = 294 × 0·7 = 206 kPa

Indicated power = 2 × p_m × a × l × N watts

$$= 2 \times 206 \times 10^3 \times \frac{\pi}{4} \times (0{\cdot}3)^2 \times \frac{38}{100} \times \frac{150}{60}$$

$$= 27{,}670 \text{ watt or } \mathbf{27{\cdot}67\ kW}$$

Problem-8 : *The following data relate to a test on a double-acting, single-cylinder steam engine :*

Indicated power, 80 kW; engine speed, 140 r.p.m.; cylinder diameter, 30 cm; piston stroke, 45 cm; steam pressure at admission, 1,100 kPa; cut-off at ⅓rd stroke and back pressure, 40 kPa. Neglecting the effect of clearance volume, calculate the diagram factor.

Here, expansion ratio, r = 1/⅓ = 3, p_1 = 1,100 kPa, and p_b = 40 kPa.

Using eqn. (9.2), $$\text{Hypothetical m.e.p.} = \frac{p_1}{r}\left[1 + \log_e(r)\right] - p_b$$

$$= \frac{1{,}100}{3}\left[1 + \log_e(3)\right] - 40 = \mathbf{729{\cdot}48\ kPa}$$

Indicated power = 2 × p_m × a × l × N W

$$\text{i.e. } 80 \times 10^3 = 2 \times p_m \times \frac{\pi}{4} \times (0{\cdot}3)^2 \times 0{\cdot}45 \times \frac{140}{60}$$

$$\therefore \text{Actual m.e.p., } p_m = \frac{80 \times 10^3 \times 4 \times 60}{2 \times 3{\cdot}14 \times (0{\cdot}3)^2 \times 0{\cdot}45 \times 140} = 5{,}39{,}210 \text{ Pa or } 539{\cdot}21 \text{ kPa}$$

$$\text{Diagram factor, } f = \frac{\text{Actual m.e.p. in kPa}}{\text{Hypothetical m.e.p. in kPa}} = \frac{539{\cdot}21}{729{\cdot}48} = \mathbf{0{\cdot}739}$$

Problem-9 : *A double-acting, steam engine receives steam at a pressure of 7 bar. The cut-off takes place at ¼th stroke and the diagram factor is 0·7. The indicated power of the engine is 50 kW and the engine runs at 3 revolutions per second. Find the dimensions of the steam engine if the piston stroke is 1·5 times the diameter of the cylinder. The back pressure is 1·1 bar. Neglect clearance effect.*

Here, Expansion ratio, $r = \frac{1}{\text{cut–off}} = \frac{1}{1/4} = 4$, p_1 = 7 bar, p_b = 1·1 bar and f = 0·7.

Using eqn. (9.2), $$\text{Hypothetical m.e.p.} = \frac{p_1}{r}\left[1 + \log_e(r)\right] - p_b$$

$$= \frac{7}{4}\left[1 + \log_e(4)\right] - 1{\cdot}1 = 3{\cdot}08 \text{ bar}$$

Actual m.e.p., p_m = f × theoretical m.e.p. = 0·7 × 3·08 = 2·156 bar

Indicated power = $2 \times p_m \times a \times l \times N$ W.

i.e. $50 \times 10^3 = 2 \times (2.156 \times 10^5) \times \frac{\pi}{4} d^2 \times 1.5d \times 3$

$$\therefore\ d^3 = \frac{50 \times 10^3 \times 4}{2 \times 2.156 \times 10^5 \times 3.14 \times 1.5 \times 3} = 0.0328$$

$\therefore\ d = \sqrt[3]{0.0328}$ = **0.32 m, or 32 cm**

$\therefore$ Piston stroke, $l = 1.5d = 1.5 \times 32 =$ **48 cm**

Problem-10 : *A simple, double-acting steam engine having a cylinder diameter 35 cm and stroke 53 cm takes steam at a pressure of 800 kPa and exhaust takes place at 100 kPa. Find the increase in power when cut-off is changed from 0.4 to 0.5 of the stroke. The engine speed is 200 r.p.m. and the diagram factor can be taken as 0.85 in both the cases. Neglect clearance effect.*

(i) Here, Expansion ratio, $r = 1/0.4 = 2.5$, $f = 0.85$, $P_1 = 800$ kPa and $p_b = 100$ kPa

Using eqn. (9.2), Hypothetical m.e.p. $= \frac{p_1}{r}\,[1 + \log_e (r)] - p_b$

$$= \frac{800}{2.5}\,[1 + \log_e (2.5)] - 100 = 513 \text{ kPa.}$$

$\therefore$ Actual m.e.p., p_m = hypothetical m.e.p. $\times f = 513 \times 0.85 = 436$ kPa

Indicated power = $2 \times p_m \times a \times l \times N$ W

$$= 2 \times 436 \times 10^3 \times \frac{\pi}{4}\,(0.35)^2 \times \frac{53}{100} \times \frac{200}{60}$$

= 1,48,200 W or 148.2 kW

(ii) Expansion ratio, $r = 1/1.5 = 2$, $p_1 = 800$ kPa, $p_b = 100$ kPa, $f = 0.85$

Hypothetical m.e.p. $= \frac{P_1}{r}[1 + \log_e (r)] - p_b = \frac{800}{2}\,[1 + \log_e(2)] - 100 = 578$ kPa

$\therefore$ Actual m.e.p., $p_m = 578 \times 0.85 = 491$ kPa

Indicated power = $2 \times p_m \times a \times l \times N$ W

$$= 2 \times 491 \times 10^3 \times \frac{\pi}{4} \times (0.35)^2 \times \frac{53}{100} \times \frac{200}{60}$$

= 1,66,900 W or 166.9 kW

$\therefore$ Increase in power when cut-off is changed from 0.4 to 0.5

= 166.9 − 148.2 = **18.7 kW**

Problem-11 : *Calculate the theoretical steam consumption in kg per kW per hour of a single-cylinder, double-acting steam engine from the following data :*

Diameter of cylinder. 25 cm; stroke, 45 cm; r.p.s., 3; steam pressure, 1,100 kPa; dryness fraction of steam supplied, 0.95; back pressure, 110 kPa; and cut-off takes place at 1/3rd stroke for both sides. Assume a diagram factor of 0.8 and neglect clearance.

Here, Expansion ratio, $r = \frac{1}{\text{cut–off}} = \frac{1}{1/3} = 3$, $p_1 = 1{,}100$ kPa, and $p_b = 110$ kPa

Hypothetical m.e.p. $= \frac{p_1}{r}\,[1 + \log_e(r)] - p_b$

$$= \frac{1,100}{3}\,[1 + \log_e 3] - 110 = 660 \text{ kPa.}$$

Actual m.e.p., $p_m = 660 \times 0{\cdot}8 = 528$ kPa

Indicated power $= 2 \times p_m \times a \times l \times N$ W

$$= 2 \times 528 \times 10^3 \times \frac{\pi}{4}(0{\cdot}25)^2 \times \frac{45}{100} \times 3$$

$$= 70,000 \text{ W or } 70 \text{ kW}$$

Volume of cylinder $= \frac{\pi}{4}\,d^2 \times l = \frac{\pi}{4}\left(\frac{25}{100}\right)^2 \times \frac{45}{100} = 0{\cdot}0221 \text{ m}^3.$

Since cut-off takes place at ⅓rd stroke,

Volume of steam admitted per stroke $= \frac{0{\cdot}0221}{3} = 0{\cdot}00737 \text{ m}^3.$

Since, the engine is double-acting,

Volume of steam admitted per revolution $= 0{\cdot}00737 \times 2 = 0{\cdot}01474 \text{ m}^3.$

$\therefore$ Volume of steam admitted per hour $= 0{\cdot}01474 \times 3 \times 3,600 = 159 \text{ m}^3.$

From steam tables at 1,100 kPa (11 bar), $v_s = 0{\cdot}1775 \text{ m}^3/\text{kg}.$

$\therefore$ Volume of 1 kg of wet steam admitted $= xv_s = 0{\cdot}95 \times 0{\cdot}1775 = 0{\cdot}1686 \text{ m}^3.$

$\therefore$ Theoretical steam consumption in kg per hour

$$= \frac{\text{Volume of steam admitted/hr}}{\text{Volume of 1 kg of wet steam}} = \frac{159}{0{\cdot}1686} = \mathbf{943 \text{ kg.}}$$

$\therefore$ Theoretical steam consumption in kg per kW per hour $= \frac{943}{70} =$ **13·47 kg/kW-hr.**

Problem-12 : *The following data were obtained during the trial of single-cylinder, double-acting steam engine :*

Cylinder diameter, 28 cm; stroke, 40 cm; speed. 4 r.p.s.; area of the indicator diagram, 10 cm², length of indicator diagram, 7·6 cm; scale of indicator spring, 200 kPa/cm; dead load on the brake, 2,119 newtons; spring balance reading, 157 newtons; circumference of brake wheel, 5 m; circumference of brake rope, 8 cm; steam supplied per hour, 435 kg.

Determine the indicated power, brake power, mechanical efficiency, steam consumption per kW per hour on indicated power basis, steam consumption per kW per hour on brake power basis, and power lost in friction.

Actual m.e.p., $p_m = \frac{\text{area of the indicator diagram}}{\text{length of the indicator diagram}} \times \text{spring scale}$

$$= \frac{10}{7{\cdot}6} \times 200 = 263 \text{ kPa}$$

Indicated power $= 2 \times p_m \times a \times l \times N$ W

$$= 2 \times 263 \times 10^3 \times \frac{\pi}{4}(0{\cdot}28)^2 \times \frac{40}{100} \times 4 = 51{,}820 \text{ W or } \mathbf{51{\cdot}82 \text{ kW}}$$

Dia. of brake wheel $= \frac{5 \times 100}{\pi} = 159$ cm, and Dia. of brake rope $= \frac{8}{\pi} = 2{\cdot}54$ cm.

Effective radius of brake wheel, $R = \frac{159 + 2{\cdot}54}{2 \times 100} = 0{\cdot}8077$ m.

Using eqn. (9.13), Brake Power $= \frac{(W - S) \times 2\pi \times R \times N}{1{,}000}$

$$= \frac{(2{,}119 - 157) \times 2\pi \times 0{\cdot}8077 \times 4}{1{,}000} = \mathbf{39{\cdot}82 \text{ kW}}$$

Using eqn. (9.15), Mechanical efficiency, $\eta_m = \frac{\text{Brake Power}}{\text{Indicated Power}}$

$$= \frac{39{\cdot}82}{51{\cdot}82} = 0{\cdot}768 \text{ or } \mathbf{76{\cdot}8\%}$$

Steam consumption per kW per hour on Indicated power basis $= \frac{435}{51{\cdot}82} =$ **8·394 kg**

Steam consumption per kW per hour on Brake power basis $= \frac{435}{39{\cdot}82} =$ **10·924 kg**

Using eqn. (9.14), Power lost in friction

= Indicated Power − Brake Power

$= 51{\cdot}82 - 39{\cdot}82 =$ **12 kW**

Problem-13 : *The following data were obtained during the trial of a single-cylinder, double-acting steam engine :*

Speed	.. *104·5 r.p.m*
Cylinder diameter	.. *22 cm*
Piston stroke	.. *30 cm*
M.E.P. (each end)	.. *120 kPa*
Effective brake radius	.. *60 cm*
Dead load on the brake	.. *726 newtons*
Spring balance reading	.. *176 newtons*
Pressure of steam supplied	.. *700 kPa (7 bar)*
Steam used per hour	.. *57 kg*
Condensate temperature	.. *50°C*

Steam supplied was dry and saturated. Calculate the mechanical efficiency, and the indicated and brake thermal efficiencies of the engine.

indicated power $= 2 \times p_m \times a \times l \times N$ watts

$$= 2 \times 120 \times 10^3 \times \frac{\pi}{4}(0{\cdot}22)^2 \times \frac{30}{100} \times \frac{104{\cdot}5}{60}$$

$$= 4{,}767 \text{ W or } 4{\cdot}767 \text{ kW}$$

$$\text{Brake Power} = \frac{(W-S)\,R \times 2\pi \times N}{1{,}000}\ \text{kW}$$

$$= \frac{(726-176) \times \frac{60}{100} \times 2\pi \times \frac{104{\cdot}5}{60}}{1{,}000} = 3{\cdot}607\ \text{kW}$$

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{Brake Power}}{\text{Indicated Power}} = \frac{3{\cdot}607}{4{\cdot}767} = 0{\cdot}7567 \text{ or } \mathbf{75{\cdot}67\%}$$

At 7 bar (from steam tables) H_1 = 2,763·5 kJ/kg.

Heat remaining in condensate, $h_2 = 4{\cdot}187 \times (50 - 0) = 209{\cdot}3$ kJ/kg.

Steam used per second, $m_s = \frac{57}{3{,}600} = 0{\cdot}0158$ kg/sec.

∴ Heat supplied in steam in kJ per second = $m_s\,(H_1 - h_2)$ kJ/sec.

$$= 0{\cdot}0158(2{,}763{\cdot}5 - 209{\cdot}3) = \mathbf{40{\cdot}36\ kJ/sec.}$$

Heat equivalent of indicated power = 4·767 kJ/sec.

Using eqn. (9.16), Indicated thermal efficiency,

$$\eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Heat supplied in steam in kJ per sec.}} = \frac{4{\cdot}767}{40{\cdot}36} = 0{\cdot}1181 \text{ or } \mathbf{11{\cdot}81\%}$$

Using eqn. (9.17), Brake thermal efficiency,

$$\eta_b = \frac{\text{Heat equivalent of brake power in kJ per sec.}}{\text{Heat supplied in steam in kJ per sec.}} = \frac{3{\cdot}607}{40{\cdot}36} = 0{\cdot}0894 \text{ or } \mathbf{8{\cdot}94\%}$$

Problem-14 : *A single-cylinder, double-acting steam engine of 28 cm bore and 45 cm stroke works between a supply pressure of 980 kPa (9·8 bar) and a back pressure of 15 kPa (0·15 bar). Assuming a diagram factor of 0·7 and neglecting clearance volume, estimate the indicated power at 3 r.p.s. if the cut-off is at ⅓rd stroke.*

If the above engine consumes 900 kg of dry saturated steam per hour, determine the indicated thermal efficiency of the engine.

Here, Expansion ratio, $r = \frac{1}{\text{cut–off}} = \frac{1}{1/3} = 3$, $p_1 = 980$ kPa, $p_b = 15$ kPa, and $f = 0{\cdot}7$.

$$\text{Actual m.e.p. } p_m = f\left[\frac{p_1}{r}(1 + \log_e r) - p_b\right]$$

$$= 0{\cdot}7\left[\frac{980}{3}(1 + \log_e 3) - 15\right] = 469{\cdot}37\ \text{kPa}$$

$$\text{Indicated power} = 2 \times p_m \times a \times l \times N$$

$$= 2 \times 469{\cdot}37 \times 10^3 \times \frac{\pi}{4}\left(\frac{28}{100}\right)^2 \times \frac{45}{100} \times 3$$

$$= 77{,}980\ \text{W or } \mathbf{77{\cdot}98\ kW}$$

$$\text{Indicated thermal efficiency, } \eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec}}{\text{Heat supplied in steam in kJ per sec}}$$

Heat equivalent of indicated power in kJ per sec. = 77·98 kJ/sec.

Heat supplied in steam per sec. = m_s (H_1 − h_2) kJ/sec.

where, m_s = steam supplied per sec. = 900/3,600 = 0·25 kg.

H_1 = enthalpy of 1 kg of dry saturated steam at 9·8 bar = 2,777·3 kJ/kg, and

h_2 = enthalpy of 1 kg of water of exhaust steam at 0·15 bar = 225·94 kJ/kg.

Heat supplied in the steam/sec. = 0·25 × (2,777·3 − 225·94) = 637·84 kJ/sec.

$$\therefore \text{ Indicated thermal eff., } \eta_i = \frac{77{\cdot}98}{637{\cdot}84} = 0{\cdot}1222 \text{ or } \mathbf{12{\cdot}22\%}$$

Problem-15 : *The following data were obtained during the trial of a single-cylinder, double-acting steam engine :*

Engine speed, 4 r.p.s.; Cylinder diameter, 20 cm; Piston stroke, 30 cm; Indicated mean effective pressure (both ends), 1·2 bar; Effective brake radius, 60 cm; Net load on the brake wheel, 490 newtons; Pressure of steam supplied, 7 bar; Steam used per hour, 120 kg; Condensate temperature, 50°C. Steam supplied is dry saturated. Calculate : (a) the mechanical efficiency, (b) the indicated thermal efficiency, (c) the brake thermal efficiency, and (d) the steam consumption in kg per kW-hr on brake power basis.

(a) Indicated power = 2 × p_m × a × l × N

$$= 2 \times 120 \times 10^3 \times \frac{\pi}{4} \times (0{\cdot}2)^2 \times 0{\cdot}3 \times 4 = 9{,}042 \text{ W or } 9{\cdot}042 \text{ kW}$$

$$\text{Brake power} = \frac{(W-S)\,R \times 2\pi N}{1{,}000}$$

$$= \frac{490 \times 0{\cdot}6 \times 2\pi \times 4}{1{,}000} = 7{\cdot}392 \text{ kW}$$

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{7{\cdot}392}{9{\cdot}042} = 0{\cdot}8165 \text{ or } \mathbf{81{\cdot}65\%}$$

$$\text{(b) Indicated thermal efficiency, } \eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Heat supplied in steam in kJ per sec.}}$$

Heat equivalent of indicated power = 9·042 kJ/sec.

Heat supplied in steam = m_s × (H_1 − h_2) kJ/sec,

where H_1 = Enthalpy of 1 kg of steam at 7 bar from steam tables = 2,763·5 kJ/kg.

h_2 = Enthalpy of condensate/kg = (50 − 0) × 4·187 = 209·35 kJ/kg, and

$$m_s = \text{Steam supplied in kg per sec.} = \frac{120}{3{,}600}$$

$$\therefore \text{ Heat supplied in steam, } = \frac{120}{3{,}600}\,(2{,}763{\cdot}5 - 209{\cdot}35) = 85{\cdot}13 \text{ kJ/sec.}$$

$$\therefore \text{ Indicated thermal efficiency, } \eta_i = \frac{9{\cdot}042}{85{\cdot}13} = 0{\cdot}1062 \text{ or } \mathbf{10{\cdot}62\%}$$

$$\text{(c) Brake thermal efficiency, } \eta_b = \frac{\text{Heat equivalent of brake power in kJ per sec.}}{\text{Heat in steam supplied in kJ per sec.}}$$

$$= \frac{7{\cdot}392}{85{\cdot}13} = 0{\cdot}0868 \text{ or } \mathbf{8{\cdot}68\%}$$

(d) Steam consumption on brake power basis in kg/kW-hr.

$$= \frac{\text{steam consumption in kg per hour}}{\text{brake power in kW}} = \frac{120}{7{\cdot}392} = \mathbf{16{\cdot}23\ kg/kW\text{-}hr.}$$

Problem-16 : *A single-cylinder, double-acting steam engine admits steam at a pressure of 800 kPa (8 bar) and temperature 200°C. The cut-off takes place at 0·4 of the stroke and engine runs at 200 r.p.m. Assuming a diagram factor of 0·85, calculate the indicated power of the engine when the cylinder diameter is 35 cm and stroke is 53 cm. The back pressure is 100 kPa (1 bar).*

If the above engine consumes 1,960 kg of steam per hour, determine the indicated thermal efficiency of the engine. Take k_p for superheated steam as 2·1 kJ/kg K.

Here, Expansion ratio, $r = 1/0{\cdot}4 = 2{\cdot}5$, $p_1 = 800$ kPa, $p_b = 100$ kPa.

Hypothetical m.e.p. $= \frac{p_1}{r}\,[1 + \log_e (r)] - p_b$

$$= \frac{800}{2{\cdot}5}\,[1 + \log_e (2{\cdot}5)] - 100 = 513 \text{ kPa}$$

Actual m.e.p., p_m = hypothetical m.e.p. $\times f = 513 \times 0{\cdot}85 = 436$ kPa

Indicated power $= 2 \times p_m \times a \times l \times N$

$$= 2 \times 436 \times 10^3 \times \frac{\pi}{4}(0{\cdot}35)^2 \times \frac{53}{100} \times \frac{200}{60}$$

$$= 1{,}48{,}200 \text{ W or } = 148{\cdot}2 \text{ kW}$$

Heat equivalent of indicated power = 148·2 kJ/sec.

Mass of steam supplied in kg per second, $m_s = \frac{1{,}960}{3{,}600} = 0{\cdot}544$ kg

From steam tables, at 8 bar, $H_s = 2{,}769{\cdot}1$ kJ/kg, $t_s = 170{\cdot}43$°C,

and at 1 bar, enthalpy of saturated water, $h_2 = 417{\cdot}46$ kJ/kg.

$H_1 = H_s + k_p\,(t_{sup} - t_s) = 2{,}769{\cdot}1 + 2{\cdot}1\,(200 - 170{\cdot}43) = 2{,}831{\cdot}2$ kJ/kg.

Heat supplied in steam in kJ per second $= m_s \times (H_1 - h_2)$

$$= 0{\cdot}544\,(2{,}831{\cdot}2 - 417{\cdot}46) = \mathbf{1{,}313{\cdot}1\ kJ/sec.}$$

Indicated thermal efficiency,

$$\eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Heat supplied in steam in kJ per sec.}} = \frac{148{\cdot}2}{1{,}313{\cdot}1} = 0{\cdot}113 \text{ or } \mathbf{11{\cdot}3\%}$$

Problem-17 : *Dry saturated steam at 8 bar (800 kPa) is admitted into the cylinder of a single-cylinder, double-acting steam engine. The cylinder diameter is 30 cm and stroke is 60 cm and cut-off is at 50%$ of the stroke. The back pressure is 1 bar (100 kPa). Taking the diameter of piston rod as 4 cm and assuming a diagram factor of 0·75 and mechanical efficiency of 70%, calculate the brake power of the engine at 250 r.p.m.*

If the indicated thermal efficiency of the above engine is 15 per cent, calculate the steam consumption in kg per kW per hour on indicated power basis.

Here, Expansion ratio, $r = 1/0{\cdot}5 = 2$, $p_1 = 800$ kPa, $p_b = 100$ kPa.

Hypothetical m.e.p. $= \frac{p_1}{r}\,(1 + \log_e r) - p_b$

$$= \frac{800}{2}\ [1 + \log_e 2] - 100 = 578 \text{ kPa}$$

$\therefore$ Actual *m.e.p.*, p_m = theoretical m.e.p. × f = 578 × 0·75 = 433 kPa (for each end)

Area of cylinder (cover end) $= \frac{\pi}{4} d^2 = \frac{\pi}{4}(30)^2 = 707 \text{ cm}^2$

Effective area of cylinder (crank end) $= \frac{\pi}{4}(d^2 - d_1^2) = \frac{\pi}{4}(30^2 - 4^2) = 695 \text{ cm}^2$

Indicated power of engine = indicated power cover end + indicated power crank end

$$= \left[433 \times 10^3 \times \frac{707}{10^4} \times \frac{60}{100} \times \frac{250}{60}\right] + \left[433 \times 10^3 \times \frac{695}{10^4} \times \frac{60}{100} \times \frac{250}{60}\right]$$

= 76,500 + 75,200 = 1,51,700 W or 151·7 kW

Brake power = Mechanical efficiency × Indicated power = 0·7 × 151·7 = **106·19 kW**

From steam tables, at 8 bar, H_1 = 2,769·1 kJ/kg, and

at 1 bar, heat remaining in condensate, h_2 = 417·46 kJ/kg.

Heat supplied in steam per second = $m_s(H_1 - h_2) = m_s$ (2,769·1 − 417·46) **kJ/sec.**

$$\text{Indicated thermal efficiency, } \eta_i = \frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\text{Heat supplied in steam in kJ/sec.}}$$

$$\text{i.e. } 0{\cdot}15 = \frac{151{\cdot}7}{m_s(2{,}769{\cdot}1 - 417{\cdot}46)}$$

$$\therefore\ m_s = 0{\cdot}43 \text{ kg/sec.}$$

$\therefore$ Steam consumption per hour = 0·43 × 3,600 = 1,548 kg/hour.

Steam consumption in kg per kW per hour on indicated power basis,

$$= \frac{1{,}548}{151{\cdot}7} = \mathbf{10{\cdot}2\ kg/kW\text{-}hr.}$$

Problem-18 : *The following are the average readings taken during a trial on a steam engine plant :*

Indicated Power, 30 kW; Brake Power, 22·5 kW; Pressure of steam supplied, 5·5 bar; Quality of steam supplied, 5% wet; Condenser vacuum, 647·5 mm of Hg; Barometer reading, 760 mm of Hg; Steam consumption, 360 kg per hour; 45 kg of coal with a calorific value of 33,500 kJ/kg is supplied per hour. Calculate : (a) the mechanical and indicated thermal efficiencies of the steam engine, and (b) the overall efficiency of steam engine plant (from coal to brake).

(a) Mechanical efficiency, $\eta_m = \frac{\text{Brake Power}}{\text{Indicated Power}} = \frac{22{\cdot}5}{30} = 0{\cdot}75$ or 75%

Mass of steam supplied per second, $m_s = \frac{360}{3{,}600} = 0{\cdot}1$kg

At 5·5 bar, h = 655·93 kJ/kg and L = 2,097 kJ/kg (from steam tables).

Enthalpy of 1 kg of wet steam at 5·5 bar,

$$H_1 = h_1 + x_1 L_1 = 655{\cdot}93 + 0{\cdot}95 \times 2{,}097 = 2{,}648 \text{ kJ/kg.}$$

Condenser or back pressure = (760 − 647·5) 0·1333 = 15 kPa (∵ 1 mm of Hg = 0·1333 kPa)

At 15 kPa (0·15 bar), heat remaining in condensate,

h_2 = 225·94 kJ/kg (from steam tables).

Heat supplied in steam/sec. = $m_s(H_1 - h_2)$ = 0·1 (2,648 − 225·94) = 242·2 kJ/sec.

Indicated thermal efficiency,

$$\eta_i = \frac{\text{Heat equivalent of Indicated power in kJ/sec.}}{\text{Heat supplied by steam in kJ/sec.}} = \frac{30}{242{\cdot}2} = 0{\cdot}1239 \text{ or } 12{\cdot}39\%$$

(b) Using eqn. (9.19), overall efficiency of the steam engine plant

$$= \frac{\text{Heat equivalent of brake power in kJ/sec.}}{\text{Heat supplied by coal in kJ/sec.}}$$

$$= \frac{\text{Brake power in kW}}{m_f \times \text{C.V.}}$$

where m_f = Mass of coal supplied in kg per sec., and

$C.V$ = Calorific value of coal used in the boiler in kJ/kg.

$$\therefore \text{ Overall efficiency of steam engine plant} = \frac{22{\cdot}5}{\frac{45}{3{,}600} \times 33{,}500} = 0{\cdot}0537 \text{ or } 5{\cdot}37\%$$

9.10 Heat Engine Cycle

A heat engine cycle is a series of thermodynamic processes through which a working fluid or substance passes in a certain sequence. At the completion of the cycle, the working fluid returns to its original thermodynamic state, i.e. the working fluid at the end of 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 working fluid. It is then the object of the heat engine cycle to convert as much of this heat energy as possible into useful work. The heat energy which is, thus, not converted, is rejected by the working fluid during some process of the cycle.

Any machine designed to carry out a thermodynamic cycle and thus to convert 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*.

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 supplied and the heat rejected 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,

Q_1 = heat received during each cycle, and

Q_2 = heat rejected during each cycle.

Then, $Q = Q_1 - Q_2$

If W = net work done during the cycle,

Then, $W = Q = Q_1 - Q_2$

The efficiency of a heat engine cycle is defined as the ratio of the available heat energy of the cycle for during work, and the heat received during the cycle. Thus,

$$\text{Efficiency, } \eta = \frac{\text{Heat equivalent of the net work of the cycle}}{\text{Heat received during the cycle}}$$

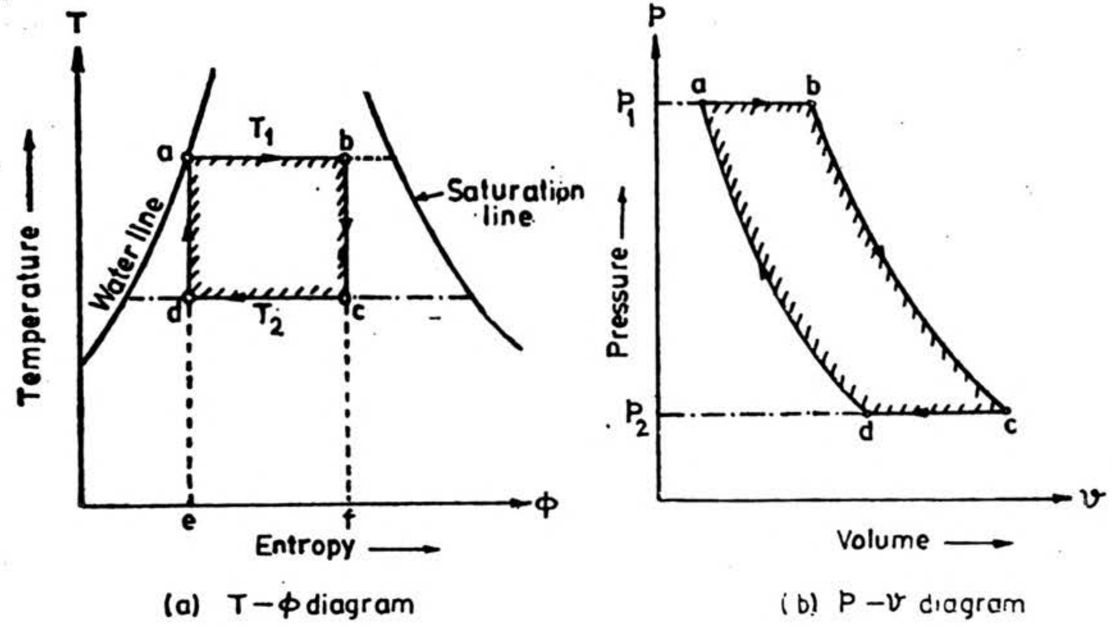
$$= \frac{Q}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{W}{Q_1} \quad ..(9.20)$$

The definition of efficiency given above is applicable to any type of heat engine cycle. Hence, the expression for efficiency given by the eqn. (9.20) is known as *theoretical 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.

9.11 Steam Power Cycles

Heat engine cycles, using steam as the working fluid, are Carnot and Rankine cycles. Carnot cycle is a theoretical cycle which serves as a yard-stick for comparison purposes. All actual steam engines operate on theoretical cycle, known as the *Rankine cycle.*

9.11.1 Carnot Cycle : The cycle is represented in fig. 9-11 on $T - \Phi$ and $p - v$ diagrams. It consists of following four operations – two constant pressure (isothermal) operations (*a–b*) and (*c–d*), and two frictionless adiabatic or isentropic operations (*b–c*) and (*d–a*) :



(a) T – Φ diagram (b) P – ʋ diagram

Fig. 9-11. Carnot cycle for steam (in wet region).

- — Heat is supplied at constant temperature T_1 and constant pressure p_1, where water at its saturation temperature T_1 is heated to form wet steam of dryness fraction x_1. This operation (isothermal expansion) is represented by (*a–b*).
- — In the next operation (*b–c*), this steam is expanded isentropically to temperature T_2 and pressure p_2. The condition of steam after expansion is represented by point *c*.
- — Then heat is rejected at constant pressure p_2 and constant temperature T_2. Steam becomes wetter as it is exhausted and cooled from *c* to *d*. This operation (isothermal compression) is represented by (*c–d*).
- — Lastly, the wet steam at *d* is compressed isentropically till the steam comes to its initial state of temperature T_1 and pressure p_1. This operation is represented by (*d-a*).

Thus, the cycle is completed.

Referring to the $T - \Phi$ diagram of fig. 9-11(a), heat supplied at constant temperature T_1 during (*a-b*) is represented by area *a-b-f-e* and is equal to T_1 $(\Phi_b - \Phi_a)$ or T_1 $(\Phi_c - \Phi_d)$.

The amount of heat rejected (*c-d*) at constant temperature T_2 is represented by area *c-d-e-f* and is equal to T_2 $(\Phi_c - \Phi_d)$.

As there is no exchange of heat during isentropic operations (*b-c*) and (*d-a*),

Net work done = heat supplied during operation (*a-b*) – heat rejected during operation (*c-d*)

$$= T_1 \ (\Phi_c - \Phi_d) \ - \ T_2 \ (\Phi_c - \Phi_d) \ = \ (T_1 \ - \ T_2) \ (\Phi_c - \Phi_d).$$

$$\therefore \text{ Efficiency of the Carnot cycle } = \frac{\text{work done}}{\text{heat supplied}} = \frac{(T_1 - T_2)(\Phi_c - \Phi_d)}{T_1 (\Phi_c - \Phi_d)} = \frac{T_1 - T_2}{T_1} \quad ..(9.21)$$

Although Carnot cycle is thermodynamically simple, yet it is extremely difficult to operate in practice, because the isothermal compression (*c-d*) must be stopped at *d*, so that subsequent isentropic compression (*d-a*) restores the fluid to its initial state *a*.

If superheated steam is used, the cycle would be still more difficult to operate in practice, owing to the necessity of supplying the superheat at constant temperature instead of constant pressure, as it is customary. In a practical cycle, limits of pressure and volume are far more easily realised than limits of temperature *so that no practical engine operates on the Carnot cycle, although all modern cycles aspire to achieve it as it is more efficient than Rankine cycle.* So it is an ideal cycle for comparison purposes.

9.11.2 Rankine Cycle : In a steam plant, the supply of heat and rejection of heat is more easily performed at constant pressure than at constant temperature, and although engines have operated on this principle since the time of Watt, yet it was not until 1844, that an attempt was made by Rankine to calculate the maximum possible work that could be developed by an engine using dry saturated steam, between the pressure limits of the boiler and the condenser. Two years later, Clausius developed a more general expression for the maximum thermal efficiency of a steam engine by allowing for the steam being wet initially.

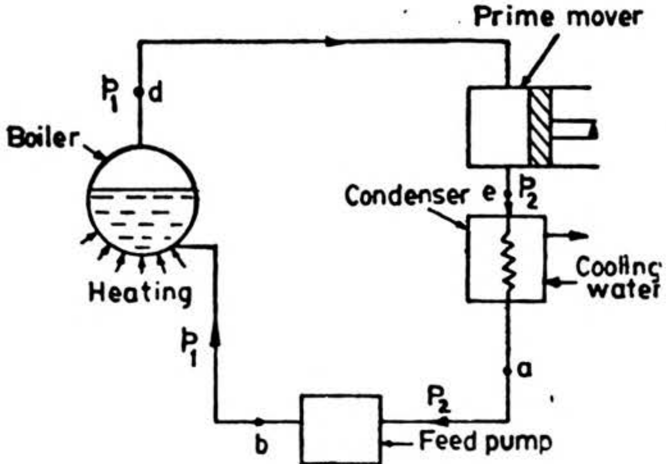


Fig. 9-12. Organs of the Rankine cycle.

The ideal Rankine cycle is used as a yard-stick for the determination of the best performance obtainable in a simple steam engine cycle operating under the specific steam pressure.

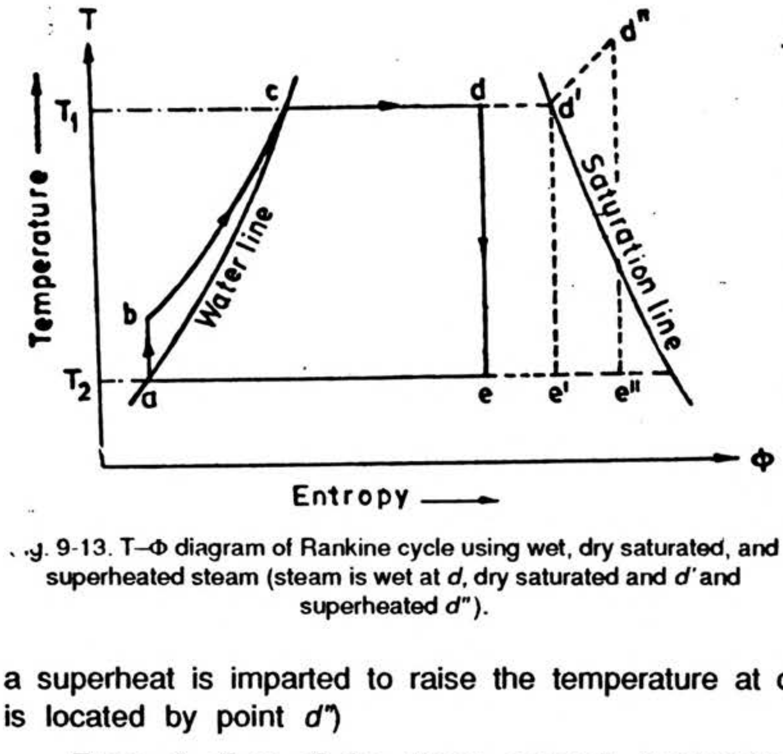
Rankine cycle is a modified Carnot cycle. *Except for the isentropic compression (d-a) on the Carnot cycle (fig. 9-11), the Rankine and Carnot cycle, are the same.*

The ideal Rankine cycle is represented on $T - \Phi$ diagram in fig. 9-13. A flow diagram of a condensing steam plant working on the Rankine cycle is shown in fig. 9-12.

The following assumptions are made in the working of the Rankine cycle :

- — The working fluid (water) is pumped into the boiler, evaporated into steam in the boiler, expanded in the prime-mover (steam engine or steam turbine), condensed in the condenser and returned to the feed pump and circulated again in a closed circuit.
- — Heat is added only in the boiler and is rejected only in the condenser. There is no transfer of heat between the working fluid and the surroundings at any place except the boiler and the condenser.
- — There is no pressure drop in the piping system.
- — Expansion in the prime-mover occurs without friction or heat transfer, i.e. the expansion is isentropic, in which case the entropy of the working fluid entering and leaving the prime-mover is same.
- — The working fluid (water) is not undercooled in the condenser, i.e. temperature of the water leaving the condenser is same as the saturation temperature corresponding to the exhaust pressure.

In the Rankine cycle, we commence with one kilogram of water at the lower temperature and pressure T_2 and p_2 respectively as shown at point *a* in fig. 9-13. The pressure of the water is then raised to p_1 by frictionless adiabatic compression (*a-b*) in the feed pump. The increase in temperature consequent on this compression may be of the order of a few degrees, and is represented by vertical line (*a-b*) on $T - \Phi$ diagram. From *b* to *c* the liquid receives sensible heat at constant pressure p_1 (temperature increases to T_1) to be followed by evaporation at constant pressure p_1, which may be partial at *d* (i.e. wet steam) or complete at *d′* (i.e. dry saturated steam) or frequently a superheat is imparted to raise the temperature at constant pressure p_1 to T_{sup} (which is located by point *d″*)



g. 9-13. T–Φ diagram of Rankine cycle using wet, dry saturated, and superheated steam (steam is wet at *d*, dry saturated and *d′* and superheated *d″*).

From *d, d′* or *d″* the steam expands isentropically to *e, e′* or *e″* respectively until its pressure becomes p_2 and temperature T_2. The last operation (*ea, e′a* or *e″a*) is condensation at constant temperature T_2 and constant pressure p_2 (i.e. isothermal compression), until the working fluid is returned to its original state at *a*.

The process *abc* has been greatly exaggerated on the $T - \Phi$ diagram (fig. 9-13) to illustrate the process. If plotted actually, it is difficult to differentiate between *abc* and *ac* (along the water line), and in normal practice, it is assumed to be *ac*.

During the operation of the cycle, heat is added in the boiler, the work is performed in the prime-mover (steam engine), heat is rejected in the condenser, and the work is done on water as it passes through the feed pump. The heat equivalent of work done in pumping feed-water into the boiler is so small that it is usually neglected, i.e. triangular strip *abc* in the liquid region is neglected.

Let H_1 = Enthalpy of steam at pressure p_1 as it enters the prime-mover at condition d, d' or d''.

H_2 = Enthalpy of steam at pressure p_2 as it leaves the prime-mover at condition e, e' or e'', and

h_2 = Enthalpy of water at pressure p_2 as it enters the feed pump at condition a.

Work done by the prime-mover supplied with wet steam (at condition d)

$$= H_d - H_e = H_1 - H_2 \text{ kJ per kg of steam.}$$

Heat rejected in the condenser $= H_e - H_a = H_2 - h_2$ kJ per kg of steam.

Heat supplied = work done + heat rejected

$$= (H_1 - H_2) + (H_2 - h_2) = H_1 - h_2 \text{ kJ per kg of steam.}$$

$$\text{Efficiency of the Rankine cycle} = \frac{\text{Work done in kJ per kg of steam}}{\text{Heat supplied in kJ per kg of steam}}$$

$$= \frac{H_1 - H_2}{H_1 - h_2} \qquad \text{.. (9.22)}$$

From the given values of p_1, p_2 and condition of steam before entering the prime-mover (at point d), the condition of steam after isentropic expansion d-e at point e (fig. 9-13) is evaluted by equating entropies at d and e and using entropy values from steam tables or can be directly obtained from $H - \Phi$ chart by drawing vertical line as shown in fig. 9-21.

The Rankine cycle, *ignoring the effect of feed pump work,* is represented on $p - v$ diagram in fig. 9-14. The work done per cycle is given by area a-b-c-d. This area can be divided in three parts viz. work done by steam during admission (b-c) work done by steam during isentropic expansion (c-d), and work done on the steam during exhaust (d-a). Thus,

Work done by steam on the piston during admission (b-c) $= p_1v_1$

$$\text{Work done during isentropic expansion } (c\text{-}d) = \frac{p_1v_1 - p_2v_2}{n - 1}$$

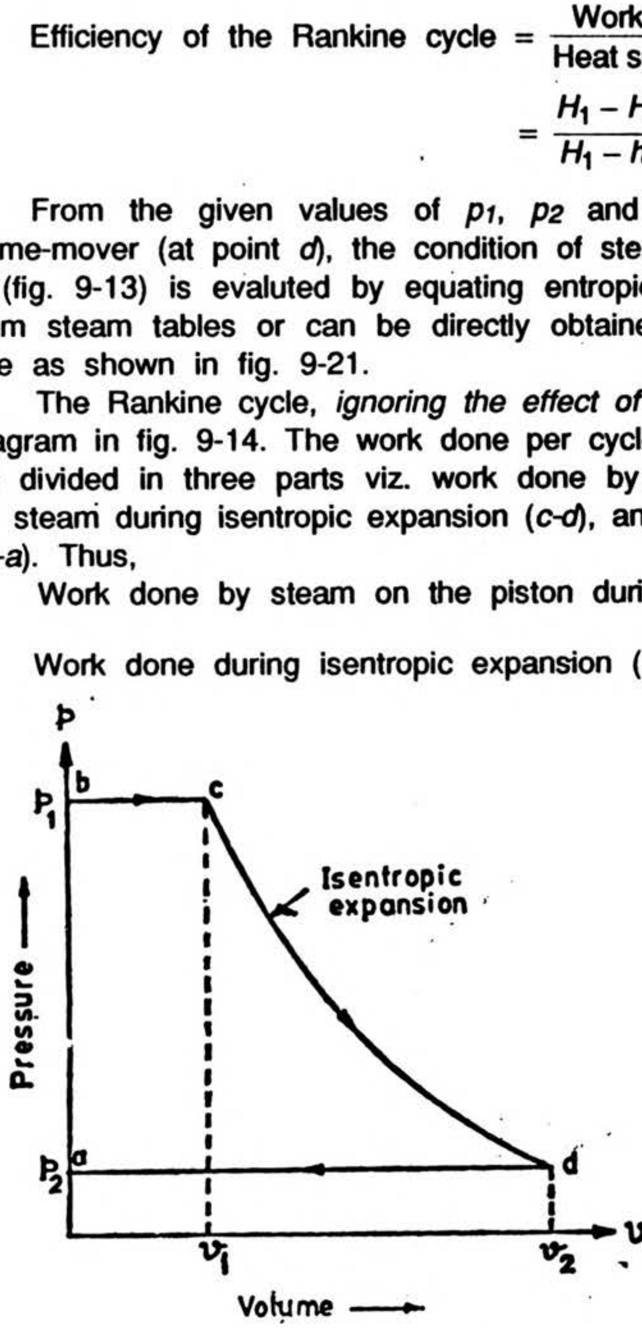


Fig. 9-14. $p - v$ diagram of Rankine cycle.

Work done on the steam during exhaust stroke $(d - a) = p_2v_2$

$\therefore$ Net work done during the cycle

$$= p_1v_1 + \frac{p_1v_1 - p_2v_2}{n - 1} - p_2v_2$$

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

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

kJ per kg of steam ..(9.23)

where p_1 and p_2 are in kPa and v_1 and v_2 are in m^3/kg.

The value of n can be evaluated from equation given by Dr. Zeuner or may be taken as 1·135 for steam initially wet and 1·3 for steam initially superheated.

Alternatively, the work done during isentropic expansion (*c-d*) can be found from the definition of non-flow adiabatic process viz.

Work done during expansion = change of internal energy during expansion = $u_1 - u_2$

$$= (H_1 - p_1v_1) - (H_2 - p_2v_2) = H_1 - H_2 - (p_1v_1 - p_2v_2) \text{ kJ per kg of steam}$$

$\therefore$ Net work done during the cycle $= p_1v_1 + [H_1 - H_2 - (p_1v_1 - p_2v_2)] - p_2v_2$

$$= H_1 - H_2 \text{ kJ/kg (same as before).}$$

Heat equivalent of one kW-hour = 3,600 kJ ($\because$ 1 kW = 1 kJ/sec.)

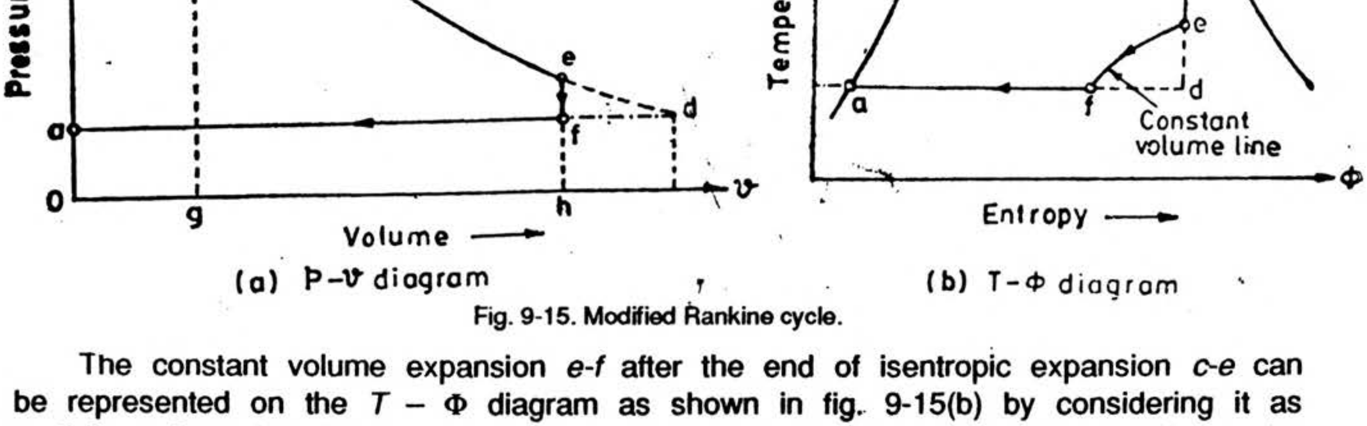
Work done per kg of steam in Rankine engine = $H_1 - H_2$ kJ

$$\text{Steam consumption} = \frac{\text{Heat equivalent of one kW–hour in kJ}}{\text{Work done per kg of steam in kJ}}$$

$$= \frac{3{,}600}{H_1 - H_2} \text{ kg per kW-hour} \qquad \text{.. (9.24)}$$

9.11.3 Modified Rankine Cycle : In practice, it is not economical to expand the steam to the extreme toe of the *p-v* diagram, which is represented by the point *d* (fig. 9-15). It is obvious that the diagram is very narrow at the toe. For this reason, the expansion is terminated (stopped) at point *e* and the cylinder is connected either to the atmosphere or to the condenser through exhaust port. Steam rushes out of the cylinder, causing drop in pressure upto the exhaust pressure. The expansion of the steam is thus completed by the constant volume line *e-f* as shown in fig. 9-15(a).

The loss of work due to incomplete expansion is represented by the area *e-f-d*. This cutting-off the toe occurs in reciprocating steam engine cylinder where the piston stroke is limited. In fact, the extra work (area *e-f-d*) obtained by complete expansion is not sufficient to overcome the friction of the moving parts of the engine.



(a) p–v diagram (b) T–Φ diagram

Fig. 9-15. Modified Rankine cycle.

The constant volume expansion *e-f* after the end of isentropic expansion *c-e* can be represented on the $T - \Phi$ diagram as shown in fig. 9-15(b) by considering it as partial condensation. There will, therefore, be a reduction in dryness fraction during the operation *e-f*.

As the value of specific volume of steam at point *e* is known, the value of dryness fraction for any other pressure between *e* and *f* may be calculated.

The work done during the modified Rankine cycle shown in fig. 9-15(a), may be calculated from the area *a-b-c-e-f* of the *p-v* diagram.

Let p_1, v_1, H_1 and u_1 apply to initial condition of steam at *c*.

p_2, v_2, H_2 and u_2 apply to condition of steam at *e*, and

p_3, h_3 apply to condition of water at *a*,

Then, work done during cycle per kg of steam

= area *a-b-c-e-f* of fig. 9-15(a)

= area *b-c-g-o* plus area *c-e-h-g* *minus* are *a-f-h-o*

= Work done during admission (*b-c*) *plus* work done during isentropic expansion (*c-e*) *minus* work done during exhaust (*f-a*)

$$= p_1 v_1 + \frac{p_1 v_1 - p_2 v_2}{n-1} - p_3 v_2$$

Alternatively, writing the work done during isentropic expansion *c-e* as $u_1 - u_2$, i.e. change of internal energy,

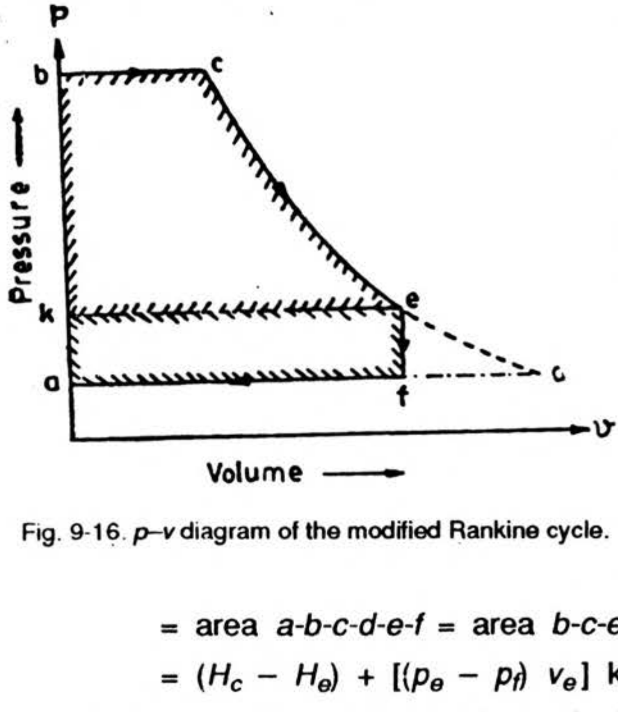


Fig. 9-16. *p–v* diagram of the modified Rankine cycle.

Work done during cycle per kg of steam

$= p_1 v_1 + (u_1 - u_2) - p_3 v_2$ kJ

where p_1 and p_3 are in kPa, and v_1 and v_2 are in m^3/kg

Heat supplied per kg of steam = $H_1 - h_3$ kJ

Modified Rankine cycle efficiency,

$$= \frac{\text{Work done during cycle in kJ per kg of steam}}{\text{Heat supplied in kJ per kg of steam}}$$

$$= \frac{p_1 v_1 + (u_1 - u_2) - p_3 v_2}{H_1 - h_3} \quad \text{.. (9.25)}$$

Alternatively, referring to fig. 9-16, Work done during cycle per kg of steam

= area *a-b-c-d-e-f* = area *b-c-e-k* plus area *k-e-f-a*

$$= (H_c - H_e) + [(p_e - p_f)\ v_e] \text{ kJ/kg} \quad \text{..(9.26a)}$$

where p_e = Pressure at release in kPa, p_f = Back pressure in kPa,

H_c = Enthalpy of steam at inlet (before isentropic expansion) in kJ/kg,

H_e = Enthalpy of steam after isentropic expansion in kJ/kg, and

v_e = Volume occupied by steam at the end of isentropic expansion i.e. at *e* in m^3/kg.

Heat supplied per kg of steam = $H_c - h_a$ kJ/kg

where h_a = enthalpy (sensible heat) of 1 kg of water of exhaust steam (condensate) in kJ/kg.

$$\text{Efficiency of modified Rankine cycle} = \frac{\text{Work done per kg of steam in kJ}}{\text{Heat supplied per kg of steam in kJ}}$$

$$= \frac{(H_c - H_e) + (p_e - p_f)\, v_e}{H_c - h_a} \qquad \text{.. (9.26b)}$$

9.11.4 Relative Efficiency : The relative efficiency of a steam engine is the ratio between the indicated thermal efficiency and Rankine cycle efficiency operating between the same pressure limits. It is also known as *efficiency ratio.*

$$\text{Rankine cycle efficiency} = \frac{H_1 - H_2}{H_1 - h_2} \quad \text{[from eqn. (9.22)]}$$

$$\text{Thus, Relative efficiency} = \frac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}} \qquad \text{.. (9.27)}$$

When brake thermal efficiency is used instead of indicated thermal efficiency in eqn. (9.27), the efficiency is termed *overall efficiency ratio.*

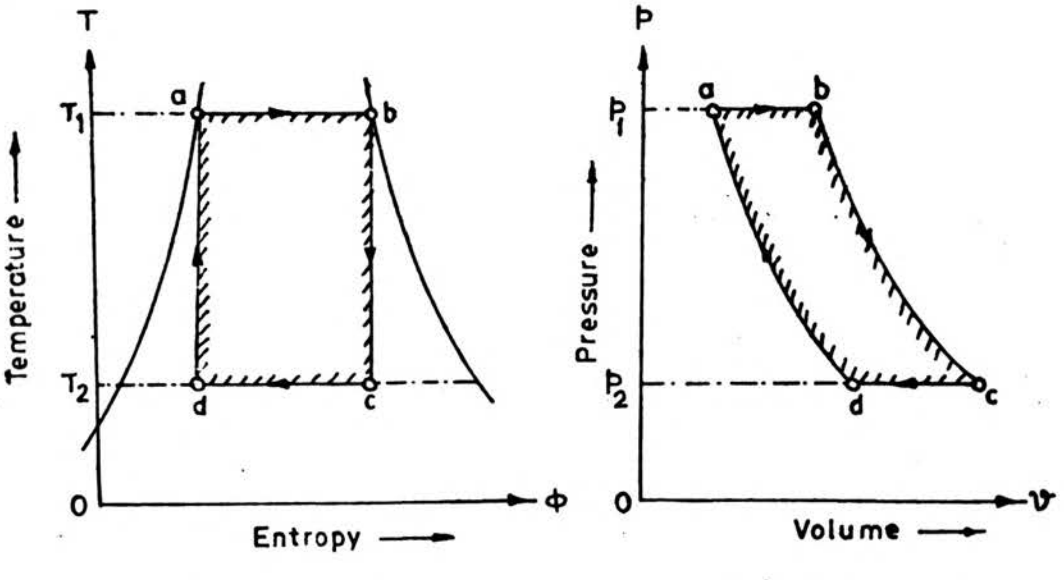
The efficiency ratio for a given steam engine indicates the percentage of the available heat converted into mechanical work. It also provides a measure of the excellence of design of the steam engine for the steam conditions under which it is operating and of its mechanical condition.

Problem-19 : *A Carnot engine receives dry saturated steam at 35 bar (3,500 kPa) and exhausts at 0·7 bar (70 kPa). Calculate : (a) the heat supplied per kg of steam, (b) the heat rejected per kg of steam, (c) the maximum work obtained per kg of steam, and (d) the Carnot cycle efficiency.*

From steam tables :

p bar	t_s °C	Φ_w kJ/kg K	Φ_s kJ/kg K
35	242·6	2·7253	6·1253
0·7	89·95	—	—

$T_1 = 242{\cdot}6 + 273 = 515{\cdot}6$ K; $T_2 = 89{\cdot}95 + 273 = 362{\cdot}95$ K



(a) T-Φ diagram (b) p-v diagram

Fig. 9·17.

Referring to fig. 9-17(a),

(a) Heat supplied = T_1 $(\Phi_b - \Phi_a)$ = 515·6 (6·1253 − 2·7253) = **1,753·04 kJ/kg**

(b) Heat rejected = T_2 $(\Phi_c - \Phi_d)$

= 362·95 (6·1253 − 2·7253) [$\because \Phi_c = \Phi_b$ and $\Phi_d = \Phi_a$)

= 362·95 × 3·4 = **1,234·03 kJ/kg.**

(c) Work obtained = Heat supplied − Heat rejected

= 1,753·04 − 1,234·03 = **519·01 kJ/kg**

(d) Carnot cycle efficiency or thermal efficiency

$$= \frac{\text{Work done per kg}}{\text{Heat supplied per kg}} = \frac{519{\cdot}01}{1{,}753{\cdot}04} = 0{\cdot}2961 \text{ or } 29{\cdot}61\%$$

Alternatively, using eqn. (9.21), Carnot cycle efficiency

$$= \frac{T_1 - T_2}{T_1} = \frac{515{\cdot}6 - 362{\cdot}95}{515{\cdot}6} = 0{\cdot}2961 \text{ or } \mathbf{29{\cdot}61\%\ (same\ as\ before)}$$

Problem-20 : *A steam power plant is supplied with dry saturated steam at a pressure of 14 bar and exhausts into a condenser at a pressure of 0·3 bar. Using the steam tables, calculate the efficiency of Carnot cycle and Rankine cycle for these conditions. Draw T − Φ diagrams for the cycles.*

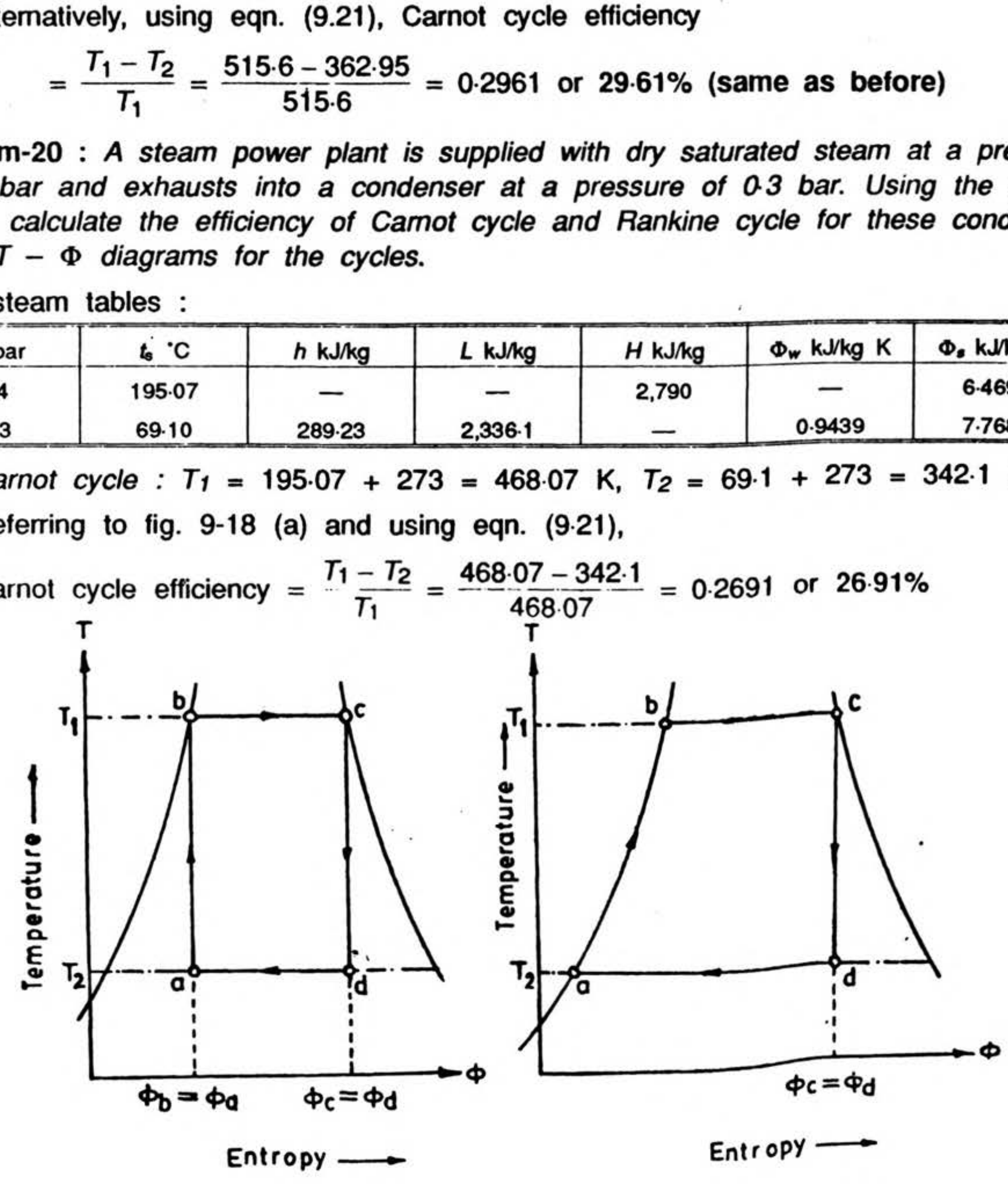
From steam tables :

p bar	t_s °C	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
14	195·07	—	—	2,790	—	6·4693
0·3	69·10	289·23	2,336·1	—	0·9439	7·7686

Carnot cycle : T_1 = 195·07 + 273 = 468·07 K, T_2 = 69·1 + 273 = 342·1 K

Referring to fig. 9-18 (a) and using eqn. (9·21),

$$\text{Carnot cycle efficiency} = \frac{T_1 - T_2}{T_1} = \frac{468{\cdot}07 - 342{\cdot}1}{468{\cdot}07} = 0{\cdot}2691 \text{ or } 26{\cdot}91\%$$



(a) Carnot cycle Fig. 9-18. (b) Rankine cycle

Rankine cycle : Referring to fig. 9-18 (b) and considering isentropic expansion *c-d*,

Entropy at c = Entropy at d i.e. $\Phi_{sc} = \Phi_{wd} + x_d(\Phi_{sd} - \Phi_{wd})$

i.e. $6{\cdot}4693 = 0{\cdot}9439 + x_d\,(7{\cdot}7686 - 0{\cdot}9439)$ $\therefore\ x_d = 0{\cdot}81$

$H_d = h_d + x_d L_d = 289{\cdot}23 + 0{\cdot}81 \times 2{,}336{\cdot}1 = 2{,}181{\cdot}47$ kJ/kg.

$H_c = 2{,}790$ kJ/kg (from steam tables).

Work done $= H_c - H_d = 2{,}790 - 2{,}181{\cdot}47 = 608{\cdot}53$ kJ/kg.

Heat supplied $= H_c - h_a = 2{,}790 - 289{\cdot}23 = 2{,}500{\cdot}77$ kJ/kg.

Using eqn. (9.22), Rankine cycle efficiency

$$= \frac{\text{work done in kJ per kg}}{\text{heat supplied in kJ per kg}} = \frac{608{\cdot}53}{2{,}500{\cdot}77} = 0{\cdot}2433 \text{ or } \mathbf{24{\cdot}33\%}$$

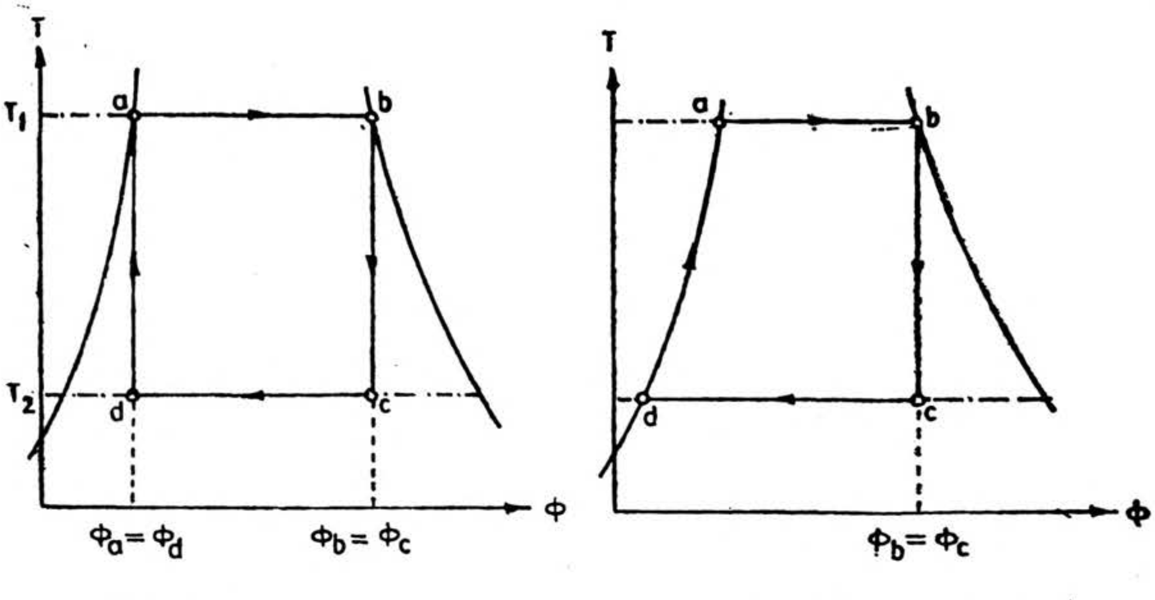
Alternatively, the value of x_d and $H_c - H_d$ can be directly obtained from $H - \Phi$ chart by drawing vertical line from the point on dry saturated line intersected by 14 bar line, to meet the line of 0·3 bar. The method of finding out x_d and $H_c - H_d$ from $H - \Phi$ chart is shown in fig. 9-21. Hence, the Rankine cycle efficiency.

It may be noted that the Carnot cycle is more efficient than Rankine cycle.

Problem-21 : *In a Carnot cycle the upper and lower limits of temperature correspond to steam pressures of 28 bar and 0·15 bar respectively. Dry saturated steam is supplied to the engine. Evaluate : (a) the dryness fraction of steam at the begining of isentropic compression, (b) the work done per kg of steam, (c) the heat supplied per kg of steam, and (d) the Carnot cycle efficiency. Compare items (b), (c) and (d) with the corresponding figures obtained for an engine working on the Rankine cycle between the same limits of pressure and temperature.*

From steam tables :

p bar	t °C	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
28	230·1	990·59	—	2,804	2·6109	6·2139
0·15	53·97	225·94	2,373·1	—	0·7549	8·0085



(a) Carnot cycle Fig. 9-19. T–Φ diagram. (b) Rankine cycle

Carnot cycle : (a) $T_1 = 230{\cdot}1 + 273 = 503{\cdot}1$ K; $T_2 = 53{\cdot}97 + 273 = 326{\cdot}97$ K

Referring to T–Φ diagram of fig. 9-19(a) and considering isentropic compression *d–a*

Entropy at *a* = Entropy at *d* i.e. $\Phi w_a = \Phi w_d + x_d\ (\Phi s_d - \Phi w_d)$

i.e. $2{\cdot}6109 = 0{\cdot}7549 + x_d\ (8{\cdot}0085 - 0{\cdot}7549)$ $\therefore\ x_d = \mathbf{0{\cdot}256}$

(b) Work done $= (T_1 - T_2)\ (\Phi_{sc} - \Phi_{wd})$

$= (T_1 - T_2)\ (\Phi_{sb} - \Phi_{wa})$

$= (503{\cdot}1 - 326{\cdot}97)\ (6{\cdot}2139 - 2{\cdot}6109) =$ **634·6 kJ/kg.**

(c) Heat supplied $= H_b - h_a = 2{,}804 - 990{\cdot}59 =$ **1,813·41 kJ/kg**

(d) Carnot cycle efficiency $= \dfrac{\text{work done}}{\text{heat supplied}} = \dfrac{634{\cdot}6}{1{,}813{\cdot}41} \times 100 =$ **35%**

Alternatively, using eqn. (9.21),

Carnot cycle efficiency $= \dfrac{T_1 - T_2}{T_1} = \dfrac{503{\cdot}1 - 326{\cdot}97}{503{\cdot}1} \times 100 =$ **35%** (same as before)

Rankine cycle :

(b) Referring to fig. 9-19(b), Entropy at *b* = Entropy at *c*

i.e. $\Phi_{sb} = \Phi_{wc} + x_c\,(\Phi_{sc} - \Phi_{wc})$

i.e. $6{\cdot}2139 = 0{\cdot}7549 + x_c\ (8{\cdot}0085 - 0{\cdot}7549)$

$\therefore\ x_c = 0{\cdot}753$ (dryness fraction at *c*)

Now, $H_c = h_c + x_c L_c = 225{\cdot}94 + 0{\cdot}753 \times 2{,}373{\cdot}1 = 2{,}012{\cdot}88$ kJ/kg

Work done $= H_b - H_c = 2{,}804 - 2{,}012{\cdot}88 =$ **791·12 kJ/kg of steam.**

(c) Heat supplied $= H_b - h_d = 2{,}804 - 225{\cdot}94 =$ **2,578·06 kJ/kg of steam.**

(d) Rankine cycle efficiency $= \dfrac{\text{work done in kJ/kg}}{\text{heat supplied in kJ/kg}} = \dfrac{791{\cdot}12}{2{,}578{\cdot}06} \times 100 =$ **30·7%**

Problem-22 : *Steam at 14 bar and 315°C is supplied to an engine working on the Rankine cycle. The steam is exhausted at 0·07 bar. Draw the $T - \Phi$ and $p - v$ diagrams for the cycle and calculate using the steam tables : (a) the condition of steam after isentropic expansion, (b) the Rankine cycle efficiency, (c) the mean effective pressure, (d) the ideal steam consumption per kW hour, and (e) the actual steam consumption per kW hour of the engine if the relative efficiency is 0·6. Take k_p for superheated steam as 2·1 kJ/kg K.*

From steam tables, at 14 bar, $H = 2{,}790$ kJ/kg, $\Phi_s = 6{\cdot}4693$ kJ/kg K, [illegible] 195·07°C, and at 0·07 bar, $h = 163{\cdot}4$ kJ/kg, $L = 2{,}409{\cdot}1$ kJ/kg, $v_s = 20{\cdot}53$ m^3/kg, $\Phi_w = 0{\cdot}5592$ kJ/kg K and $\Phi_s = 8{\cdot}2758$ kJ/kg K.

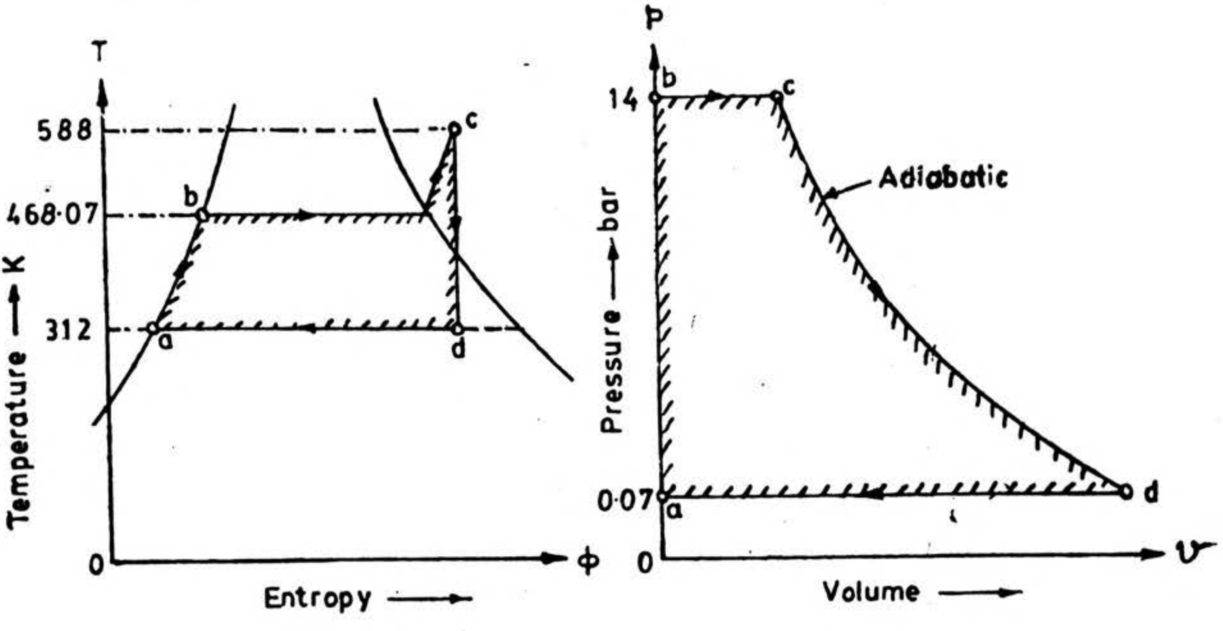
(a) Referring to T–Φ diagram of fig. 9-20 and considering isentropic expansion *c-d,*

Entropy at *c* = Entropy at *d*

i.e. $\Phi_{sc} + k_p \log_e\left(\dfrac{T_{supc}}{T_{satc}}\right) = \Phi_{wd} + x_d\,(\Phi_{sd} - \Phi_{wd})$

i.e. $6{\cdot}4693 + 2{\cdot}1\ \log_e\left(\dfrac{315 + 273}{195{\cdot}07 + 273}\right) = 0{\cdot}5592 + x_d\,(8{\cdot}2758 - 0{\cdot}5592)$ $\therefore\ x_d = \mathbf{0{\cdot}828}$

$$\therefore \quad x_d = 0{\cdot}828$$



(a) T-φ diagram (b) p-v diagram

Fig. 9-20. Rankine cycle diagrams.

(b) $H_c = 2{,}790 + 2{\cdot}1\ (315 - 195{\cdot}07) = 3{,}041{\cdot}85$ kJ/kg.

$H_d = 163{\cdot}4 + 0{\cdot}828 \times 2{,}409{\cdot}1 = 2{,}158$ kJ/kg

$h_a = 163{\cdot}4$ kJ/kg (from steam tables.).

Work done $= H_c - H_d = 3{,}041{\cdot}85 - 2{,}158 = 883{\cdot}85$ kJ/kg.

Heat supplied $= H_c - h_a = 3{,}041{\cdot}85 - 163{\cdot}4 = 2{,}878{\cdot}45$ kJ/kg

Now, Rankine cycle efficiency,

$$= \frac{\text{work done in kJ per kg}}{\text{heat supplied in kJ per kg}} = \frac{883{\cdot}85}{2{,}878{\cdot}45} \times 100 = 30{\cdot}7\%$$

(c) Specific volume, v_s at 0·07 bar = 20·53 m^3/kg (from steam tables),

Volume, $v_d = x_d \times v_s = 0{\cdot}828 \times 20{\cdot}53$ m^3/kg.

$$\text{Mean effective pressure} = \frac{\text{work done in kJ or kN.m per kg of steam}}{\text{stroke volume in m}^3\text{ per kg of steam } (v_d)}$$

$$= \frac{883{\cdot}85}{0{\cdot}828 \times 20{\cdot}53} = 52 \text{ kPa or } 52 \text{ kN/m}^2$$

(d) Work done in Rankine cycle $= H_c - H_d = 3{,}041{\cdot}85 - 2{,}158 = 883{\cdot}85$ kJ/kg.

Using eqn. (9.24) ideal steam consumption in kg per kW-hr.

$$= \frac{\text{Heat equivalent of one kW–hr.}}{\text{Work done per kg of steam}} = \frac{1 \times 3{,}600}{H_c - H_d} = \frac{3{,}600}{883{\cdot}85} = \mathbf{4{\cdot}078 \text{ kg per kW-hr.}}$$

(e) Using eqn. (9.27), Relative efficiency $= \dfrac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}}$

$\therefore$ Indicated thermal efficiency, $\eta_i = 0{\cdot}6 \times 0{\cdot}307 = 0{\cdot}1842$ or 18·42%

Indicated thermal efficiency, $\eta_i = \dfrac{\text{Heat equivalent of one kW–hr in kJ}}{M_s\,(H_c - h_a) \text{ in kJ}}$

(where M_s is the actual steam consumption in kg per kW-hr.)

$$\text{i.e. } 0{\cdot}1842 = \frac{1 \times 3{,}600}{M_s \times (3{,}041{\cdot}85 - 163{\cdot}4)}$$

$$\therefore\ M_s = \frac{1 \times 3{,}600}{0{\cdot}1842 \times (3{,}041{\cdot}85 - 163{\cdot}4)} = \textbf{6·79 kg per kW-hr.}$$

Alternatively, $M_s = \dfrac{4{\cdot}078}{0{\cdot}6}$ = **6·79** kg per kW-hr.(same as before)

Problem-23 : *A steam engine is supplied with dry saturated steam at 7 bar and exhausts at 1·4 bar. The steam consumption was found to be 2 kg per minute when developing indicated power of 4·5 kW. Using the steam tables or H – Φ chart, find the relative efficiency.*

From steam tables, at 7 bar, H = 2,763·5 kJ/kg, Φ_s = 6·7080 kJ/kg K; and at 1·4 bar, h = 458·39 kJ/kg, L = 2,232·1 kJ/kg, Φ_W = 1·4109 kJ/kg K and Φ_s = 7·2464 kJ/kg K.

Indicated thermal efficiency, $\eta_i = \dfrac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Net heat supplied in kJ per sec.}}$

$$= \frac{\text{kW or kJ sec.}}{m_s\,(H_1 - h_2)} = \frac{4{\cdot}5}{\frac{2}{60}\,(2{,}763{\cdot}5 - 458{\cdot}39)} = 0{\cdot}0587 \text{ or } 5{\cdot}87\%$$

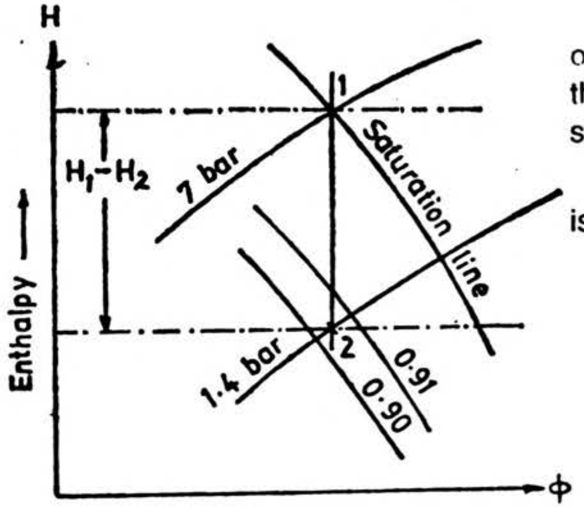


Fig. 9-21. Isentropic heat drop, $H_1 - H_2$ from $H - \Phi$ chart.

In order to determine the relative efficiency or efficiency ratio, it is first necessary to determine the Rankine cycle efficiency operating between same pressure limits, i.e. 7 bar and 1·4 bar.

Using the steam tables and considering isentropic expansion 1 – 2, as shown in fig. 9-21,

Entropy before expansion, Φ_1

= Entropy after expansion, Φ_2

$$\Phi_{s1} = \Phi_{w2} + x_2\,(\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e. } 6{\cdot}7080 = 1{\cdot}4109 + x_2\,(7{\cdot}2464 - 1{\cdot}4109)$$

$$\therefore\ x_2 = 0{\cdot}908$$

Thus, enthalpy after isentropic expansion,

$$H_2 = h_2 + x_2 L_2$$

$$= 458{\cdot}39 + 0{\cdot}908 \times 2{,}232{\cdot}1 = 2{,}485{\cdot}14 \text{ kJ/kg}$$

Enthalpy before isentropic expansion, H_1 = 2,763·5 kJ/kg.

$\therefore$ Isentropic enthalpy drop = Work done

$= H_1 - H_2$ = 2,763·5 – 2,485·14 = 278·36 kJ/kg of steam.

Alternatively, the value of $H_1 - H_2$ (isentropic enthalpy drop) can be directly obtained from $H - \Phi$ chart by drawing vertical line 1-2 from point on saturation line intersected by 7 bar line, to meet line of 1·4 bar as shown in fig. 9-21.

Heat supplied = $H_1 - h_2$ = 2,763·5 – 458·39 = 2,305·1 kJ/kg of steam.

Rankine cycle efficiency = $\dfrac{\text{Work done in kJ per kg of steam}}{\text{Heat supplied in kJ per kg of steam}}$

$$= \frac{H_1 - H_2}{H_1 - h_2} = \frac{278{\cdot}36}{2{,}305{\cdot}1} = 0{\cdot}12077 \text{ or } \mathbf{12{\cdot}077\%}$$

Using eqn. (9.27), Relative efficiency or efficiency ratio

$$= \frac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}} = \frac{0{\cdot}0587}{0{\cdot}12077} = 0{\cdot}486 \text{ or } \mathbf{48{\cdot}6\%}$$

Problem-24 : *In a modified Rankine cycle steam is supplied at 5·5 bar and 190°C. It expands isentropically to 2·8 bar when it is released at constant volume to exhaust pressure of 1 bar.*

Using the steam tables only, determine : (a) the work done per kg of steam supplied, (d) the modified Rankine cycle efficiency, (c) the ideal steam consumption per kW-hr and (d) the volume of steam to be supplied per kW-hr. Take kp of superheated steam as 2·1 kJ/kg K.

From steam tables, at 5·5 bar, Φ_s = 6·7893 kJ/kg K, H = 2,753·0 kJ/kg, v_s = 0·3427 m^3/kg, t_s = 155·48°C; at 2·8 bar, Φ_w = 1·6472 kJ/kg K, Φ_s = 7·0149 kJ/kg K, h = 551·48 kJ/kg, L = 2,170·7 kJ/kg, v_s = 0·6463 m^3/kg, and at 1 bar, h = 417·46 kJ/kg.

(a) Referring to fig. 9-22, and considering isentropic expansion c-e,

Entropy at c = Entropy at e

$$\text{i.e. } 6{\cdot}7893 + 2{\cdot}1 \log_e\left(\frac{190 + 273}{155{\cdot}48 + 273}\right)$$

$$= 1{\cdot}6472 + x_e\,(7{\cdot}0149 - 1{\cdot}6472)$$

$\therefore\ x_e = 0{\cdot}988$ (dryness fraction at e)

Now, $H_c = 2{,}753 + 2{\cdot}1\,(190 - 155{\cdot}48)$

$= 2{,}825{\cdot}5$ kJ/kg

$H_e = h_e + x_e L_e$

$= 551{\cdot}48 + 0{\cdot}988 \times 2{,}170{\cdot}7$

$= 2{,}696$ kJ/kg

$v_e = x_e \times v_s$

$= 0{\cdot}988 \times 0{\cdot}6463 = 0{\cdot}6385$ m^3/kg.

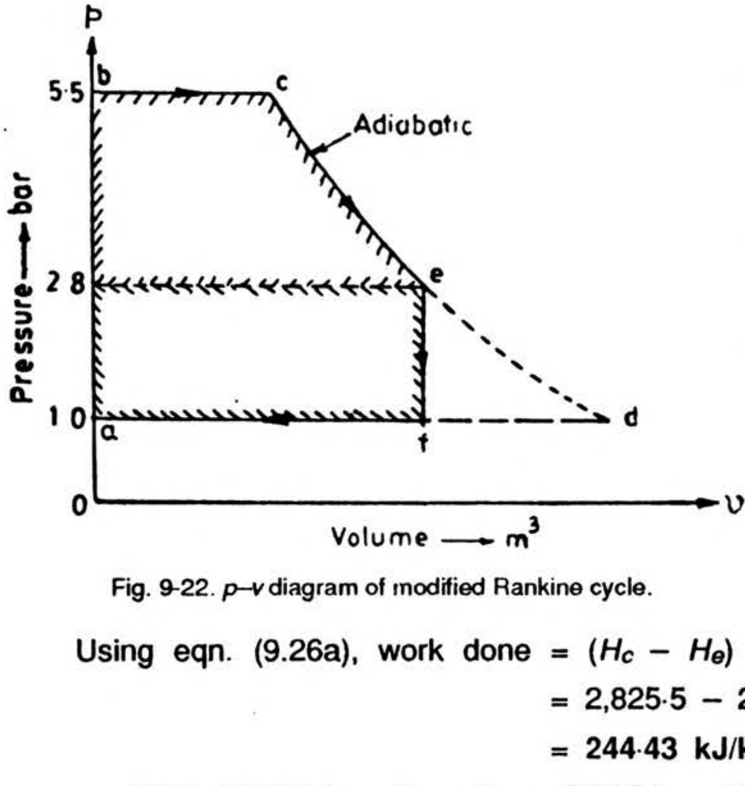


Fig. 9-22. p–v diagram of modified Rankine cycle.

Using eqn. (9.26a), work done = $(H_c - H_e) + [(p_e - p_f)\ v_e]$

= 2,825·5 – 2,696) + [(280 – 100) 0·6385]

= **244·43 kJ/kg of steam**

Heat supplied = $H_c - h_a$ = 2,825·5 – 417·46 = 2,408·04 kJ/kg of steam.

(b) Using eqn. (9.26b), Modified Rankine cycle efficiency

$$= \frac{\text{work done in kg per kg of steam}}{\text{heat supplied in kJ per kg of steam}} = \frac{244{\cdot}43}{2408{\cdot}04} = 0{\cdot}1015 \text{ or } \mathbf{10{\cdot}15\%}$$

(c) Using eqn. (9.24), Ideal steam consumption per kW-hour

$$= \frac{\text{heat equivalent of one kW–hour}}{\text{Work done per kg of steam}} = \frac{3{,}600}{244{\cdot}43} = \mathbf{14{\cdot}73 \text{ kg per kW-hr.}}$$

(d) Volume of 1 kg of superheated steam at 5·5 bar and 190°C

$$v_{sup} = v_s \times \frac{T_{sup}}{T_{sat}} = 0{\cdot}3427 \times \left(\frac{190+273}{155{\cdot}48+273}\right) = 0{\cdot}3703 \text{ m}^3\text{/kg}$$

$\therefore$ Volume of steam supplied = 14·73 × 0·3703 = **5·4545 $\mathbf{m^3}$ per kW-hr.**

Problem-25 : *A steam engine is supplied with dry saturated steam at a pressure of 14 bar (1·4 MPa). Pressure at release is 0·6 bar (60 kPa) and exhaust takes place at 0·15 bar (15 kPa). Assuming isentropic expansion with constant volume conditions between release and commencement of exhaust, determine using steam tables only, (a) the work done per kg of steam (b) the mean effective pressure, and (c) the efficiency of the cycle.*

Compare these values with those for Rankine cycle between the same pressure limits.

From steam tables :

p bar	v_s m^3/kg	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
14·0	—	—	—	2,790	—	6·4693
0·6	2·732	359·86	2,293·6	—	1·1453	7·532
0·15	10·022	225·94	2,373·1	—	0·7549	8·0085

(a) Referring to fig. 9-23, and considering isentropic expansion 1-2 from 14 bar to 0·6 bar and then release at constant volume 2-4 from 0·6 bar to 0·15 bar (*modified Rankine cycle*) :

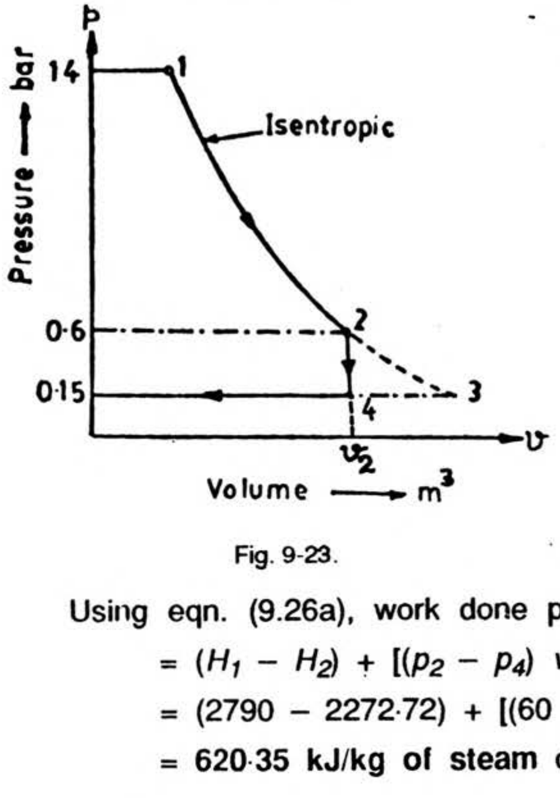


Fig. 9-23.

Considering isentropic expansion 1 – 2,

$$\Phi_{s1} = \Phi_{w2} + x_2(\Phi_{s2} - \Phi_{w2})$$

i.e. $6{\cdot}4693 = 1{\cdot}1453 + x_2(7{\cdot}532 - 1{\cdot}1453)$

$\therefore \ x_2 = 0{\cdot}834$

From steam tables, H_1 = 2,790 kJ/kg;

$H_2 = h_2 + x_2L_2$

$= 359{\cdot}86 + 0{\cdot}834 \times 2{,}293{\cdot}6 = 2{,}272{\cdot}72$ kJ/kg

Specific volume at 0·6 bar = 2·732 m^3/kg (from steam tables).

$\therefore$ Volume $v_2 = x_2v_{s2} = 0{\cdot}834 \times 2{\cdot}732$ m^3/kg $= v_4$

Using eqn. (9.26a), work done per kg of steam in modified Rankine cycle

$= (H_1 - H_2) + [(p_2 - p_4)\ v_2]$

$= (2790 - 2272{\cdot}72) + [(60 - 15) \times 0{\cdot}834 \times 2{\cdot}732\ ]$

= **620·35 kJ/kg of steam** or = 6,20,350 N.m/kg of steam

(b) Mean effective pressure $= \dfrac{\text{Work done per of steam in N.m}}{\text{Stroke volume in m}^3 \text{ per kg of steam } (v_4)}$

$$= \frac{6{,}20{,}350}{0{\cdot}834 \times 2{\cdot}732} \text{ N/m}^2 \text{ or (Pa)} = \frac{6{,}20{,}350}{10^5 \times 0{\cdot}834 \times 2{\cdot}732} = 2{\cdot}72 \text{ bar}$$

($\because$ 1 bar = 10^5 Pa)

(c) Modified Rankine cycle efficiency $= \dfrac{\text{Work done per kg of steam in kJ}}{\text{Heat supplied per kg of steam in kJ}}$

$$= \frac{\text{Work done}}{H_1 - h_3} = \frac{620{\cdot}35}{2{,}790 - 225{\cdot}95} = 0{\cdot}2419 \text{ or } 24{\cdot}19\%$$

Considering Rankine cycle :

(a) Referring to fig. 9-23, and considering complete isentropic expansion 1-3 from 14 bar to 0·15 bar,

$$\Phi_{s1} = \Phi_{w3} + x_3\,(\Phi_{s3} - \Phi_{w3})$$

i.e. $6{\cdot}4693 = 0{\cdot}7549 + x_3\,(8{\cdot}0085 - 0{\cdot}7549)$ $\therefore$ $x_3 = 0{\cdot}788$

Now, $H_1 = 2{,}790$ kJ/kg (steam tables), and

$$H_3 = h_3 + x_3\,L_3 = 225{\cdot}94 + 0{\cdot}788 \times 2{,}373{\cdot}1 = 2{,}095{\cdot}94 \text{ kJ/kg}$$

Work done $= H_1 - H_3 = 2{,}790 - 2{,}095{\cdot}94 =$ **694·06 kJ/kg.**

$= 6{,}94{,}060$ N.m/kg of steam.

(b) Mean effective pressure $= \dfrac{\text{Work done in N.m per kg of steam}}{\text{stroke volume in m}^3 \text{ per kg of steam } (v_3)}$

$$= \frac{6{,}94{,}060}{0{\cdot}788 \times 10{\cdot}022} \text{ N.m}^2 \text{ or} = \frac{6{,}94{,}060}{10^5 \times 0{\cdot}788 \times 10{\cdot}022} = \mathbf{0{\cdot}879 \text{ bar}} \;(\because 1 \text{ bar} = 10^5 \text{ N/m}^2)$$

(c) Rankine cycle efficiency $= \dfrac{\text{Work done per kg of steam in kJ}}{\text{Heat supplied per kg of steam in kJ}}$

$$= \frac{H_1 - H_3}{H_1 - h_3} = \frac{694{\cdot}06}{2{,}790 - 225{\cdot}94} \times 100 = 27{\cdot}07\%$$

Problem-26 : *An engine is supplied with dry saturated steam at 8·5 bar. Expansion is carried out at constant entropy to a pressure of 1 bar (100 kPa) and the exhaust pressure is 0·15 bar (15 kPa). Sketch the ideal p – v and T – Φ diagrams for the cycle and indicate there on the work done and the condition of steam at exhaust. Using the steam tables only, find the work done per kg of steam, the modified Rankine cycle efficiency, and the index of expansion.*

Taking the Rankine cycle as standard, find the percentage loss of efficiency due the incomplete expansion.

p bar	v_s m^3/kg	h kJ/kg	L kJ/kg	H kJ/kg	Φ_w kJ/kg K	Φ_s kJ/kg K
8·5	0·227	—	—	2,771·6	—	6·6421
1	1·694	417·46	2,258	—	1·3026	7·3594
0·15	—	225·94	2,373·1	—	0·7549	8·0085

Considering modified Rankine cycle :

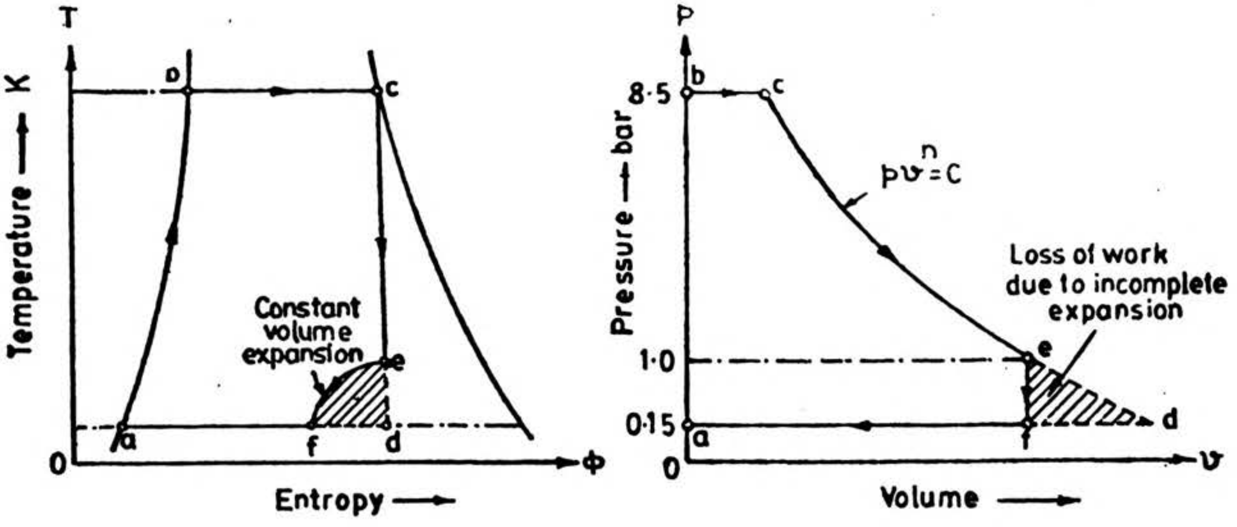
Referring to fig. 9-24(a) and considering isentropic expansion *c-e*,

Entropy at c = Entropy at e

i.e. $6{\cdot}6421 = 1{\cdot}3026 + x_e(7{\cdot}3594 - 1{\cdot}3026)$ $\therefore$ $x_e = 0{\cdot}882$ (dryness fraction at e)

$v_e = x_e \times v_{se} = 0{\cdot}822 \times 1{\cdot}694 = 1{\cdot}5$ m^3/kg; $v_c = 0{\cdot}227$ m^3/kg (from steam tables)

$\therefore$ $H_e = h_e + x_e L_e = 417{\cdot}46 + 0{\cdot}882 \times 2{,}258 = 2{,}409$ kJ/kg



(a) T-φ diagram (b) p-v diagram

Fig. 9·24. Modified Rankine cycle.

H_c = 2,771·6 kJ/kg and h_a = 225·94 kJ/kg (from 'steam tables)

Using eqn. (9.26a), Work done in modified Rankine cycle

$$= (H_c - H_e) + [(p_e - p_f)\ v_2]$$

$$= (2{,}771{\cdot}6 - 2{,}409) + [(100 - 15)\ 1{\cdot}5] = \mathbf{490{\cdot}1\ kJ/kg.}$$

Heat supplied = $H_c - h_a$ = 2,771·6 − 225·94 = 2,545·66 kJ/kg.

Modified Rankine cycle efficiency,

$$= \frac{\text{Work done in kJ/kg}}{\text{Heat supplied in kJ/kg}} = \frac{490{\cdot}1}{2{,}545{\cdot}66} = \mathbf{0{\cdot}1925\ or\ 19{\cdot}25\%}$$

The isentropic expansion *c-e* of fig. 9-24 (b) is represented by equation $pv^n = C$.

Now, $p_e v_e^n = p_c v_c^n$ i.e. $\dfrac{p_c}{p_e} = \left(\dfrac{v_e}{v_c}\right)^n$

$$\therefore\ n = \frac{\log\left(\dfrac{p_c}{p_e}\right)}{\log\left(\dfrac{v_e}{v_c}\right)} = \frac{\log\left(\dfrac{8{\cdot}5}{1}\right)}{\log\left(\dfrac{1{\cdot}5}{0{\cdot}227}\right)} = \mathbf{1{\cdot}134}$$

Considering Rankine cycle :

Referring to fig. 9-24(a) and considering isentropic expansion *c-d*.

Entropy at *c* = Entropy at *d*

i.e. 6·6421 = 0·7549 + x_d (8·0085 − 0·7549) ∴ x_d = 0·812 (dryness fraction at *d*)

Now, $H_d = h_d + x_d L_d$ = 225·94 + 0·812 × 2,373·1 = 2,152·9 kJ/kg

H_c = 2,771·6 kJ/kg and h_a = 225·94 kJ/kg (from steam tables).

Work done = $H_c - H_d$ = 2,771·6 − 2,152·9 = 618·7 kJ/kg

Heat supplied = $H_c - h_a$ = 2,771·6 − 225·94 = 2,545·66 kJ/kg

$$\text{Rankine cycle efficiency} = \frac{\text{Work done in kJ/kg}}{\text{Heat supplied in kJ/kg}} = \frac{618{\cdot}7}{2{,}545{\cdot}66} = \mathbf{0{\cdot}243\ or\ 24{\cdot}3\%}$$

Percentage loss of efficiency due to incomplete expansion (in modified Rankine cycle)

$$= \frac{24{\cdot}3 - 19{\cdot}25}{24{\cdot}3} \times 100 = \mathbf{20{\cdot}78\%}$$

9.12 Estimation of Missing Quantity

Mass of steam in the cylinder or the total steam present in the cylinder during expansion stroke consists of :

.. mass of steam admitted per stroke, called the *cylinder feed*, and

.. mass of steam contained in the clearance space before admission, called the *cushion steam.*

The mass of cylinder feed per stroke m_f can be determined experimentally by condensing exhaust steam in the condenser. From the measurement, cylinder feed per stroke,

$$\left.\begin{aligned} m_f &= \frac{\text{total mass of exhaust steam condensed per minute}}{\text{number of strokes made in exhausting the steam per min}} \\ \text{or } &= \frac{\text{cylinder feed per min}}{2 \times \text{r.p.m.}} \text{ (for double-acting steam engine)} \end{aligned}\right] \quad .. (9.28)$$

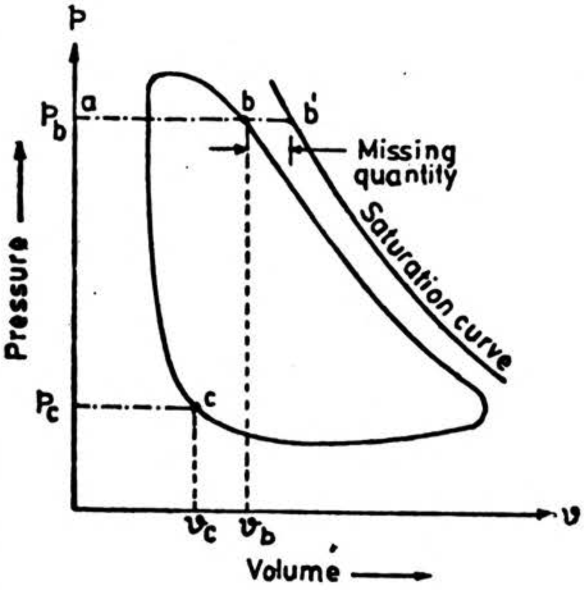


Fig. 9-25. Saturation curve on p–v diagram

Cushion steam is the steam trapped in the clearance space during compression (after the closing of the exhaust valve), and it may be regarded as steam never exhausted. Assuming that the cushion steam is dry saturated after closing of exhaust valve, its mass m_c may be estimated by dividing the total volume (actual indicated volume) v_c of steam at some convenient point on the compression curve of the indicator diagram, say point c (fig. 9-25), by the specific volume of dry saturated steam v_{sc} at the corresponding pressure p_c obtained from steam tables. This assumption is justified as the cushion steam is small in quantity compared to the cylinder feed per stroke and only error of the second order is involved,

i.e. mass of cushion steam, m_c

$$= \frac{\text{actual indicated volume at } c}{\text{volume of 1 kg of dry saturated steam}}$$

$$= \frac{v_c}{v_{sc}} \quad .. (9.29)$$

The mass of steam present in the cylinder during expansion,

m_s = cylinder feed per stroke, m_f + mass of cushion steam, m_c.

Now, consider any point b on the expansion curve (fig. 9-25) and read off from the diagram the pressure and volume of steam at point b.

Let p_b = pressure of steam in bar at b, and v_b = indicated volume of steam in m^3 at b.

From steam tables obtain the specific volume v_{sb} of dry saturated steam at pressure p_b. Then, the volume, the steam at *b* would occupy if dry saturated = $m_s \times v_{sb}$. This volume is represented by *ab'* to the volume scale of the indicator diagram. Thus, the point *b'* represents the volume which the steam at *b* would occupy if it was dry saturated.

Similarly, for every point on the expansion curve, a point corresponding to dry saturated condition can be obtained. A curve drawn through all these points is known as the *saturation curve.* The saturation curve shows at a glance the wetness (dryness fraction) of steam in the cylinder and volume of the missing quantity at any part of the expansion stroke.

$$\text{Dryness fraction of steam at } b = \frac{\text{actual indicated vol. at } b}{\text{dry steam volume}} = \frac{ab}{ab'} = \frac{v_b}{m_s \times v_{sb}} \quad ..(9.30)$$

The distance *bb'* (fig. 9-25) is the volume of missing quantity for the point *b*.

It may be noticed from fig. 9-25 that the steam is wet at the beginning of the expansion stroke and becomes drier towards the end of the stroke. This is due to the fact that the high pressure steam at the commencement of the expansion stroke is hotter than the cylinder walls; this causes steam to condense. The steam pressure falls during the expansion stroke, and towards the end of the stroke the cylinder walls will be hotter than the steam; this will cause re-evaporation of condensed steam and the dryness fraction of steam will consequently increase (Refer illustrative Problem no.28).

Referring to indicator diagram of fig. 9-25, the point *b'* on saturation curve represents the volume, the steam would occupy, if it was dry saturated while the actual volume occupied is represented by the point *b* on the expansion curve. The horizontal distance between the saturation curve and expansion curve at any part of the stroke is known as *missing quantity* and is represented by *bb'* m^3 at point *b*. *Alternatively, the missing quantity in kg per stroke may be regarded as the difference between the actual total mass of steam present in the cylinder during expansion stroke* ($m_f + m_c$) *and the indicated dry mass of steam at the point b.*

Referring to fig. 9-25,

Let p_b = pressure of steam in bar at *b*.

v_b = volume of steam (actual indicated volume) in m^3 at b, and

v_{sb} = specific volume in m^3/kg at pressure p_b (from steam tables).

Then, indicated dry mass of steam at *b*

$$= \frac{\text{actual indicated volume in m}^3 \text{ at } b}{\text{specific volume in m}^3 \text{ of 1 kg of dry saturated steam at pressure } p_b}$$

$$= \frac{v_b}{v_{sb}} \text{ kg} \quad .. (9.31)$$

This mass is called the "indicated mass" because it is calculated by means of data obtained from a curve drawn by an indicator.

The dryness fraction of steam at *b* may also be expressed as

$$\frac{\text{indicated dry mass of steam at } b}{\text{total mass of steam present in the cylinder during expansion } = m_f + m_c} \quad .. (9.32)$$

Missing quantity per stroke at b = actual total mass of steam present in the cylinder during the expansion stroke $(m_f + m_c)$ *minus* indicated dry mass of steam at b. .. (9.33)

The area between the saturation curve and expansion curve represents the work lost due to missing quantity.

The missing quantity is mainly due to condensation (initially and during expansion) but a small amount will be due to leakage of steam past the piston rings and valves. The missing quantity may be of the order of 3% of the cylinder feed. This can be reduced by taking anti-condensation measures (methods) as discussed earlier in art. 9.7.

Problem-27 : *The following results were obtained by measurements taken on an indicator card from a double-acting steam engine :*

	volume m^3	*pressure bar*
At point x on the expansion curve immediately after cut-off	*0.1038*	*13*
At point y on the compression curve	*0.0285*	*4*

The engine was running at 105 r.p.m. and the quantity of steam supplied per minute was 210 kg. Estimate the missing quantity in kg per minute and as a percentage of actual steam supplied, and the dryness fraction of steam at point x.

Volume of 1 kg of dry saturated steam at 4 bar is 0·4625 m^3 (from steam tables).

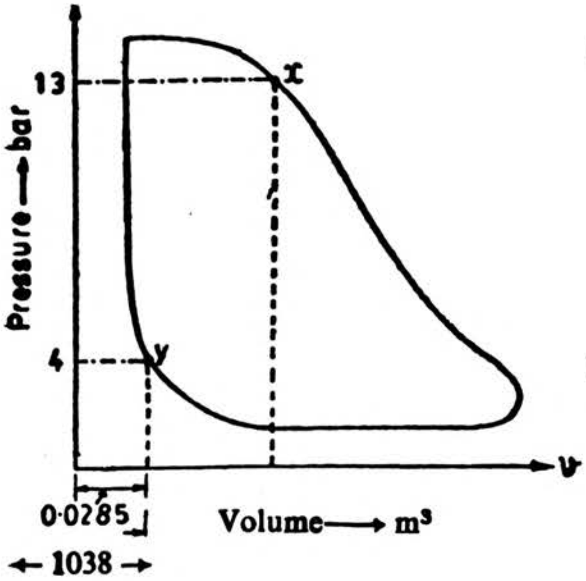


Fig. 9-26.

Assuming steam to be dry saturated at point y (fig. 9-26) and using eqn. (9.29),

Mass of cushion steam = m_c

$$= \frac{\text{actual indicated volume at } y}{\text{vol. of 1 kg of dry saturated steam}}$$

$$= \frac{0{\cdot}0285}{0{\cdot}4625} = 0{\cdot}0616 \text{ kg/stroke.}$$

Mass of steam admitted per stroke (cylinder feed per stroke),

$$m_f = \frac{\text{mass of steam supplied per min.}}{\text{no. of strokes (cycles) per min.}}$$

$$= \frac{210}{2 \times 105} = 1 \text{ kg.}$$

∴ The mass of steam present in the cylinder during expansion,

$m_s = m_f + m_c = 1 + 0{\cdot}0616 = 1{\cdot}0616$ kg/stroke.

Pressure at point x is 13 bar and one kg of dry saturated steam at this pressure has a volume of 0·1513 m^3 (for steam tables).

Using eqn. (9.31), indicated dry mass of steam at point x

$$= \frac{\text{actual indicated volume at } x}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}1038}{0{\cdot}1513} = 0{\cdot}686 \text{ kg/stroke.}$$

Using eqn. (9.33), missing quantity at point x

= mass of steam present in the cylinder during expansion *minus* indicated dry mass of steam point x.

= 1·0616 − 0·686 = 0·3756 kg/stroke.

∴ Missing quantity/min. at point x = 0·3756 (2 × 105) = 78·88 kg per min. (engine being double-acting).

Missing quantity at point x as a percentage of actual mass of steam supplied

$$= \frac{78{\cdot}88}{210} \times 100 = \mathbf{37{\cdot}56\%}$$

Using eqn. (9.32), dryness fraction of steam at point x

$$= \frac{\text{indicated dry mass of steam at point } x}{\text{mass of steam present in the cylinder during expansion}}$$

$$= \frac{0{\cdot}686}{1{\cdot}0616} = 0{\cdot}646 \text{ or } \mathbf{64{\cdot}6\% \ dry.}$$

Problem-28 : *The high-pressure cylinder of a double-acting steam engine running at 100 r.p.m. has an effective area of 250 cm² and a stroke of 60 cm and the clearance volume is 7 per cent of the stroke volume. During a trial the cylinder feed was 9 kg per minute. The indicator diagram showed that cut-off and release occurred at 30 and 90 per cent of the outward (expansion) stroke while the compression commenced at 70 per cent of the return (exhaust) stroke. The indicated pressures at cut-off, release and commencement of the compression are 14, 5·5 and 3 bar respectively. Find the quality of steam at cut-off and at release.*

Referring to fig. 9-27, the swept volume, $v_s = \frac{250}{10^4} \times \frac{60}{100} = 0{\cdot}015 \text{ m}^3$.

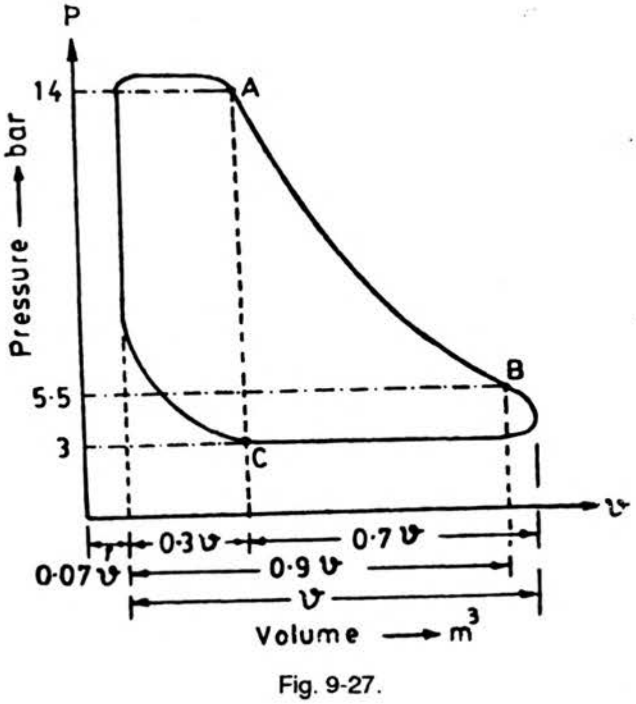


Fig. 9-27.

Clearance volume = $0{\cdot}07 v_s$

$$= \frac{7}{100} \times 0{\cdot}015 = 0{\cdot}00105 \text{ m}^3.$$

Total volume (indicated) at A (at cut-off)

$$= \left(\frac{30}{100} \times 0{\cdot}015\right) + 0{\cdot}00105 = 0{\cdot}00555 \text{ m}^3$$

Total volume (indicated) at B (at release)

$$= \left(\frac{90}{100} \times 0{\cdot}015\right) + 0{\cdot}00105 = 0{\cdot}01455 \text{ m}^3.$$

Total volume (indicated) at C (at commencement of compression)

$$= \left[\frac{100-70}{100} \times 0{\cdot}015\right] + 0{\cdot}00105$$

$$= 0{\cdot}00555 \text{ m}^3$$

Mass of steam admitted per stroke (cylinder feed per stroke),

$$m_f = \frac{9}{2 \times 100} = 0{\cdot}045 \text{ kg (engine being double-acting)}$$

The specific volume of steam at 3 bar = 0·6058 m^3/kg (from steam tables).

Using eqn. (9.29), the mass of cushion steam assumed to be dry saturated per stroke.

$$m_c = \frac{\text{actual indicated volume at point C}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}00555}{0{\cdot}6058} = 0{\cdot}00916 \text{ kg per stroke.}$$

∴ The mass of steam present in the cylinder during the expansion,

$$m_s = m_f + m_c = 0{\cdot}045 + 0{\cdot}00916 = 0{\cdot}05416 \text{ kg.}$$

At cut-off (Point A) :

Total mass of steam present in the cylinder during expansion stroke = 0·05416 kg. The specific volume of steam at 14 bar = 0·1408 m^3/kg (from steam tables). Hence, the steam if dry saturated at cut-off in the cylinder at 14 bar would have a volume of

$$0{\cdot}05416 \times 0{\cdot}1408 = 0{\cdot}00763 \text{ m}^3.$$

The total volume (actual indicated volume) of steam in the cylinder at point A (at cut-off) is found to be 0·00555 m^3.

Using eqn. (9.30), quality (dryness fraction) of steam at point A (at cut-off)

$$= \frac{\text{actual indicated volume at A}}{\text{dry steam volume}} = \frac{0{\cdot}00555}{0{\cdot}00763} = 0{\cdot}727 \text{ or } 72{\cdot}7\% \text{ dry.}$$

At release (point B) :

Total mass of steam in the cylinder during expansion stroke = 0·05416 kg. The specific volume of steam at 5·5 bar = 0·3427 m^3/kg (steam tables). Hence, the volume of steam if dry saturated at release in the cylinder at 5·5 bar = 0·05416 × 0·3427 = 0·01856 m^3.

The total volume (actual indicated volume) of steam at release (at point B) in the cylinder is found to be 0·01455 m^3.

Using eqn. (9.30), quality (dryness fraction) of steam at point B (at release)

$$= \frac{\text{actual indicated volume at B}}{\text{dry steam volume}} = \frac{0{\cdot}01455}{0{\cdot}01856} = 0{\cdot}784 \text{ or } 78{\cdot}4\% \text{ dry.}$$

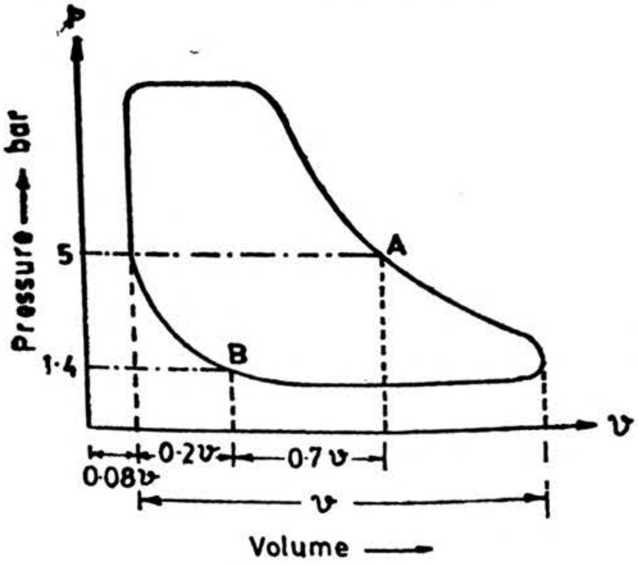


Fig. 9-28.

Problem-29 : *Determine the dryness fraction of steam and the missing quantity per hour at 0·7 of the expansion stroke, from the following data :*

Diameter of cylinder, 40 cm; stroke, 60 cm; r.p.m., 100; cut-off at 50% of the stroke; clearance volume, 8% of swept volume; steam condensed per min., 40 kg; pressure of steam at 0·7 of the expansion stroke, 5 bar; pressure of steam at 0·8 of the return stroke on the compression curve, 1·4 bar. State any assumptions made,

Referring to fig. 9-28, the swept volume,

$$v = \frac{\pi}{4}\left(\frac{40}{100}\right)^2 \times \frac{60}{100} = 0{\cdot}0754 \text{ m}^3$$

The clearance volume = $0{\cdot}08\,v = 0{\cdot}08 \times 0{\cdot}0754 = 0{\cdot}00603$ m^3.

Total volume (actual indicated volume) of steam at A, which is at 0·7 of the expansion stroke,

$$= 0{\cdot}7 \times \text{swept volume} + \text{clearance volume}$$

$$= (0{\cdot}7 \times 0{\cdot}0754) + 0{\cdot}00603 = 0{\cdot}05881 \text{ m3.}$$

Total volume (actual indicated volume) of steam at B, which is at 0.8 of the return stroke on the compression curve

$$= (1 - 0{\cdot}8) \times \text{swept vol.} + \text{clearance vol.}$$

$$= (0{\cdot}2 \times 0{\cdot}0754) + 0{\cdot}00603 = 0{\cdot}02111 \text{ m}^3.$$

The specific volume of dry saturated steam at 1·4 bar = 1·2366 m^3/kg (from steam tables).

Using eqn.(9.29), the mass of cushion steam assumed to be dry saturated per stroke,

$$m_c = \frac{\text{actural indicated volume at B}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}02111}{1{\cdot}2366} = 0{\cdot}01707 \text{ kg.}$$

Using eqn. (9.28), mass of steam admitted per stroke (cylinder feed per stroke),

$$m_f = \frac{\text{mass of steam condensed per minute}}{\text{no. of strokes (cycles) per minute}}$$

$$= \frac{40}{2 \times 100} = 0{\cdot}2 \text{ kg (assuming double-acting steam engine)}$$

The mass of steam present in the cylinder during expansion,

$$m_s = \text{mass of cushion steam + mass of steam admitted/per stroke}$$

$$= m_c + m_f = 0{\cdot}01707 + 0{\cdot}2 = 0{\cdot}21707 \text{ kg.}$$

Considering point A on the indicator diagram i.e. at 0·7 of the expansion stroke :

Pressure at A is 5 bar and 1 kg of steam dry saturated at this pressure has a volume of 0·3749 m^3 (from steam tables).

Hence, if steam dry and saturated, the steam at point A would have a volume of $0{\cdot}21707 \times 0{\cdot}3749 = 0{\cdot}08138$ m^3.

But, from the indicator diagram the actual indicated volume of steam at point A is 0·05881 m^3.

Using eqn. (9.30), Dryness fraction of steam at A

$$= \frac{\text{actual indicated volume at A}}{\text{dry steam volume}} = \frac{0{\cdot}05881}{0{\cdot}08138} = 0{\cdot}7226 \text{ or } \mathbf{72{\cdot}26\%\ dry}$$

Using eqn. (9.31), Indicated dry mass of steam at A

$$= \frac{\text{actual indicated volume at A}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}05881}{0{\cdot}3749} = 0{\cdot}15686 \text{ kg/stroke.}$$

Using eqn. (9.33), Missing quantity per stroke at A

= total mass of steam present in the cylinder during expansion *minus* indicated dry mass of steam at A = 0·21707 – 0·15686 = 0·06021 kg per stroke,

$\therefore$ Missing quantity at A per min = 0·06021(2 × 100) = 12·042 kg/min.

= **722·52 kg/hr.**

Alternatively, using eqn. (9.32),

$$\left\{\begin{array}{l}\text{Dryness fraction of steam at A}\\ \text{i.e. at 0·7 of the expansion stroke}\end{array}\right\} = \frac{\text{indicated dry mass of steam at A}}{\left[\begin{array}{c}\text{mass of steam present in the}\\ \text{cylinder during expansion}\end{array}\right]}$$

$$= \frac{0{\cdot}15686}{0{\cdot}21707} = 0{\cdot}7726 \text{ or } 77{\cdot}26\% \text{ (same as before)}$$

Problem-30 : *A double-acting steam engine, having a diameter of 70 cm and stroke 120 cm runs at 75 r.p.m.. Clearance volume is 7% of the swept volume. The cylinder feed is 1·1 kg per stroke. From the indicator card, the following readings were obtained*

(i) At a point on compression curve – pressure 3·5 bar and volume 0·1133 m^3.

(ii) Cut-off at ⅓ stroke – pressure and volume just after cut-off are 11 bar and 0·2 m^3 *respectively.*

(iii) At half stroke – pressure 8 bar (on expansion curve).

(iv) At release – pressure 4·5 bar, volume 0·485 m^3.

Estimate the dryness fraction and missing quantity in kg per hour at (a) cut-off, (b) half–stroke, and (c) release.

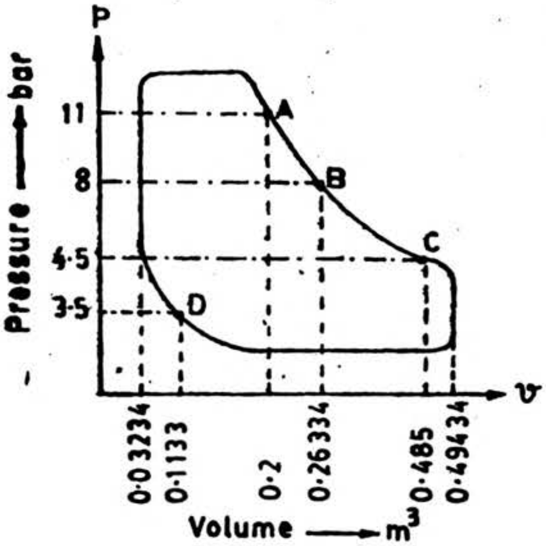


Fig. 9-29.

Referring to fig. 9-29, actual indicated volume at $D = 0{\cdot}1133 \text{ m}^3$ (given).

1 kg of dry saturated steam at 3·5 bar has a volume of 0·5243 m^3 (from steam tables).

Using eqn. (9.29), mass of cushion steam assumed to be dry saturated per stroke,

$$m_c = \frac{\text{Actual indicated volume at D}}{\text{volume of 1 kg of dry saturated steam}}$$

$$= \frac{0{\cdot}1133}{0{\cdot}5243} = 0{\cdot}216 \text{ kg/stroke.}$$

The cylinder feed per stroke,

$m_f = 1{\cdot}1$ kg (given)

The mass of steam present in the cylinder during expansion,

$m_s = m_f + m_c = 1{\cdot}1 + 0{\cdot}216 = 1{\cdot}316$ kg/stroke.

(a) *At cut-off (Point A) :*

Volume of 1 kg of steam dry saturated at 11 bar = 0·1775 m^3 (from steam tables). Hence, if steam is dry saturated, the steam at this pressure would have a volume of $1{\cdot}316 \times 0{\cdot}1775 = 0{\cdot}2336 \text{ m}^3$.

Actual indicated volume of steam in the cylinder at A = 0·2 m^3 (given).

Using eqn. (9.30), Dryness fraction of steam at A (at cut-off)

$$= \frac{\text{actual indicated volume at A}}{\text{dry steam volume}} = \frac{0{\cdot}2}{0{\cdot}2336} = 0{\cdot}856 \text{ or } 85{\cdot}6\% \text{ dry.}$$

Using eqn. (9.31), Indicated dry mass of steam at A

$$= \frac{\text{actual indicated volume at A}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}2}{0{\cdot}1775} = 1{\cdot}1268 \text{ kg/stroke.}$$

Using eqn. (9.33), Missing quantity at A (at cut-off)

$$= 1{\cdot}316 - 1{\cdot}1268 = 0{\cdot}1892 \text{ kg/stroke or } 0{\cdot}1892\ (2 \times 75) \times 60 = \mathbf{1{,}703\ kg/hr.}$$

(b) *At half-stroke (Point B) :*

$$\text{Swept volume, } V = \frac{\pi}{4}\left(\frac{70}{100}\right) \times \frac{120}{100} = 0{\cdot}462 \text{ m}^3.$$

Clearance volume $= 0{\cdot}07\ V = 0{\cdot}07 \times 0{\cdot}462 = 0{\cdot}03234$ m^3.

$\therefore$ Total indicated volume of steam at half stroke $= (0{\cdot}462 \times 0{\cdot}5) + 0{\cdot}03234 = 0{\cdot}26334$ m^3

1 kg of dry saturated steam at 8 bar has a volume Of $0{\cdot}2404$ m^3 (from steam tables). Hence, if steam is dry saturated, the steam at this pressure would have a volume of $1{\cdot}316 \times 0{\cdot}2404 = 0{\cdot}3164$ m^3.

Using eqn. (9.30), Dryness fraction of steam at half-stroke (at point B)

$$= \frac{\text{actual indicated volume at B}}{\text{dry steam volume}} = \frac{0{\cdot}26334}{0{\cdot}3164} = 0{\cdot}832 \text{ or } \mathbf{83{\cdot}2\%\ dry.}$$

Using eqn. (9.31), Indicated dry mass of steam at B

$$= \frac{\text{actual indicated volume at B}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}26334}{0{\cdot}2404} = 1{\cdot}095 \text{ kg/stroke.}$$

$\therefore$ Missing quantity at B (half – stroke) $= 1{\cdot}316 - 1{\cdot}095 = 0{\cdot}221$ kg/stroke

$$= 0{\cdot}221 \times (2 \times 75) \times 60 = \mathbf{1{,}989\ kg/hr.}$$

(c) *At release (Point C) :*

One kg of dry saturatred steam at 4·5 bar has a volume of $0{\cdot}414$ m^3 (from steam tables). Hence, if steam is dry saturated, the steam at this pressure would have a volume of $1{\cdot}316 \times 0{\cdot}414 = 0{\cdot}5448$ m^3.

Indicated volume of steam in the cylinder at C $= 0{\cdot}485$ m^3 (given)

Dryness fraction of steam at C (at release)

$$= \frac{\text{actual indicated volume at C}}{\text{dry steam volume}} = \frac{0{\cdot}485}{0{\cdot}5448} = 0{\cdot}89 \text{ i.e. } 89\% \text{ dry}$$

Indicated dry mass of steam at C (at release)

$$= \frac{\text{actual indicated volume at C}}{\text{volume of 1 kg of dry saturated steam}} = \frac{0{\cdot}485}{0{\cdot}414} = 1{\cdot}1715 \text{ kg/stroke.}$$

Missing quantity at C (at release) $= 1{\cdot}316 - 1{\cdot}1715 = 0{\cdot}1445$ kg/stroke.

$$= 0{\cdot}1445 \times (2 \times 75) \times 60 = \mathbf{1{,}300{\cdot}5\ kg/hr.}$$

9.13 Methods of Governing Steam Engines

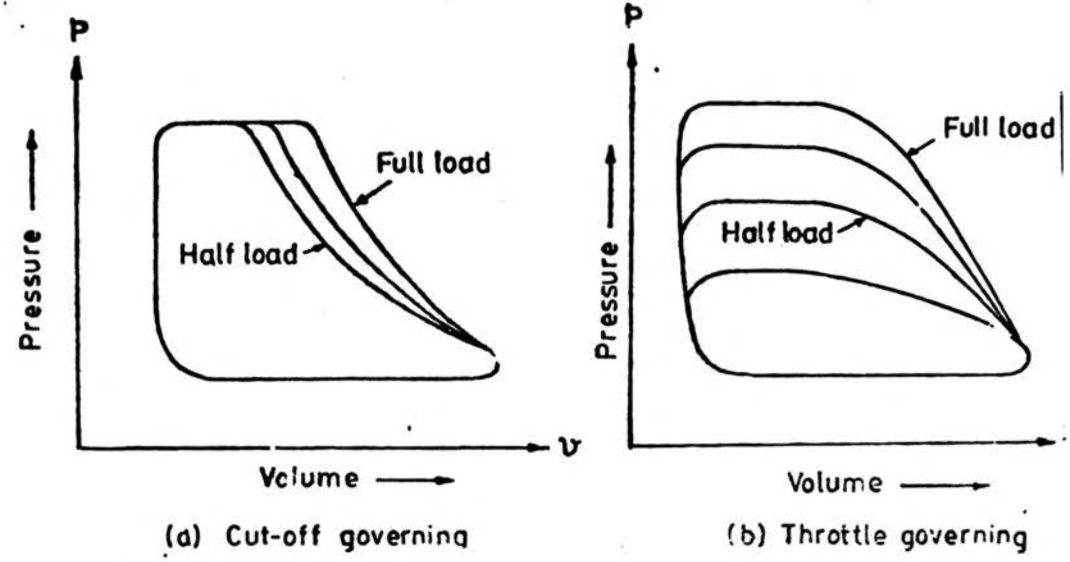
The function of a governor is to maintain the speed of the engine constant by either controlling the quantity of steam or the pressure of the steam supplied to the engine according to the load on the engine.

There are two methods of maintaining the speed of steam engines constant by the action of the governor :

.. By varying the point of cut-off (altering the period of admission of steam to the engine cylinder). This method of governing is known as *cut-off governing.*

.. By throttling the entering steam. This method is known as *throttle governing.*

In *cut-off governing,* the volume of steam supplied to the engine is altered by changing the point of cut-off by a special slide valve working under the control of the governor. In the *throttle governing,* the supply of steam is throttled to a lower pressure by partial closure of throttle valve under the control of a governor. The indicator diagram showing the effect of cut-off governing and throttle governing at various loads are shown in fig. 9-30.



(a) Cut-off governing (b) Throttle governing

Fig. 9-30. Effects of cut-off and throttle governing on P–V diagram at various loads.

9.13.1 Willan's Law : During a test on steam engine, the indicated power is varied by altering the load, and the rate of steam consumption is obtained from the mass of steam condensed by the condenser.

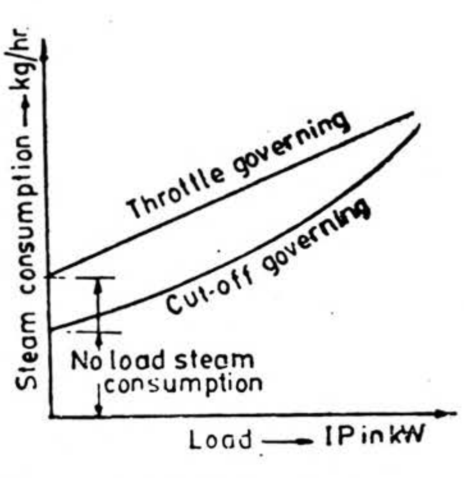


Fig. 9-31. Variatioon of steam consumption with indicated power (I.P.).

Figure 9-31 shows graphs of steam consumption in kg/hr. plotted against load in terms of indicated power in kW for both throttle governing and cut-off governing. *It will be noticed that the graph is a straight line* in case of throttle governing and a smooth curve in case of cut-off governing. The *straight line graph in case of throttle governing indicates that steam consumption per hour is directly proportional to the indicated power. This is known as Willan's law, and holds good only for a throttle-governed engine running at constant speed.* In general the straight line can be obtained for the steam consumption graph only if the ratio of expansion or cut-off is constant; this condition is not fulfilled in a cut-off governed engine.

The intercepts of the steam consumption lines on the vertical axis at no load, represent the steam consumption required for idle (no load) running of the engine, i.e. for overcoming frictional power of the engine.

For an engine governed by throttle governing, *Willan's line* is a straight line and may therefore be expressed in the form of

$$S = a + b\,I \qquad ..\ (9.34)$$

where S = steam consumption in kg/hr.,

a = steam consumption at no load in kg/hr. (constant),

b = slope of Willan's line (constant), and

I = load on the engine in terms of indicated power in kW.

Problem-31 : *A throttled governed engine running at constant speed uses 310 kg of steam per hour when developing indicated power of 15 kW and 574 kg of steam per hour when developing indicated power of 37 kW. Estimate the steam consumption in kg/hour when developing indicated power of 22 kW.*

For a throttled governed engine, running at constant speed, steam consumption in kg/hr. (S) can be expressed in terms of indicated power (I) in kW as

$S = a + b\,I$ where a and b are constants.

Substituting the values of steam consumption for the two loads in the above equation we have,

$$310 = a + b \times 15 \qquad ..\ (a)$$

$$574 = a + b \times 37 \qquad ..\ (b)$$

Subtracting (a) from (b), we have,

$$264 = 22b \quad \therefore\ b = 12$$

Substituting value of b in equation (a), we have,

$$310 = a + 12 \times 15 \quad \therefore\ a = 150$$

Now, steam consumption when developing indicated power of 22 kW,

$$S = a + b \times 22$$

$$= 150 + 12 \times 22 = 414 \text{ kg/hr.}$$

Problem-32 : *A small steam engine uses 115 kg of steam per hour when developing indicated power of 4 kW. The speed is kept constant by throttle governor and the steam consumption is 195 kg per hour when developing indicated power of 9·34 kW. Calculate the indicated thermal efficiency of the engine at the same speed when developing indicated power of 7·35 kW. The steam supplied is at a pressure of 4 bar and 0·95 dry. The temperature of condensate is 44°C.*

Compare the indicated thermal efficiency with the Rankine cycle efficiency if the condenser pressure is 0·2 bar.

For a throttle governed steam engine running at constant speed, steam consumption in kg/hr (S) can be expressed in terms of indicated power (I) in kW as

$S = a + b\,I$ where a and b are constants

Substituting the values of the steam consumption for two loads in the above equation, we have,

$$115 = a + b \times 4{\cdot}0 \qquad ..\ (a)$$

$$195 = a + b \times 9{\cdot}34 \qquad ..\ (b)$$

Subtracting (a) from (b), we have, $80 = 5·34\ b$ $\quad\therefore\ b = 14·98$

Substituting value of b in eqn. (b), we have, $195 = a + 14·98 \times 9·34\ \therefore\ a = 55$

Using eqn. (9.34), steam consumption when developing 7·35. kW indicated power,

$S = a + b \times 7·35 = 55 + 14·98 \times 7·35 = 165$ kg per hour.

At 4 bar, $h = 604·74$ kJ/kg. $L = 2{,}133·8$ kJ/kg (from steam tables).

Now, $H_1 = h_1 + x_1L_1 = 604·75 + 0·95 \times 2{,}133·8 = 2{,}631·85$ kJ/kg.

Heat remaining in condensate, $h_2 = 4·187\ (t_c - 0) = 4·187\ (44 - 0) = 184$ k

Heat supplied/hr. $= S\ (H_1 - h_2) = 165\ (2{,}631·85 - 184)$ kJ/hr.

$$\text{Indicated thermal efficiency} = \frac{\text{Heat equivalent of indicated power in kJ per hr.}}{\text{Heat supplied in kJ per hr.}}$$

$$= \frac{\text{kW} \times 3{,}600}{S\,(H_1 - h_2)} = \frac{7·35 \times 3{,}600}{165\,(2{,}631·85 - 184)} = 0·0655 \text{ or } 6·55\%$$

Rankine cycle : Rankine heat drop (work done) from 4 bar and 0·95 dry to 0·2 bar $= H_1 - H_2 = 445$ kJ/kg (from $H - \Phi$ chart).

At 0·2 bar, $h_2 = 251·4$ kJ/kg (from steam tables)

(where h_2 = Enthalpy of 1 kg of water of exhaust steam).

Heat supplied per kg of steam $= H_1 - h_2 = 2{,}631·85 - 251·4 = 2{,}380·45$ kJ/kg.

$$\text{Rankine cycle efficiency} = \frac{\text{heat drop (work done) per kg of steam}}{\text{heat supplied per kg of steam}}$$

$$= \frac{H_1 - H_2}{H_1 - h_2} = \frac{445}{2{,}380·45} = 0·1869 \text{ or } 18·69\%$$

$$\text{Efficiency ratio} = \frac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}} = \frac{6·55}{18·69} \times 100 = 35·0\%$$

Problem-33 : *A throttle governed steam turbine, running at constant speed develops indicated power of 926·5 kW while consuming 7,430 kg of steam per hour. When the steam consumption is 10,000 kg per hour it develops indicated power of 1,323 kW. If the steam supply is at 24 bar with 100°C of superheat, find (without using Mollier chart) the efficiency ratio for the turbine while developing indicated power of 1,103 kW. The back pressure is 0·1 bar. Take* K_p *of superheated steam as 2·35 kJ/kg K.*

For a throttle governed steam turbine running at constant speed, steam consumption in kg/hour (S) can be expressed in terms of indicated power (I) in kW as

$S = a + b\,I$ where a and b are constants

Substituting the values of the steam consumption for two loads in the above equation

$7{,}430 = a + b \times 926·5$ .. (a)

$10{,}000 = a + b \times 1{,}323$ .. (b)

Subtracting (a) from (b), we have, $2{,}570 = 396·5\ b$ $\quad\therefore\ b = 6·48$

Substituting value of b in eqn. (a), we get, $7{,}430 = a + 6·48 \times 926·5\ \therefore\ a = 1{,}426·3$

Using eqn. (9.34), steam consumption when developing indicated power of 1,103 kW.

$S = a + b \times 1{,}103 = 1{,}426·3 + 6·48 \times 1{,}103 = 8{,}574$ kg/hr.

At 24 bar, H_s = 2,802·6 kJ/kg (from steam tables).

Now, $H_1 = H_s - K_p\ (t_{sup} - t_{sat})$ = 2,802·6 + 2·35 (100) = 3,037·6 kJ/kg.

At 0·1 bar, h_2 = 191·83 kJ/kg (from steam tables).

Heat supplied/hour = $S\ (H_1 - h_2)$ = 8,574 (3,0337·6 − 191·83) kJ per hour

Now, indicated thermal efficiency $\eta_1 = \dfrac{\text{heat equivalent of indicated power in kJ per hr}}{\text{heat supplied in kJ/hr}}$

$$= \frac{\text{kW} \times 3{,}600}{S\,(H_1 - H_2)} = \frac{1{,}103 \times 3{,}600}{8{,}574\,(3{,}037{\cdot}6 - 191{\cdot}83)} = 0{\cdot}1627 \text{ or } \mathbf{16{\cdot}27\%}$$

Rankine cycle : From steam tables, at 24 bar, Φ_s = 6·2729 kJ/kg K, t_s = 221·83°C, and at 0·1 bar Φ_w = 0·6493 kJ/kg K, Φ_s = 8·1502 kJ/kg K, h = 191·83 kJ/kg, L = 2,584·7 kJ/kg

Now, entropy before expansion, Φ_1 = entropy after expansion, Φ_2

$$\text{i.e. } \Phi_{s1} + K_p \log_e \frac{T_{sup}}{T_{sat}} = \Phi_{w2} + x_2\,(\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e. } 6{\cdot}2729 + 2{\cdot}35 \log_e \left(\frac{100 + 221{\cdot}83 + 273}{221{\cdot}83 + 273}\right) = 0{\cdot}6493 + x_2\ (8{\cdot}1502 - 0{\cdot}6493)$$

$$\therefore\ x_2 = 0{\cdot}8073$$

$H_2 = h_2 + x_2L_2$ = 191·83 + 0·8073 × 2,584·7 = 2,278·5 kJ/kg.

Rankine cycle efficiency, $= \dfrac{\text{work done per kg of steam}}{\text{heat supplied per kg of steam}}$

$$= \frac{H_1 - H_2}{H_1 - h_2} = \frac{3{,}037{\cdot}6 - 2{,}278{\cdot}5}{3{,}037{\cdot}6 - 191{\cdot}83} = \frac{759{\cdot}1}{2{,}845{\cdot}77} = 0{\cdot}267 \text{ or } 26{\cdot}7\%$$

$$\therefore \text{ Efficiency ratio} = \frac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}} = \frac{16{\cdot}27}{26{\cdot}7} \times 100 = \mathbf{61\%}$$

Problem-34 : *In a steam engine plant the boiler supplies the engine dry saturated steam at 14 bar and the condenser pressure is 0·3 bar. Calculate the Rankine cycle efficiency of the engine. Using the steam tables only and selecting your own values for the relative efficiency, mechanical efficiency of the engine, and boiler thermal efficiency, estimate the probable overall efficiency of the steam engine plant from coal to brake.*

From steam tables at 14 bar, H_s = 2,790 kJ/kg, Φ_s = 6·4693 kJ/kg K, and at 0·3 bar, h = 289·23 kJ/kg, L = 2,336·1 kJ/kg, Φ_w = 0·9439 kJ/kg K, Φ_s = 7·7686 kJ/kg K.

Entropy before expansion, Φ_1 = Entropy after expansion, Φ_2

$$\text{i.e. } \Phi_{s1} = \Phi_{w2} + x_2\,(\Phi_{s2} - \Phi_{w2})$$

$$\text{i.e. } 6{\cdot}4693 = 0{\cdot}9439 + x_2\ (7{\cdot}7686 - 0{\cdot}9439) \quad \therefore\ x_2 = 0{\cdot}81$$

Now, H_1 = 2,790 kJ/kg (from steam tables),

h_2 = 289·23 kJ/kg (from steam tables).

∴ $H_2 = h_2 + x_2L_2$ = 289·23 + 0·81 × 2,336·1 = 2,181·7 kJ/kg

Work done = $H_1 - H_2$ = 2,790 — 2,181·7 = 608·3 kJ/kg.

Heat supplied $= H_1 - h_2 = 2{,}790 - 289{\cdot}23 = 2{,}500{\cdot}77$ kJ/kg.

$$\text{Rankine cycle efficiency} = \frac{\text{work done per kg of steam}}{\text{heat supplied per kg of steam}}$$

$$= \frac{H_1 - H_2}{H_1 - h_2} = \frac{608{\cdot}3}{2{,}500{\cdot}77} = 0{\cdot}2432 \text{ or } 24{\cdot}32\%$$

Taking relative efficiency as 60%, we have, $0{\cdot}6 = \dfrac{\text{Indicated thermal efficiency}}{\text{Rankine cycle efficiency}}$

∴ Indicated thermal efficiency, $\eta_i = 0{\cdot}6 \times 0{\cdot}2432 = 0{\cdot}146$ or $14{\cdot}6\%$

Now, brake thermal efficiency, η_b = indicated thermal efficiency × mechanical efficiency

Taking mechanical efficiency, η_m as 80%,

Brake thermal efficiency, $\eta_b = 0{\cdot}146 \times 0{\cdot}8 = 0{\cdot}1168$ or $11{\cdot}68\%$

Using eqn. (9.17), Brake thermal efficiency, $\eta_b = \dfrac{\text{Brake power in kW}}{\text{mass of steam/sec.} \times (H_1 - h_2)}$

Assuming that heat of condensate h_2 is not returned to hot-well, i.e. neglecting heat of condensate, h_2, we have,

$$\text{Brake thermal efficiency, } \eta_b = \frac{\text{brake power in kW}}{\text{mass of steam sec.} \times \text{enthalpy of 1 kg of steam}}$$

$$\text{Boiler thermal efficiency} = \frac{\text{mass of steam per sec.} \times \text{enthalpy of 1 kg of steam}}{\text{mass of coal/sec.} \times \text{C.V. of coal}}$$

∴ Brake thermal efficiency of the engine × boiler thermal efficiency

$$= \left[\frac{\text{brake power in kW}}{\text{mass of steam/sec.} \times \text{enthalpy of 1 kg of steam}}\right] \times \left[\frac{\text{mass of steam/sec.} \times \text{enthalpy of 1 kg of steam}}{\text{mass of coal/sec.} \times \text{C.V. of coal}}\right]$$

$$= \frac{\text{brake power in kW}}{\text{mass of coal/sec.} \times \text{C.V. of coal}}$$

Now, over-all efficiency of the steam plant from coal to brake

$$= \frac{\text{brake power in kW}}{\text{mass of coal/sec.} \times \text{C.V. of coal}}$$

= brake thermal efficiency of the engine × boiler thermal efficiency

Taking boiler thermal efficiency as 75%,

Overall efficiency of the plant from coal to brake $= 0{\cdot}1168 \times 0{\cdot}75$

$= 0{\cdot}0876$ or $8{\cdot}76\%$.

Tutorial – 9

1. (a) What is a heat engine?
   (b) Why is steam engine known as a prime mover?
   (c) Distinguish between external combustion and internal combustion engines.
2. Describe briefly the function of the following parts of steam engine :
   (a) Eccentric, (b) Piston, (c) Stuffing box and gland, (d) Crosshead, (e) Crank, (f) Flywheel, (g) Connecting rod, (h) Gudgeon pin, and (i) D-slide valve.

3. Explain the following terms as applied to steam engines :

   (i) Condensing and non-condensing, (ii) Single-acting and double-acting, (iii) Simple and compound, (iv) Piston displacement, (v) Dead centre, and (vi) Crank throw.

4. (a) Show that hypothetical m.e.p. of steam engine may be expressed as

   $$\text{Hypothetical m.e.p.} = \left[\frac{p_1}{r}(1 + \log_e r)\right] - p_b$$

   State clearly the meaning of each symbol.

   (b) Steam is admitted to the cylinder of steam engine at a pressure of 735 kPa and cut-off takes place at 0·4 of the stroke. The back pressure is 29·5 kPa. Calculate the hypothetical mean effective pressure on the piston during the stroke.

   [533 89 kPa]

5. (a) Explain the following terms as applied to steam engines :

   (i) Mean effective pressure, (ii) Back pressure, (iii) Brake power, (iv) Indicated power, and (v) Diagram factor.

   (b) Sketch and explain the working of any one type of steam engine indicator you know.

6. (a) Explain why steam engines are provided with steam jacket.

   (b) Write short notes on the following and explain them with the help of sketches wherever necessary :

   (i) Indicator, (ii) Mechanical efficiency, (iii) Indicated thermal efficiency, (iv) Brake thermal efficiency, and (v) Overall efficiency of a steam engine plant.

7. What do you understand by the term "diagram factor"? Give its average value.

   A single-cylinder, double-acting steam engine of 20 cm diameter and 40 cm stroke is supplied with steam at 8·5 bar and exhausts at 0·15 bar. Cut-off takes place at ⅓ rd of the stroke and the engine runs at 120 r.p.m. Using the diagram factor of 0·7, estimate the actual m.e.p. and indicated power of the engine.

   [4·06 bar; 20·41 kW]

8. A double-acting, simple steam engine, having a slide valve, has a bore and stroke, 30 cm and 38 cm respectively. It runs at an average speed 200 r.p.m. and is supplied with steam at 7 bar. The vacuum gauge of its condenser indicates 650 mm of Hg, while barometer indicates 750 mm of Hg. If the valve is set to give a constant cut-off at 40% of the stroke and a diagram factor of 0.8, find indicated power of the engine.

   [74·959 kW]

9. Obtain an expression for the hypothetical mean effective pressure in a steam engine cylinder, neglecting the clearance volume, stating clearly the assumptions you make.

   A single-cylinder, double-acting, steam engine is to develop indicated power of 185 kW at 120 r.p.m., the engine stroke being twice the diameter of the cylinder. Steam supply is at 10 bar and the exhaust pressure is 0·2 bar. Cut-off takes place at ⅖th of the stroke. Using a diagram factor of 0·85, calculate the required diameter of the cylinder and piston stroke.

   [35·89 cm; 71·75 cm]

10. (a) Explain what is meant by the mean effective pressure of a steam engine and show how its value is obtained from an indicator card.

    (b) A single-cylinder, double-acting steam engine has a bore and stroke, 25 cm and 35 cm respectively. Cut-off is at ⅓rd of stroke. The initial steam pressure is 10 bar, and the exhaust pressure is 1·1 bar and the engine is found to develop indicated power of 30 KW at 120 r.p.m. Calculate the diagram factor of the engine.

    [0·741]

11. (a) Derive an expression for the hypothetical mean effective pressure in a steam engine cylinder, in terms of steam supply pressure p_1, back pressured p_b and expansion ratio r.

    (b) A single-cylinder, double-acting steam engine having cylinder diameter 20 cm and stroke 30 cm, is working with 40% cut-off and running at 200 r.p.m. The engine is supplied with dry saturated steam at 8·5 bar. The back pressure is 0·25 bar. Calculate the brake power of the engine, assuming the mechanical efficiency of 80% and a diagram factor of 0·75.

    [23·62 kW]

12. What is meant by the term clearance?

    A steam engine has a cylinder diameter 30 cm and stroke 40 cm and has a clearance volume of 7·5 per cent of stroke volume. If the cut-off takes place at ¼th of the stroke, what is the true or real ratio of expansion ?

    [3·308]

13. Find the diameter of the cylinder of a single-cylinder, double-acting steam engine developing indicated power of 75 kW at a piston speed of 180 metres/minute and operating under the following conditions : Initial pressure, 10 bar, back pressure, 1·5 bar, cut-off at ⅙ th of stroke; clearance volume, 10% of the piston displacement. A diagram factor of 0·85 may be assumed and effect of compression be neglected.

[29·17 cm]

14. A single-cylinder, double-acting steam engine has a cylinder diameter of 75 cm and a stroke of 120 cm. Dry saturated steam is admitted to the engine at a pressure of 6·5 bar and cut-off takes place at 50% of the stroke. The back pressure is 1·1 bar. The crank is rotating at 300 r.p.m. Neglecting clearance and diameter of the piston rod and assuming a diagram factor of 0·8, calculate the m.e.p., indicated power and theoretical steam consumption in kg per kW per hour based on indicated power.

[3·52 bar; 1,866 kW; 17·47 kg]

15. A single-cylinder, double-acting steam engine has the cylinder diameter 30 cm and stroke 50 cm. Steam is admitted at a pressure of 7 bar and cut-off is at 0.35 stroke. The back pressure is 1·1 bar. The brake power and mechanical efficiency of the engine at a speed of 150 r.p.m. are 45 kW and 81% respectively. Find the diagram factor of the engine.

[0·801]

16. A double-acting, simple steam engine receives steam at a pressure of 7 bar. The cut-off takes placed at ½ stroke. Find the indicated power developed by the engine at 210 r.p.m. The diameter of the piston and the piston rod are 20 cm and 6 cm respectively and the stroke is 50 cm. Assume a diagram factor of 0·75 for both ends. Back pressure may be taken as 1·2 bar.

[37·215 kW]

17. The following observations were made during a test of a single-cylinder, double-acting steam engine :

Area of indicator diagram 14·5cm^2, length of indicator diagram 10 cm, indicator spring scale 200 kPa/cm, cylinder diameter 17 cm, stroke 30 cm, r.p.m. 250. Calculate the m.e.p. and indicator power of the engine.

[290 kPa; 16·456 kW]

18. A single-cylinder, double-acting steam engine of 30 cm cylinder diameter and 38 cm piston stroke, is supplied with steam at a pressure of 875 kPa. Cut-off takes place at 40 per cent of the stroke and the engine runs at 3 r.p.s. (revolutions per second). The back pressure is 14·7 kPa. Using the diagram factor of 0·8, estimate the indicated m.e.p. and indicated power of the engine.

[524·81 kPa; 84·58 kW]

19. During a test on a double-acting, single-cylinder steam engine running at 140 r.p.m., the indicated power was 81 kW, cylinder diameter 30 cm, stroke 45 cm, cut-off at /3rd stroke, initial steam pressure 1,100 kPa, back pressure 40 kPa. Find the diagram factor.

[0·748]

20. In a trial on a single-cylinder, double-acting vertical steam engine, the following observations were made :

Cylinder diameter, 17 cm; Piston rod diameter, 5 cm; Stroke, 30 cm; r.p.m. 240; Length of both indicator diagrams, 10 cm; Area of top end indicator diagram, 14·5 cm^2; Area of bottom end indicator diagram, 13 cm^2; Indicator spring scale, 250 kPa/cm; Circumference of brake wheel, 4·25 metres; Circumferences of brake rope, 3 cm; Dead load on the brake, 900 newtons; Reading of spring balance, 75 newtons.

Calculate the indicated power, brake power and mechanical efficiency of the engine.

[17·96 kW; 14·124 kW; 78·64%]

21. Explain the term "Indicated thermal efficiency" as applied to a steam engine.

A single-cylinder, double-acting steam engine of 28 cm bore and 45 cm stroke, works between a supply pressure of 10 bar and back pressure of 0·15 bar. Assuming diagram factor of 0·7 and neglecting the clearance volume, estimate the indicated power developed at 180 r.p.m., if the cut-off occurs at ⅓rd stroke.

If the above engine consumes 900 kg of dry saturated steam per hour, determine the indicated thermal efficiency of the engine.

[79·66 kW; 12·48%]

22. A double-acting, single-cylinder steam engine with cylinder 35 cm diameter and 53 cm stroke, is to develop indicated power of 147·8 kW at 200 r.p.m. with a cut-off at o·4 stroke. Determine the admission pressure of steam if the exhaust pressure is 25 kPa and the diagram factor is 0·85.

If the above engine consumes 1,980 kg of steam per hour and the steam at admission is dry saturated, determine the indicated thermal efficiency of the engine.

[700 kPa; 10·79%]

23. Explain the term "brake thermal efficiency" as applied to a steam engine.

The following results refer to a test on a double-acting, simple steam engine with a cylinder 22 cm in diameter and stroke 30 cm : Engine speed, 2 r.p.s; Mean effective pressure 245 kPa; Brake wheel diameter, 1.4 metres; Net load on the brake wheel, 981 newtons; Steam supplied at a pressure of 7 bar, dry saturated; Exhaust pressure, 1 bar; Steam used per hour, 180 kg. Determine : (i) the indicated power, (ii) the brake power (power output), (iii) the mechanical efficiency, (iv) the brake thermal efficiency, and (v) the indicated thermal efficiency of the engine. [(i) 11·17 kW; (ii) 8·63 kW; (iii) 77·26%; (iv) 7·36%; (v) 9·52%]

24. Define the terms "diagram factor" and "indicated thermal efficiency" as applied to the reciprocating steam engines and explain the practical use of these terms.

The admission pressure of steam to a single-cylinder, double-acting steam engine is 10 bar. Cut-off takes place at 0·4 of the stroke and back pressure is 1·1 bar. The cylinder diameter is 35 cm and the stroke is 53 cm. Taking a diagram factor of 0·75, calculate the m.e.p. Assuming mechanical efficiency of 80 per cent, estimate the brake power of the engine at 200 r.p.m. Neglect clearance.

If the indicated thermal efficiency of the above engine is 14%, calculate the steam supplied in kg per kW per hour based on indicated power. The engine is supplied with dry saturated steam.

[4·924 bar; 133·912 kW; 10·946 kg]

25. The following results refer to a trial on a double-acting, simple steam engine with a cylinder 22 cm in diameter and stroke 30 cm, brake wheel 1·4 metres in diameter : dead load on the brake, 1,177 newtons; spring balance reading, 196 newtons; engine speed, 140 r.p.m.; mean effective pressure, 2·5 bar; steam supplied is dry saturated at a pressure of 5 bar; exhaust pressure, 1·1 bar; steam used per hour, 180 kg. Determine the mechanical efficiency, brake thermal efficiency and indicated thermal efficiency of the engine,

[75·67%; 8·68%; 11·479%]

26. The following data were obtained during a test on a single cylinder, double-acting, steam engine having 21 cm cylinder diameter, and 26 cm piston stroke :

Effective radius of the brake wheel, 38 cm; engine speed, 5 r.p.s.; net brake load, 1,334 newtons; m.e.p., 235 kPa; pressure of steam supplied, 8 bar; dryness fraction of steam supplied, 0·9; steam consumption, 180 kg per hour; condensate temperature, 60°C. Calculate : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the brake thermal efficiency, and (e) the indicated thermal efficiency.

[(a) 15·92 kW; (b) 21·14 kW; (c) 75·3%; (d) 13·79%; (e) 18·31%]

27. The following data were obtained during a test on a steam engine plant :

Indicated m.e.p., 2·5 bar; bore, 25 cm; stroke, 30 cm; r.p.m., 104; brake-torque, 932 N-m; pressure of steam supplied, 7 bar; steam supplied dry saturated; condenser vacuum, 610 mm of Hg; barometer reading, 760 mm of Hg; steam consumption, 2·7 kg per minute; 22.5 kg of coal with calorific value of 33,500 kJ/kg is supplied per hour. The steam engine is single-cylinder and double-acting.

Determine the mechanical efficiency and brake thermal efficiency of the steam engine. Also determine the overall efficiency of the steam engine plant (from coal to brake). [79·53%; 8·98%; 4·85%]

28. The following data were obtained during a test of single-cylinder, double-acting steam engine having 21 cm cylinder diameter and 26 cm stroke :

Effective radius of brake wheel, 38 cm; Speed, 300 r.p.m.; Net brake load, 1,335 newtons; Mean effective pressure 2·4 bar; Pressure of steam supplied, 8 bar; Dryness fraction of steam supplied, 0·97; Steam consumption, 3·6 kg/minute; Condensate temperature, 60°C.

Calculate : (a) Brake power, (b) Indicated power, (c) Mechanical efficiency, (d) Brake thermal efficiency, (e) Indicated thermal efficiency, and (f) Steam consumption in kg per kW per hour based on brake power.

[(a) 15·937 kW; (b) 21·613 kW; (c) 73·74%; (d) 10·81%; (e) 14·66%; (f) 13·55 kg]

29. A double-acting, simple steam engine runs at 180 r.p.m. The cylinder diameter is 30 cm, the piston rod diameter is 5 cm, and the stroke length is 45 cm. Indicator diagrams taken on the engine show that the mean effective pressure is 4·5 bar for the in-stroke cycle and 4·7 bar for the out-stroke cycle. The engine is supplied with 1,000 kg of dry saturated steam per hour at 10 bar. The back pressure is 0·15 bar.

Calculate : (a) the indicated power, and (b) the indicated thermal efficiency of the engine.

[(a) 86·571 kW; (b) 12·21%]

30. Compare the Carnot and Rankine cycles using steam. Sketch $p - v$ and $T - \Phi$ diagrams for each cycle. Explain why Rankine cycle is employed rather than Carnot cycle as the standard of comparison for actual steam engine.

Find the efficiency of Carnot cycle, using water and steam, between temperature limits corresponding to pressure of 14 bar and 1·4 bar. Compare the results with the efficiency of an engine following the Rankine cycle and working between the same pressure limits and using steam 90 per cent dry.

What influence has increasing wetness upon the efficiency of the Rankine cycle?

[Carnot η = 18·32%; Rankine η = 16·91%]

31. In what respect does the Rankine cycle differ from the Carnot cycle?

A prime-mover operating on the Rankine cycle is supplied with steam at pressure of 9 bar and dryness fraction 0·9, and exhausts at 0·7 bar. Find the work done in kJ per kg of steam and the efficiency of the Rankine cycle. Use the steam tables only. [385·14 kJ/kg; 17·56%]

32. Describe briefly the Rankine cycle with complete expansion and sketch the p–v and T–Φ diagrams for the cycle.

Calculate, *using the steam tables only,* (a) the ideal steam consumption in kg/hr. and (b) the thermal efficiency of the engine working on Rankine cycle and developing indicated power of 1,471 kW. The steam supplied is dry saturated at 24 bar and exhaust is at 0·15 bar.

[(a) 6,870 kg/hr; (b) 29·91%]

33. A multiple expansion steam engine develops indicated power of 1,180 kW and uses 10,000 kg of steam per hour. The steam is supplied at 16 bar and 295°C and exhaust is at 647·5 mm of Hg vacuum (barometer 760 mm of Hg). Estimate : (a) the indicated thermal efficiency, (b) the efficiency of the Rankine cycle, and (c) the efficiency ratio or relative efficiency. Take k_p of superheated steam as 2·3 kJ/kg K.

[(a) 15·26%; (b) 28·56%; (c) 53·43%]

34. Explain briefly, with the help of $P - v$ and $T - \Phi$ diagrams, the essential differences between the Carnot and Rankine steam engine cycles.

An engine working on Rankine cycle receives dry saturated steam at 14 bar directly from the boiler and exhausts at 0·07 bar. Find the work done per kg of steam and the thermal (Rankine) efficiency. If the steam is throttled to 8·5 bar before entering the engine, find the reduction in work done per kg of steam and in thermal efficiency by means of steam tables and H–Φ chart. Take Kp of superheated steam as 2·1 kJ/kg K.

[781·2 kJ/kg; 29·74%; 65·8 kJ/kg, 2·48%]

35. A simple Rankine cycle steam power plant operates between the temperature limits of 260°C and 95°C. Steam is supplied to the turbine in a condition of dry saturated and the expansion in the turbine is isentropic. Draw the ideal Rankine cycle and the Carnot cycle using steam and capable of working between the same temperature limits on a $T - \Phi$ diagram. Estimate and compare the efficiencies of the two cycles while working between the given conditions.

Give reasons for difference in the values of the efficiencies of the two cycles.

[Carnot η = 30·9%; Rankine η = 27%]

36. A steam engine working on the incomplete Rankine cycle is supplied with dry saturated steam at 10 bar. Pressure at release is 2·5 bar and exhaust takes place at 1·2 bar. Assuming isentropic condition with constant volume conditions between release and commencement of exhaust, determine using steam tables, (a) the work done per kg of steam, (b) the mean effective pressure, and (c) the efficiency of the cycle.

Compare these values with those of Rankine cycle working between the same pressure limits.

[Incomplete Rankine : (a) 333·41 kJ/kg; (b) 5·074 bar, (c) 14·26%;
Rankine : (a) 363·88 kJ/kg; (b) 2·895 bar, (c) 15·56%]

37. An engine receives steam at 18 bar with 50°C superheat. Expansion takes place at constant entropy to pressure of 2.8 bar. Exhaust is at 0·15 bar. Estimate : (a) the heat received per kg of steam, (b) the work done per kg of steam, (c) the heat rejected per kg of steam, and (d) the thermal efficiency of the cycle. Take k_p of superheated steam as 2·1 kJ/kg K.

[(a) 2,676·17 kJ/ kg; (b) 511·17 kJ/kg; (c) 2,165 kJ/kg; (d) 19·1%]

38. In a modified Rankine cycle, steam is supplied at 14 bar with 38°C superheat. It expands isentropically to 2 bar and then released at constant volume to the exhaust pressure of 1 bar. Determine, using steam or H–Φ chart, (a) the thermal efficiency of the cycle, and (b) ideal steam consumption in kg per kW per hour. Take k_p of superheated steam as 2·1 kJ/kg K.

[(a) 17·85%; (b) 8·22 kg]

39. An engine is supplied with dry saturated steam at 11 bar. Expansion is carried out at constant entropy to a pressure of 2 bar and then released at constant volume to an exhaust pressure of 0·15 bar. Sketch the ideal pressure-volume and temperature-entropy diagrams for the cycle and indicate thereon the work done and the condition of steam at exhaust. Using the steam tables only, find the index of expansion, and the work done in kJ per kg of steam. Taking Rankine cycle as standard, find the percentage loss of work due to incomplete expansion.

[1·137; 446·85 kJ/kg; 32·02%]

40. (a) Discuss the causes of loss of thermal efficiency in a steam engine.

(b) State the cause and effect of "initial condensation" in the cylinder of a steam engine.

(c) How do you calibrate an indicator diagram of a steam engine? What is a saturation curve?

41. Explain what do you understand by the term missing quantity as supplied to a steam engine? What are the causes of missing quantity and what measures are taken to reduce in its practice?

The total mass of steam in an engine cylinder is 0·037 kg. From the indicator card it is found that at a point during expansion stroke the pressure is 3·5 bar and volume occupied by steam is 0·014 m^3. Calculate the dryness fraction of the steam and also the missing quantity in kg per stroke at this point.

[0·7216; 0·0103 kg]

42. Discuss in detail the different methods employed for reducing cylinder condensation in reciprocating steam engines.

In a double-acting steam engine, the measured mass of steam per minute was 245 kg, and speed was 110 r.p.m. The volume immediately after cut-off was 0·13 m^3 and the pressure at this point was 12 bar. The volume immediately after compression had begun was 0·04 m^3 and the pressure was 3·5 bar. Find the missing quantity in kg per minute and as a percentage of the actual cylinder feed at cut-off. The volumes given in the question are the total volumes at the points referred to.

[86·68 kg; 35·38%]

43. Explain the term "missing quantity" and state its causes and methods adopted for reducing it.

A non-condensing, double-acting steam engine has a cylinder diameter of 25 cm and 50 cm stroke and the clearance volume is 9 per cent of the swept volume. The steam supplied is dry saturated and the engine runs at 100 r.p.m. At point A on the expansion curve immediately after cut-off and at 40 per cent of the outward stroke, the pressure taken from the indicator card is 5 bar. At a point B on the compression curve at 90 per cent of the return (exhaust) stroke the pressure is 1·4 bar.

Determine the indicated and actual steam consumptions per hour if the missing quantity at point A is 0·01 kg per stroke.

[338·88; 458·88]

44. A double-acting, steam engine running at 75 r.p.m. has a piston diameter of 70 cm and 120 cm stroke and the clearance volume is 7 per cent of the swept volume. The pressures at cut-off and beginning of compression are 10 bar and 3.5 bar respectively.

Assuming that the expansion follows the law $PV^{1·2}$ = C and the engine is supplied with 150 kg of steam per minute, estimate : (i) the pressure of steam at 0·5 of the expansion stroke, and (ii) the dryness fraction of the steam and the missing quantity per hour at 0·5 of the expansion stroke.

[(i) 7·185 bar; (ii) 0·803, 2,137·6 kg/hr]

45. Draw graphs showing how the steam consumption is kg/hour varies with indicated power in a simple steam engine governed (a) by throttling the entering steam, and (b) by varying the point of cut-off.

46. Sketch the shape of graph you would expect to obtain if steam consumption in kg/hour were plotted to a base of indicated power in kW during the trial of a throttle governed steam engine running at constant speed.

A throttle governed steam engine running at constant speed uses 350 kg of steam per hour when developing indicated power of 18·4 kW and 700 kg per hour when developing indicated power of 44 kW. Esimtate the indicated thermal efficiency of the engine developing indicated power of 22 kW assuming that the steam supplied is dry saturated at 11 bar and exhaust is at 0·3 bar.

[7·96%]

47. State what is meant by "Willan's line"

A steam engine governed by throttling and running at constant speed uses 600 kg of steam per hour when developing indicated power of 41 kW and 2,500 kg of steam per hour when developing indicated power of 214 kW. Derive the Willan's law of the engine and find the steam consumption (i) in kg per hour, and (ii) in kg per kW-hour when developing indicating power of 148 kW.

[S = 150·28 + 10·98 I (i) 1,775·32 kg/hr.; (ii) 11·995 kg.kW-hr.]

48. State what is "Willan's straight line law" ?

A throttle governed steam engine running at a constant speed uses 14·3 kg of steam per kW per hour when developing indicated power of 7·4 kW and 10·9 kg per kW per hour when developing indicated power of 37 kW. Calculate the indicated thermal efficiency of the engine when developing indicated power of 22 kW. Dry saturated steam at 13 bar is supplied and temperature of condensate is 52°C.

[12·2%]

49. Explain the terms : (i) Willan's law, and (ii) Willan's line.

A throttle governed steam engine running at a constant speed consumes 1,638 kg of steam per hour when developing indicated power of 206 kW and 2,725 kg of steam per hour when developing indicated power of 375 kW. Determine the efficiency ratio for the steam engine when developing indicated power of 295 kW. The steam is supplied at 16 bar and 300°C and the back pressure is 0·4 bar. Take, k_p of the superheated steam as 2·3 kJ/kg K.

[71·76%]

10

INTERNAL COMBUSTION ENGINES - RECIPROCATING

10.1 Introduction

Heat engine in which the combustion of fuel takes place in the engine cylinder is known as internal combustion engine. As the combustion takes place inside the engine cylinder, very high temperature is produced in the cylinder. It is therefore, necessary to abstract or remove some of the heat from the cylinder to prevent damage to the metal of the cylinder, by circulating water through jacket, surrounding the cylinder. The cylinder of a motor cycle or an aero-engine is cooled by atmospheric air. It may be noted that the cylinder of a steam engine requires to be heated by supplying steam (from the boiler) in the jacket surrounding the cylinder to reduce condensation of steam in the cylinder.

The thermal efficiency of an internal combustion engine is much higher than that of a steam engine plant, as in I.C. engine there is no apparatus corresponding to boiler and no losses corresponding to the boiler losses. A best modern I.C. engine converts about 30 to 35 per cent of the heat of combustion of fuel into work (i.e. the thermal efficiency is about 30 to 35 per cent), whereas an ordinary steam engine plant converts only 8 to 10 per cent and a best modern steam turbine plant converts only 15 to 25 per cent of heat of combustion of fuel into work, i.e., the overall efficiency of a modern steam plant (boiler and turbine combined) is about 15 to 25 per cent.

Reciprocating internal combustion engines are most commonly single-acting whereas reciprocating steam engines are nearly always double-acting. All large size I.C. engines and marine I.C. engines are double-acting.

As the combustion of fuel takes place inside the engine cylinder of internal combustion engine, they are relatively smaller in size as compared to steam engine plant. A steam engine plant needs a boiler, a condenser, and an economiser. Internal combustion engine can be started quickly within a short time, whereas a steam engine plant will require much more time as steam has to be generated in the boiler before the steam engine can be started.

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 his famous Otto gas engine. The operating cycle of this engine was based upon principles first laid down in 1860 by a French engineer named Mr. Bea be Rochas. The majority of modern I.C. engines operate according to these principles.

The development of the well known Diesel engine began about 1893 by Mr. Rudolf Diesel. Although this engine differs in many important aspects from the Otto engine, the operating cycle of modern high speed Diesel engines is thermodynamically very similar to the Otto engines.

10.2 Classification

Internal combustion engines may be classified according to the :

- *Type of fuel used* - Diesel oil engine, Petrol engine, Gas engine, and Light oil (Kerosene) engine.
- *Nature of combustion* - Otto or Constant Volume combustion cycle engine, Diesel or Constant pressure combustion cycle engine and Dual-combustion cycle (combustion partly at constant volume and partly at constant pressure) engine.
- *Cycle of operation (number of strokes required to complete each cycle)* - Two-stroke cycle engine and four-stroke cycle engine.
- *Method of igniting fuel* - Spark ignition (S.I.) engine, and Compression ignition (C.I.) engine.
- *Method of fuel supply to the engine cylinder* - Carburettor engine, Air injection engine, and Solid or Airless injection engine.
- *Arrangement of cylinder* - Horizontal engine, Vertical engine, V-type engine, and Radial engine.
- *Speed of the engine* - Low speed engine, Medium speed engine, and High speed engine.
- *Method of cooling the cylinder* - Air cooled engine and Water cooled engine.
- *Method of governing the engine* - Hit and miss, Quality, and Quantity governed engine.

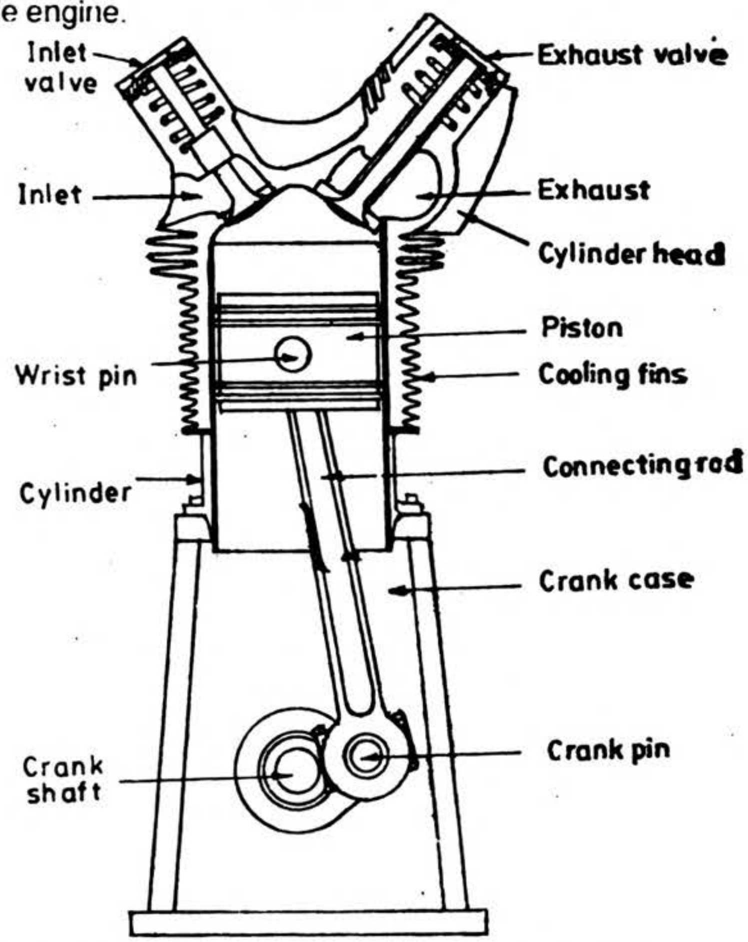


Fig. 10-1. Cross-section of air cooled, single cylinder, single-acting, vertical petrol engine.

- *Application of the engine* - Stationary engine, Portable engine, Marine engine, Automobile engine, Tractor engine, Locomotive engine and Aero engine.
- *Number of cylinders* - Single-cylinder engine, Twin-cylinder engine and multi-cylinder engine, etc.
- *Action of product of combustion upon the piston* - Single-acting engine and Double-acting engine.
- *Suction pressure* - Naturally aspirated engine and Supercharged engine.

The engine shown in fig. 10-1 is commonly known as petrol engine, but it can be more properly described as : 20 B.H.P., petrol, Otto four-stroke cycle, spark ignition, single-cylinder, single-acting, vertical, high speed, air cooled, quantity governed, stationary engine.

10.3 Engine Parts and Terms

Internal combustion engines have one or more cylinders in which combustion of fuel takes place. A cross-section of an air cooled, single-cylinder, petrol engine (Otto cycle engine) with principal parts is shown in fig. 10-1. One end, i.e., top end of the cylinder is closed by means of a cover known as **cylinder head** as shown in fig. 10-2, which contains the **inlet or admission valve** for admitting the mixture of air and petrol into the cylinder, an **exhaust valve** for removing the products of combustion (exhaust gases) from the cylinder, and a **spark plug** for igniting or exploding the mixture of petrol and air.

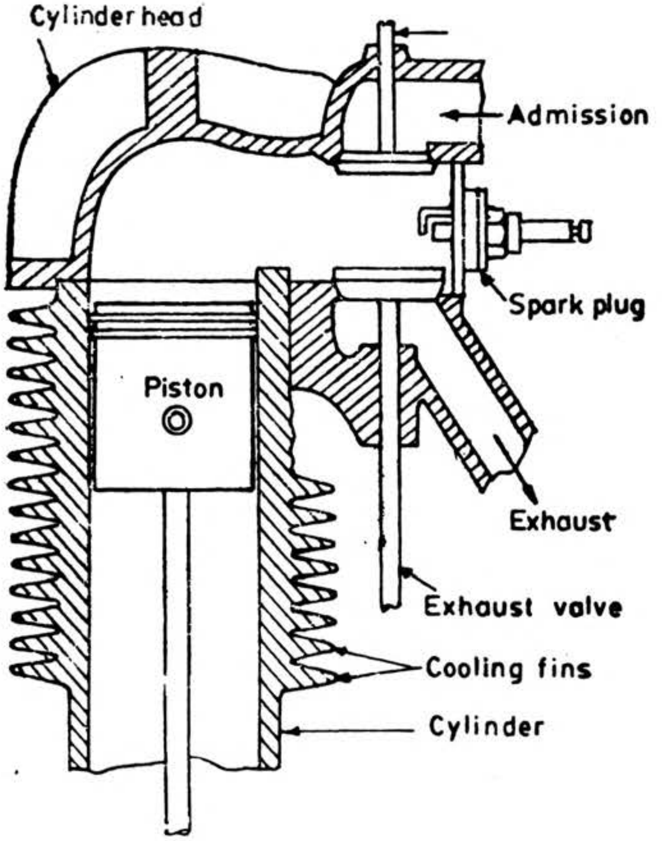


Fig. 10-2. Section of a vertical, air-cooled cylinder of a petrol engine.

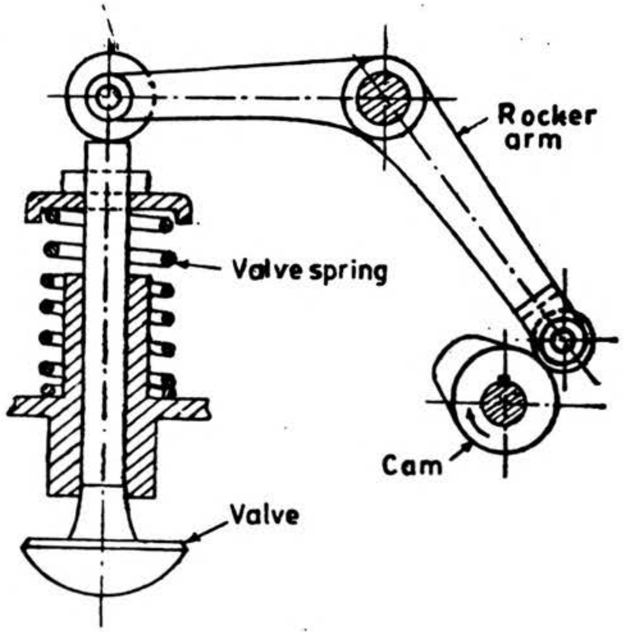


Fig. 10-3. Cam and rocker mechanism for opening and closing air and exhaust valves.

The two valves are kept closed by means of springs and are opened mechanically by means of **levers or rocker arms and cams as** shown in fig. 10-3. The cams are fitted on the **camshaft**. The camshaft is driven by the crankshaft through gear wheels fitted on each shaft. The passages in the cylinder head leading to and from the valves are called **ports.** If the inlet ports of the various cylinders of a multi-cylinder engine are connected to a common inlet pipe of engine, the pipe is called the **inlet manifold.** If the exhaust ports are similarly connected to a common exhaust pipe, the pipe is called the **exhaust manifold.**

The **piston** shown in fig. 10-2 is a gas tight movable cylindrical disc which slides up and down in the cylinder against which the combustion pressure acts, to cause the crankshaft to rotate by means of connecting rod which is connected to the piston at the other end. The piston is given a gas tight fitting in the cylinder by means of **piston rings** shown in fig. 10-1.

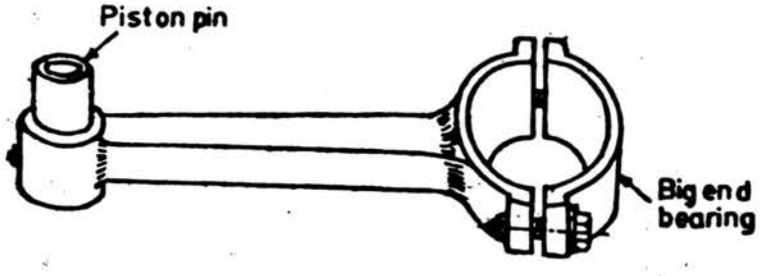


Fig. 10-4. Connecting rod.

The **connecting rod** (figs. 10-1 and 10-4) connects the crank pin with the piston and transmits the force due to the pressure of the cylinder gases on the piston head, down to the crankshaft. To provide for the swinging operation of the connecting rod, the upper (small) end of the connecting rod is fitted to the piston by means of cylindrical pin called the **piston pin** or **gudgeon pin** or **wrist pin** (figs. 10-1 and 10-4).

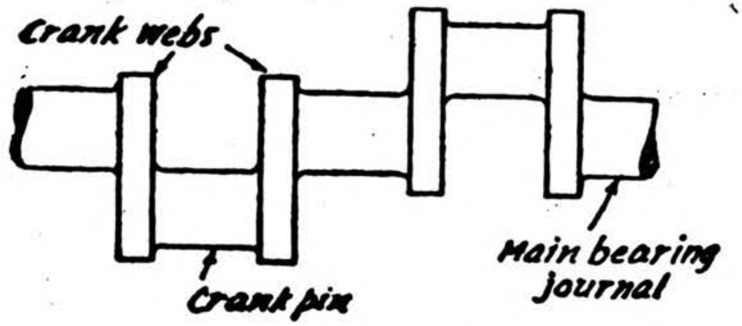


Fig. 10-5. Crankshaft of a multi-cylinder engine.

The lower (big) end of the connecting rod is fitted to the **crank pin** (figs. 10-1 and 10-5). The **crankshaft** is the principal rotating part of the engine. This shaft transmits the reciprocating motion of the piston to the driven unit (say electric generator) in the form of rotating motion. This shaft is built with one or more eccentric portions called **cranks.** Crankshaft with two cranks is shown in fig. 10-5. The part of crank to which the big end of connecting rod is fitted, is called **crank pin.**

The main body of the engine which contains the crankshaft and main bearings, is called the **crankcase** (fig. 10-1). This part hold (supports) the crankshaft and other engine parts in alignment (in line) and resists the explosive and inertia forces produced during working of the engine.

The piston in fig 10-2 is shown in top dead centre position which is the top most position of the piston during upward movement. At this position piston comes to rest. The piston travels from **top dead centre** to a point near the bottom end of the cylinder, called **bottom dead** centre. The distance the piston travels during one stroke, from top dead centre to bottom dead *centre*, is called the **length of stroke** or **piston stroke**. The inside diameter of the cylinder is called the **bore**.

The volume swept by the piston during one stroke, i.e., while moving from top dead centre to bottom dead centre, is called the **displacement volume of the cylinder.** The volume or space between the piston and the cylinder head, when the piston remains at the top dead centre position, is called the **clearance volume**. This clearance space forms the combustion chamber where the combustion of mixture of fuel and air takes place.

Some familiarity with the terms and functions of parts explained above is necessary for understanding the basic principles of engine operation. The other important parts, namely, fuel pump, fuel injector (fuel valve), carburettor, electric-spark ignition system, and governor will be described later in the chapter.

The following sequence of events (operations) is required to take place in any I.C. engine to complete the cycle :

.. The mixture of gas or fuel vapour and air in correct proportion in the case of gas or petrol engine, or pure air only in the case of Diesel engine, must be supplied to the engine cylinder.

.. The mixture of fuel and air in case of petrol engine, or pure air only in case of Diesel engine, must be compressed in the engine cylinder during the compression stroke, and in Diesel engine the fuel must be pumped into the cylinder through the fuel valve when compression of air is complete.

.. The compressed mixture of fuel and air in case of petrol engine or fuel oil in case of Diesel engine must be fired when compression is complete. In case of petrol engine the moderately compressed petrol-air mixture is fired by an electric spark, and in Diesel engine the fuel oil is fired when it comes in contact with highly compressed hot air at the end of compression stroke.

.. The resulting pressure rise, due to combustion of fuel and the expansion of combustion products, drive the piston out (i.e. the piston moves away from the dead centre) on its power stroke and rotates the crankshaft. The crankshaft in turn drives the machine connected to it.

.. When the expansion of combustion products is complete, the burnt-out gases must be cleared or removed from the engine cylinder to make room for fresh mixture of fuel and air in case of petrol engine or only air in case of Diesel engine to enter the cylinder.

10.4 Cycles of Operations

In any internal combustion engine, all the above mentioned operations for completing a cycle are carried out either in two strokes or four strokes. If an engine requires four strokes of the piston or two revolutions of the crankshaft to complete the cycle of the operations, it is termed a *four-stroke cycle engine.* If on the other hand, the cycle of operations is completed in *two-strokes of the piston* or in one revolution of the crankshaft, the engine is termed a *two-stroke cycle engine.*

Further any I.C. engine will work on one of the following three combustion cycles :

— Constant volume combustion or Otto cycle,

— Constant pressure combustion or Diesel cycle, and

— Partly constant volume and partly constant pressure combustion or Dual-combustion cycle.

These may be operated on either two-stroke or four-stroke cycle.

In **Otto engines** mixture of fuel and air is compressed to a moderate pressure of about 700 to 800 kPa (to prevent pre-ignition of the mixture during the compression stroke) and mixture is ignited by means of an electric spark while the piston remains on, or close to, the dead centre so that ignition takes place theoretically at constant volume. Petrol, gas, light oil (paraffin or kerosene), and heavy oil engines work on this cycle. In petrol and gas engines, the mixing of fuel and air takes place outside the cylinder (petrol is vapourised and mixed with air in correct proportion in the carburettor in case of petrol engines) and then the mixture is drawn in the cylinder during the suction stroke. In light oil engines, the fuel is vapourised and converted into vapour outside the cylinder by spraying oil into the vapouriser which is heated continuously by exhaust gases of the engine and then the mixture is drawn during the suction stroke. In heavy oil engines, the oil is vapourised by spraying oil into hot bulb or hot combustion chamber which is fitted on the top of the cylinder. In this case, the mixing of fuel and air is done in the cylinder before the compression of mixture starts. The hot combustion chamber requires heating by stove only at the time of starting the engine.

In **Diesel cycle engines**, only air is drawn in the cylinder during the suction stroke and compressed by the piston during the compression stroke to a high pressure (about 3,500 kPa) and to a temperature above the ignition temperature of the fuel oil. The fuel oil is then forced (pumped) into the cylinder under pressure just before the end of the compression stroke. Due to high temperature of the compressed air, the fuel oil is fired. The injection or admission of fuel oil is continued during the small part of the working stroke. The duration of the fuel injection is regulated by the governor. The fuel injection, during the working stroke is regulated in such a manner that the burning of the fuel takes place theoretically at constant pressure. Diesel cycle engine needs no spark plug or any other ignition equipment as the ignition takes place due to heat of the compressed air. It is, therefore, called compression-ignition engine. Heavy oil engines work on this cycle. Paraffin or light oil engines can also work on this cycle.

In **Dual-combustion cycle engines** 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 2,500 kPa. The heat of the compressed air (which is alone not sufficient to ignite the fuel) together with the heat of the hot combustion chamber ignites the fuel. The fuel is injected or sprayed into the hot combustion chamber just before the end of the compressure stroke, as shown in fig. 10-20, where it immediately gets ignited. The injection of fuel is continued during the small 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. Engines working on this cycle are sometimes known as **semi-Diesel engines**. This cycle is much used in heavy oil engines. Paraffin engines can also work on dual-combustion cycle. Modern high speed Diesel engines operate on this cycle.

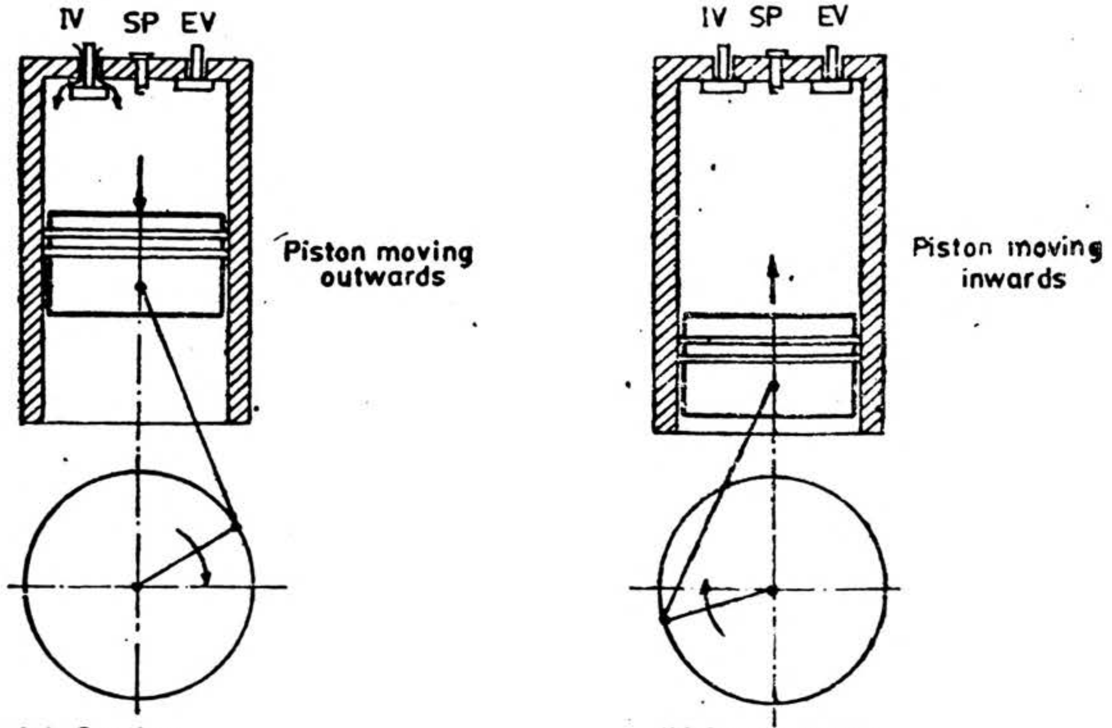
10.5 Four-Stroke Cycle Engines

10.5.1 Otto Engine : The working of the Otto four-stroke cycle as applied to gas engines, petrol engines, or light oil engines, is as under :

First stroke – Suction stroke,
Second stroke – Compression stroke,
Third stroke – Expansion stroke, and
Fourth stroke – Exhaust stroke.

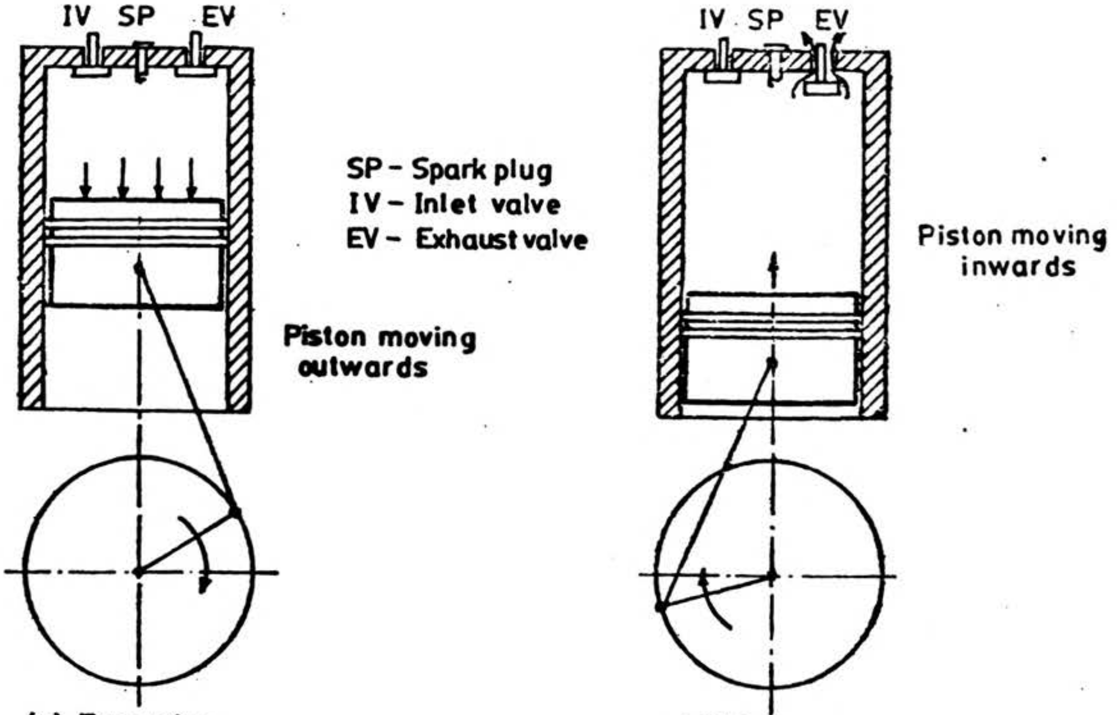
The suction stroke (fig. 10-6(a) starts with the piston at or very near the top dead centre when the inlet valve is open and the exhaust valve is closed. As the piston moves down-wards or outwards, suction is produced in the cylinder and fresh charge of fuel-air mixture enters the cylinder through the open inlet valve.

During compression stroke, both valves are closed, the piston moves up in the cylinder and compresses the charge of fuel-air mixture into the clearances space to a pressure of about 800 kPa when the piston reaches the top dead centre. This operation is represented by the line *ab* on the indicator diagram. Just before the end of the compression stroke, the mixture is ignited by an electric spark and ignition takes place rapidly which results in rapid rise in pressure of gases and combustion takes place at constant volume as shown by the line *bc* on the indicator diagram. The spark plug is fitted on the top of the cylinder head.



(a) Suction

(b) Compression



(c) Expansion

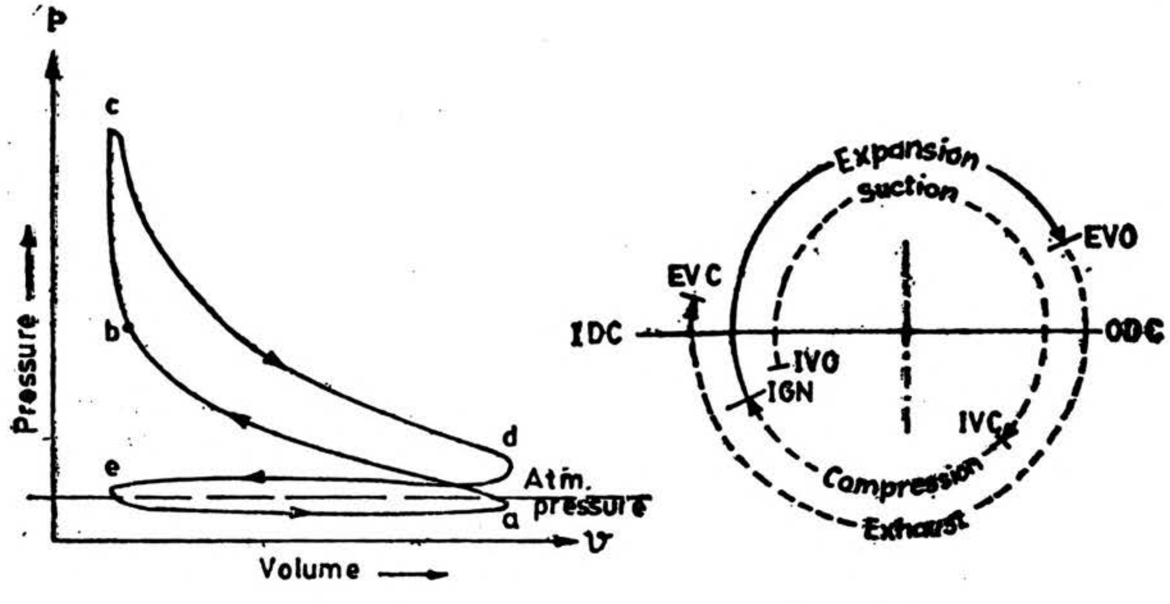
(d) Exhaust

Fig. 10-6. Operations of a vertical, Otto, four-stroke cycle engine.

During the start of **expansion stroke** [fig. 10-6(c)], the exhaust valve remain closed, and the hot high pressure gases drive the piston down on the power stroke back to bottom dead centre. As the gases expand and give up their energy to the piston, they become cooler and the pressure decreases.

During **exhaust stroke,** (fig. 10-6(d), the exhaust valve is open and inlet valve is closed. The piston now moves up in the cylinder again and burnt gases are driven out through the open exhaust valve. The exhaust valve closes after the piston reaches the top dead centre.

The indicator diagram of Otto four-stroke cycle engine is shown in fig. 10-7(a). The suction line *ea* lies below the atmospheric pressure line. This fall of pressure below the atmospheric pressure is as a result of the restricted area of the inlet valve passages. Due to the restricted area the entering mixture cannot flow into the cylinder in sufficient quantity to keep up the pressure (atmospheric) with the rapidly moving piston.



(a) Indicator diagram

(b) Valve timing diagram

Fig. 10-7. Horizontal, Otto, four-stroke cycle engine.

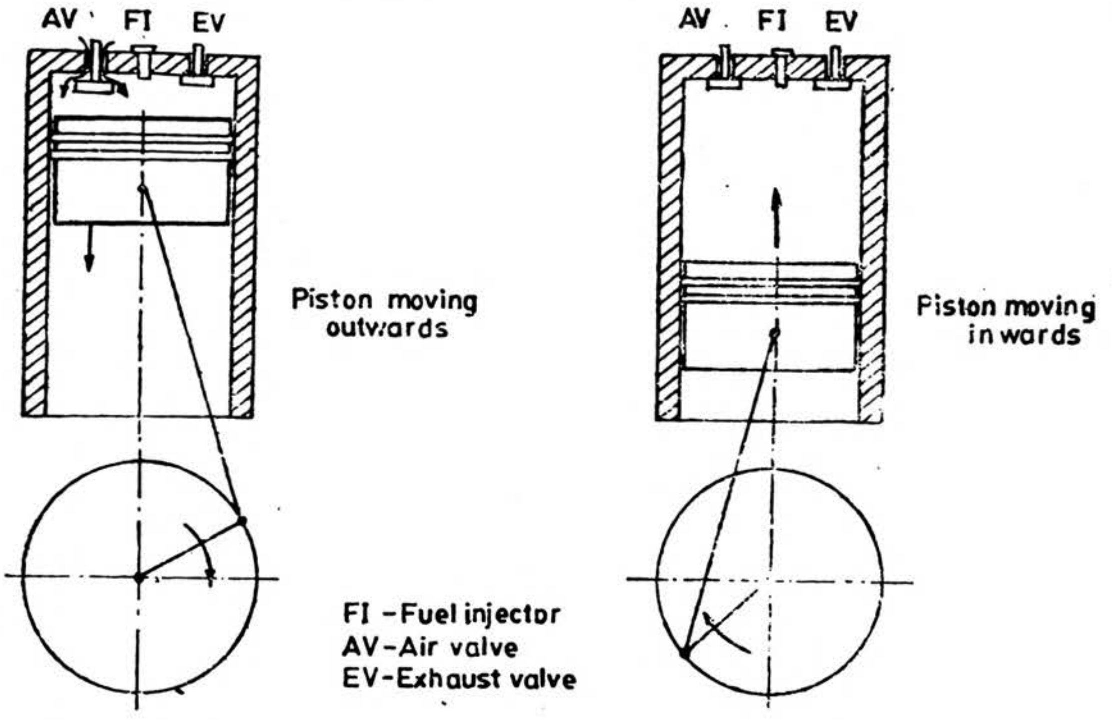
The exhaust line *de* does not coincide with atmospheric pressure line but it rises slightly above it. This is due to the restricted area of the exhaust passages which do not allow the gases to move out of the cylinder quickly as a result of which the exhaust pressure remains somewhat higher than the atmospheric pressure and works as a back pressure on the piston.

The negative loop, i.e. area *d-e-a* gives the pumping loss due to admission of fresh charge and removal of exhaust gases. The large area *abcd* (positive area) represents the gross work done by the piston during the cycle. The negative work is to be deducted from the gross work developed to get the net work done.

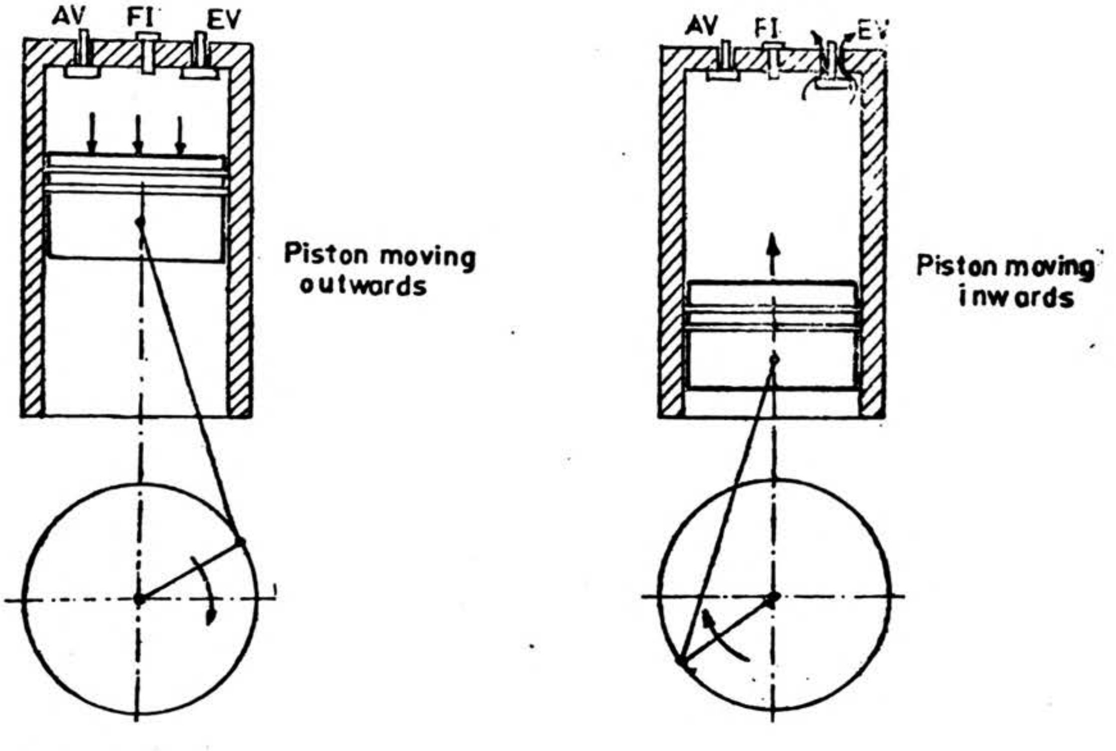
The cycle of operations is also shown by the *valve timing diagram* [fig. 10-7(b)]. The diagram shows the crank position from the dead centres when the various operations, i.e. suction, compression, expansion, and exhaust, begin and end.

It is to be noted here that positive work is done on the piston during only one of the four strokes of the cycle. Energy required to keep the crankshaft turning during the suction, compression and exhaust strokes is supplied by the flywheel. The engine flywheel stores energy received from the gases during the power or expansion stroke and releases it during the other three strokes.

10.5.2 Diesel Engine : An engine working on the Diesel four-stroke cycle has, for normal working, three valves - the air admission valve, the fuel valve and the exhaust valve. As in four-stroke Otto cycle, there are four distinct operations (suction, compression, expansion and exhaust) in the four-stroke cycle Diesel cycle.



(a) Suction (b) Compression



(c) Expansion (d) Exhaust

Fig. 10 8. Operations of a vertical, Diesel, four-stroke cycle engine.

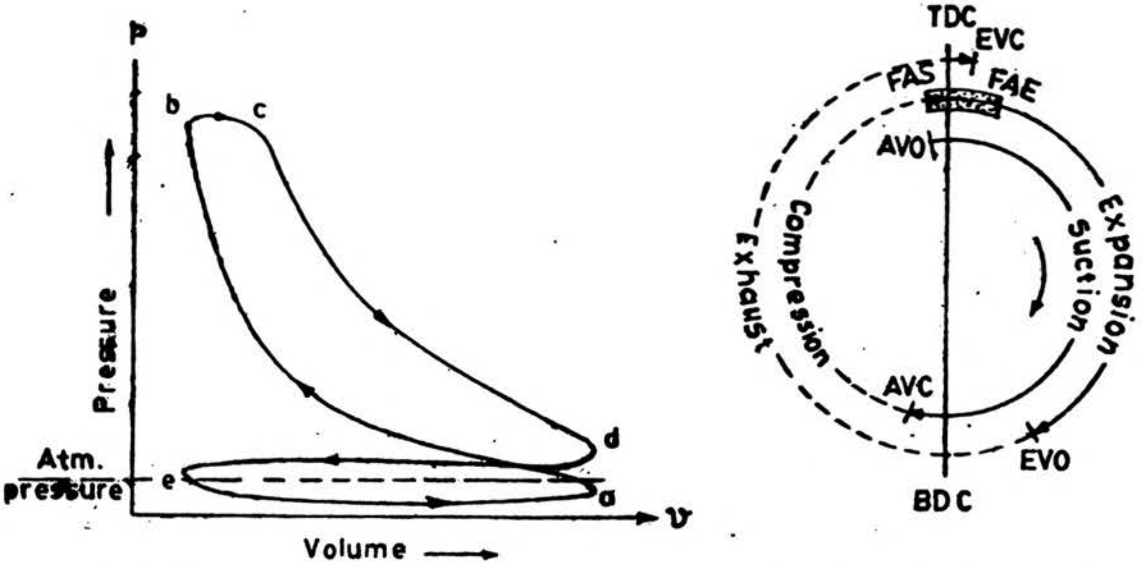
The suction starts with the piston on the top dead centre. With the air valve open and exhaust valve closed, the downward or outward movement of the piston causes suction in the cylinder [fig. 10-8(a)] which draws in a fresh charge of pure air from atmosphere at atmospheric pressure.

Compression stroke starts [fig. 10-8(b)] with both valves closed, the piston moves up in the cylinder and compresses the air in the clearance space. The air taken in during suction stroke is at nearly atmospheric pressure and is compressed to a high pressure and temperature (about 3,500 kPa an 600°C).

During expansion stroke [fig. 10-8(c)] the air and exhaust valves remain closed. The fuel valve opens just before the beginning of the third stroke and remains open during a small part of the expansion stroke. The fuel is admitted through the fuel valve in the form of a fine spray into this hot air. The heat produced by the high compression of air raises the temperature of air sufficiently to ignite the fuel as soon as it is injected into the cylinder, and combustion goes on atleast as long as the fuel valve is open. The fuel is admitted gradually in such a manner that the pressure is maintained constant as long as the fuel valve remains open, i.e., during the burning period. The hot high pressure gases drive the piston down on the power stroke to the bottom dead centre.

With exhaust valve open [fig.10-8(d)] and air and fuel valves closed, the piston now moves up in the cylinder again driving out the burnt gases through the open exhaust valve, in order to make room for fresh air to enter the cylinder. The exhaust valve closes after the piston reaches the top dead centre.

The cycle is thus completed.



(a) Indicator diagram (b) Valve timing diagram

Fig. 10-9. Vertical, four-stroke cycle Diesel engine.

The indicator diagram of a Diesel four-stroke cycle engine is shown in fig. 10-9(a). The suction line *ea* lies below the atmospheric pressure line. This fall of pressure below the atmospheric pressure is a result of the restricted area of the inlet passages. Due to the restricted area, the entering air cannot flow into the cylinder in sufficient quantity to keep up the pressure (atmospheric) with the rapidly moving piston.

The exhaust line *de* does not coincide with the atmospheric pressure line but it rises slightly above it. This is due to the restricted area of the exhaust passages which do not allow the gases to move out of the cylinder quickly as a result of which the exhaust pressure remains somewhat higher than the atmospheric pressure and works as a back pressure on the piston.

The negative loop, i.e., area *d-e-a*, gives the pumping loss due to admission of fresh charge of air and removal of exhaust gases.

The area *d-e-a* is called the negative work done or pumping work. The large area *a-b-c-d* represents the gross work done by the piston during the cycle. The net work done is the difference of positive area *a-b-c-d* and negative area *d-e-a*.

The cycle of operations is also shown by the **valve timing diagram** [fig. 10-9(b)].

10.6 Valve Timing (Setting) Diagram of Four-Stroke Cycle Engines

This is the most useful diagram which shows the correct crank positions for the opening and closing of inlet and exhaust valves, and beginning of ignition in case of Otto engines and beginning of fuel injection in case of Diesel engines, in order to obtain the best performance from the engine.

Figure 10-7(b) shows the valve timing diagram of a four-stroke, high speed, horizontal petrol engine. The inlet valve opens about 5° before the inner dead centre (I.D.C.). The inlet valve closes about 50° after the outer dead centre (O.D.C.) position is passed. The closing of the inlet valve much after the O.D.C. enables greater charge per cycle to enter the cylinder.

Similarly, to obtain the best clearing (scavenging) of the exhaust gases it is usual to have the exhaust valve opening about 50° before the crank reaches (O.D.C.) on the expansion stroke. In this way the speed of the exhaust gases through the exhaust valves is increased, and the improved scavenging is obtained at the cost of power from the expansion (power) stroke. Closing of the exhaust valve is about 10° after I.D.C.

It takes some times after the occurance of the spark to ignite the fuel. The spark in petrol engine is therefore arranged to occur about 38° before I.D.C. (fig. 10-7b) so that the actual combustion takes place at the moment piston reaches the inner dead centre and starts the next stroke. If the spark is advanced too far, complete ignition may take place before the crank reaches the inner dead centre and causes a back explosion and this in turn will cause the engine to run in the reversed direction of rotation.

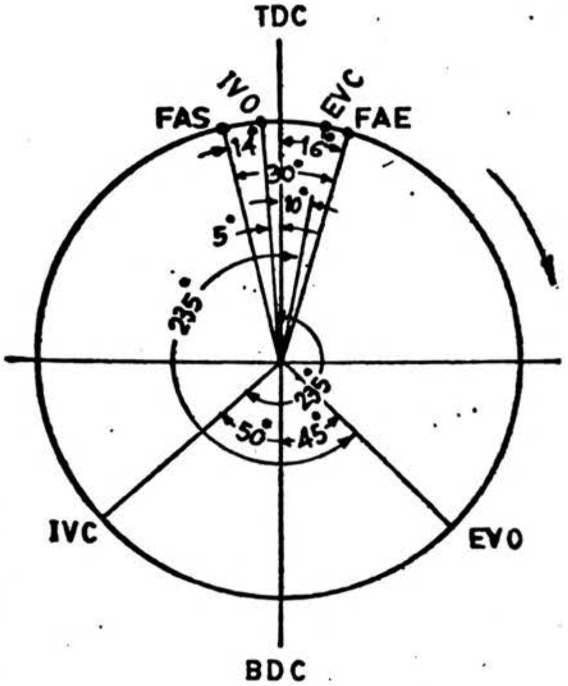


Fig. 10-10. Typical valve timing diagram of a vertical Diesel, four-stroke cycle engine.

Figure 10-10 shows the typical valve timing diagram of a four-stroke, high speed, vertical Diesel engine. The inlet (air) valve opens 5° before T.D.C. and exhaust valve closes 10° after T.D.C. It will be seen from the diagram that both the valves (inlet and exhaust) remain open at the same time for some part of the cycle near the T.D.C. The period i.e., 15° for which both valves remain open at the same time is called "*overlap*".

For quick removal of the exhaust gases from the cylinder and their replacement by fresh air from atmosphere, 15° overlapping between exhaust and inlet valves near T.D.C. is essential. The process of removing the exhaust gases with the help of incoming fresh charge of air is called **scavenging.**

The inlet valve closes late (50° after, B.D.C.) to make use of high velocity of air induced in the air intake pipe and thus to obtain greater charge of air into the cylinder.

The exhaust valve opens early (45° before B.D.C.) to reduce the pressure in the cylinder, as near the atmospheric pressure as possible, before the exhaust stroke starts.

There is a brief interval of time for the fuel oil to mix with hot compressed air in the cylinder and ignite. The injection of fuel oil is, therefore, started before the end of the compression stroke, i.e. 40° before T.D.C., so that the actual combustion takes place at the moment piston reaches the T.D.C. and starts the next stroke (i.e. expansion stroke).

It may be noted that the valve timing is a function of the engine speed. With increase of the engine speed the inlet valve is closed later, exhaust valve is opened earlier, and spark is arranged to occur earlier in Otto cycle engines, and injection of fuel oil is timed to begin earlier in case of Diesel engines.

10.7 Two-Stroke Cycle Engines

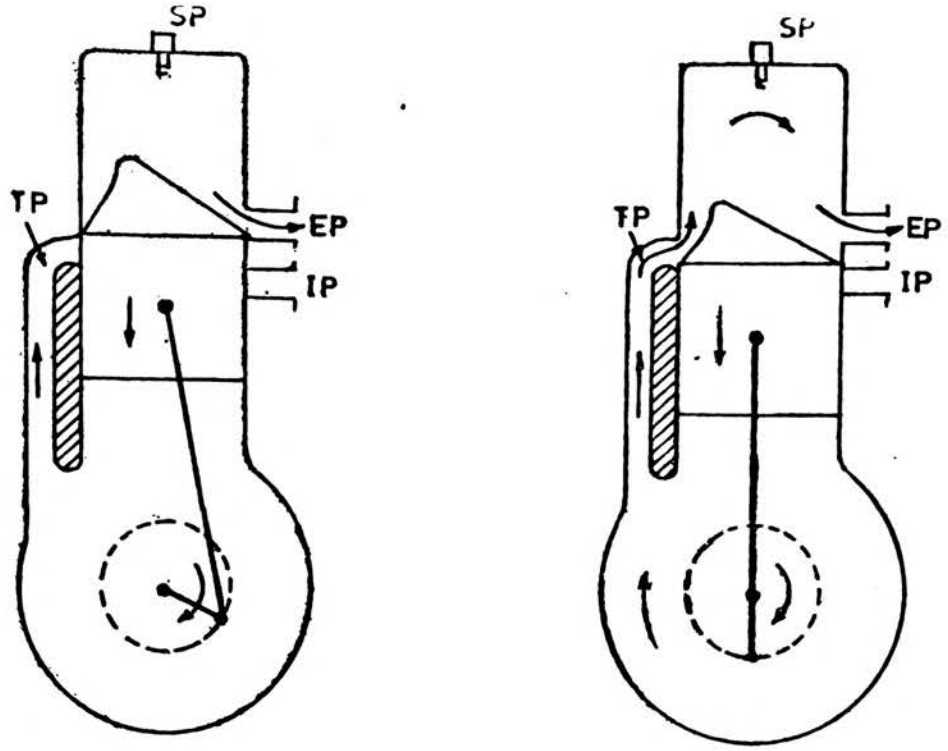
This cycle was invented by Sir Dugald Clerk. In this cycle, the operations of charging the cylinder with mixture of air and vapourised fuel, or air alone in case of Diesel engines, compression of mixture, or air, expansion of gases and scavenging of the cylinder (removing the exhaust gases from the cylinder), are carried out in two strokes of the piston and in one revolution of the crankshaft. To achieve this object, the following two different methods of *scavenging* are used in two-stroke cycle engines :

.. To have a separate pump outside the cylinder, in which the air, or mixture of air and fuel, is compressed and then it is forced into the cylinder under pressure. The pump usually, is a part of the engine and is driven by it. The method is used by large engines.

.. The crankcase of the engine is designed as a compressor in which the air, or mixture is compressed, as shown in fig. 10-11, and then it is forced into the cylinder. The cylinder has diametrically opposite holes or ports in the cylinder walls at the crank end of the cylinder instead of valves. The piston is specially shaped to help the action of charging and scavenging. This method is used by small engines.

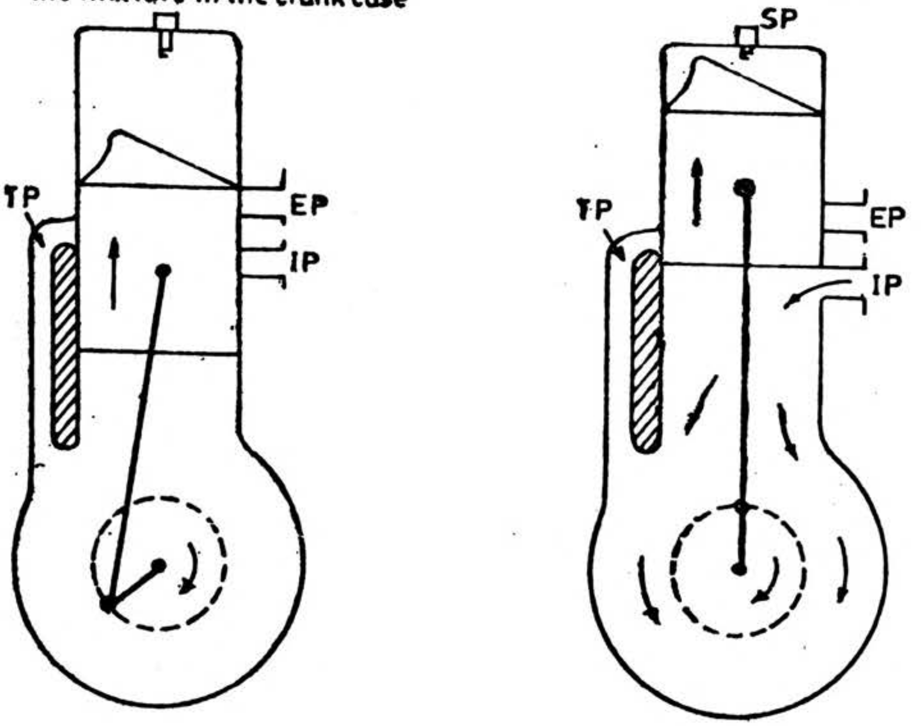
10.7.1 Otto Two-Stroke Cycle Engine: Figure 10-11, illustrates the two-stroke cycle petrol (Otto cycle) engine, employing crankcase compression. In this engine the charge of air-fuel mixture is compressed in the crankcase by the underside of the piston during the expansion or working stroke.

It will be easier to describe the cycle beginning at the point when the piston has uncovered (opened) the exhaust port during the out-stroke (down stroke) as shown in fig. 10-11(a). This is indicated by point *d* on the indicator diagram of fig. 10-12(a). When the piston is in this position the space above it contains the expanded gases. The expanded gases leave the cylinder through the exhaust port after having done work on the piston. A little later (fig. 10-11 b), a second port (transfer port) is uncovered by the piston. In this position of the piston, i.e. dead centre position, the transfer port (T.P.) connects the cylinder with the crankcase which contains slightly compressed charge of air mixed with fuel vapour at a pressure of about 130 kPa. This charge is transferred to the upper part of the cylinder through the transfer port. The piston is so shaped that the fresh charge of fuel and air will sweep up (i.e. move up) to the top of the cylinder and push out the remaining exhaust gases through the exhaust port. This is for the purpose of the scavenging or clearing the upper part of the cylinder of combustion



(a) Expansion of products in the cylinder and compression of the mixture in the crank case

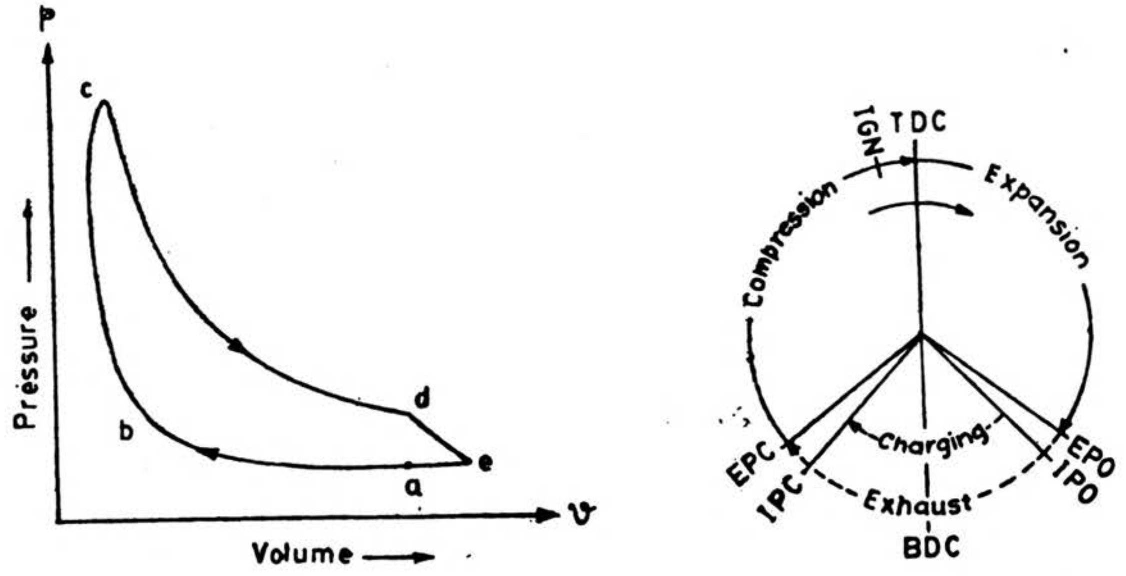
(b) Exhaust of gases and transfer of mixture



(c) Compression of mixture

(d) Mixture induced in the crank case

Fig. 10-11. Operations of a vertical, Otto, two-stroke cycle engine.



(a) Indicator diagram (b) Valve timing diagram

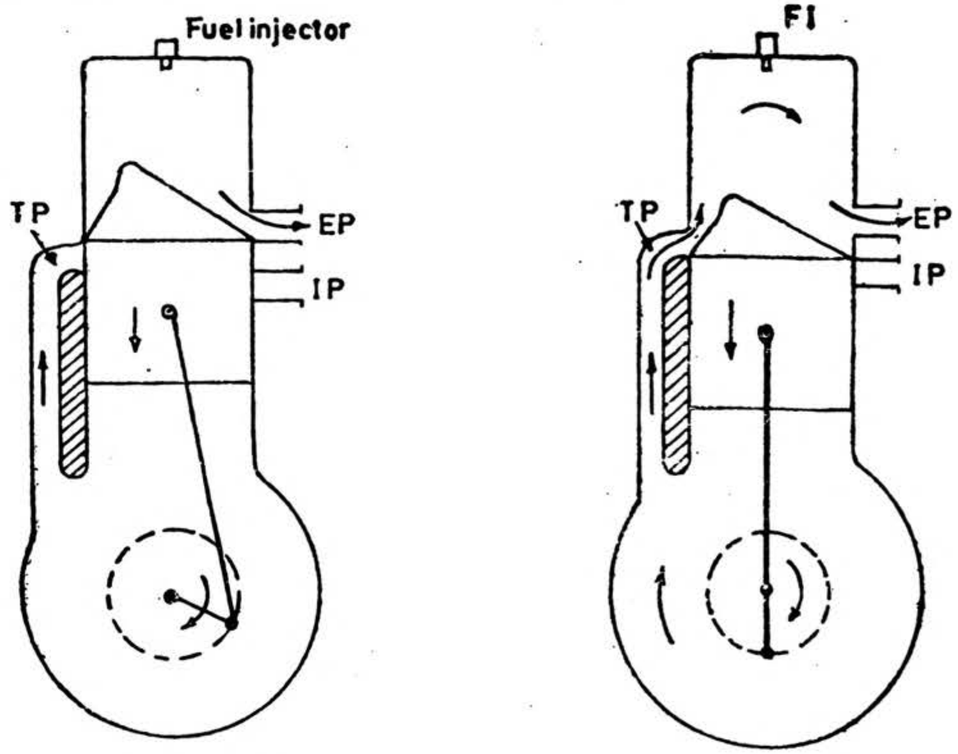
Fig. 10-12. Vertical, Otto, two-stroke cycle engine.

products (exhaust gases), and also to prevent the fresh charge of fuel and air from going out directly through the exhaust port and being lost. The projection on the piston is called *deflector*. There is, however, a loss of unburnt fuel through the exhaust port (E.P.)while scavenging and charging the cylinder with fresh charge.

During the in-stroke (up stroke) of the piston, compression of charge begins at point *a* [fig. 10-12(a)] as soon as the exhaust port is covered by the ascending (moving) piston. It may be noted that the transfer port closes first and then the exhaust port as shown in fig. 10-11(c). Compression starts at point *a* and goes on upto the end of in-stroke. The upward movement of the piston during compression stroke lowers the pressure (below atmospheric) in the crankcase so that fresh charge of air mixed with fuel vapour is induced (drawn) in the crankcase through the inlet port uncovered by the piston as shown in fig. 10-11(d). The charge compressed to about 700 to 800 kPa pressure in the cylinder during the compression, is then ignited by an electric spark which occurs slightly before the piston reaches the top dead centre. The hot high pressure gases drive the piston down on the power stroke. The cycle is then repeated. Indicator diagram of two-stroke Otto cycle engine is shown in fig. 10-12(a) in which *ab* is the compression line, *bc* is the constant volume ignition line, *cd* is the expansion line, *dea* shows the exhaust and charging (suction) period.

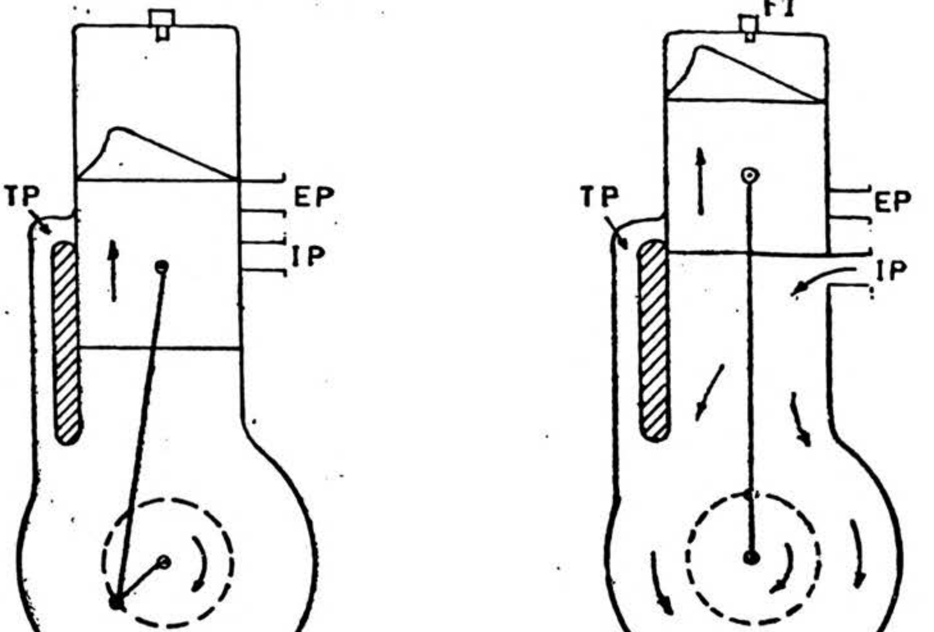
The cycle of operations is also shown by the valve timing diagram [fig. 10-12(b)]. The diagram shows the crank positions from the dead centres when the various operations (i.e., compression, expansion etc.) begin and end.

10.7.2 Diesel Two-Stroke Cycle Engine : One of the simplest type of Diesel two-stroke cycle engine is shown in fig. 10-13. In this engine the scavenging air is compressed in the crankcase by the underside of the piston during the expansion stroke. For this reason it is called a crankcase-scavenged engine. As the piston moves down on the power stroke (down stroke), it first uncovers the exhaust port as shown in fig. 10-13(a), and the cylinder pressure drops to atmospheric pressure as the combustion products leave the cylinder. Further movement of the piston uncovers the transfer port as shown in fig. 10-13(b). As soon as the transfer port opens, the slightly compressed



(a) Expansion of products in the cylinder and compression of air in the crank case

(b) Exaust of gases and transfer of air



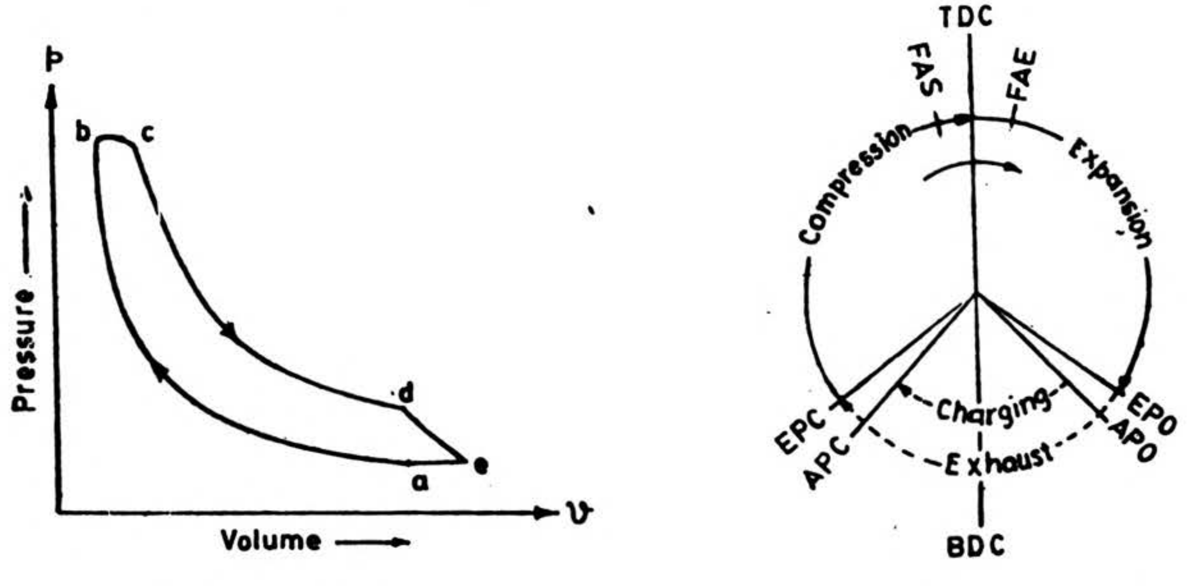
(c) Compression of air

(d) Air induced in the crank case

Fig. 10-13. Operation of a vertical, Diesel, two-stroke cycle engine.

air in the crankcase enters the engine cylinder at a pressure of about 130 kPa. The piston is so shaped that the fresh air will sweep up (or move up) to the top of the cylinder and push out the remaining exhaust gases through the exhaust port. This is for the purpose of scavenging or cleaning the upper part of the cylinder of combustion products, and also to prevent the fresh air from flowing directly to the exhaust port and being lost. The projection on the piston is called *deflector.*

During the in-stroke or compression stroke, first the transfer port closes and then the exhaust port is closed at *a* fig. 10-14(a). As soon as the exhaust port closes compression of the air begins as shown in fig. 10-13(c). Upward movement of the piston during compression stroke lowers the pressure in the crankcase so that the fresh air is drawn into the crankcase through the open inlet port as shown in fig. 10-13(d). Just before the end of the compression stroke the fuel is pumped, i.e., forced under pressure, in the form of a fine spray into this hot air. The heat produced by the high compression raises the temperature of air sufficiently to ignite the fuel as soon as it is injected into the cylinder, and combustion goes on atleast as long as the injection nozzle (fuel valve) is open. The rate of fuel injection is such as to maintain the pressure of gases approximately constant during the combustion period. The hot high pressure gases approximately constant during the combustion period. The hot high pressure gases drive the piston on the power stroke. When the piston is near the bottom of stroke the piston uncovers exhaust port in the cylinder wall at *d* [fig. 10-14(a)] which permits the gases to flow out of the cylinder. The cycle is repeated. Indicator diagram of Diesel two-stroke cycle engine is shown in fig. 10-14(a) in which *ab* is the compression line, *bc* is the constant pressure combustion line, *cd* is the expansion line, and *dea* shows the exhaust and charging (suction) period.



(a) Indicator diagram (b) Valve timing diagram

Fig. 10-14. Vertical, Diesel, two-stroke cycle engine.

The cycle of operations is also shown by the valve timing diagram fig. 10-14(b). The diagram shows the crank positions from the dead centres when the various operations begin and end.

10.8 Comparison between Two-Stroke Cycle and Four-Stroke Cycle Engines

Advantages of four-stroke cycle engines over two-stroke cycle engines are as under :

.. Scavenging (removing exhaust gases) is better because there is a separate exhaust stroke.

- Consumption of fuel oil and lubricating oil is low
- More time is available for removing the heat from the cylinder and, therefore the engine runs cooler.
- Since the engine runs cooler, compression is better maintained.
- Wear of the cylinder is less.
- The thermal efficiency is higher than two-stroke cycle engine because two-stroke cycle engine has increased fuel consumption (kg/kW/hr) owing to fuel losses through the exhaust port in case of Otto cycle engines.

The four-stroke cycle engine is not easily reversible, and therefore requires some special arrangement for the purpose. This is a *disadvantage.*

Following are advantages of two-stroke cycle engines over four-stroke cycle engines :

- The power developed will be twice that of a four-stroke engine of the same dimensions as it gives twice as many working strokes as the four-stroke cycle engine at the same engine speed or number of revolutions of the crankshaft. If separate scavenging pump is used it takes some power for its running. Therefore, extra power of a two-stroke cycle engine is only 80 to 90 per cent instead of 100 per cent.
- There is one working stroke for each revolution of the crankshaft. This reduces the size of the flywheel.
- It is lighter than the four-stroke cycle engine for the same power. Reduction of weight and space occupied, makes it suitable as marine engine.
- Absence of valves and the valve gear gives simplicity in construction and consequently low initial cost.
- It is simple to maintain.
- It is easily reversible.
- Fewer spare parts are required.

Disadvantages of two-stroke cycle engine over four-stroke cycle engine are as under :

- Scavenging is poor as there is no separate exhaust stroke to help scavenging work, as in the case of four-stroke cycle engine.
- The period during which the exhaust ports remain open is very short. This results in incomplete scavenging, leading to dilution of the fresh charge with burnt gases. Dilution of the fresh charge means loss of power output.
- In case of Otto cycle engine there is a loss of unburnt fuel through the exhaust ports while scavenging and charging the cylinder with fresh charge.
- Due to loss of fresh charge (unburnt fuel) through the exhaust ports, the fuel consumption is comparatively high, resulting in reduced thermal efficiency.

However, with increasing experience in designing and operation of the two-stroke cycle engine, its economy in fuel consumption has much improved and is closely approaching the economy of the four-stroke cycle engine.

- The fresh charge admission period is short, and therefore, there is less mass of fresh charge admitted to the cylinder. This reduces the power output.
- Consumption of lubricating oil is high.
- As the number of power strokes are double, the cooling system presents difficulty.
- Due to higher output, more wear and tear takes place for the same engine size and speed.

10.9 Fuel Supply in Petrol Engines

Most petrol engines operate on four-stroke, Otto cycle principle. The air and petrol are mixed in the correct proportion in the **carburettor** and are drawn into the cylinder during the suction stroke. The mixture of air and petrol is compressed during the compression stroke. Near the end of the compression stroke the charge is fired by a **spark**, and as the heat is released from the petrol, the pressure in the combustion space rises rapidly. The high pressure gases drive out the piston on the expansion stroke, and upon the opening of the exhaust valve, the exhaust gases are discharged to the atmosphere.

The apparatus used for vapourising petrol and other spirit fuels is called a carburettor. In this apparatus no heating is necessary. The main functions of the carburettor are :

- To maintain a small reserve of petrol at a constant level in the flat chamber,
- To supply a fine spray of petrol to the air entering the cylinder,
- To vapourise the petrol, and to produce homogeneous (uniform) air-petrol mixture, and
- To maintain a correct mixture of petrol and air at all speeds.

The simplest design of a carburettor is a **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, and a throttle valve which controls the amount of mixture delivered to the engine.

Petrol enters the float chamber (fuel chamber) through a filter and a valve. The level of petrol is maintained constant at correct height by the float. When the correct level shown in dotted lines is reached, the float rises and force the valve spindle downwards by means of levers and shuts off the petrol supply.

The suction of the engine causes air supply to rush through the choke tube which is shaped as a venturi cone. The choke tube surrounding the top of the fuel jet is

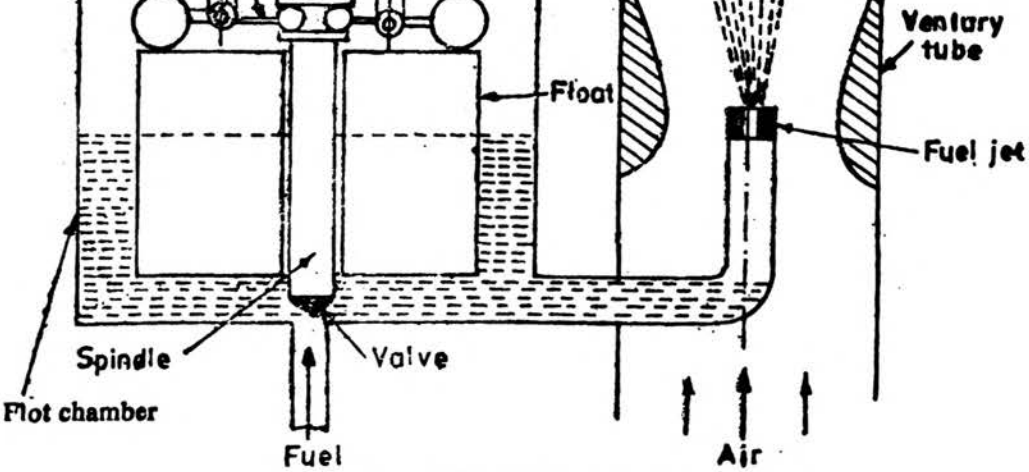


Fig. 10-15. Simple carburettor.

re-duced in dia-meter, 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, being produced by air vent (hole). The difference between the atmospheric pressure and the pressure around the top of the fuel jet causes the petrol to flow into air stream at the throat of choke tube and gets vapourised. The tip of jet is placed higher by 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 shown in fig. 10-15 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 fuel jet have been so designed as to permit the passage of 15 parts of air and one part of petrol by weight under certain conditions of suction. A mixture of proper proportion will be drawn into the engine. 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 the case. Petrol is more responsive to suction than air. The laws governing the flow of liquid 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 the engine increases faster than the flow of air, the mixture becoming too rich at high speeds. Thus, in a given example, if the velocity of air past the jet be doubled, the flow of petrol will be increased by about 2½ times.

Many different devices have been used for balancing or compensating this action of the simple single-jet carburettor 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 jet and the compensating jet.

10.10 Methods of Igniting Fuel

The energy of the fuel of I.C. engine is locked up in the fuel in the form of chemical energy. Some means have to be employed whereby this energy can be released and made available to run the engine. In addition to the fuel for the purpose of combustion, two things are necessary—the oxygen supplied with air and some means for igniting the fuel.

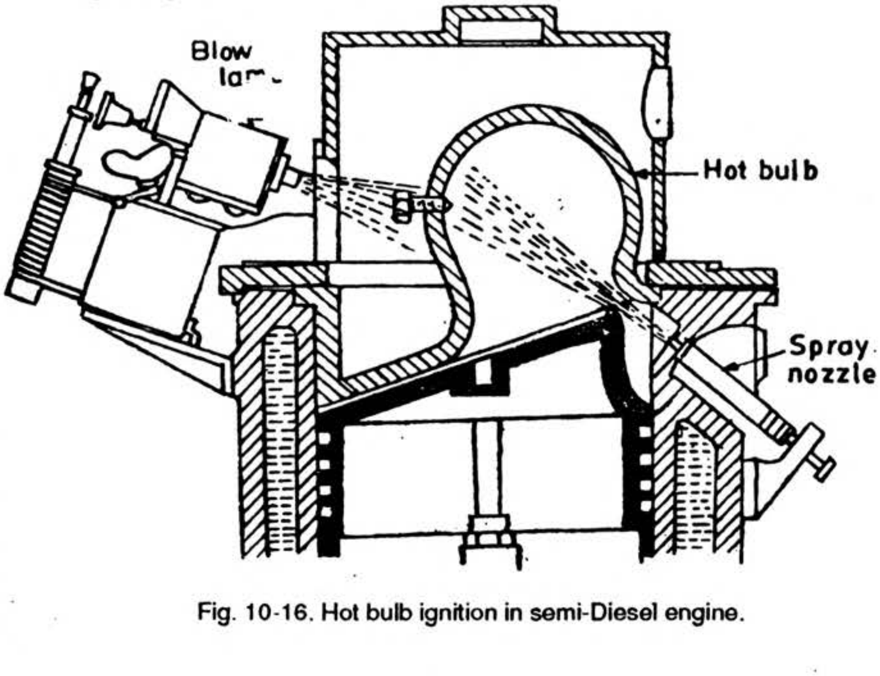


Fig. 10-16. Hot bulb ignition in semi-Diesel engine.

All petrol and gas engines use *electric spark ignition.* Diesel engines, on the other hand use the *heat of compressed charge of air* alone to ignite the fuel. Before the successful introduction of electric spark ignition, petrol and gas engines used *hot tube ignition.*

Semi-Diesel engines use *hot bulb ignition.* In this system a chamber of bulb shape is attached to the cylinder head as shown in fig. 10-16. This bulb is unjacked (not water cooled) 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 partly due to heat of the compressed air and partly by the heat of hot bulb.

All petrol engines use **electric spark** ignition. The compressed air-petrol mixture in a petrol engine must be fired at the correct instant so that the resulting rise in pressure acts on the piston when it (piston) is at the top dead centre. A high voltage is required to jump the gap of a spark plug and give a spark of sufficient energy to ignite the mixture, and this is produced by an induction coil. Spark ignition is also used in gas and light oil engines.

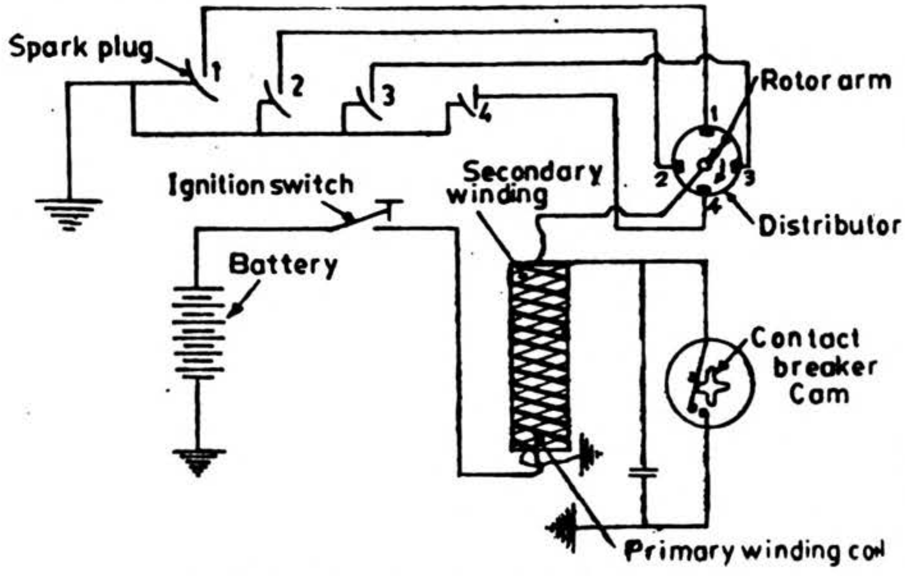


Fig. 10-17. Coil ignition of a four-stroke petrol engine.

The coil ignition circuit of a four-cylinder petrol engine working on the four-stroke cycle is shown in fig. 10-17. The primary circuit consists of a 12 volt *battery*, ignition switch, primary winding of the coil, and contact breaker. The *secondary* circuit consists of secondary winding of the coil, distributor and four spark plugs. *The primary winding* of the coil consists of a comparatively few turns of thick wire wound around an iron core. Around this is wound the *secondary winding* consisting of a large number of turns of thin wire. The contact breaker is worked by a cam driven at half the engine speed (for four-stroke engines).

To begin with, the *ignition switch* is switched on and the engine is cranked (turned by hand). When the contacts touch, the current flows from the battery through the switch and through the primary winding of the *induction coil,* to the contact breaker points and returns to the battery through the ground. A *condenser* is connected across the terminals of the contact breaker points. This prevents sparking at the breaker points. Immediately after this the moving cam break opens the contacts. The breaking of the primary circuit causes a change of magnetic field and this induces a very high voltage in the secondary winding. The ratio of number of turns of secondary to primary has been so adjusted to give a voltage of 8,000 to 12,000V across the secondary terminals. The high voltage passes to the distributor and then to the individual spark plugs in the order 1,3,4,2, which are screwed in the cylinder head. The high voltage is applied across the spark plug gap. Due to high voltage the spark jumps across the gap, causing ignition of petrol-air mixture in the cylinder.

The principle differences as regards fuel supply and ignition (combustion) between engines working on Diesel cycle and Otto cycle are :

.. The petrol engine draws a *mixture of air and petrol* into the cylinder during suction stroke; the Diesel engine draws in a charge of *air only,* and fuel (diesel oil) is forced under pressure into the cylinder through a fuel valve at the end of compression stroke.

.. The pressure at the end of compression in petrol engine is low, ranging from 1,00 kPa downwards to as low as 700 kPa; in the Diesel engine, the pressure at the end of compression is about 3,500 kPa, i.e., compression ratio is higher in Diesel engines.

.. As the compression ratio is higher in Diesel cycle engines, these engines are heavier than Otto cycle engines for same power developed and consequently the initial cost of Diesel cycle engines is higher.

.. A spark ignites the air-petrol mixture in petrol engines, while in the Diesel engines compression of air to about 3,500 kPa pressure, raises the air temperature to about 600°C — high enough to ignite the fuel when it is introduced in the cylinder.

.. Combustion or burning of fuel takes place approximately *at constant volume* in petrol engines (burning of the mixture takes place while piston remains at the top dead centre), while in Diesel engines combustion takes place approximately at *constant pressure* (rate of fuel injection is such as to maintain the gases at approximately constant pressure during the combustion period).

.. Power cost is lower in Diesel engines as diesel oil is cheaper than petrol.

10.11 Method of Supplying Fuel in Diesel Engines

Engines working on constant pressure (Diesel cycle) or dual combustion cycle (semi-Diesel cycle), both of which require pure air of compression, must have some external source of forcing the fuel oil into the cylinder. For these types of engines there are two distinct methods of fuel injection :

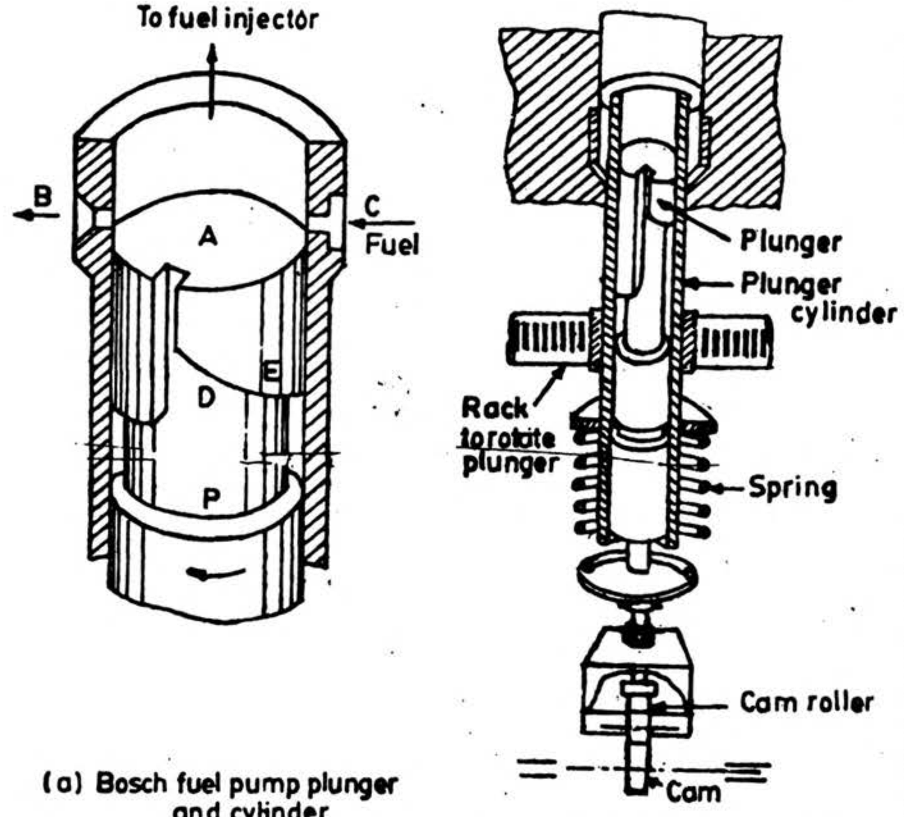
- *Air injection*, and
- *Airless injection* or *Solid injection.*

For these types of engines the following units or parts are necessary :

- A fuel pump to deliver fuel oil under pressure to the injection nozzle (injector),
- An injector or atomiser to inject fuel oil into the cylinder in a finely atomised state (fuel oil is injected in very small particles),
- A governor to regulate the fuel supply according to load, and
- An air compressor in case of air injection engine.

In **air injection** method of fuel injection, the fuel is injected through the fuel nozzle 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 6,500 kPa is applied to it. When the fuel injection valve is mechanically opened by the cam and rocker lever arrangement, the fuel is driven into combustion space (cylinder) at high velocity by the high pressure air. The high pressure air is supplied from storage air bottles which are kept charged (filled) by air compressor driven by the engine itself.

Solid injection is also termed as *airless* or *mechanical injection*. This method employs a mechanically operated fuel pump which supplies quantity of fuel required for the working stroke and to inject it through a fuel injection nozzle shown in fig. 10-19 under high pressure, with a view to atomise it (fuel) or break it into very small particles and so inject the fuel particles at a high velocity into the mass of compressed air in the combustion space. The injection pressure varies from 10,000 to 10,500 kPa and in some cases even more than this. The desired pressure is produced by the fuel pump of the plunger type shown in fig. 10-18.



(a) Bosch fuel pump plunger and cylinder

(b) Bosch fuel pump assembly

Fig. 10-18.

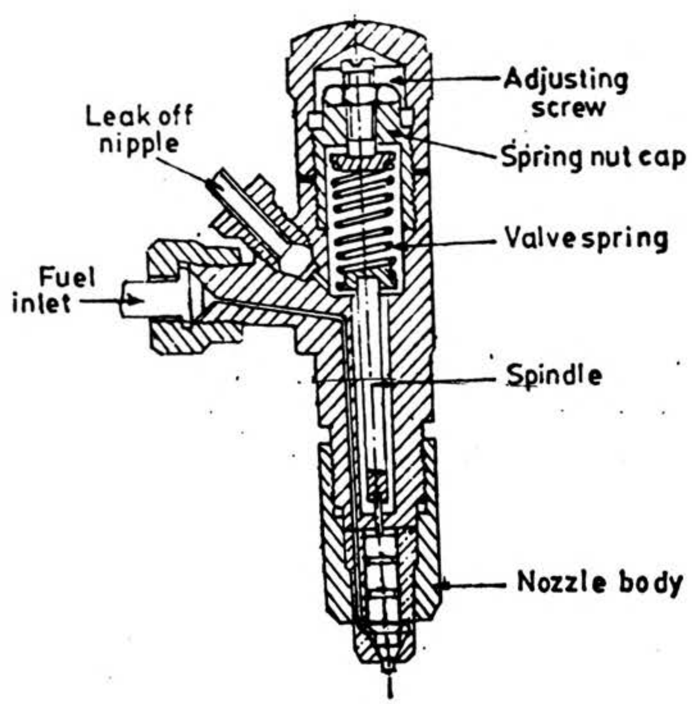
The fuel pump has two main functions as under :

- Start the fuel injection at the proper crank angle, late in the compression stroke, and
- Force through the nozzle into the cylinder, the exact quantity of fuel needed to produce the desired power.

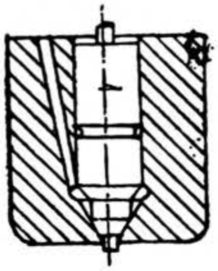
The Bosch fuel pump, shown in fig. 10-18, is widely used in modern Diesel engines. The plunger of the pump which is operated by a cam has a constant stroke, and can be rotated in the plunger cylinder so as to control the amount of fuel pumped to the nozzle. The vertical groove A of the plunger leads into the helical groove D. Fuel oil flows to the fuel pump under gravity when the fuel pump plunger uncovers the suction ports B and C on the downward stroke as shown in fig. 10-18(a), 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 the suction ports B and C. From then on, the fuel is put under pressure and the pump begins to force it through the delivery valve to the engine cylinder via the nozzle. Delivery begins as soon as the plunger has covered the ports on the way up, and ends as soon as the sloping edge E of the helical groove D opens the ports C on the right side and permits the fuel to escape from the pressure space above the plunger to the suction space through the port C. The pressure is then relieved and the delivery stops. The plunger P is rotated by the rack shown in fig. 10-18(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 engine is also varied.

The purpose of fuel injector is to atomise or break the fuel oil into fine particles and to direct the spray into the combustion chamber so that every fuel particle mixes with the air. In order to achieve this, injection takes place through very fine holes in the nozzle body at a pressure of about 15 MPa. The desired pressure is produced by the fuel pump of the plunger type (Bosch fuel pump) shown in fig. 10-18(b).

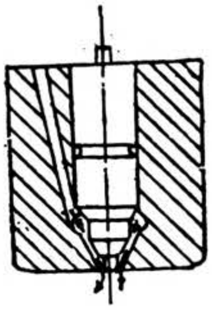
The fuel injector shown in fig. 10-19, is a simple spring loaded valve, consisting of a nozzle valve that fits in the nozzle body. The nozzle body valve is held down on to a conical seating by a spring that exerts pressure on the valve through the nozzle



(a) Section of a fuel injector



(b) Closing position



(c) Opening position

Fig. 10 19. Fuel injector and two different types of nozzles.

valve rod. The pressure at which the nozzle valve will be lifted, depends upon the amount of compression placed on the spring which is adjustable by a screw. The nozzle valve is usually set to open at 14 to 17·5 MPa pressure. When the fuel from the fuel pump enters the pressure chamber around the base of the nozzle valve, pressure rises until it is sufficient to raise the valve from its seating, allowing the fuel to be sprayed into the combustion chamber (cylinder) through the hole or holes, in the end of the nozzle body.

Any leakage of fuel that may accumulate above the nozzle valve, can be led away to a drain tank by means of a pipe connected to the leak-off pipe. The leakage of fuel occurs when the nozzle valve is worn out. The fuel injector is sometimes known *as atomiser.*

10.12 Methods of Governing

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 decreases, the speed of the engine will begin to increase, because the fuel supply is more for the decreased load. As the speed of the engine increases, the centrifugal force on the rotating weights of the governor will also increase and will move 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 of 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 :

.. Completely cutting off the fuel supply for one or more cycles - This is called *hit and miss method.*

.. 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.

.. 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.*

.. Combination of the quality and quantity method - This is called *combination method.*

Hit and Miss Governing 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 will be number of explosions omitted. The usual method of missing an explosion is to omit the opening of the gas valve in case of gas engines, or putting the plunger of the oil pump out of action in case of oil engines, so that no fuel is admitted and the engine performs an ideal stroke.

As applied to gas engines, the **Quality Governing** is effected by reducing the quantity of gas supplied to the engine. This is done by varying the lift (opening) 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, thereby controlling the quantity of gas supplied.

As applied to oil engines, quality governing is effected by varying the amount of fuel oil entering the engine cylinder per cycle. This is done by altering the angular position of the helical groove of the 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 solid injection, compression-ignition, high speed engines.

Quantity Governing may be accomplished by varying the amount of air-fuel mixture entering the cylinder, while the proportion of the mixture and number of working cycles are constant. It is applied to petrol engines by having a throttle valve in the pipe leading from the carburettor to the cylinder. The motor car engine is governed by hand by controlling the quantity of mixture entering the cylinders, the proportion of petrol to air remaining the same for a given carburettor adjustment.

The **Combination Governing** may be obtained by combining any two of the above systems. For instance, 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.

10.13 Methods of Cooling Cylinders

Very high temperature is developed in the cylinder of an I.C. engine as a result of the combustion of fuel taking place inside the cylinder. It is, therefore, necessary to extract some of the heat from the cylinder to avoid damage 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 of 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. Too much cooling on the other hand, will reduce the thermal efficiency of the engine. The object of cooling is achieved by any one of the two methods —

Very high temperature is developed in the

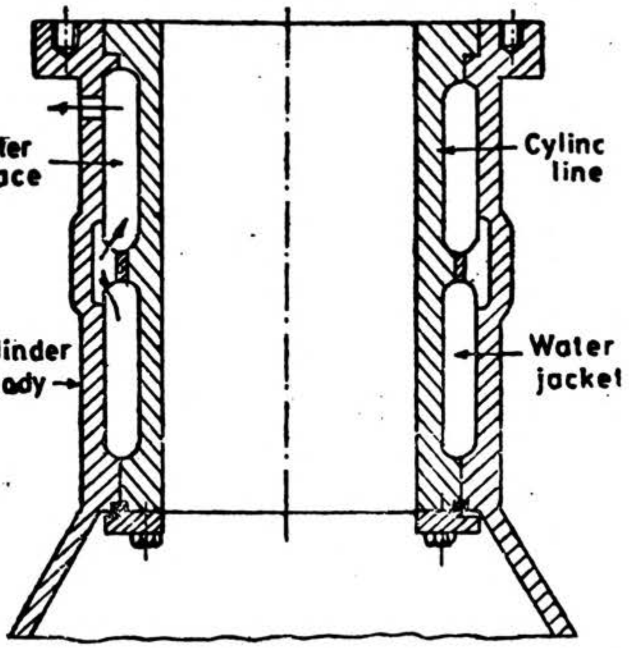


Fig. 10-20. Section of a water cooled cylinder.

Air cooling and *Water cooling*.

Air cooling is the simplest method in which the heat is carried away by the air flowing over and around the cylinder. In this method, fins are cast on the cylinder head and cylinder barrel with the object of providing additional conductive and radiating surface as illustrated in fig. 10-2. 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 the movement of the engine itself as in motor cycle, automobile, and aeroplane engines.

In water cooling method, the advantage of superior convective and conductive properties of water is taken. The cylinder is provided with an annular space called *water jacket* (fig. 10-20) 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 the lubricating oil. The water space should be wide in large cylinders and cleaning doors should be provided for cleaning the **water jacket**.

10.14 Power and Efficiency

10.14.1 Indicated Power : The power developed in the cylinder of an engine can be determined if the engine cylinder diameter and speed are known, and if an indicator diagram with its spring scale is available. The first step is to find the mean effective pressure.

The mean effective pressure is the average effective pressure acting on the piston for one stroke. The mean effective pressure depends on the mean height of indicator diagram and the scale of the spring used in the indicator. Thus, the mean height of the indicator diagram is first determined. The base length of the indicator diagram is carefully measured from boundary to boundary and is divided into any convenient number of equal divisions. This will divide the area into small strips. The mid-ordinates are then drawn for each strip. The length of each mid-ordinate is measured and then added together. The sum of the mid-ordinates is then divided by the number of mid-ordinates, giving the mean height of the indicator diagram as a result.

$$\text{Thus, mean height, } h \text{ in mm} = \frac{a_1 + a_2 + a_3}{\text{number of mid–ordinates}} \quad .. (10.1)$$

where a_1, a_2, a_3 are the lengths of mid-ordinates of the strips

The mean effective pressure, p_m in kN/m^2 or in kPa is then

$= h$ (mm) × spring scale or spring number (kN per m^2 per mm)

Another method of determining the mean height of the indicator diagram is to measure the area of the indicator diagram with the help of a planimeter. This area divided by the base length of the diagram will give the mean height of the indicator diagram.

The area may also be measured by counting the number of squares. If the square method is used, the smaller squares the more accurate is the result.

Thus, mean height, h in mm $= \dfrac{\text{area of the indicator diagram in mm}^2}{\text{base length of indicator diagram in mm}}$

The mean effective pressure, p_m in kN/m^2 is then

= mean height, h (mm) × spring scale (kN/m^2 per mm)

$$\therefore\ p_m = \frac{\text{area of indicator diagram (mm}^2)}{\text{length of indicator diagram (mm)}} \times \text{spring scale} \quad ..\ (10.2)$$

Let p_m = mean effective pressure in N/m^2,

a = area of the piston in m^2,

l = length of piston stroke in metre, and

n = number of working strokes or working cycles or explosions per second

The mean force on the piston $= p_m \times a$ newtons

∴ Work done per working stroke or cycle $= p_m \times a \times l$ N.m or J

Thus work done per second $= p_m \times a \times l \times n$ N.m/sec. or J/sec.

Indicated power $= p_m \times a \times l \times n$ W (for a single-acting I.C. engine)

$$\text{or} = \frac{p_m \times a \times l \times n}{1{,}000}\ \text{kW} \quad ..\ (10.3a)$$

In the case of an engine governed by varying the fuel supply (i.e. explosion occuring every cycle), the number of working strokes or working cycles or explosions per second (n) will be taken equal to half the number of revolutions per seconds for a single-acting, four-stroke cycle engine. For a single-acting, two-stroke cycle engine, the number of explosions or working stroke or working cycles per second (n) will be equal to the number of revolutions per second.

Thus, for a single-acting four-stroke-cycle engine, number of working strokes or working cycles,

$n = \dfrac{N}{2}$, and for a double-acting four-stroke cycle engine, $n = N$, and

For a single-acting two-stroke cycle engine, $n = N$, and

for a double-acting, two-stroke cycle engine $n = 2N$

where N = r.p.s., and n = no. of working strokes per sec.

For double-acting engines, the work done on the other side (crank end side) of the piston must be added. While calculating the area of the piston, allowance must be made for the area of the piston rod which reduces the effective area of the piston on the crank end side.

In the case of an engine governed by hit and miss method, the explosions per second (n) must be counted during the test. This may be done by listening the sound of explosiona and counting the explosions or firing strokes per second.

Thus, for a single-acting, four-stroke cycle engine, governed by hit and miss method, number of firing strokes, $n = \frac{N}{2}$ *minus* number of missed cycles per sec.

As in steam engine, the indicated power of a multi-cylinder internal combustion engine will be the sum of the indicated power developed in the individual cylinder,

i.e. total indicated power = number of cylinders × indicated power of one cylinder.

10.14.2 Pumping Power : Internal combustion engines lose a small amount of work due to exhausting of burnt gases from the cylinder and the admission of fresh charge into the cylinder. The indicator diagram of a diesel four-stroke cycle engine (fig. 10-21) consists of two enclosed areas. The large area (+ ve area) represents the *gross work done.* The smaller (–ve area) shaded area, formed by the suction and exhaust operations is called pumping loop and represents the loss of work due to exhausting of burnt gases and admission of new unburnt gas (charge). This negative work is to be deducted from the gross work (obtained from the + ve area) to obtain the net work done. The negative area or pumping loop area is too small to be measured on ordinary indicator diagram. In practice a weak spring is used in the indicator to obtain the indicator diagram for obtaining the pumping m.e.p. The pumping loop is shown magnified in fig. 10-21 for explanation purpose.

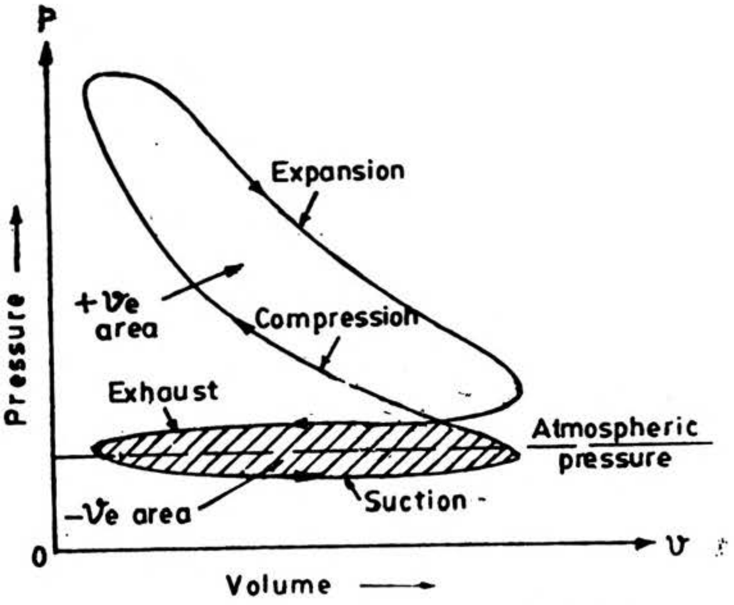


Fig. 10-21. Actural indicator diagram of a Diesel, four-stroke cycle engine.

$$\begin{bmatrix}\text{Mean height of the} \\ \textit{Pumping}\text{ loop, } h \text{ in mm}\end{bmatrix} = \left[\frac{\text{area of pumping loop of the indicator diagram in mm}^2}{\text{base length of the indicator diagram in mm}}\right]$$

$$\text{Pumping m.e.p. in kN/m}^2 = \begin{bmatrix} h\text{ (mm)} \times \text{ spring scale (kN/m}^2\text{ per mm)} \\ \text{of the weak spring used in the indicator}\end{bmatrix}$$

To obtain the net m.e.p., the pumping m.e.p. is to be deducted from the gross m.e.p.

i.e. Net m.e.p. = gross m.e.p. – pumping m.e.p, .. (10.3b)

(See illustrative Problem -2)

Let pumping m.e.p. = pumping mean effective pressure in N/m^2,

a = area of the piston in m^2,

l = length of piston stroke in metre, and

n = number of working strokes per second.

Then Pumping power = pumping m.e.p. × a × l × n W .. (10.3c)

In two-stroke cycle engines, the pumping power is the power lost in driving the air compressor or in compressing the charge in the crank-case which is designed as a compressor.

10.14.3 Brake Power : The manufactures of internal combustion engines generally work on brake power basis, on account of the difficulties experienced in accurately measuring the indicated power. The accuracy of indicator diagram may be affected by high speed, high temperature and high pressure of the engine. Special indicators such as optical and diaphragm types are generally used to obtain indicator diagrams when the speed exceeds 500 r.p.m. Devices for measuring the brake power of an engine are of two kinds :

- — Absorption dynamometers-those absorbing the power by friction and dissipating it as heat (used for low and medium speed engines).
- — Transmission dynamometers-those transmitting or passing on the power they measure and only a very small portion is wasted (used for high speed engines).

One of the simplest form of *absorption dynamometer* is the *rope brake* explained in chapter 9. *The brake power of the engine is determined in exactly the same way as for steam engine explained in chapter 9.*

Brake power = $(W - S)\ R \times 2\pi \times N$ W

$$= \frac{(W-S)\ R \times 2\pi \times N}{1{,}000} \text{ kW} \qquad \text{.. (10.4a)}$$

or Brake power = $(W - S) \times (2\pi R) \times N$ W .. (10.4b)

or Brake power = $T \times 2\pi \times N$ W .. (10.4c)

where $(W - S)$ = *net load* on the brake wheel in N,

R = *effective radius* of the brake wheel in m

$$= \frac{D+d}{2} \text{ in m}$$

where D = diameter of the brake wheel, and d = diameter of the brake rope,

$2\pi R$ = *effective circumference* of the brake wheel in m,

$T = (W - S)\ R$ = braking torque or resisting torque in N–m,

N = number of revolutions of crankshaft per second (r.p.s.), and

$2\pi N$ = number of radians per second.

One of the most convenient method of measuring the brake power of high speed engines (by means of transmitting the power), is to connect an electric dynamometer or electric generator to the crankshaft of the engine. If the efficiency of the generator is known at the particular speed and output at which it is operated, an accurate method of measuring brake power becomes available.

Brake mean effective pressure (b.m.e.p.) : The brake power of an I.C. engine is frequently expressed as

Brake power = $b.m.e.p. \times a \times l \times n$ W

where $b.m.e.p$ = brake mean effective pressure in N/m^2,

a = area of the piston in m^2,

l = length of the piston stroke in m, and

n = number of working strokes per sec.

The mean effective pressure based on brake power is called the brake mean effective pressure. If the brake power of the engine is known, the *b.m.e.p.* can be calculated in the following manner :

$$b.m.e.p. = \frac{\text{Brake power in watts}}{a \times l \times n} \ \text{N/m}^2 \quad \text{.. (10.4d)}$$

10.14.4 Mechanically Efficiency : There is a certain amount of work needed to overcome the friction of the moving parts of the engine itself and therefore, the amount of work available at the shaft is less than that produced in the cylinder. In other words, mechanical efficiency measures the efficiency with which an engine converts the power produced in its cylinder into useful power available at the engine crankshaft.

Mechanical efficiency (η_m) may be defined as the ratio of useful power available at the engine crankshaft and power developed in the engine cylinder,

i.e. Mechanical efficiency, $\eta_m = \dfrac{\text{Brake power}}{\text{Indicated power}}$ .. (10.5)

The difference between the indicated power and brake power represents the loss of power due to the friction of the rotating and sliding parts of the engine, a loss called the friction power,

i.e. Friction power = Indicated power − Brake power .. (10.6)

10.14.5 Thermal Efficiency : No engine can convert all the heat energy supplied by fuel to it into work. The fraction which is converted, is the 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 of heat equivalent of power produced in the cylinder (indicated power) in unit time and heat supplied to the engine in unit time. *The unit of time must be the same for the heat equivalent of power produced and heat supplied to the engine. This is very important.*

∴ Indicated thermal efficiency,

$$\eta_i = \frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\text{Heat supplied to the engine in kJ/sec.}}$$

$$= \frac{\text{Indicated power in kW}}{m_f \times C.V.} \quad \text{.. (10.7)}$$

where m_f = fuel oil supplied in kg per sec.

= fuel oil supplied in litres per sec. × specific gravity of fuel oil, and

$C.V.$ = calorific value of fuel oil in kJ/kg.

In case of gas engine, indicated thermal efficiency,

$$\eta_i = \frac{\text{Indicated power in kW (kJ/sec)}}{V_g \times C.V}. \quad \text{.. (10.8)}$$

where V_g = volume of gas supplied per sec. in m^3, and

$C.V.$ = calorific value of gas in kJ per m^3.

Brake thermal efficiency is denoted by η_b and is defined as the ratio of heat equivalent of brake power in unit time and heat supplied to the engine in unit time.

$\therefore$ Brake thermal efficiency, $\eta_b = \dfrac{\text{Brake power in kW (kJ/sec.)}}{m_f \times C.V.}$ .. (10.9a)

where m_f = fuel oil supplied in kg per sec. and

$C.V.$ = calorific value of fuel oil in kJ/kg.

In case of gas engines, brake thermal efficiency

$$\eta_b = \frac{\text{Brake power in kW (kJ/sec)}}{V_g \times C.V.} \quad \text{.. (10.9b)}$$

where V_g = volume of gas supplied per sec.

Now, indicated thermal efficiency, $\eta_i = \dfrac{\text{Indicated power}}{m_f \times C.V.}$

and Mechanical efficiency, $\eta_m = \dfrac{\text{Brake power}}{\text{Indicated power}}$

$\therefore$ Indicated thermal efficiency × Mechanical efficiency

$$= \frac{\text{Indicated power}}{m_f \times C.V.} \times \frac{\text{Brake power}}{\text{Indicated power}} = \frac{\text{Brake power}}{m_f \times C.V.}$$

Now, Brake thermal efficiency $= \dfrac{\text{Brake power}}{m_f \times C.V.}$

$\therefore$ Brake thermal efficiency, $\eta_b = \eta_i \times \eta_m$ .. (10.10)

Brake thermal efficiency is also termed as *overall efficiency.*

Relative efficiency is designated by η_r and is defined as the ratio of indicated thermal efficiency and air standard efficiency or theoretical thermal efficiency,

i.e. $\eta_r = \dfrac{\text{Indicated thermal efficiency}}{\text{Air–standard efficiency}}$ .. (10.11)

Relative efficiency shows how close the actual engine comes to the theoretically possible performance.

Relative efficiency on the basis of brake thermal efficiency is defined as the ratio of brake thermal efficiency and air standard efficiency,

i.e. $\eta_r = \dfrac{\text{Brake thermal efficiency}}{\text{Air–standard efficiency or Ideal thermal efficiency}}$ .. (10.12)

For air standard efficiency and relative efficiency, please refer volume II.

Problem-1 : *In a test on a single-cylinder oil engine working on the four-stroke cycle, the following readings were taken :*

Indicated mean effective pressure	*.. 540 kPa (5.4 bar)*
Diameter of engine cylinder	*.. 30 cm*
Piston stroke length	*.. 45 cm*
Engine speed	*.. 4 r.p.s.*

Calculate the indicated power of the engine.

Using eqn. (10.3a), Indicated power = $p_m \times a \times l \times n$ W

where p_m = Indicated m.e.p. in pascals (or N/m^2) = 540×10^3 Pa

a = area of the piston in square metre = $\frac{\pi}{4}\left(\frac{30}{100}\right)^2$ m^2

l = length of piston stroke in metre = 45/100 metre

n = number of working cycles or working strokes per second = $\frac{r.p.s}{2} = \frac{4}{2}$

$$\left(n = \frac{r.p.s.}{2} \text{ as the engine is four-stroke cycle}\right)$$

On substituting the above values, we have,

$$\text{Indicated power} = (540 \times 10^3) \times \frac{\pi}{4}\left(\frac{30}{100}\right)^2 \times \frac{45}{100} \times \frac{4}{2}$$

$$= 34{,}330 \text{ W or } = \mathbf{34{\cdot}33 \text{ kW}}$$

If the indicated m.e.p. is taken as 5·4 bar,

$$\text{Indicated power} = (5{\cdot}4 \times 10^5) \times \frac{\pi}{4}\left(\frac{30}{100}\right)^2 \times \frac{45}{100} \times \frac{4}{2}$$

$$= 34{,}330 \text{ W or } = \mathbf{34{\cdot}33 \text{ kW}} \text{ (same as before)}$$

Note : 1 kPa = 1 kN/m^2 = 10^3 Pa, 1 bar = 10^5 Pa and 1 MPa = 10^6 Pa

Problem-2 : *Calculate the indicated mean effective pressure when the indicator diagrams taken on a four-stroke cycle oil engine gave the following results :*

(i) Area of the positive loop, 820 mm^2; base length of the diagram, 50 mm; main spring used in the indicator had a stiffness of 50 kN/m^2 per 1 mm elongation.

(ii) Area of the negative loop 1,470 mm^2 ; base length of the diagram, 50 mm; spring used in the indicator had stiffness of 2 kN/m^2 per 1 mm elongation.

Using eqn. (10.2), indicated mean effective pressure,

$$p_m = \frac{\text{Area of the indicator diagram}}{\text{Base length of indicator diagram}} \times \text{Spring scale}$$

$$\therefore \text{ Gross m.e.p.} = \frac{820}{50} \times 50 = 820 \text{ kN/m}^2 \text{ and}$$

$$\text{Pumping m.e.p.} = \frac{1{,}470}{50} \times 2 = 58{\cdot}8 \text{ kN/m}^2.$$

Referring to fig. (10.21) and using eqn. (10.3b),

Net indicated m.e.p., p_m = gross m.e.p. – pumping m.e.p. = 820 – 58·8 = **761·2 kN/m^2.**

Problem-3 : *In a test on a single-cylinder oil engine working on the four-stroke cycle the following readings were taken :*

Indicated m.e.p., 755 kPa (0·755 MPa) ; cylinder diameter, 10 cm; piston stroke, 15 cm; engine speed, 8 r.p.s.; brake wheel diameter, 62·5 cm; net load on the brake wheel, 170 newtons. Calculate : (a) the indicated power, (b) the brake power, and (c) the mechanical efficiency of the engine.

$$\text{(a) Indicated power} = p_m \times a \times l \times n = (755 \times 10^3) \times \frac{\pi}{4}\left(\frac{10}{100}\right)^2 \times \frac{15}{100} \times \frac{8}{2}$$

$$= 3{,}560 \text{ W or } = \mathbf{3{\cdot}56 \text{ kW}}$$

If the indicated m.e.p. is taken as 0·775 MPa,

$$\text{Indicated power} = (0{\cdot}755 \times 10^6) \times \frac{\pi}{4}\left(\frac{10}{100}\right)^2 \times \frac{15}{100} \times \frac{8}{2}$$

$$= 3{,}560 \text{ W or } = \mathbf{3{\cdot}56 \text{ kW}} \text{ (same as before)}$$

(b) Using eqn. (10.4a),

Brake power = $(W - S) \times R \times 2\pi N$

where $(W - S)$ = net load on the brake wheel in newtons = 170 N

R = effective radius of the brake wheel in metre = $\frac{62{\cdot}5}{200}$ m

$2\pi N$ = number of radians per second = $2\pi \times 8$

N = number of revolutions per second = 8

On substituting the above values,

$$\text{Brake power} = 170 \times \frac{62{\cdot}5}{200} \times 2\pi \times 8 = 2{,}699 \text{ W or } = \mathbf{2{\cdot}669 \text{ kW}}$$

(c) Using eqn. (10.5),

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{2{\cdot}669}{3{\cdot}56} = 0{\cdot}75 \text{ or } \mathbf{75\%}$$

Problem-4 : *The following data refer to a test on a four-stroke cycle, double-acting single-cylinder Diesel engine having cylinder diameter 20 cm and piston stroke 30 cm :*

M.E.P. on cover side	*.. 590 kPa*
M.E.P. on crank side	*.. 645 kPa*
Speed	*.. 7 r.p.s.*
Diameter of piston rod	*.. 2.2 cm*
Dead load on the brake	*.. 1,324 N*
Spring balance reading	*.. 131 N*
Brake wheel diameter	*.. 1 m*
Brake rope diameter	*.. 2 cm*

Calculate the mechanical efficiency of the engine.

a_1 = area of the cylinder on the cover side = $\frac{\pi}{4} d^2 = \frac{\pi}{4}\left(\frac{20}{100}\right)^2$ m²

a_2 = effective area of the cylinder on crank end side

$$= \frac{\pi}{4}(d^2 - d_1^2) = \frac{\pi}{4}\left[\left(\frac{20}{100}\right)^2 - \left(\frac{2{\cdot}2}{100}\right)^2\right] \text{ m}^2$$

where d_1 = diameter of piston rod, and d = cylinder diameter

Indicated power on the cover side = $p_m \times a_1 \times l \times n$

$$= (590 \times 10^3) \times \frac{\pi}{4}\left(\frac{20}{100}\right)^2 \times \frac{30}{100} \times \frac{7}{2}$$

$$= 19{,}450 \text{ W i.e. } 19{\cdot}45 \text{ kW}$$

Indicated power on the crank side $= p_m \times a_2 \times l \times n$

$$= (645 \times 10^3) \times \frac{\pi}{4}\left[\left(\frac{20}{100}\right)^2 - \left(\frac{2{\cdot}2}{100}\right)^2\right] \times \left(\frac{30}{100}\right) \times \frac{7}{2}$$

$$= 21{,}063 \text{ W or } = 21{\cdot}063 \text{ kW}$$

As the engine is double-acting, total indicated power

= Indicated power on cover side + Indicated power on crank side

$= 19{\cdot}45 + 21{\cdot}063 = 40{\cdot}513$ kW

Brake power $= (W - S)\ R \times 2\pi N$

where $(W - S)$ = net load on the brake wheel $= 1{,}324 - 131 = 1{,}193$ N,

R = effective brake wheel radius $= \frac{1}{2} + \frac{2}{200} = 0{\cdot}51$ m, and

N = revolution per second made by the engine = 7 r.p.s.

Brake power $= 1{,}193 \times 0{\cdot}51 \times 2\pi \times 7 = 26{,}760$ W or $26{\cdot}76$ kW

Mechanical efficiency $= \dfrac{\text{Brake power}}{\text{Indicated power}} = \dfrac{26{\cdot}76}{40{\cdot}513} = 0{\cdot}6605$ or **66·05%**

Problem-5 : *A four cylinder four-stroke cycle petrol engine is to be designed to give a brake power of 185 kW at 35 r.p.s. The stroke bore ratio is to be 1.5 to 1. Assuming a mechanical efficiency of 75 per cent and indicated mean effective pressure of 830 kPa, determine the required bore and stroke.*

Indicated power $= \dfrac{\text{Brake power}}{\text{Mechanical efficiency}} = \dfrac{185}{0{\cdot}75} = 246{\cdot}67$ kW i.e. 2,46,670 W

$\dfrac{\text{Indicated power}}{4} = p_m \times l \times a \times n$ (indicated power per cylinder)

i.e. $$\frac{2{,}46{,}670}{4} = (830 \times 10^3) \times \frac{\pi}{4}\left(\frac{d}{100}\right)^2 \times \frac{1{\cdot}5d}{100} \times \frac{35}{2}$$

$$\therefore\ d^3 = \frac{2{,}46{,}670 \times 4 \times 10^4 \times 10^2 \times 2}{4 \times 830 \times 10^3 \times 3{\cdot}14 \times 1{\cdot}5 \times 35} = 5{,}405$$

$\therefore$ Diameter of cylinder, $d = \sqrt[3]{5{,}405} = $ **17·54 cm**

Piston stroke, $r = 1{\cdot}5\ d = 1{\cdot}5 \times 17{\cdot}54 = $ **26·32 cm.**

Problem-6 : *The following readings were taken during the test on a single-cylinder four-stroke cycle oil engine :*

Cylinder diameter, 19 cm; stroke, 35 cm; gross mean effective pressure, 748 kPa; pumping mean effective pressure, 36 kPa; engine speed, 240 r.p.m. fuel used per hour 3·5 kg, calorific value of fuel oil, 46,000 kJ/kg. Determine : (i) the indicated mean effective pressure, (ii) the indicated power, and (iii) the indicated thermal efficiency of the engine.

(i) Gross *m.e.p.* = 748 kPa; Pumping *m.e.p.* = 36 kPa.

Using eqn. (10.3b),

Net indicated mean effective pressure. $p_m = 748 - 36 = 712$ kPa

(ii) Using eqn. 10.3(a),

$$\text{Indicated power} = p_m \times a \times l \times n = (712 \times 10^3) \times \frac{\pi}{4}\left(\frac{19}{100}\right)^2 \times \frac{35}{100} \times \frac{4}{2}$$

$$= 14{,}124 \text{ W or } \mathbf{14{\cdot}124 \text{ kW}}.$$

(iii) Here, Indicated power = 14·124 kW, W_f = 3·5 kg/hr and $C.V.$ = 46,000 kJ/kg.

Heat equivalent of indicated power = 14·124 kJ/sec.

$$\text{Heat supplied} = \frac{3{\cdot}5}{3{,}600} \times 46{,}000 = 44{\cdot}722 \text{ kJ/sec,}$$

$$\text{Indicated thermal efficiency, } \eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Heat supplied in kJ per sec.}}$$

$$= \frac{14{\cdot}124}{44{\cdot}722} = 0{\cdot}3158 \text{ or } \mathbf{31{\cdot}58\%}$$

Problem-7 : *In a test on a single-cylinder oil engine working on the four-stroke cycle and fitted with a simple rope brake, the following readings were taken :*

Brake wheel diameter, 60 cm; brake rope diameter, 2·5 cm; dead load on the brake, 200 newtons; spring balance readings, 30 newtons; speed, 8 r.p.s.; area of the indicator diagram, 420 mm^2; length of indicator diagram, 60 mm; spring used had a stiffness of 110 kN/m^2 per mm elongation; diameter of the cylinder, 10 cm; stroke, 15 cm; fuel used, 0·32 kg/kW/hour on brake power basis of calorific value 44,000 kJ/kg.

Calculate the brake power, indicated power, mechanical efficiency and indicated thermal efficiency of the engine.

$$\text{Effective radius of the brake wheel, } R = \frac{D+d}{2} = \frac{60+2.5}{2} = 31{\cdot}25 \text{ cm} = 0{\cdot}3125 \text{ m}$$

(where D = Diameter of brake wheel in m, and d = diameter of brake rope in m)

Using eqn. (10.4a), Brake power = $(W - S) \times 2\pi \times R \times N$

where $(W - S)$ = Net load on the brake wheel = 200 − 30 = 170 N,

N = r.p.s. = 8 and R = Effective radius of the brake wheel = 0·3125 m

Substituting the above values in eqn. (10.4a),

Brake power = 170 × 2π × 0·3125 × 8 = 2,670 W = **2·67 kW**

Using eqn. (10.2), indicated mean effective pressure,

$$p_m = \frac{\text{area of the indicator diagram}}{\text{length of the indicator diagram}} \times \text{ spring number}$$

$$= \frac{420}{60} \times 110 = 770 \text{ kPa}$$

Indicated power = $p_m \times a \times l \times n$ W

$$= (770 \times 10^3) \times \frac{\pi}{4}\left(\frac{10}{100}\right)^2 \times \frac{15}{100} \times \frac{8}{2} = 3{,}627 \text{ W or } = \mathbf{3{\cdot}627 \text{ kW}}$$

Using eqn. (10.5), Mechanical efficiency,

$$\eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{2{\cdot}67}{3{\cdot}627} = 0{\cdot}7316 \text{ or } \mathbf{73{\cdot}16\%}$$

Fuel consumption per hour = 2·67 × 0·32 = 0·854 kg/hr.

Using eqn. (10.7), Indicated thermal efficiency,

$$\eta_i = \frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\text{Heat supplied to engine in kJ/sec.}}$$

$$= \frac{3{\cdot}627}{\frac{0{\cdot}854}{3{,}600} \times 44{,}000} = \frac{3{\cdot}627}{10{\cdot}437} = 0{\cdot}3475 \text{ or } \mathbf{34{\cdot}75\%}$$

Problem-8 : *The following data and test results refer to a test on a single-cylinder two-stroke cycle oil engine :*

Indicated m.e.p., 550 kPa; cylinder diameter, 21 cm; piston stroke, 28 cm; engine speed, 6 r.p.s. brake torque, 628 N.m; oil consumption, 8·16 kg/hr; calorific value of oil, 42,700 kJ/kg. Calculate : (a) the mechanical efficiency, (b) the indicated thermal efficiency, (c) the brake thermal efficiency and (d) brake specific fuel consumption in kg/kW/hr.

(a) Indicated power = $p_m \times a \times l \times N$

$$= (550 \times 10^3) \times \frac{\pi}{4}\left(\frac{21}{100}\right)^2 \times \frac{28}{100} \times 6 = 32{,}000 \text{ W or } \mathbf{32\ kW}$$

Brake power = $(W - S)\ R \times 2\pi N = T \times 2\pi N$ W

where $T = (W - S)\ R$ = Brake torque = 628 N.m,

N = r.p.s. = 6, and

$2\pi N$ = number of radians per second = $2\pi \times 6$.

Substituting the above values, we have,

Brake power = 628 × 2 × 3·14 × 6 = 23,663 W or 23·663 kW

$$\text{Mechanical efficiency} = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{23{\cdot}663}{32} = 0{\cdot}7398 \text{ or } \mathbf{73{\cdot}98\%}$$

$$\text{(b) Indicated thermal efficiency} = \frac{\text{Heat equivalent of indicated power in kJ/sec.}}{\text{Heat supplied to the engine in kJ/sec.}}$$

Heat equivalent of indicated power = 32 kJ/sec. (∵ 1 kW = 1 kJ/sec.)

$$\text{Heat supplied/sec.} = \frac{8{\cdot}16}{3{,}600} \times 42{,}700 \text{ kJ/sec.}$$

$$\text{Indicated thermal efficiency} = \frac{32}{\frac{8{\cdot}16}{3{,}600} \times 42{,}700} \times 100 = \mathbf{33{\cdot}06\%}$$

$$\text{(c) Brake thermal efficiency} = \frac{23{\cdot}663}{\frac{8{\cdot}16}{3{,}600} \times 42{,}700} \times 100 = \mathbf{24{\cdot}4\%}$$

$$\text{(d) Brake specific fuel consumption (B.S.F.C.)} = \frac{\text{Fuel consumption in kg per hour}}{\text{Brake power in kW}}$$

$$= \frac{8{\cdot}16}{23{\cdot}663} = \mathbf{0{\cdot}3449\ kg/kW/hr.}$$

Problem-9 : *The following data was obtained from a trial of an single-cylinder gas engine working on the four-stroke cycle and governed by hit and miss method of governing; Diameter of cylinder, 18 cm; stroke, 38 cm; effective brake radius, 74 cm; indicated mean effective pressure, 440 kPa; average revolutions per second, 3; average explosions (working cycles) per minute, 82; net load on the brake, 235 newtons; gas used per hour, 4·2 m^3; and calorific value of gas, 15,490 kJ/m^3. Calculate : (a) the indicated power, (b) the brake power, (c) the indicated thermal efficiency and (d) gas consumption in m^3 per kW per hour based on brake power.*

(a) Indicated power = $p_m \times a \times l \times n$ (where n = no. of explosions per sec.)

$$= (440 \times 10^3) \times \frac{\pi}{4}\left(\frac{18}{100}\right)^2 \times \frac{38}{100} \times \frac{82}{60}$$

$$= 5{,}810 \text{ W or } = \mathbf{5{\cdot}81 \text{ kW}}$$

(b) Brake power = $(W - S)\ R \times 2\pi N = 235 \times \frac{74}{100} \times 2 \times 3{\cdot}14 \times 3$

$$\risingdotseq 3{,}277 \text{ W or } \mathbf{3{\cdot}277 \text{ kW}}$$

(c) Indicated thermal efficiency, $= \dfrac{5{\cdot}81}{\dfrac{4{\cdot}2}{3{,}600} \times 15{,}490} = 0{\cdot}3215$ or **32·15%**

(d) Gas consumption in m^3 per kW Hr. on brake power basis

$$= \frac{4{\cdot}2}{3{\cdot}277} = \mathbf{1{\cdot}28\ m^3/kW\ Hr.}$$

Problem-10 : *The following data and test results refer to a four-stroke cycle oil engine :*

Number of cylinders = 4; bore = 10 cm; stroke = 15 cm; indicated mean effective pressure = 670 kPa; speed = 34 r.p.s.; brake-torque = 185 N.m; oil consumption = 12·89 kg/hr; calorific value of fuel oil = 42,000 kJ/kg.

Calculate : (a) the brake power, (b) the brake thermal efficiency, (c) the specific fuel consumption on brake power basis (B.S.F.C.), (d) the indicated power, (e) the mechanical efficiency of the engine, and (f) the brake mean effective pressure.

(a) Using eqn. (10.4c), Brake power = $T \times 2\pi \times N$

where T = Brake-torque = 185 N.m and N = *r.p.s.* = 34

$\therefore$ Brake power = $185 \times 2\pi \times 34$ = 39,520 W or **39·52 kW**

(b) Using eqn. (10.9a), Brake thermal efficiency,

$$\eta_b = \frac{\text{Heat equivalent of brake power in kJ/sec.}}{\text{Heat supplied to engine in kJ/sec.}}$$

$$= \frac{39{\cdot}52}{\dfrac{12{\cdot}89}{3{,}600} \times 42{,}000} = \frac{39{\cdot}52}{150{\cdot}38} = 0{\cdot}2628 \text{ or } \mathbf{26{\cdot}28\%}$$

(c) Specific fuel consumption on brake power basis or Brake specific fuel consumption,

$$\text{B.S.F.C.} = \frac{12{\cdot}89}{39{\cdot}52} = \mathbf{0{\cdot}326} \text{ kg per kW per hr.}$$

(d) Indicated power per cylinder $= p_m \times a \times l \times n$

$$= (670 \times 10^3)\ \frac{\pi}{4} \times \left(\frac{10}{100}\right)^2 \times \frac{15}{100} \times \frac{34}{2}$$

$$= 13{,}140 \text{ W or } 13{\cdot}41 \text{ kW}$$

$\therefore$ Total indicated power of four cylinders (engine) $= 13{\cdot}41 \times 4 =$ **53·64 kW**

(e) Mechanical efficiency, $\eta_m = \frac{\text{Brake power}}{\text{Indicated power}} = \frac{39{\cdot}52}{53{\cdot}64} = 0{\cdot}7367$ or **73·67%**

(f) Brake power per cylinder $= \frac{39{\cdot}52}{4} = 9{\cdot}88$ kW

Using eqn. (10.4d), Brake power $= b.m.e.p. \times a \times l \times n$

where $b.m.e.p.$ = brake mean effective pressure in kPa, a = area of the piston in m^2, l = length of piston stroke in m, and n = no. of working strokes per sec.

i.e. $9{\cdot}88 \times 10^3 = b.m.e.p. \times \frac{\pi}{4}\left(\frac{10}{100}\right)^2 \times \frac{15}{100} \times \frac{34}{2}$

$\therefore$ $b.m.e.p.$ per cylinder $= 4{,}94{,}000$ Pa or **494 kPa**

Alternatively Brake mean effective pressure ($b.m.e.p.$)

= Indicated mean effective pressure × Mechanical efficiency

$= 670 \times 0{\cdot}7367 = 494$ kPa (same as before)

Problem-11 : *A single-cylinder, four-stroke cycle oil engine is to develop 31 kW at 300 r.p.m. If the indicated mean effective pressure is 675 kPa and speed of the piston is 183 metres per minute, what is the diameter of cylinder and piston stroke ?*

If the indicated thermal efficiency of engine is 30 per cent and the calorifc value of the oil is 42,000 kJ/kg, what is the fuel consumption in kg per kW per hour on indicated power basis ?

Piston speed in m/min. $= 2lN$

[where l = length of piston stroke in m, and

N = number of revolutions of the engine/min. (r.p.m.)]

i.e. $183 = 2 \times l \times 300$

$\therefore$ Piston stroke, $l =$ **0·305 m**

Using eqn. (10-3a), Indicated power $= p_m \times a \times l \times n$

i.e. $(31 \times 10^3) = (675 \times 10^3) \times \frac{\pi}{4}\left(\frac{d}{100}\right)^2 \times 0{\cdot}305 \times \left(\frac{300}{60 \times 2}\right)$

$\therefore D^2 = 767{\cdot}27$

$\therefore$ Diameter of the cylinder, $d = \sqrt{767{\cdot}27} =$ **27·67 cm.**

Using eqn. (10.7) and taking indicated power as 1 (one),

Indicated thermal efficiency, $\eta_i = \frac{\text{Heat equivalent of indicated power in kJ per sec.}}{\text{Heat supplied in kJ per sec.}}$

$$\text{i.e. } 0{\cdot}3 = \frac{1}{\frac{M_f}{3{,}600} \times 42{,}000}$$

$\therefore$ Fuel consumption, $M_f = 0{\cdot}2857$ kg/kW/hr.

i.e. Fuel consumption per kW per hour on indicated power basis = 0·2857 kg.

Tutorial - 10

1. Fill in the gaps to complete the following statements :

   (a) All large size I.C. engines and marine engines are ....

   (b) The development of the Diesel engine began about ....

   (c) In case of Otto cycle engine, the mixture pressure at the end of compression stroke is about .... bar.

   (d) In Diesel cycle engine .... is compressed to about .... bar at the end of compression stroke.

   (e) Dual-combustion cycle engine is also known as .... engine.

   (f) In the petrol engine the fuel and air charge is fired by ....

   (g) Lubricating oil consumption in two-stroke cycle engines is relatively .... than that of four-stroke cycle engines.

   [(a) double-acting, (b) 1,893, (c) 7, (d) air, 35, (e) Semi-Diesel, (f) spark, (g) higher]

2. What is meant by "Internal combustion engine" ? Briefly describe the history of its development.

3. What is meant by cycle of operations of an internal combustion engine ? Explain briefly the cycle of operations of (i) a four-stroke cycle petrol engine, and (ii) a four-stroke cycle Diesel engine. Draw indicator diagram in each case.

4. Sketch a typical indicator diagram of a four-stroke cycle petrol engine and explain briefly each phase of the cycle.

5. Describe with the help of neat sketches a two-stroke cycle Diesel engine. Sketch the indicator and valve timing diagrams of such an engine.

6. Describe with the help of neat sketches the working of a two-stroke cycle petrol engine giving probable indicator and valve timing diagrams.

7. (a) Describe the cycle of operations of : (i) a two-stroke cycle petrol engine, and (ii) a two-stroke cycle Diesel engine.

   (b) What are the relative advantages and disadvantages of I.C. engines working on four-stroke and two-stroke cycles ?

8. (a) State the distinguishing features between four-stroke cycle and two-stroke cycle internal combustion engines.

   (b) Discuss the merits and demerits of four-stroke cycle versus two-stroke cycle internal combustion engines.

9. What is meant by cycle of operations in an internal combustion engine ? Explain clearly with the help of neat sketches the working of four-stroke cycle and two-stroke cycle Diesel engines. Why two-stroke cycle is specially suitable for Diesel engines ?

10. (a) Describe with a neat sketch, the ignition system used in a four-cylinder four-stroke cycle petrol engine.

    (b) Describe with a neat sketch, the working of a fuel pump for a Diesel cycle engine.

11. (a) Enumerate the essential functions of a carburettor. Explain briefly, giving suitable sketches, the working of a simple carburettor.

    (b) Sketch and explain the working of a fuel injector for a Diesel cycle engine.

12. Explain giving reasons why

    (i) Internal combustion engine cylinders need cooling.

    (ii) Internal combustion engines need cranking to start them.

    (iii) Otto cycle engines need some method of ignition but Diesel cycle engines generally do not need ignition devices.

    (iv) The camshaft speed is half that of the crankshaft in four-stroke cycle-engines.

13. (a) Distinguish between a water cooled I.C. engine and an air cooled I.C. engine.

    (b) Describe an experimental method for determining brake power of an I.C. engine.

14. (a) Discuss the field of applications of I.C. engines.

    (b) What is the difference between naturally aspirated I.C. engine and supercharged I.C. Engine.

15. (a) State the factors according to which I.C. engines can be classified.

    (b) Explain the basic difference between a petrol engine and a Diesel engine.

16. Answer the following briefly :

    (i) Explain the term clearance volume.

    (ii) What is meant by swept volume or capacity of a cylinder ?

    (iii) What are the usual compression ratios for petrol engines ?

    (iv) What are the usual compression ratios for Diesel engines ?

    (v) How is the cylinder scavenged of burnt gases in case of a two-stroke cycle I.C. engine and in case of a four-stroke cycle I.C. engine ?

    (vi) What are the functions of flywheel and governor in the operation of I.C. engines ?

17. What is meant by governing of an I.C. engine ? What are the methods of governing I.C. engines ?

18. What is meant by indicator, Indicator card, indicator diagram, Indicated mean effective pressure, Spring number, and Indicated power of an I.C. engine ?

19. Explain the following terms as applied to I.C. engines :

    (i) Pumping power,

    (ii) Indicated power,

    (iii) Brake power, and

    (iv) Friction power.

20. Explain the following terms as applied to I.C. engines :

    (i) Mechanical efficiency,

    (ii) Indicated thermal efficiency,

    (iii) Brake thermal efficiency, and

    (iv) Relative efficiency.

21. State in the following cases whether the engine is a petrol engine or a Diesel engine: (i) Fuel injector is used, (ii) Spark plug is used, (iii) It is used in motor car engine, (iv) Compression ratio is 14, (v) combustion takes place at constant volume, (vi) Pressure in the cylinder at the end of compression is 35 bar, (vii) Carburettor is used, and (viii) Mixture of fuel and air is taken during the suction stroke.

    [(i) Diesel, (ii) Petrol, (iii) Petrol, (iv) Diesel, (v) Petrol, (vi) Diesel, (vii) Petrol, (viii) Petrol]

22. Write a short notes on the following giving neat sketches wherever necessary :

    (i) Methods of ignition used in I.C. engines,

    (ii) Methods of cooling I.C. engine cylinder,

    (iii) Methods of governing I.C. engines,

    (iv) Methods of fuel injection employed in Diesel engines, and

    (v) Simple carburettor.

23. The diameter of the brake wheel fitted to a single-cylinder, four-stroke cycle oil engine on test is 1.48 metres and the brake rope diameter is 5 cm. The dead load on the brake is 804 newtons and the spring balance reading is 88 newtons and the engine is running at 3 r.p.s. From indicator card, the indicated m.e.p. is 687 kPa. The fuel consumption is 0·31 kg per kW per hour on brake power basis (brake specific fuel consumption) of calorific value 42,000 kJ/kg. The cylinder diameter is 23 cm and the piston stroke is 30 cm. Calculate : (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, and (d) the indicated thermal efficiency of the engine.

    [(a) 10·324 kW; (b) 12·844 kW; (c) 80·38%; (d) 34·4%]

24. The following readings were taken during the test of a single-cylinder, four-stroke cycle oil engine :

Cylinder diameter	.. 280 mm
Piston stroke	.. 425 mm
Gross m.e.p.	.. 724 kPa
Pumping m.e.p.	.. 40 kPa
Engine speed	.. 210 r.p.m.
Net load on the brake wheel	.. 1,300 newtons
Effective diameter of the brake wheel	.. 1·6 metres

Fuel used per hour	.. 9 kg
Calorific value of fuel oil	.. 42,000 kJ/kg

Calculate the indicated power, brake power, mechanical efficiency, and indicated and brake thermal efficiencies of the engine.

[31·325 kW ; 22·87 kw; 73% 29·83%; 21·78%]

25. The following results were obtained during the test on a single-cylinder oil engine :

Duration of test 50 minutes; fuel oil used, 2·633 kg; average indicated power 13·32 kW; average brake power, 10·48 kW; calorific value of fuel oil, 44,400 kJ/kg. Calculate : (a) the indicated thermal efficiency, and (b) the brake thermal efficiency of the engine. [(a) 34·18%; (b) 26·89%]

26. The following results were obtained from a test on a single-cylinder oil engine : Brake specific fuel consumption, 0·31 kg/kW/hr; Calorific value of fuel oil, 43,960 kJ/kg; Brake power, 26·1 kW; Indicated power, 32·68 kW. Calculate : (a) the mechanical efficiency, (b) the indicated thermal efficiency, and (c) the brake thermal efficiency of the engine.

[(a) 79·86%; (b) 33·08%; (c) 26·42%]

27. The following observations were made during a trial on a single-cylinder two-stroke cycle oil engine having a cylinder diameter 20 cm and piston stroke 30 cm :

Duration of trial, 40 minutes; total number of revolutions, 12,000; indicated m.e.p., 580 kPa; net load on the brake, 1,240 newtons; effective radius of the brake wheel, 0·58 metre; fuel consumption, 4·32 kg; calorific value of fuel oil, 44,000 kJ/kg. Calculate : (a) the mechanical efficiency, (b) the indicated thermal efficiency, and (c) the brake thermal efficiency of the engine.

[(a) 82·65%; (b) 34·48%; (c) 28·51%]

28. The following observations were made during a test on a single-cylinder, two-stroke cycle oil engine : Cylinder dimensions : 20 cm bore (diameter), 26 cm stroke; speed, 6 r.p.s.; effective diameter of the brake wheel, 1.2 metres; net load on the brake wheel, 471 newtons; indicated mean effective pressure, 275 kPa; fuel oil consumption, 3·36 kg per hour; calorific value of the fuel oil, 41,870 kJ/kg. Calculate : (a) the indicated power, (b) the brake power, (c) the mechanical efficiency, (d) the indicated thermal efficiency, and (e) fuel consumption in kg per kW per hour on brake power basis.

[(a) 13·47 kW; (b) 10·648 kW; (c) 79%; (d) 34·47%; (e) 0·3155 kg/kW per hour]

29. A four-cylinder, four-stroke cycle petrol engine is to be designed to develop a brake power of 185 kW at 2,100 r.p.m. The stroke bore ratio is to be 1·5 to 1. Assuming mechanical efficiency of 75 per cent and indicated mean effective pressure of 830 kPa, determine the required bore and stroke.

If the indicated thermal efficiency of the engine is 31·5% and the calorific value of petrol is 42,000 kJ/kg calculate the petrol consumption in litres per hour and brake specific petrol consumption in litre per kW per hour. Take specific gravity of petrol as 0·73.

[15·332 cm; 23 cm; 91·947 litres/hr; 0·497 litre per kW/hour]

30. The following data refer to a test on a four-stroke cycle oil engine :

Number of cylinders = 4; cylinder bore = 9 cm; piston stroke = 13 cm; speed = 2,100 r.p.m.; brake torque = 167 N.m; fuel oil consumption = 13·5 litres/hr.; calorific value of fuel oil = 44,000 kJ/kg; specific gravity of fuel oil = 0·82. Calculate : (a) the brake mean effective pressure (b.m.e.p.), (b) the brake thermal efficiency, and (c) the brake specific fuel consumption in litre per kW/hr.

[(a) 634·4 kPa; (b) 27·14%; (c) 0·3676 litre/kW.hr.]

31. During a test of 40 minutes duration on a single-cylinder gas engine of 22 cm cylinder bore and 43 cm stroke, working on four-stroke cycle and governed by hit and miss method of governing, the following readings were taken :

Total number of revolutions, 9,600; total number of explosions, 4,400; indicated mean effective pressure, 667 kPa; net load on the brake wheel, 805 newtons; brake wheel diameter; 1·5 metres; brake rope diameter, 2·5 cm; calorific value of gas, 16,750 kJ/m^3; gas used, 9·2 m^3. Calculate : (a) the indicated power, (b) the brake power, (c) the indicated thermal efficiency, and (d) the brake thermal efficiency of the engine.

[(a) 19·98 kW; (b) 15·418 kW; (c) 31·11%; (d) 24·01%]

32. The following data was obtained from a trial of single-cylinder gas engine working on the four-stroke cycle and governed by hit and miss method of governing :

Diameter of the cylinder, 18 cm; stroke, 28 cm; effective brake wheel radius, 74 cm; average indicated mean effective pressure, 410 kPa; average revolutions per minute. 195; average explosions (or working strokes) per minute, 78; net load on the brake, 235 newtons; gas used per hour, 4m^3; calorific value of gas, 15,000 kJ/m^3. Calculate :

(a) Brake specific gas consumption in m^3/kW/hour,

(a) Brake specific gas consumption in m^3/kW/hour,

(b) Gas consumption in m^3/kW/hour on indicated power basis; and

(c) The indicated thermal efficiency of the engine.

[(a) 1·1264 m^3; (b) 0·7764 m^3; (c) 30·91%]

33. A single-cylinder, four-stroke cycle gas engine, having a bore and stroke of 37 cm and 70 cm respectively and running at 200 r.p.m., fires 75 times per min. The mean area of the indicator diagram is 480 mm^2 and the length of the indicator diagram is 75 mm. The strength of the spring used in the indicator is 100 kN/m^2/mm and the calorific value of the gas is 17,000 kJ per m^3. Determine the indicated power and the gas consumption in m^3 per kW per hour on indicated power basis, if the indicated thermal efficiency of the engine is 31%.

[60·212 kW; 0·6831]

11

AIR COMPRESSORS

11.1 Introduction

Air compressors are used for supplying high-pressure air. There are many uses of high-pressure air in the industry. The main uses of high-pressure (compressed) air are :

.. to drive compressed air engines (air motors) used in coal mines,

.. to inject or spray fuel into the cylinder of a Diesel engine (air injection Diesel engine),

.. to operate drills, hammers, air brakes for locomotives and railway carriages, water pumps and paint sprays,

.. to start large (heavy) Diesel engines,

.. to clean workshop machines, generators, automobile vehicles, etc.,

.. to operate blast furnaces, gas turbine plants, Bessemer convertors used in steel plants, etc.,

.. to cool large buildings and air crafts, and

.. to supercharge I.C. engines.

There are mainly two types of air compressors viz. *reciprocating air compressors and rotary air compressors.* Reciprocating air compressors are similar to reciprocating engines where a piston reciprocates inside a cylinder. In rotary air compressors, air is compressed due to rotation of impeller or blades inside a casing. Air compressors are driven by engines or electric motors. In this chapter the theory of reciprocating air compressors is discussed in details and principles of working of reciprocating compressed air motors and rotary compressors are explained in brief.

11.2 Reciprocating Air Compressors

The principal parts of a reciprocating air compressor are the same as that for a engine. The reciprocating air compressor may be *single-acting* (air is admitted to one side of the piston only) or *double-acting* (air is admitted to each side of the piston alternatively), and may be *single-stage* or *multi-stage.* In a multi-stage compressor, the air is compressed in several stages instead of compressing the air fully in a single cylinder. This is equivalent to a number of compressors arranged in series. The pressure of air is increased in each stage. Single-stage compressors are used for delivery pressures upto 10 bar, three-stage compressors for pressure upto 200 bar and two-stage compressors for pressures in between 10 to 200 bar. The average piston speed of a reciprocating air compressor is limited to about 300 to 400 metres per minute to reduce friction wear.

11.3 Single-stage Air Compressor

The sectional view of an air cooled, single-stage, single-acting reciprocating air compressor is shown in fig. 11-1. Both intake (suction) and discharge (delivery) valves

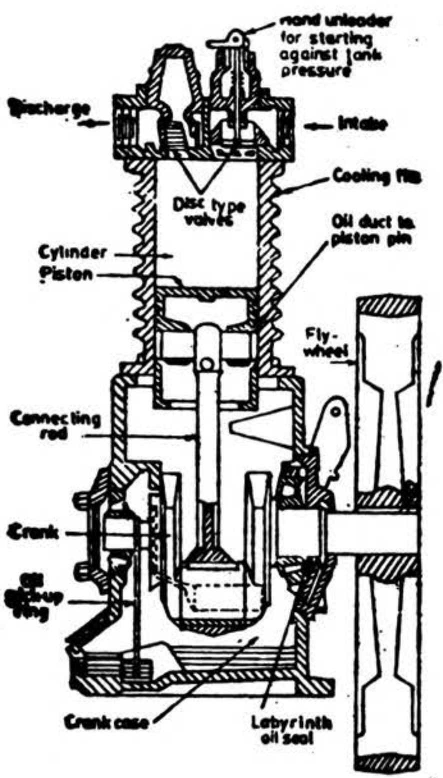


Fig. 11-1. Sectional view of a single-acting air cooled, reciprocating air compressor.

are disc type and are automatic in their action. They are opened and closed by difference in the air pressure acting on their two sides. When the pressures are equal on their two sides, they are kept closed by light springs. During the outward or *suction stroke*, the pressure in the cylinder falls below the atmospheric pressure as a result of which the intake valve opens and air is drawn from the atmosphere into the cylinder. During the inward or *compression stroke*, as a result of the piston action the pressure of the air in the cylinder gradually increases and reaches a value sufficiently above the receiver pressure. The high pressure of air, thus produced, overcomes the resistance of the spring on the discharge valve and causes the valve to open and discharge takes place from the cylinder to the receiver. The receiver is a simple vessel which acts as a storage tank. The compressor is driven by some form of prime mover (electric motor or engine). When the compressor is to be started against tank (receiver) pressure, the prime mover will have to supply very high starting torque. To avoid this, hand unloader (fig. 11-1) is used for releasing pressure from the compressor cylinder when the compressor is stopped.

11.3.1 Indicator Diagram : The events described above can be conveniently represented by *p–v* diagram shown in fig. 11-2. The diagram is drawn for a compressor without clearance. During the suction stroke the charge of air is drawn into the cylinder along line 4-1 at constant pressure p_1, which is slightly below that of the atmosphere. At point 1, the piston completes the suction (outward) stroke and starts on its return (compression) stroke. All valves being closed, the air is now compressed along the compression curve 1-2. At point 2, pressure p_2 is reached which is slightly higher than the pressure in the receiver. The discharge valve at this point opens and the delivery of the compressed air takes place along line 2-3 at pressure p_2. The piston has now reached the left hand end of the cylinder and again starts on its suction stroke and the pressure in the cylinder will be lowered again to p_1 and the cycle of operations will be repeated. The net work required for compression and delivery of the air per cycle is represented by the area 1-2-3-4 (fig. 11-2).

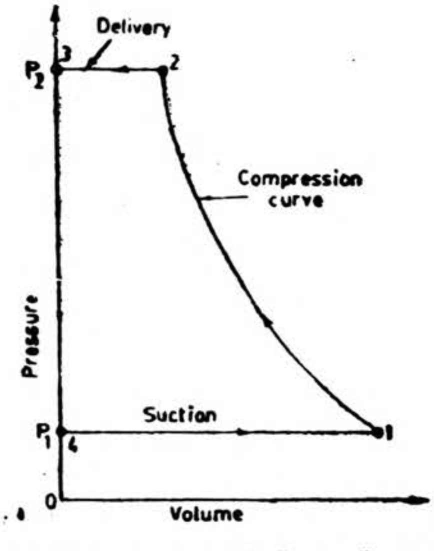


Fig. 11-2. Theoretical indicator diagram of a single-stage air compressor.

The amount of work done on the air will depend upon the nature of the compression curve. If the compression occurs very rapidly in a non-conducting cylinder so that there is no heat transfer, the compression will be practically *isentropic*. If it is carried out slowly so that the heat of the compression is extracted from the air by the jacket cooling water, the compression will approach *isothermal*. However, in actual practice neither of these conditions can be fulfilled and the actual compression will be between isentropic and isothermal.

11.3.2 Isothermal Compression Versus Isentropic Compression : The slope of the compression curve, represented by the law $pv^n = C$, depends upon the value of the index n. A large value of n will give comparatively a steeper curve. The law for an isothermal or hyperbolic compression is $pv = C$, where the value of index n is unity. The law for an isentropic compression is $pv^\gamma = C$. Since the value of γ for air is 1.4, the isentropic curve will be steeper than isothermal curve. Figure 11-3 shows curves representing an isentropic compression (1-2') and an isothermal compression (1-2"). The middle curve (1-2) shows curve, which is obtained in actual practice. The curve is polytropic ($pv^n = C$) having a value of n nearly equal to 1·3 for the water cooled cylinder.

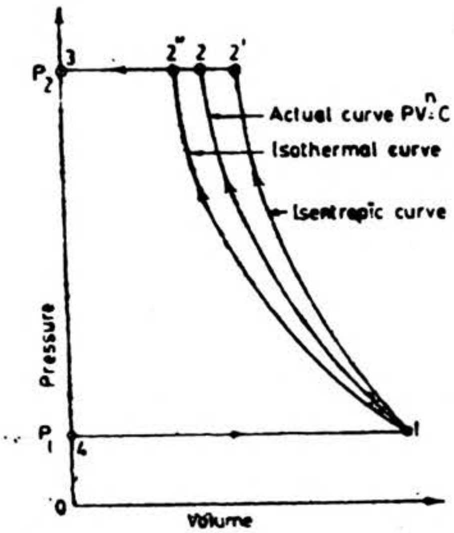


Fig. 11-3. Effect of nature of compression curve on work required for compression.

The isentropic work required to be done per cycle on the air is represented by the area 4-1-2'-3 (fig. 11-3). If the compression carried out had been isothermal, the slope of the compression curve would be less than that of isentropic and the isothermal work done would be represented by the area 4-1-2"-3 which is evidently less than the isentropic work done represented by the area 4-1-2'-3. *Therefore, it follows that an isothermal compression is economical and efficient, since less work is required to carry it out,* while an isentropic compression requires more amount of work to be supplied. Compression curve with values of index n between 1 and 1.4 will fall within the isothermal and isentropic curves. Thus, it will be seen that the work required for compression and delivery of air per cycle decreases as the value of n decreases.

The theoretical indicator diagram for a single-stage compressor without clearance is shown in fig. 11-4. Let p_1 in N/m^2 (newtons per square metre) or Pa (pascals) and v_1 in m^3 represent initial condition of the air before compression, and p_2 in N/m^2 = Pa, the final delivery pressure after compression, Then,

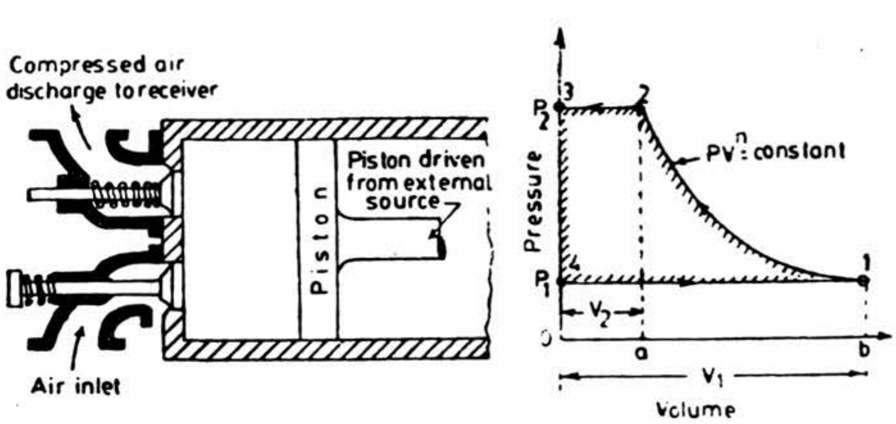


Fig. 11-4. Single-stage air compressor without clearance.

(a) Work required to be done on the air W, *per cycle* assuming compression curve to be polytropic,i.e., $pv^n = C$, is given by area 1-2-3-4 of fig. 11-4.

Now area, 1-2-3-4 = area 0-a-2-3 plus area *a*-2-1-*b* *minus* area *b*-1-4-0

$$= p_2V_2 + \frac{p_2v_2 - p_1v_1}{n-1} - p_1v_1$$

$$= \frac{n\,p_2v_2 - p_2v_2 + p_2v_2 - p_1v_1 - n\,p_1v_1 + p_1v_1}{n-1}$$

$$\therefore\ W = \frac{n}{n-1}\,(p_2v_2 - p_1v_1) \text{ Joule per cycle} \qquad \text{.. (11.1a)}$$

$$\text{or } W = \frac{n}{n-1}\,mR(T_2 - T_1) \text{ Joule per cycle} \qquad \text{.. (11.1b)}$$

From eqn. (11.1a) taking p_1v_1 outside the bracket,

Work required, $W = \frac{n}{n-1}\,p_1v_1\left[\frac{p_2v_2}{p_1v_1} - 1\right]$

But for polytropic compression, $p_1v_1^n = p_2v_2^n$. Hence, $\frac{v_2}{v_1} = \left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}$

Substituting the value of $\frac{v_2}{v_1}$ in the above equation,

Work required per cycle (or per revolution, if compressor is single-acting),

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

$$= \frac{n}{n-1}\,p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ Joule per cycle} \qquad \text{.. (11.2)}$$

This equation gives the work required in Joules per cycle (or per revolution, if the compressor is single-acting) in compressing and delivering the air.

It should be noted that units of pressure and volume in eqn. (11.2) are N/m^2 or Pa and m^3 respectively.

Indicated power of the compressor $= \frac{W \times N}{60}$ J/sec. or W ...(11.3)

where W = work required in Joules per cycle, and

N = No. of cycles performed per minute = r.p.m. for single-acting compressor.

if p_1v_1 in eqn. (11.2) is substituted by mRT_1, then work required per cycle,

$$W = \frac{n}{n-1}\,mRT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ Joule per cycle and}$$

Work required *per kg of air,*

$$W = \frac{n}{n-1}\,RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ Joule} \qquad \text{...(11.4)}$$

Indicated power of the compressor

= W × mass of air delivered per second J/sec. or W ... (11.5)

where W = work required in Joules per kg of air.

(b) If the compression is isentropic ($pv^{\gamma} = C$), the index n will be replaced by γ in eqn. (11.2) and eqn. (11.4), and then

Work required, $$W = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \text{ Joule per cycle} \quad \text{... (11.6a)}$$

Work required *per kg of air,*

$$W = \frac{\gamma}{\gamma - 1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \text{ Joule} \quad \text{... (11.6b)}$$

(c) If the compression is isothermal *($pv = C$),* then work required per cycle,

W = area 1-2-3-4 (fig. 11-4)

= area *a*-2-1-*b* *plus* area 0-*a*-2-3 *minus* area *b*-1-4-0

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

$$= p_1 v_1 \log_e \left(\frac{p_2}{p_1} \right) \text{ Joule per cycle} \quad \text{... (11.7a)}$$

Work required *per kg of air,* $W = RT_1 \log_e \left(\frac{p_2}{p_1} \right)$ Joule ... (11.7b)

11.3.3 Approximation of Isothermal Compression : Although isothermal compression is economical, it is not possible to achieve it in practice. To have an isothermal compression, the compressor will have to be run extremely slow, while in practice it is driven at high speed so that as much air as possible in compressed in a given time.

Since, there is saving of work by compressing air isothermally, it is necessary to make an attempt to obtain approximately an isothermal compression. Three methods are adopted to achieve this object while still running the compressor at high speed. The three methods adopted are :

- — cooling the air during compression by spraying cold water into the cylinder,
- — cooling the air during compression by circulating cold water through the cylinder jacket, and
- — adopting multi-stage compression with inter-stage cooling.

Cold water spray : In this method, cold water is sprayed into the cylinder during compression. The cooling, thus done, reduces the temperature of the air and the compression curve will be approximately of the form $pv^{1.2} = C$. This means that the compression is brought nearer to isothermal which results in the saving of work.

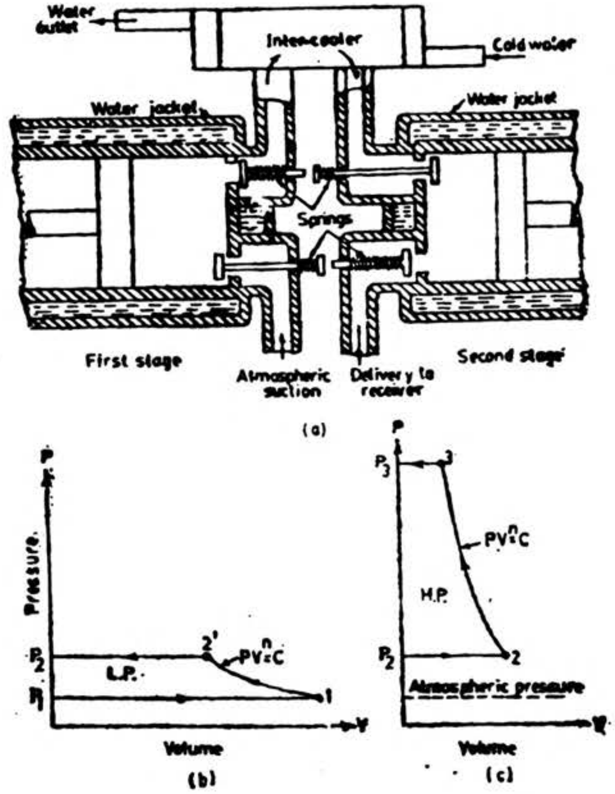
Water jacket : In this method, the heat of compression is extracted by circulating cold water in the cylinder jacket thereby keeping the temperature rise as small as possible. This keeps the compression near to isothermal as shown in fig. 11-3.

Multi-stage compression : In this method, the compression of air is carried out in two or more stages in separate cylinders. The pressure of the air is increased in each stage. It is a common practice to provide intercoolers between the cylinders of multi-stage compressor, for the purpose of cooling the compressed air to atmospheric (intake) temperature before entering the succeeding (next) stage. It is this cooling between the cylinders that keeps the compression very near to isothermal as shown in fig. 11-6.

11.4 Two-stage Air Compressor

A two-stage air compressor with water jacketed cylinders and intercooler is shown in fig. 11-5(a). The suction in the L.P. cylinder (fig. 11-5b) ends at 1 and the air is drawn in the cylinder at pressure p_1. The air is then compressed polytropically to 2'. The L.P. cylinder then discharges (delivers) the air along line 2-p_2 to the intercooler where air is cooled at constant pressure p_2, to the original (intake) temperature corresponding to point 1 by the circulating cold water. When air is cooled in the intercooler to intake temperature corresponding to point 1, the cooling is perfect. The air in cooling at constant pressure suffers a reduction of volume from 2'–p_2 to p_2–2. The cooled air is then drawn into the H.P. cylinder (fig. 11-5c) along line p_2–2 for the second stage compression, where it is compressed polytropically to the final pressure p_3 along line 2-3, and then delivered to the receiver (not shown) at constant pressure p_3 along line 3-p_3.

In fig. 11-6, the combined ideal indicator diagram is shown for the low pressure and high pressure cylinders of a single-acting, two-stage air compressor with



(a) Arrangement of cylinders.
(b) Indicator diagram of H.P. cylinder,
(c) Indicator diagram of L.P. cylinder.

Fig. 11-5. Two-stage air compressor with intercooler.

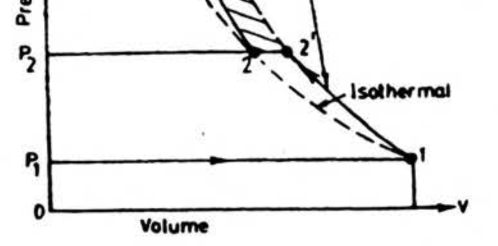


Fig. 11-6. Combined ideal indicator diagram of two-stage compression with perfect-intercooling.

perfect-intercooling. The low pressure cylinder diagram is shown as p_1–1–2'–p_2, and high pressure cylinder diagram as p_2–2–3–p_3. The reduction of work required due to intercooling is shown by the shaded are 2-3-3'-2' (fig. 11-6). When cooling is perfect, i.e., when air is cooled to intake temperature in the intercooler ($T_1 = T_2$), the point 2 will lie on the isothermal line 1 – 3" as shown in fig. 11-6.

It may be noted that each stage will increase the pressure of air while the intake temperature T_1 (corresponding to point 1) is maintained same at the end. The isothermal line during the process has been approximated as shown by the diagram, and the shaded area 2-3-3'-2' shows the saving of work as a result of this approximated isothermal.

11.4.1 Imperfect-Intercooling : Figure 11-7 represents the indicator diagram of a two-stage air compression with *imperfect-intercooling.* Let the compression follow the law pv^n = constant and the intercooling be *incomplete (imperfect)* so that the point 2 has not reached the isothermal line, i.e., point 2 does not lie on the isothermal curve 1-3".

Let p_1 in N/m^2 = Pa and v_1 in m^3 represent condition of air entering low-pressure cylinder, and p_2 and v_2 represent condition of air entering high-pressure cylinder, and p_3 be final delivery pressure of air, then the total work done for compression and delivery of air per cycle will be the sum of the work done in each cylinder. Work done in L.P. cylinder is shown by the area p_1-1-2'-p_2 and in H.P. cylinder by the area p_2-2-3-p_3 (fig. 11-7). The saving of work done due to imperfect intercooling is shown by the shaded area 2-3-3'-2'.

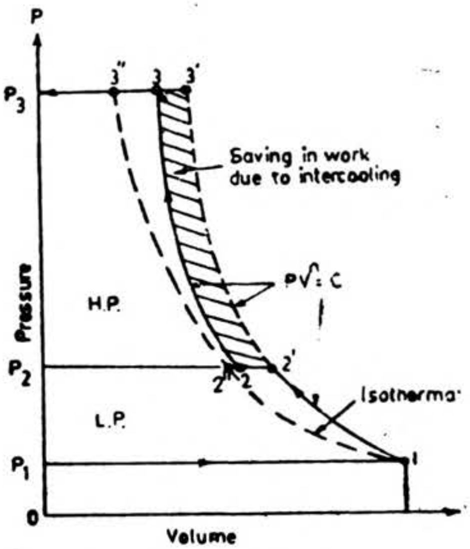


Fig. 11-7. Combined indicator diagram of two-stage compression with imperfect intercooling.

Hence, from eqn. (11.2),

$$\text{Work required in L.P. cylinder per cycle} = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{Joule.}$$

$$\text{Work required in H.P. cylinder per cycle} = \frac{n}{n-1} p_2 v_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right] \text{Joule.}$$

∴ Total work required per cycle,

$$W = \frac{n}{n-1} \left[p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} + p_2 v_2 \left\{ \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right\} \right] \text{Joule} \quad \text{... (11.8a)}$$

$$\text{Indicated power of the compressor} = \frac{W \times N}{60} \text{ J/sec. or watt.}$$

where N = No. of cycles per min. = r.p.m. for single-acting compressor, and

W = work required in Joules per cycle.

If p_1v_1 and p_2v_2 in eqn. (11.8a) are substituted by mRT_1 and mRT_2 respectively, then work done *per kg of air* can be written as

$$W = \frac{n}{n-1}\left[RT_1\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} + RT_2\left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right\}\right] \text{ Joules.} \quad \text{... (11.8b)}$$

If the compression is isentropic, γ should be substituted for n in eqns. (11.8a) and (11.8b).

11.4.2 Perfect-Intercooling : If intercooling is *perfect or complete* (fig. 11-6), the point 2 will lie on the isothermal line, i.e., point 2 will coincide with point 2", then $p_1v_1 = p_2v_2$.

Substituting this in eqn. (11.8a), total work required per cycle,

$$W = \frac{n}{n-1}p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{ Joules.} \quad \text{... (11.9a)}$$

Indicated power of the compressor $= \frac{W \times N}{60}$ J/sec. or W.

where, N = no. of cycles per min. = r.p.m. for single-acting compressor,

and W = work required in Joules per cycle.

If p_1v_1 in eqn. (11.9a) is substituted by mRT_1, then work required *per kg of air* may be written as

$$W = \frac{n}{n-1}RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \text{ Joules.} \quad \text{... (11.9b)}$$

Indicated power of the compressor

= W × mass of air delivered per second J/sec or watt. ... (11.9c)

where, W = work required in Joule per kg of air.

Referring to fig. 11-6,

Heat rejected to intercooler per min. = $mk_p\,(T_2' - T_2)$ kJ ... (11.10)

where, m = mass of air compressed per minute,

k_p = specific heat of air at constant pressure,

T_2' = temperature of air before entering the intercooler, and

T_2 = temperature of air after leaving the intercooler.

11.4.3 Ideal Intercooler Pressure : It may be noted from fig. 11-6 that saving in work increases as intercooling is increased. When intercooling is perfect, i.e., when air is cooled to intake temperature in the intercooler, point 2 lies on isothermal curve and there is maximum saving. In this case, work required is given by eqn. (11.9a). It may be further noted that this saving in work required also varies with the chosen intercooler pressure p_2. When the initial pressure p_1 and final pressure p_3 are fixed, the best value of the intercooler pressure p_2 shall be fixed to give minimum work. This value of p_2 can be found by differentiating expression of W (eqn. 11.9a) with respect to p_2 and equating it to zero.

The eqn. (11.9a) can be re-written by putting $\frac{n-1}{n} = y$.

$$W = \text{constant} \times \left[\left(\frac{p_2}{p_1}\right)^y + \left(\frac{p_3}{p_2}\right)^y - 2\right]$$

Differentiating and equating it to zero for minimum work,

$$\frac{dW}{dP_2} = \frac{y\,(p_2)^{y-1}}{(p_1)^y} - \frac{y\,(p_3)^y}{(p_2)^{y+1}} = 0$$

Dividing throughout by y and re-arranging, $\frac{(p_2)^{y-1}}{(p_1)^y} = \frac{(p_3)^y}{(p_2)^{y+1}}$

$$\therefore\ p_2^{2y} = (p_1p_3)^y$$

Taking y^{th} root throughout (i.e. both sides),

$$p_2^2 = p_1p_3 \text{ or } \frac{p_2}{p_1} = \frac{p_3}{p_2} \text{ or } p_2 = \sqrt{p_1p_3} \qquad \text{... (11.11)}$$

This shows that for minimum work required or maximum efficiency, the intercooler pressure is the geometric mean of the initial and final pressures, or pressure ratio in each stage is the same.

This (eqn. 11.11) gives the best value of p_2 when p_1 and p_3 are given for minimum work or maximum efficiency.

Substituting $\frac{p_2}{p_1}$ for $\frac{p_3}{p_2}$ in eqn. (11.9a),

$$\text{Min. work required per cycle, } W = 2\frac{n}{n-1}p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ Joule} \qquad \text{.. (11.12a)}$$

Since, $\frac{p_2}{p_1} = \frac{p_3}{p_2}$, then $\left(\frac{p_2}{p_1}\right)^2 = \frac{p_2}{p_1} \times \frac{p_3}{p_2} = \frac{p_3}{p_1}$

$$\therefore\ \left(\frac{p_3}{p_1}\right)^{1/2} = \frac{p_2}{p_1} = \frac{p_3}{p_2}$$

Substituting $\left(\frac{p_3}{p_1}\right)^{1/2}$ for $\left(\frac{p_2}{p_1}\right)$ in eqn. (11.12a),

Minimum work required per cycle (in terms of p_1 and p_3) can be written as

$$W = 2\frac{n}{n-1}p_1v_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \text{ Joule per cycle.} \qquad \text{...(11.12b)}$$

Minimum indicated power of the compressor $= W \times \frac{N}{60}$ J/sec. or watt. .. (11.12c)

If p_1v_1 in eqns. (11.12a) and (11.12b) is substituted by mRT_1, then minimum work required *per kg of air* may be written as

$$W = 2\frac{n}{n-1}RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{Joule} \qquad \text{...(11.13a)}$$

and $$W = 2\frac{n}{n-1}RT_1\left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \text{Joule} \qquad \text{.. (11.13b)}$$

Minimum indicated power of the compressor

= $W \times$ mass of air delivered per sec. J/sec. or W ..(11.13c)

Thus, conditions for maximum efficiency or minimum work required are :

.. The air is cooled to initial (intake) temperature in the intercooler, i.e., intercooling is perfect.

.. The pressure ratio in each stage is the same.

.. The work required for each stage is the same.

These conditions can also be extended for three-stage air compressors.

11.5 Three-stage Air Compressor

A three-stage air compressor with L.P. and I.P. intercoolers is shown in fig. 11-8.

In fig. 11-9, the combined indicator diagram for a three-stage air compressor is shown. The air having volume v_1 and pressure p_1 is compressed polytropically to pressure p_2 in the first or low-pressure cylinder, then delivered through L.P. intercooler to the second or intermediate pressure cylinder at pressure p_2, its volume shrinking to v_2. The air having volume v_2 and pressure p_2 is then compressed polytropically to p_3 in the I.P. cylinder, and is then delivered through I.P. intercooler to the high-pressure cylinder (third cylinder) at pressure p_3, its volume shrinking to v_3. This air having pressure p_3 and volume v_3 is then compressed polytropically to volume v_4 in the H.P. cylinder and then delivered to the receiver at pressure p_4.

As in the case of two-stage compression, the shaded area in fig. 11-9 represents the saving of work due to using three cylinders with inter-stage cooling instead of single-stage.

(a) Work required per cycle when *intercooling is imperfect*, i.e., air is not cooled to intake temperature in the intercoolers,

$$W = \frac{n}{n-1}p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}p_2v_2\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$+ \frac{n}{n-1}p_3v_3\left[\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} - 1\right] \text{Joule} \qquad \text{.. (11.14a)}$$

If p_1v_1, p_2v_2, and p_3v_3 in eqn. (11.14a) are substituted by mRT_1, mRT_2 and mRT_3 respectively, then work required *per kg of air* can be written as

$$W = \frac{n}{n-1} RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} RT_2 \left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$+ \frac{n}{n-1} RT_3 \left[\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} - 1 \right] \text{ Joule} \quad \text{.. (11.14b)}$$

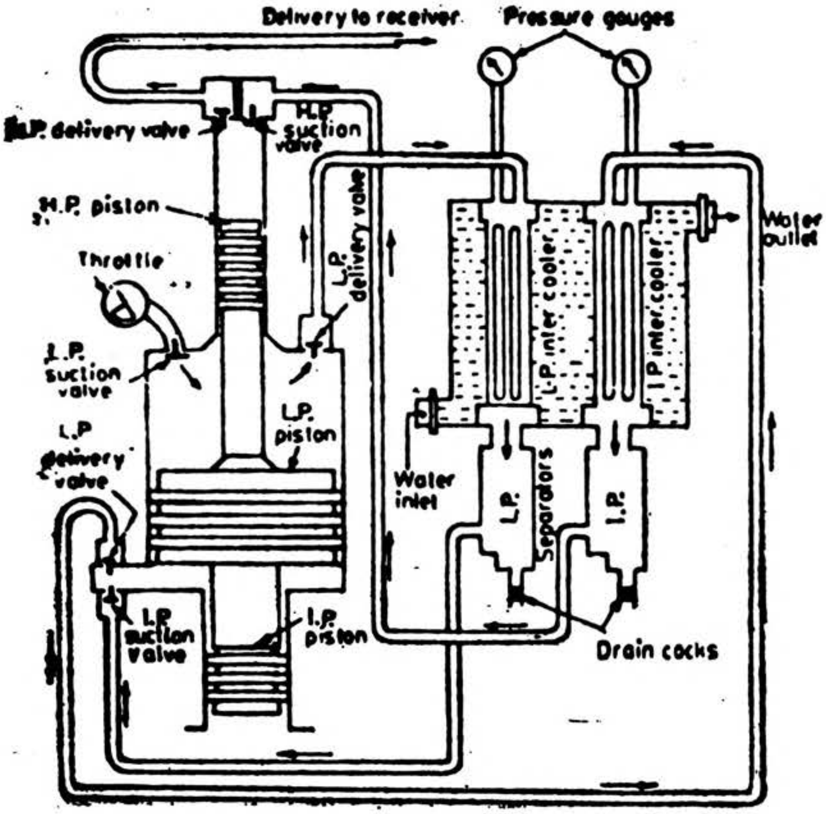


Fig. 11-8. Arrangement of three-stage air compressor with intercoolers.

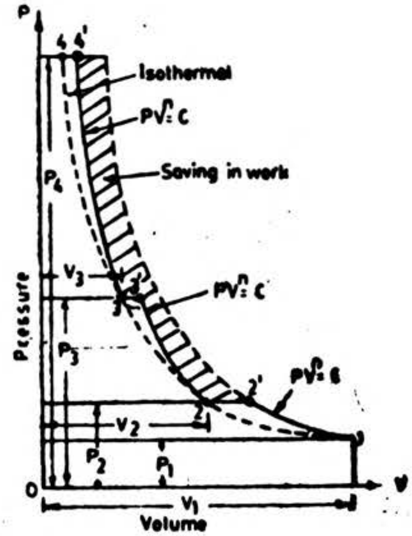


Fig. 11-9. Indicator diagram of three-stage compression with perfect intercooling.

(b) When *intercooling is perfect,* $p_1v_1 = p_2v_2 = p_3v_3$ (fig. 11-9).

Substituting p_1v_1 for p_2v_2 and p_3v_3 in eqn. (11.14a), work required per cycle,

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} + \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} - 3 \right] \text{ Joule} \quad \text{... (11.15a)}$$

If p_1v_1 in eqn. (11.15a) is substituted by mRT_1, then work required *per kg air* may be written as,

$$W = \frac{n}{n-1} RT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} + \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} - 3 \right] \text{ Joule} \quad \text{.. (11.15b)}$$

(c) *Work required is minimum* when $\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3}$

Substituting $\frac{p_2}{p_1}$ for $\frac{p_3}{p_2}$ and $\frac{p_4}{p_3}$ in eqn. (11.15a),

Min. work required per cycle, $W = \frac{3n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$ Joule .. (11.16a)

Now, for minimum work, $\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1} \right)^{1/3}$

Substituting $\left(\frac{p_4}{p_1} \right)^{1/3}$ for $\frac{p_2}{p_1}, \frac{p_3}{p_2}$ and $\frac{p_4}{p_3}$ in eqn. (11.16a),

Min. work required per cycle, $W = \frac{3n}{n-1} p_1 v_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$ Joule ..(11.16b)

Minimum indicated power of the compressor $= W \times \frac{N}{60}$ J/sec. or W .. (11.16c)

where N = no. of cycles per min. = r.p.m. for single-acting compressor, and

W = work required in Joule per cycle.

If $p_1 v_1$ in eqns. (11.16a) and (11.16b) is substituted by mRT_1,

then minimum work required *per kg of air* compressed and delivered may be written as,

$$W = \frac{3n}{n-1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \text{... (11.17a)}$$

$$W = \frac{3n}{n-1} RT_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right] \text{Joule} \quad \text{.. (11.17b)}$$

Minimum indicated power of the compressor

$= W \times$ mass of air delivered per second J/sec. or W .. (11.17c)

where W = work required in Joule per kg of air

(d) Equation (11.16b) can readily be extended to m stages.

Then $\frac{3n}{n-1}$ will be replaced by $\frac{mn}{n-1}$ and similarly $\frac{n-1}{3n}$ by $\frac{n-1}{mn}$

while $\frac{p_4}{p_1}$ will always refer to delivery and suction pressures, i.e. $\frac{p_{m+1}}{p_1}$.

Then minimum work required per cycle with complete intercooling,

$$W = \frac{mn}{n-1} p_1 v_1 \left[\left(\frac{p_{m+1}}{p_1} \right)^{\frac{n-1}{mn}} - 1 \right] \text{Joule} \quad \text{.. (11.18)}$$

All the expression derived above for work required refer to the work actually done or required to be done on the air, and the power derived from these expressions will be referred to as indicated power or air power.

11.5.1 Advantages of multi-stage compression : The advantages of multi-stage compression are as follows :

- Reduction in power required to drive the compressor owing to compression being approximated to isothermal,
- Better mechanical balance of the whole unit and uniform torque,
- Increased volumetric efficiency as a result of the lower delivery pressure in the L.P. cylinder clearance space,
- Reduced leakage loss owing to reduced pressure difference on either sides of the piston and valves,
- Less difficulty in lubrication due to the lower working temperature, and
- Lighter cylinders.

11.6 Air Compressor Terminology

The following terminology should be well understood before attempting to estimate the performance of the air compressor.

Free air delivered is the volume of air delivered under the conditions of temperature and pressure existing at the compressor intake, i.e., volume of air delivered at surrounding air temperature and pressure. In the absence of any given free air conditions, these are generally taken as 1·01325 bar and 15°C.

Capacity of a compressor is the quantity of the free air actually delivered by a compressor in cubic metres per minute.

Piston displacement is the volume in cubic metre (m^3) obtained as the product of the piston area in m^2 and the piston stroke in metre.

Displacement per minute is the product of the piston displacement and working strokes per minute. For multi-stage compressors, the displacement is based on low-pressure cylinder only, since it determines the amount of air passing through the other cylinder.

Indicated power or air power is the power determined from the actual indicator diagram taken during a test on the compressor. It is calculated in the same manner as is done in the case of a steam engine and internal combustion engine.

Shaft or brake power is the power delivered to the shaft of the compressor or the power required to drive the compressor. The compressor may be driven by an engine or an electric motor.

[Shaft or brake power] – [Air or indicated power] = [Friction power]

and Mechanical efficiency, $\eta_m = \dfrac{\text{Air (indicated) power}}{\text{Shaft (brake) power}}$

Isothermal power of a compressor is calculated from the theoretical indicator diagram drawn on the basis of an assumption that the compression is isothermal.

(a) Referring to eqn. (11.7a) for a *single-stage compressor without clearance,*

Isothermal work required per cycle, $W = p_1 v_1 \log_e \left(\dfrac{p_2}{p_1}\right)$ Joule .. (11.19a)

Isothermal power $= p_1 v_1 \log_e\left(\frac{p_2}{p_1}\right) \times \frac{N}{60}$ J/sec. or W .. (11.19b)

where N = no. of cycles per minute.

If p_1v_1 in eqn. (11.19a) is substituted by mRT_1, then isothermal work required *per kg of air* may be written as,

$$W = RT_1 \log_e\left(\frac{p_2}{p_1}\right) \text{ Joule} \quad \text{.. (11.20a)}$$

Isothermal power = W × mass of air delivered per sec. J/sec. or W .. (11.20b)

(b) For a *two-stage compressor,*

Isothermal work required per cycle, $W = p_1 v_1 \log_e\left(\frac{p_3}{p_1}\right)$ Joule .. (11.21a)

Isothermal power $= p_1 v_1 \log_e\left(\frac{p_3}{p_1}\right) \times \frac{N}{60}$ J/sec. or W .. (11.21b)

If p_1v_1 in eqn. (11.21a) is substituted by mRT_1, then isothermal work required *per kg of air* may be written as

$$W = RT_1 \log_e\left(\frac{p_3}{p_1}\right) \text{ Joule} \quad \text{.. (11.22a)}$$

Isothermal power = W × mass of air delivered per sec. J/sec. or W .. (11.22b)

(c) Similarly, for a three-stage compressor,

Isothermal work required per cycle, $W = p_1 v_1 \log_e\left(\frac{p_4}{p_1}\right)$ Joule .. (11.23a)

Isothermal power $= p_1 v_1 \log_e\left(\frac{p_4}{p_1}\right) \times \frac{N}{60}$ J/sec. or W .. (11.23b)

where N = no. of cycles per min.

Adiabatic power is calculated from a theoretical indicator diagram drawn on the basis of an assumption that the compression is an ideal adiabatic, i.e., isentropic.

Adiabatic work required, $W = \frac{\gamma}{\gamma - 1}(p_2v_2 - p_1v_1)$ Joule per cycle

This equation for the adiabatic work required may be expressed in more convenient form by writing its equivalent,

$$W = \frac{\gamma}{\gamma-1} mR(T_2 - T_1) = \frac{\gamma}{\gamma-1} mRT_1\left(\frac{T_2}{T_1} - 1\right)$$

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

∴ Adiabatic work required, $W = \frac{\gamma}{\gamma-1} mRT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$ Joule per cycle

$$= \frac{\gamma}{\gamma-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \text{ Joule per cycle} \quad ..(11.24a)$$

Thus, for a single-stage air compressor,

$$\textit{Adiabatic (isentropic) power} = \frac{\gamma}{\gamma-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \times \frac{N}{60} \text{ J/sec. or W} \quad ..(11.24b)$$

where N = no. of cycles per min.

$$\textit{Isothermal efficiency} = \frac{\text{Isothermal power in watts}}{\text{Indicated or actural power in watts}} \quad .. (11.25a)$$

$$\left[\begin{array}{l} \textit{Overall isothermal efficiency or} \\ \textit{Compressor efficiency} \end{array} \right] = \frac{\text{Isothermal power}}{\left[\begin{array}{c} \text{Shaft power or brake power required to} \\ \text{drive the compressor} \end{array} \right]}$$

$$= \frac{p_1 v_1 \log_e \left(\frac{p_2}{p_1} \right) \times \frac{N}{60} \text{ watt}}{\text{Shaft power or brake power required in watt to drive the compressor}} \quad .. (11.25b)$$

where N = no. of cycles per min.

$$\left[\begin{array}{l} \textit{Isentropic efficiency or ideal} \\ \textit{adiabatic efficiency} \end{array} \right] = \frac{\text{Isentropic power in watt}}{\text{Shaft power required in watt}}$$

$$= \frac{\frac{\gamma}{\gamma-1} p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \times \frac{N}{60} \text{ watt}}{\text{Shaft power or Brake power required in watt to drive the compressor}} \quad .. (11.26)$$

where N = no. of cycles per min.

Volumetric efficiency of an air compressor is the ratio of the actual volume of the free air at standard atmospheric conditions discharged in one delivery stroke, to the volume swept by the piston during the stroke. The standard atmospheric conditions (S.T.P.) is actually taken as pressure of 760 mm Hg (1·01325 bar) and temperature 15°C in this connection. Thus,

$$\text{Volumetric efficiency} = \frac{\text{Volume of free air delivered per stroke}}{\text{Volume swept by piston per stroke}} \quad .. (11.27)$$

The value of volumetric efficiency varies between 70 to 85 per cent according to the type of compressor.

The volumetric efficiency decreases as the clearance volume increases. Other factors that lower the volumetric efficiency are :

- Valve leakage, specially at the inlet valve,
- Obstruction at inlet valves,
- Piston ring leakage, which allows air to pass from one side of the cylinder to the other,
- Heating of air by contact with hot cylinder walls, and

- Very high speed of rotation.

With the decrease of volumetric efficiency, the capacity (quantity of free air delivered) of the compressor decreases.

Problem-1 : *A single-cylinder, single-acting reciprocating air compressor has a cylinder of 24 cm diameter and linear piston speed of 100 metres per minute. It takes in air at 100 kPa (100 kN/m*^{2}*) and delivers at 1 MPa (1 MN/m*^{2}*), Determine the indicated power of the compressor. Assume the law of compression to be* $pv^{1.25}$ *= constant. The temperature of air at inlet is 288 K. Neglect clearance effect.*

Given $p_1 = 100$ kPa $= 100 \times 10^3$ Pa; $p_2 = 1$ MPa $= 1{,}000$ kPa $= 1{,}000 \times 10^3$ Pa.

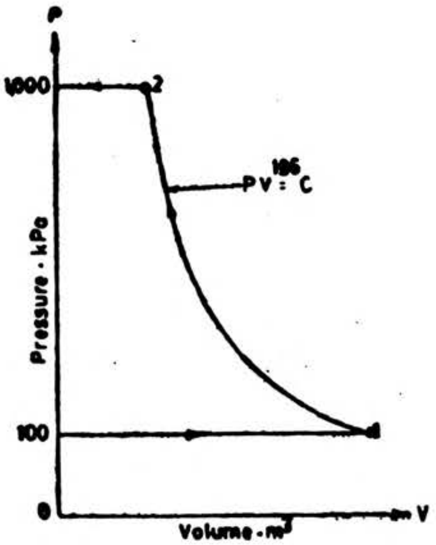


Fig. 11-10. Single stage air compressor.

$$\frac{p_2}{p_1} = \frac{1{,}000 \times 10^3}{100 \times 10^3} = \frac{1{,}000}{100} = 10$$

Swept volume in m^3/min. $= \frac{\pi}{4} d^2 \times l \times r.p.m.$

(where, l = piston strokes in metre, and d= diameter of the cylinder in metre)

$$\therefore \text{ Swept volume } = \frac{\pi}{4} \times \left(\frac{24}{100}\right)^2 \times \frac{100}{2} \text{ m}^3\text{/min.}$$

($\because$ piston speed $= 2 \times l \times r.p.m. = 100$ metres/min.)

$$= 2{\cdot}261 \text{ m}^3\text{/min.} = \frac{2{\cdot}261}{60} \text{ m}^3\text{/sec.}$$

Referring to fig. 11-10 and using eqn. (11.2), Work required per sec.,

$$W = p_1 v_1 \times \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ J/sec.}$$

[where, p_1 is pressure in Parcals (Pa) and volume of air compressed, v_1 is in m^3 per sec.]

$$\therefore\ W = (100 \times 10^3) \times \frac{2{\cdot}261}{60} \times \frac{1{\cdot}25}{1{\cdot}25 - 1}\left[\left(\frac{1{,}000}{100}\right)^{\frac{1{\cdot}25-1}{1{\cdot}25}} - 1\right]$$

$$= 1{,}886{\cdot}16 \times (1{\cdot}5848 - 1) = 11{,}030 \text{ J/sec. or } 11{,}030 \text{ W}$$

$\therefore$ Indicated power of the compressor = 11,030 W i.e. **11·03 kW**

Problem-2- : *A single-acting, single-stage air compressor developing indicated power of 11 kW, runs at 200 r.p.m. and has a linear piston speed of 100 metres per min. If the suction pressure and temperature are 100 kPa and 15°C respectively and delivery pressure is 1,000 kPa, calculate the dimensions of the compressor cylinder. Assume the law of compression to be* $pv^{1.25}$ *= constant. Neglect clearance effects.*

Referring to fig. 11-10, and considering polytropic compression 1-2, $p_1 v_1^n = p_2 v_2^n$,

$$\frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = \left(\frac{10}{1}\right)^{\frac{1}{1\cdot25}} = 6\cdot31$$

Using eqn. (11.1a), work done per cycle, $W = p_2v_2 + \frac{p_2v_2 - p_1v_1}{n-1} - p_1v_1$

$$\therefore \text{ M.E.P.} = \frac{\text{Work done per cycle in kJ, } W}{\text{Displacement volume in m}^3\text{, } v_1} \text{ kPa}$$

$$= \frac{p_2v_2 + \frac{p_2v_2 - p_1v_1}{n-1} - p_1v_1}{v_1} = \frac{p_2\,v_2}{v_1} - p_1 + \frac{\left(\frac{p_2\,v_2}{v_1} - p_1\right)}{n-1}$$

$$= 1{,}000 \times \frac{1}{6\cdot31} - 100 + \frac{1{,}000 \times \frac{1}{6\cdot31} - 100}{0\cdot25} = 292\cdot4 \text{ kPa.}$$

Piston stroke, $l = \frac{\text{piston speed per min.}}{\text{piston strokes per min. } (2 \times \text{r.p.m.})} = \frac{100}{2 \times 200}$ = **0·25 metre or 25 cm**

Indicated power of compressor = $p_m \times l \times a \times n$ watt.

$$\left(\text{where } n = \text{ no. of cycles per sec.} = \frac{200}{60}\right)$$

i.e. $11 \times 10^3 = 292\cdot4 \times 10^3 \times 0\cdot25 \times 0\cdot7854 \times \left(\frac{d}{100}\right)^2 \times \frac{200}{60}$

$$\therefore d^2 = \frac{11 \times 10^3 \times 10^4 \times 60}{292\cdot4 \times 10^3 \times 0\cdot25 \times 0\cdot7854 \times 200} = 574\cdot79$$

$$\therefore \quad d = \sqrt{574\cdot79} = 23\cdot98 \text{ cm.}$$

Problem-3 : *A single-acting, single-stage air compressor is belt driven from an electric motor at 300 r.p.m. The cylinder diameter is 20 cm and the stroke is 24 cm. The air is compressed from one atmosphere to 8 atmospheres and the law of compression is* $pv^{1\cdot25}$ *= constant. Find the power of the electric motor if the transmission efficiency is 96 per cent and the mechanical efficiency of the compressor is 85 per cent. Neglect clearance effect.*

Swept volume,

$$v_1 = \frac{\pi}{4} d_1^2 \times l = \frac{\pi}{4}\left(\frac{20}{100}\right)^2 \times \left(\frac{24}{100}\right) = 0\cdot00754 \text{ m}^3.$$

Referring to fig. 11-11 and using eqn. (11.2), work required per cycle,

$$W = \frac{n}{n-1} p_1v_1 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} \text{ Joule.}$$

$$= \frac{1\cdot25}{1\cdot25-1} \times (1\cdot01325 \times 10^5) \times 0\cdot00754 \times \left\{\left(\frac{8}{1}\right)^{\frac{1\cdot25-1}{1\cdot25}} - 1\right\}$$

$$= 5 \times (1\cdot01325 \times 10^5) \times 0\cdot00754 \times (1\cdot515 - 1) = 1{,}967\cdot27 \text{ Joule/cycle.}$$

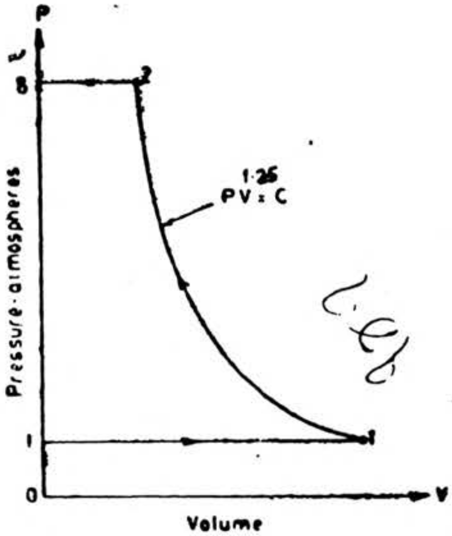


Fig. 11-11. Single-stage compression.

Work required per sec. = work done per cycle × r.p.s.

= 1,967·27 × 5 = 9,836 J/sec.

or 9,836 W

∴ Indicated power = 9·836 kW

Taking into consideration the given mechanical efficiency of the compressor only,

$$\text{Power required} = \frac{9{\cdot}836}{0{\cdot}85} = 11{\cdot}57 \text{ kW}$$

∴ Power of the electric motor. (considering the transmission efficiency also)

$$= \frac{11{\cdot}57}{0{\cdot}96} = \mathbf{12{\cdot}05 \text{ kW}}$$

Problem-4 : *The piston of an air compressor has displacement of 9·5 m³ per minute. If the pressure and temperature at the intake are 100 kPa and 25°C respectively, and the compressor in 2¼ minutes raises the pressure in 1·45 m³ capacity air receiver to 1,500 kPa and temperature 60°C, find the volumetric efficiency of the compressor.*

Assume initial pressure and temperature in the receiver as 100 kPa and 25°C. Take R = 0·287 kJ/kg K for air.

Mass of air initially in the receiver can be obtained by applying characteristic equation $p_1v_1 = m_1RT_1$,

$$\text{i.e. } m_1 = \frac{p_1v_1}{RT_1} = \frac{(100 \times 10^3) \times 1{\cdot}45}{(0{\cdot}287 \times 10^3) \times 298} = 1{\cdot}695 \text{ kg.}$$

After 2¼ minutes, the mass of the air in the receiver will be,

$$m_2 = \frac{p_2v_2}{RT_2} = \frac{(1{,}500 \times 10^3) \times 1{\cdot}45}{(0{\cdot}287 \times 10^3) \times 333} = 22{\cdot}758 \text{ kg.}$$

$$\therefore \text{ Mass of air compressed per minute, } m = \frac{m_2 - m_1}{2\,1/4} = \frac{22{\cdot}758 - 1{\cdot}695}{2{\cdot}25} = 9{\cdot}36 \text{ kg}$$

Volume which this air occupies at 100 kPa and 25°C,

$$v = \frac{mRT}{p} = \frac{9{\cdot}36 \times (0{\cdot}287 \times 10^3) \times 298}{(100 \times 10^3)} = 8 \text{ m}^3.$$

$$\text{Volumetric efficiency of the compressor} = \frac{8}{9{\cdot}5} \times 100 = \mathbf{84{\cdot}21\%}$$

Problem-5 : *It is desired to compress 17m³ of air per minute from 1 bar (100 kN/m²) and 21°C to a delivery pressure of 7 bar (700 kN/m²) in a single-stage, single-acting air compressor. Calculate the power required to drive the compressor and the heat rejected during compression to cooling water if the compression is (a) Isentropic (γ = 1·4 for air), and (b) Isothermal.*

Neglect clearance effects.

Given : $p_1 = 1 \text{ bar} = 1 \times 10^5 \text{ Pa}$; $p_2 = p_2' = 7 \text{ bar} = 7 \times 10^5 \text{ Pa}$; $\frac{p_2}{p_1} = \frac{7 \times 10^5}{1 \times 10^5} = \frac{7}{1}$

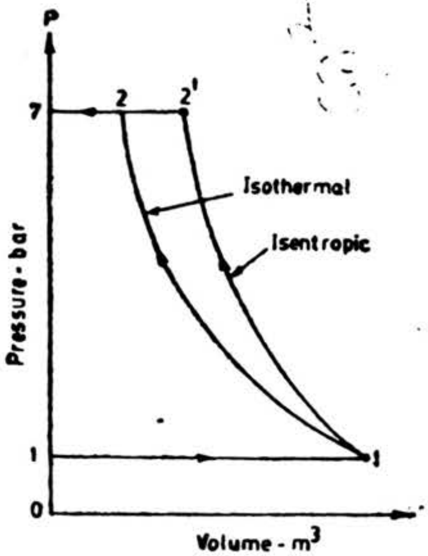


Fig. 11 12. Single stage compression.

(a) Isentropic compression :

Referring to fig. 11-12 and using eqn. (11.6a), isentropic work required per sec.,

$$\dot{W} = \frac{\gamma}{\gamma - 1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \text{ J/sec.}$$

$$= \frac{1.4}{1.4 - 1} \times (1 \times 10^5) \times \frac{17}{60} \left[\left(\frac{7}{1} \right)^{\frac{1.4 - 1}{1.4}} - 1 \right]$$

$$= \frac{1.4}{0.4} \times (1 \times 10^5) \times \frac{17}{60} \times [1.744 - 1]$$

$$= 73{,}750 \text{ J/sec. or } 73{,}750 \text{ W}$$

∴ Power required to drive the compressor

= 73,750 W i.e. **73·75 kW**

No heat is rejected during isentropic compression.

(b) Isothermal Compression

Referring to fig. 11-12 and using eqn. (11.7a), isothermal work required/sec.,

$$W = p_1 v_1 \log_e \left(\frac{p_2}{p_1} \right) = (1 \times 10^5) \times \frac{17}{60} \times \log_e \left(\frac{7}{1} \right)$$

$$= (1 \times 10^5) \times \frac{17}{60} \times 1.9459 = 55{,}080 \text{ J/sec. or } 55{,}080 \text{ W}$$

∴ Required power input = 55,080 W i.e. **55·08 kW**

In isothermal compression, as the temperature remains constant, there is no change in internal energy and the entire work of compression, i.e. 55·08 kJ/sec. is rejected to jacket cooling water. Heat rejected during isothermal compression = **55·08 kJ/sec.**

Problem-6 : *A two-stage air compressor delivers 145 m³ of free air per hour. The pressure and temperature in the cylinder at the start of compression are 1 bar and 34°C respectively. The diameter of the low-pressure cylinder is twice that of the high pressure cylinder. The air enters the high-pressure cylinder at a temperature of 40°C and is then compressed to 17·5 bar, the law of compression being $pv^{1.22}$ = constant for both stages. Neglecting the effects of clearance, estimate : (a) the intercooler pressure, (b) the air power required, and (c) the ratio of cylinder diameters for minimum work making the usual assumptions regarding the intercooler conditions.*

The free air conditions are 1·01325 bar and 15°C. Take R = 0·287 kJ/kg K for air.

(a) Since, the diameter of L.P. cylinder is twice that of the H.P. cylinder, the ratio of L.P. to H.P. cylinder volumes will be 4, i.e. $v_1 = 4v_2$.

Applying characteristic equations at points of suction in L.P. and H.P. cylinders,

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} \quad \text{i.e.} \quad \frac{1 \times 4v_2}{(34 + 273)} = \frac{p_2 \times v_2}{(40 + 273)}$$

$\therefore$ Intercooler pressure, $p_2 = \frac{1 \times 4 \times 313}{307} = \mathbf{4{\cdot}078\ bar}$

(b) Using characteristic equation for free air conditions, $pv = mRT$

i.e. $(1{\cdot}01325 \times 10^5) \times \frac{145}{60} = m \times (0{\cdot}287 \times 10^3) \times (15 + 273)$

$\therefore\ m = 2{\cdot}963$ kg of air per minute.

Referring to fig. 11-13, and using eqn. (11.8b) for imperfect intercooling,

Total work required per kg of air, $W = \begin{Bmatrix}\text{Work required in} \\ \text{H.P. cylinder}\end{Bmatrix} + \begin{Bmatrix}\text{Work required in} \\ \text{L.P. cylinder}\end{Bmatrix}$

$$\therefore\ W = RT_1 \frac{n}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} + RT_2 \frac{n}{n-1}\left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right\}$$

$$= R\frac{n}{n-1}\left[T_1\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} + T_2\left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right\}\right]$$

$$= 0{\cdot}287 \times \frac{1{\cdot}22}{1{\cdot}22 - 1}\left[307\left\{\left(\frac{4{\cdot}078}{1}\right)^{\frac{0{\cdot}22}{1{\cdot}22}} - 1\right\} + 313\left\{\left(\frac{17{\cdot}5}{4{\cdot}078}\right)^{\frac{0{\cdot}22}{1{\cdot}22}} - 1\right\}\right]$$

$= 290$ kJ per kg of air

Work done per sec. $= W \times m = 290 \times \frac{2{\cdot}963}{60} = 14{\cdot}32$ kJ/sec.

$\therefore$ Air power $= \mathbf{14{\cdot}32\ kW}$

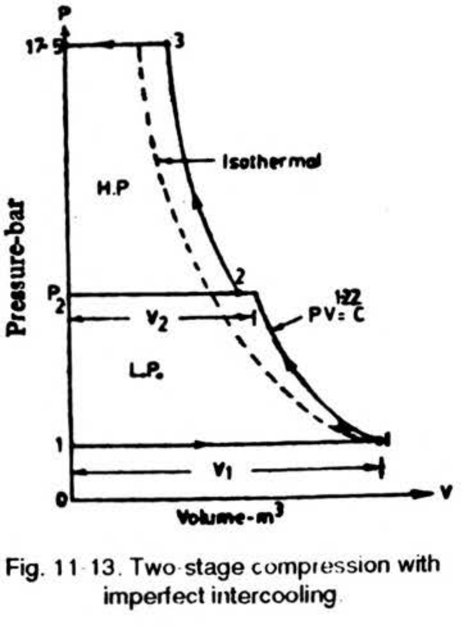


Fig. 11-13. Two stage compression with imperfect intercooling.

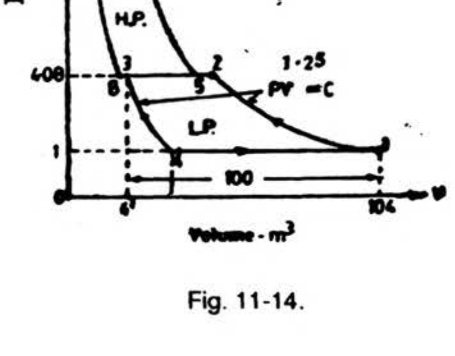


Fig. 11-14.

(c) Using eqn. (11.11),

For minimum work, intercooler pressure, $p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 17{\cdot}5} = 4{\cdot}183$ bar

Since intercooling is perfect, $T_1 = T_2$.

$\therefore\ p_1 v_1 = p_2 v_2$ i.e. $1 \times v_1 = 4{\cdot}183 \times v_2$ or $\frac{v_1}{v_2} = \frac{4{\cdot}183}{1} = 4{\cdot}183$

$\therefore$ Ratio of cylinder diameters for minimum work,

$$\frac{d_1}{d_2} = \sqrt{\frac{v_1}{v_2}} = \sqrt{4{\cdot}183} = 2{\cdot}045 \qquad (\because l_1 = l_2)$$

Problem-7 : *It is desired to compress 16 m³ of air per minute from 1 bar (100 kPa) and 294 K to 10·5 bar (1·05 MPa). Calculate : (i) the minimum power required to drive the compressor with two-stage compression and compare it with the power required for single-stage compression, (ii) the maximum temperature in the two cases. (iii) the heat to be removed in the intercooler per minute, (iv) the amount of cooling water required per minute if the inlet and outlet temperatures of cooling water to and from the intercooler are 15°C and 40°C. Assume the value of index for compression process to be 1·35 for both cases. Also assume proper intercooler pressure for minimum work and perfect intercooling. Take R = 0·287 kJ/kg K and* k_p *= 1·0035 kJ/kg K for air.*

Given : $p_1 = 1$ bar $= 1 \times 10^5$ Pa; $p_3 = 10{\cdot}5$ bar $= 10{\cdot}5 \times 10^5$ Pa;

$$\frac{p_3}{p_1} = \frac{10{\cdot}5 \times 10^5}{1 \times 10^5} = \frac{10{\cdot}5}{1}$$

Using equation (11.11) for maximum efficiency or minimum work, intercooler pressure,

$$p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 10{\cdot}5} = 3{\cdot}241 \text{ bar} = 3{\cdot}241 \times 10^5 \text{ Pa}$$

$$\therefore\ \frac{p_2}{p_1} = \frac{3{\cdot}241 \times 10^5}{1 \times 10^5} = \frac{3{\cdot}241}{1}$$

Referring to fig. 11-15 and using eqn. (11.12a),

Minimum work required per sec. for two-stage compression,

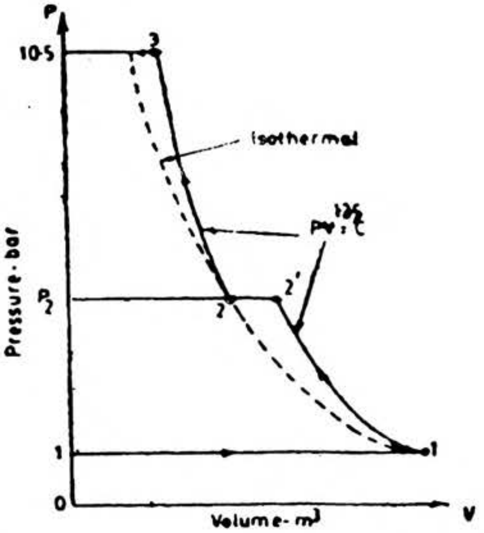


Fig. 11-15. Two-stage compression with perfect intercooling.

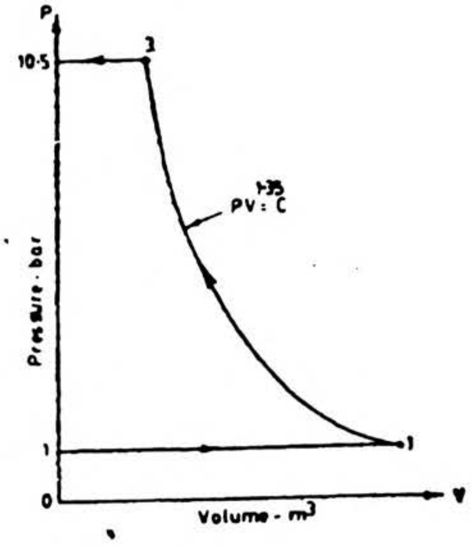


Fig. 11-16. Single-stage compression.

$$W = 2 \times \frac{n}{n-1} \times p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ J/sec.}$$

(where, p_1 is pressure in pascals (Pa) and v_1 is volume of air compressed in m^3 per sec.)

$$= 2 \times \frac{1{\cdot}35}{1{\cdot}35 - 1} \times (1 \times 10^5) \times \frac{16}{60} \left[\left(\frac{3{\cdot}241}{1} \right)^{\frac{1{\cdot}35-1}{1{\cdot}35}} - 1 \right] = 73{,}170 \text{ J/sec. or } 73{,}170 \text{ W}$$

Minimum power required for two-stage compression = 73,170 W i.e. **73·17 kW**

For *single-stage* compression (fig. 11-16), the work required/sec. is given by

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_3}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ J/sec.}$$

(where v_1 is the volume of air taken in per sec. in m^3 and p_1 is inlet pressure and p_3 is final pressure)

$$= \frac{1{\cdot}35}{1{\cdot}35 - 1} \times (1 \times 10^5) \times \frac{16}{60} \left[\left(\frac{10{\cdot}5}{1} \right)^{\frac{1{\cdot}35-1}{1{\cdot}35}} - 1 \right] = 86{,}250 \text{ J/sec. or } 86{,}250 \text{ W}$$

Minimum power required to drive the compressor for single-stage compression

= 86,250 W i.e. **86·25 kW**

(ii) For *single-stage* compression (fig.11-16), the maximum absolute temperature is T_3.

$$\text{Now, } \frac{T_3}{T_1} = \left(\frac{p_3}{p_1} \right)^{\frac{n-1}{n}}$$

$$\therefore\ T_3 = T_1 \times \left(\frac{p_3}{p_1} \right)^{\frac{n-1}{n}} = 294 \left(\frac{10{\cdot}5}{1} \right)^{\frac{1{\cdot}35-1}{1{\cdot}35}} = 541 \text{ K}$$

∴ Maximum temperature with single-stage compression = 541 − 273 = **268°C.**

For *two-stage* compression (fig. 11-15), the maximum absolute temperature is T_2'.

$$\text{Now, } \frac{T_2'}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}}$$

$$\therefore\ T_2' = T_1 \times \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} = 294 \times \left(\frac{3{\cdot}241}{1} \right)^{\frac{1{\cdot}35-1}{1{\cdot}35}} = 398{\cdot}72 \text{ K}$$

∴ Maximum temp. with two-stage compression = 398·72 − 273 = **125·72°C**

(iii) Now, mass of air compressed per min.,

$$m = \frac{p_1 v_1}{R T_1} = \frac{(1 \times 10^5) \times 16}{287 \times 294} = 18{\cdot}96 \text{ kg/min.}$$

Referring to fig. 11-15 and using eqn. (11.10),

Heat rejected by air to the intercooler water per min. with two-stage compression

$= mk_p\ (T_2' - T_2)$ kJ/min.

$= 18{\cdot}96 \times 1{\cdot}0035 \times (398{\cdot}72 - 294) = 1{,}995$ kJ/min.

Heat gained by cooling water in the intercooler per min.

$= m_1 \times 4{\cdot}187 \times (t_2 - t_1)$ where, m_1 = mass of cooling water/min.

$= m_1 \times 4{\cdot}187 \times (40 - 15)$ kJ/min.

Heat rejected by air/min. = Heat gained by cooling water/min.

i.e. $1{,}995 = m_1 \times 4{\cdot}187 \times (40 - 15)$

$\therefore\ m_1 =$ **19·06 kg/min.** (mass of cooling water per min.)

Problem-8 : *A two-stage, single-acting air compressor for a Diesel engine runs at 250 r.p.m. and takes in 6 m^3 of air per minute at a pressure of 1 bar and temperature of 15°C. It delivers the air at 70 bar and compression is carried out in each cylinder according to the law $pv^{1.3}$ = constant. Assuming complete intercooling and mechanical efficiency of 80 per cent, determine the minimum power required to drive the compressor. Calculate also the cylinder diameters and common stroke, if the average piston speed is 170 metres per minute. Neglect clearance effects and wire-drawing losses.*

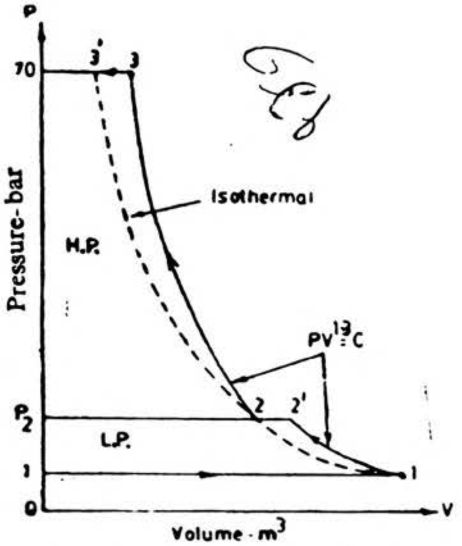


Fig. 11-17. Two-stage compression with perfect intercooling.

Referring to fig. 11-17, and using eqn. (11.12b),

Minimum indicated power

$$= \frac{2n}{n-1}\, p_1 v_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \text{ watts}$$

$$= \frac{2 \times 1{\cdot}3}{1{\cdot}3 - 1} \times (1 \times 10^5) \times \frac{6}{60}\left[\left(\frac{70}{1}\right)^{\frac{1{\cdot}3\,-\,1}{2\,\times\,1{\cdot}3}} - 1\right]$$

$= 54{,}773$ W $= 54{\cdot}773$ kW

$\therefore$ Power required to drive the compressor (i.e. brake power) taking into consideration the given mechanical efficiency.

$$= \frac{54{\cdot}773}{0{\cdot}8} = 68{\cdot}466 \text{ kW}$$

i.e. Minimum brake power required to drive the compressor = **68·466 kW**

Stroke, $l = \frac{\text{mean piston speed per min.}}{\text{no. of strokes per min.}} = \frac{170}{2 \times 250} =$ **0·34 m or 34 cm.**

L.P. Cylinder stroke volume in m^3, $v_1 = \frac{6}{250} = \frac{\pi}{4}(d_1)^2 \times l$

i.e. $\frac{6}{250} = \frac{\pi}{4}(d_1)^2 \times 0{\cdot}34$

$\therefore\ d_1^2 = 0{\cdot}0899$

$\therefore$ L.P. cylinder dia, $d_1 = \sqrt{0{\cdot}0899} \approx$ **0·3 m, or 30 cm.**

Now, for perfect-intercooling, $p_1 v_1 = p_2 v_2$,

i.e. $p_1 \times d_1^2 = p_2 \times d_2^2 \qquad (l_1 = l_2)$

$$\therefore \left(\frac{d_2}{d_1}\right)^2 = \frac{p_1}{p_2} = \frac{p_1}{\sqrt{p_1 p_3}} = \sqrt{\frac{p_1}{p_3}} = \sqrt{\frac{1}{70}} = 0{\cdot}1196$$

$$\therefore \left(\frac{d_2}{d_1}\right) = \sqrt{0{\cdot}1196} = 0{\cdot}346$$

$\therefore$ H.P. cylinder dia., $d_2 = d_1 \times 0{\cdot}346 = 0{\cdot}3 \times 0{\cdot}346 =$ **0·1038 m, or 10·38 cm.**

Problem-9 : *A four-stage air compressor works between the pressures of 1 bar and 140 bar and the index of compression in each stage is 1·23. The temperature at the start of compression in each cylinder is 48°C and the intercooler pressure are so chosen that the work is divided equally between the stages. If the clearance effect be neglected, estimate :*

(i) the volume of free air at a pressure of 1·01325 bar and temperature of 15°C which would be dealt with per kW-hour, and (ii) the isothermal efficiency referred to 15°C,

Take R = 0·287 kJ/kg K for air.

$$\text{Now, } \frac{p_5}{p_1} = \frac{140 \times 10^5}{1 \times 10^5} = \frac{140}{1}$$

(i) Using eqn. (11.18) for minimum work, four-stage work required per kg of air,

$$W = \frac{4n}{n-1} RT_1 \left[\left(\frac{p_5}{p_1}\right)^{\frac{n-1}{4n}} - 1\right]$$

$$= \frac{4 \times 1{\cdot}23}{0{\cdot}23} \times 0{\cdot}287 \times 321 \left[\left(\frac{140}{1}\right)^{\frac{0{\cdot}23}{4 \times 1{\cdot}23}} - 1\right] = 528{\cdot}145 \text{ kJ/kg of air.}$$

But, one kW-hour = 1 × 3,600 = 3,600 kJ

$$\therefore \text{Mass of air dealt with, } m \text{ per kW-hour} = \frac{3{,}600}{528{\cdot}145} = 6{\cdot}816 \text{ kg}$$

Corresponding *volume* of this (6·816 kg) air at 1·01325 bar and 15°C,

$$v = \frac{mRT}{p} = \frac{6{\cdot}816 \times 0{\cdot}287 \times 10^3 \times 288}{1{\cdot}01325 \times 10^5}$$

$$= \mathbf{5{\cdot}56 \text{ m}^3 \text{ of free air/kW-hour}}$$

(ii) Isothermal work required referred to 15°C per kg of air,

$$W = RT_1 \log_e\left(\frac{p_5}{p_1}\right)$$

$$= 0{\cdot}287 \times 288 \times \log_e\left(\frac{140}{1}\right) = 408{\cdot}454 \text{ kJ per kg of air}$$

$$\text{Isothermal efficiency} = \frac{\text{Work required when compression is isothermal}}{\text{Actual work required to be done on the air}}$$

$$= \frac{408{\cdot}454}{528{\cdot}145} = 0{\cdot}7734, \text{ or } \mathbf{77{\cdot}34\%}$$

Problem-10 : *The following particulars apply to a two-stage, single-acting air compressor :*

Stroke = 28·5 cm; Low pressure cylinder = 23 cm;

Final pressure = 25 bar; Intermediate pressure = 5 bar;

Temperature of air leaving the intercooler = 35°C

If the air drawn in the compressor is at 1 bar and 15°C, find the power expended (used) in compressing air when running at 350 r.p.m. Assume law of compression as $pv^{1.3}$ *= constant for both stages.*

Swept volume of the L.P. cylinder, $v_1 = \frac{\pi}{4} d^2 \times l = \frac{\pi}{4}\left(\frac{23}{100}\right)^2 \times \frac{28{\cdot}5}{100} = 0{\cdot}01189 \text{ m}^3.$

Referring to fig. 11-18, and considering polytropic compression 1–2′,

$p_1 v_1{}^n = p_2 v_2{}'^n$

$$\therefore \; v_2{}' = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} \times v_1 = \left(\frac{1}{5}\right)^{\frac{1}{1{\cdot}3}} \times 0{\cdot}01189 = 0{\cdot}00344 \text{ m}^3.$$

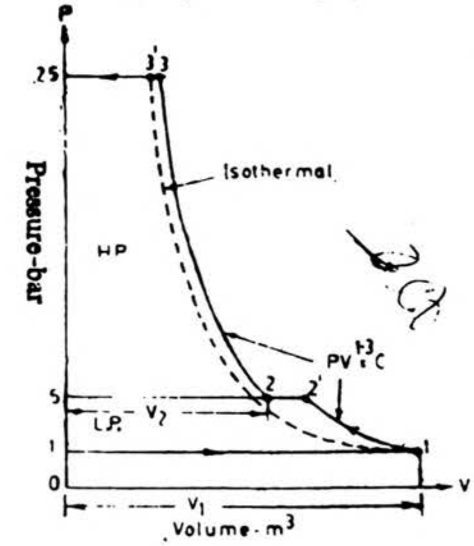


Fig. 11-18. Two-stage compression with imperfect intercooling.

Again, $\frac{T_2{}'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

$\therefore \; T_2' = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

$= (15 + 273) \times \left(\frac{5}{1}\right)^{\frac{0{\cdot}3}{1{\cdot}3}} = 417{\cdot}3 \text{ K}$

Now, $\frac{p_2 v_2}{T_2} = \frac{p_2 v_2{}'}{T_2{}'}$

$\therefore \; v_2 = T_2 \times \frac{v_2{}'}{T_2{}'} = (35 + 273) \times \frac{0{\cdot}00344}{417{\cdot}3}$

$= 0{\cdot}00254 \text{ m}^3.$

Referring to fig. 11-18 and using eqn. (11.8*a*) for two-stage compression with *imperfect* intercooling,

$$\text{Total work required per cycle, } W = \begin{Bmatrix}\text{Work done in}\\ \text{L.P. cylinder}\end{Bmatrix} + \begin{Bmatrix}\text{Work done in}\\ \text{H.P. cylinder}\end{Bmatrix}$$

$$= p_1 v_1 \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + p_2 v_2 \frac{n}{n-1}\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= (1 \times 10^5) \times 0{\cdot}01189 \times \frac{1{\cdot}3}{0{\cdot}3}\left[\left(\frac{5}{1}\right)^{\frac{0\cdot3}{1\cdot3}} - 1\right] + (5 \times 10^5) \times 0{\cdot}00254 \times \frac{1{\cdot}3}{0{\cdot}3}\left[\left(\frac{25}{5}\right)^{\frac{0\cdot3}{1\cdot3}} - 1\right]$$

$= 4{,}800$ J/cycle.

Work required per second $= W \times \frac{N}{60} = 4{,}800 \times \frac{350}{60} = 28{,}000$ J/s or 28,000 W

∴ Power required = **28 kW**

Problem-11 : *A three-stage air compressor works between pressures of 100 kPa (100 kN/m²) and 5 MPa (5 MN/m²). For one m³ of air taken in, calculate :*

(a) the work required assuming conditions to be for maximum efficiency,

(b) the isothermal work required between the same pressure limits,

(c) the work required if the compressor was one-stage only,

(d) the percentage saving in work input to the compressor by using three-stages instead of single-stage, and

(e) the isothermal efficiency.

In parts (a) and (c), assume that the index of compression in each stage, $n = 1{\cdot}3$.

Referring to fig. 11-19 and using eqn. (11.16b) for maximum efficiency or minimum work,

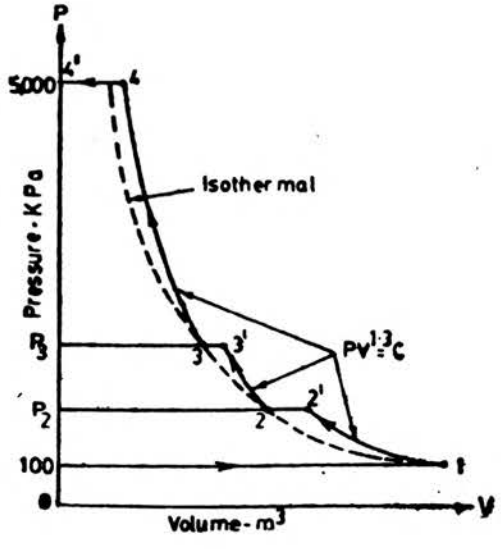


Fig. 11-19.

Work required for one m³ of air taken in,

$$W = \frac{3n}{n-1} \times p_1 v_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1\right] \text{ Joule.}$$

$$= \frac{3 \times 1{\cdot}3}{1{\cdot}3 - 1} \times (100 \times 10^3) \times 1 \left[\left(\frac{5{,}000}{100}\right)^{\frac{1\cdot3-1}{3\times1\cdot3}} - 1\right]$$

$= \mathbf{4{,}56{,}300 \ J}$

Alternative solution for work required in terms of p_1 and p_2.

Referring to fig. 11-19, pressures p_1, p_2, p_3 and p_4 are in geometric progression for maximum efficiency or minimum work.

Then, $p_4 = p_1 \times (R)^m$ where, R = pressure ratio of each stage and m = number of stages

i.e. $5{,}000 = 100 \times (R)^3 \quad \therefore \ R = \sqrt[3]{50} = 3{\cdot}69$

Using eqn. (11.16a), minimum work required for one m³ of air taken in,

$$W = \frac{3n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

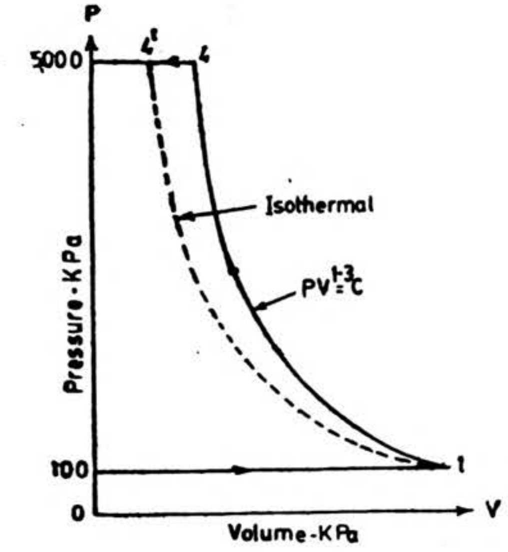


Fig. 11-20

$$= \frac{3 \times 1{\cdot}3}{1{\cdot}3 - 1} \times (100 \times 10^3) \times 1 \left[(3{\cdot}69)^{\frac{1{\cdot}3-1}{1{\cdot}3}} - 1\right]$$

$$= \mathbf{4{,}56{,}300\ J} \text{ (same as before)}$$

(b) Referring to fig. 11-20 and using eqn. (11-23), isothermal work required for one m^3 of air,

$$W = p_1 v_1 \log_3 \left(\frac{p_4}{p_1}\right)$$

$$= (100 \times 10^3) \times 1 \times \log_e \left(\frac{5{,}000}{100}\right) = \mathbf{3{,}91{,}210\ J}$$

(c) Referring to fig. 11-20, single-stage work required for one m^3 of air,

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1{\cdot}3}{1{\cdot}3 - 1} \times (100 \times 10^3) \times 1 \times \left[\left(\frac{5{,}000}{100}\right)^{\frac{1{\cdot}3-1}{1{\cdot}3}} - 1\right] = \mathbf{6{,}37{,}000\ J}$$

(d) Saving in work input to the compressor by adopting three-stages instead of single-stage compression = 6,37,000 − 4,56,300 = 1,80,700 J.

$$\therefore \quad \text{Percentage saving in work input} = \frac{1{,}80{,}700}{6{,}37{,}000} = 0{\cdot}2835 \text{ or } \mathbf{28{\cdot}35\%}$$

$$\text{(e) Isothermal efficiency} = \frac{\text{the isothermal work input}}{\text{the actual work input}} = \frac{3{,}91{,}210}{4{,}56{,}300} = 0{\cdot}856 \text{ or } \mathbf{85{\cdot}6\%}$$

11.7 Effect of Clearance on Volumetric Efficiency

The clearance space is provided in an actual compressor to safeguard the piston from striking the cylinder head or cylinder cover. The events taking place in a reciprocating compressor with clearance are the same as those taking place in a compressor without clearance. All the air compressed in the cylinder at the end of the compression stroke will not be discharged from it but some air will be left in the clearance space at the end of the delivery stroke 2-3 (fig. 11-21). The high pressure air, thus, left in the clearance space will re-expand along the curve 3-4 to the suction pressure (p_1) before the suction valve can open and the suction starts again. The volume of air drawn into the cylinder *without clearance* is equal to the dis-

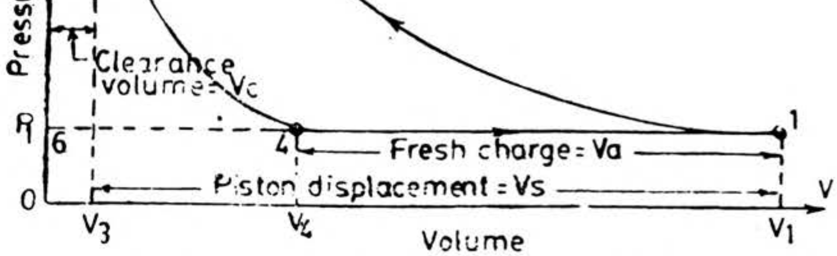


Fig. 11-21. Single-stage air compressor with clearance.

placement volume corresponding to full stroke v_s. However *with clearance,* the volume of fresh air drawn into the cylinder is only v_a corresponding to stroke 4-1.

Thus, the effect of clearance in a compressor is to decrease the amount of fresh air that can be drawn into the cylinder during the suction stroke. Therefore, there is a considerable reduction in the volumetric efficiency of the compressor due to clearance. In practice the clearance volume is limited to, two or three per cent of the displacement or swept volume (v_a) of the piston.

11.7.1 Expression for work done : Assuming that the value of the index n for expansion curve 3-4 (fig. 11-21) is the same as that for the compression curve 1-2, and making use of eqn. (11.2),

Net work required per cycle, W = area 1-2-3-4 = area 1-2-5-6 *minus* area 5-3-4-6

$$W = \frac{n}{n-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} p_1 v_4 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} p_1 (v_1 - v_4) \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{n}{n-1} p_1 v_a \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ Joule per cycle.} \qquad ..\ (11.28)$$

where pressures are measured in N/m^2 or p_a and volumes in m^3.

It may be noted that eqn. (11.28) is same as eqn. (11.2).

This shows that the work required to compress and deliver the same volume of air, v_a (volume of fresh air drawn in the cylinder) in a compressor with clearance, is same as that required in a compressor without clearance.

In other words, the introduction of clearance does not theoretically increase work of compression as the work done in compressing the clearance space air will be regained during the expansion of the clearance air from v_3 to v_4 at the beginning of the suction stroke.

Net work required in = $W \times \frac{N}{60}$ J/sec. or W where N = no. of cycles per min.

Indicated power of the compressor = $\frac{\text{watts}}{1{,}000}$ kW .. (11.29)

If $p_1 v_a$ in eqn. (11.28) is substituted by mRT_1, then net work required per *kg of air* may be written as

$$W = \frac{n}{n-1} RT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{ Joule.}$$

which is same as eqn. (11.4).

Net work required in J/sec.

= $W \times$ mass of air delivered per second J/sec. or W .. (11.30)

11.7.2 Expression for Volumetric Efficiency : Let v_c and v_s be the clearance volume and swept volume respectively of the compressor, p_2 = pressure in $N/m^2 = p_a$ of air in the clearance space, p_1 = pressure in $N/m^2 = p_a$ of clearance air at the end of expansion, and n is the index of expansion.

Referring to fig. 11-21, Volume of clearance air at the end of re-expansion,

$$v_4 = v_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = v_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

The volume of fresh charge of air, $v_a = v_1 - v_4$

$$v_a = v_1 - v_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} = v_s + v_c - v_c\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \quad [\because v_1 = v_s + v_c]$$

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

Volumetric efficiency, $\eta_v = \dfrac{v_1 - v_4}{v_s} = \dfrac{v_a}{v_s}$

$$= \frac{v_s - v_c\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1\right]}{v_s} = 1 - \frac{v_c}{v_s}\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1\right] \quad \text{.. (11.31)}$$

Thus, the volumetric efficiency depends upon the pressure ratio, p_2/p_1 *and the clearance ratio,* v_c/v_s. *Volumetric efficiency decreases as the pressure ratio increases. It also decreases as clearance volume (as a percentage of swept volume) increases or volumetric efficiency decreases as the clearance ratio increases.*

Problem-12 : *A compressor has 20 cm bore and 30 cm stroke. It has a linear clearance of 1·5 cm. Calculate the theoretical volume of air taken in per stroke when working between pressures of 1 bar and 7 bar. The index of compression and expansion is same and its value is 1·25.*

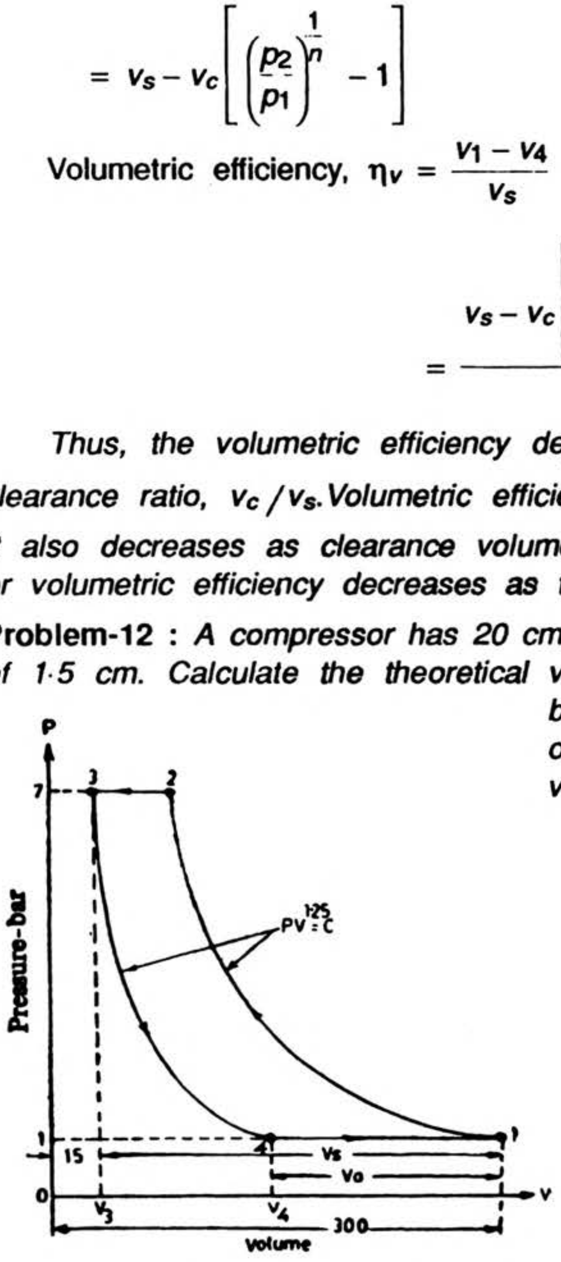


Fig. 11-22. Single-stage air compressor with clearance.

Referring to fig. 11-22,

Clearance volume $v_c = v_3$

v_3 = Area of cylinder × linear clearance.

$= \dfrac{\pi}{4} d^2 \times$ linear clearance

$= \dfrac{\pi}{4}(0{\cdot}2)^2 \times \dfrac{1{\cdot}5}{100} = 0{\cdot}000471 \text{ m}^3.$

Considering polytropic expansion (3-4),

$$p_3 v_3^{\,n} = p_4 v_4^{\,n}$$

$$\therefore\ v_4 = v_3\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}}$$

$$\therefore\ v_4 = 0{\cdot}000471 \times \left(\frac{7}{1}\right)^{\frac{1}{1{\cdot}25}} = 0{\cdot}002233\ \text{m}^3.$$

Now, v_1 = stroke volume, v_s + clearance volume $= \frac{\pi}{4} d^2 \times l + v_3$

$$= \frac{\pi}{4}(0{\cdot}2)^2 \times 0{\cdot}3 + 0{\cdot}000471 = 0{\cdot}009891\ \text{m}^3.$$

Theoretical volume of air taken in per stroke,

$$v_a = v_1 - v_4 = 0{\cdot}009891 - 0{\cdot}002233 = \mathbf{0{\cdot}007658\ m^3}.$$

Problem-13 : *The clearance volume in a single cylinder single-acting reciprocating compressor is 5% of the swept volume. Air is drawn in at 100 kPa (100 kN/m^2) and 311 K. Compression and expansion curves follow the law $pv^{1.2} = C$. The delivery pressure is 700 kPa (700 kN/m^2) and the atmospheric pressure and temperature are 101·325 kPa and 15°C respectively. Estimate :*

(a) the volumetric efficiency,

(b) the volumetric efficiency referred to atmospheric conditions, and

(c) the required work input per kg of air.

Take $R = 0{\cdot}287$ kJ/kg K for air.

(a) Referring to fig. 11-23,

Since, the index n is the same for both compression and expansion,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{But, } T_3 = T_2$$

$$\therefore\ T_4 = T_1 = 311\ \text{K}$$

Now, $p_3(v_3)^n = p_4(v_4)^n$

$$\therefore\ v_4 = v_3\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}}$$

$$= 0{\cdot}05 v_s \left(\frac{700}{100}\right)^{\frac{1}{1{\cdot}2}}$$

$$= 0{\cdot}05\ v_s \times 5{\cdot}06 = 0{\cdot}253 v_s$$

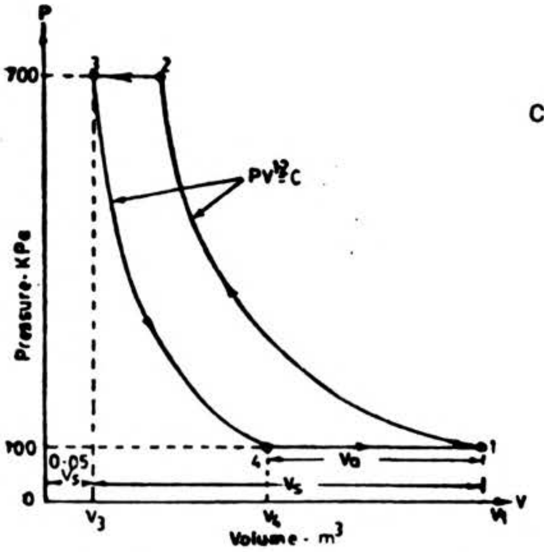


Fig. 11-23. Single-stage air compressor with clearance.

$$\therefore\ v_a = v_1 - v_4 = (1{\cdot}05 - 0{\cdot}253)v_s = 0{\cdot}797 v_s$$

Volumetric efficiency, $\eta_v = \frac{v_a}{v_s} = \frac{0{\cdot}797 v_s}{v_s}$

$$= 0{\cdot}797 \text{ or } 79{\cdot}7\%$$

Alternatively, using eqn. (11-31),

$$\text{Volumetric efficiency, } \eta_v = 1 - \frac{v_c}{v_s}\left[\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} - 1\right]$$

where p_2/p_1 is pressure ratio, v_c/v_s is the clearance ratio and n is the index of expansion.

$$\therefore\ \eta_v = 1 - \frac{0{\cdot}05v_s}{v_s}\left[\left(\frac{700}{100}\right)^{\frac{1}{1{\cdot}2}} - 1\right] = 0{\cdot}797 \text{ or } 79{\cdot}7\% \text{ (same as before)}$$

(b) Corresponding volume, v_s at 101·325 kPa and 15°C (atmospheric conditions)

$$= 0{\cdot}797v_s \times \frac{100}{101{\cdot}325} \times \frac{288}{311} = 0{\cdot}7283v_s$$

$$\left[\begin{array}{l}\text{Volumetric efficiency referred to}\\ \text{atmospheric conditions}\end{array}\right] = \frac{\text{volume of free air drawn in per stroke}}{\text{swept volume per stroke}}$$

$$= \frac{0{\cdot}7283v_s}{v_s} = 0{\cdot}7283 \text{ or } 72{\cdot}83\%$$

(c) Using eqn. (11.30) or eqn. (11.4), required work input per kg of air,

$$W = RT_1\frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \text{Joule}$$

$$= 287 \times 311 \times \frac{1{\cdot}2}{1{\cdot}2-1}\left[\left(\frac{700}{100}\right)^{\frac{1{\cdot}2-1}{1{\cdot}2}} - 1\right] = 2{,}05{,}110 \text{ J/kg of air}$$

Note : Work input/kg of air is not affected by clearance.

Problem-14 : *A double-acting, single cylinder air compressor runs at 100 r.p.m. The air is compressed from an initial pressure of 100 kPa (1 bar) to a delivery pressure of 750 kPa (7·5 bar). The stroke volume is 0·15 m^3 and law of compression and expansion is $pv^{1.25} = C$. If the clearance volume is 1/18th of the stroke volume, calculate :*

(a) the volume of air taken in per stroke, and

(b) the indicated power of the compressor.

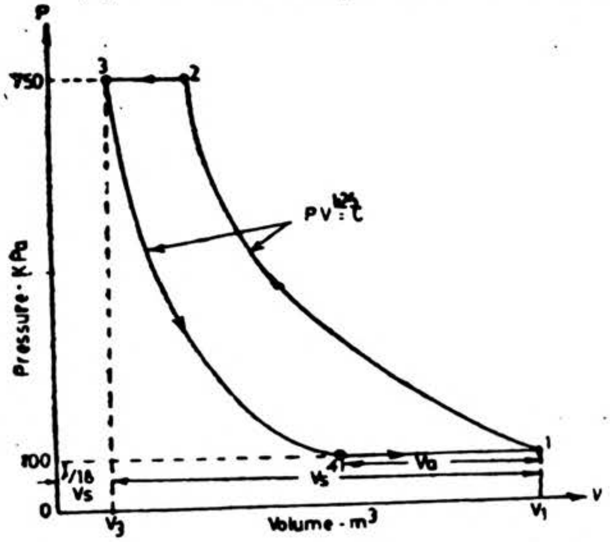


Fig. 11 24. Single air compression with clearance.

(a) Referring to fig. 11-24 and using eqn. (11.31), Volumetric efficiency,

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

$$= 1 - \frac{\frac{1}{18}v_s}{v_s}\left[\left(\frac{750}{100}\right)^{\frac{1}{1{\cdot}25}} - 1\right]$$

$$= 0{\cdot}7773 \text{ or } 77{\cdot}73\%$$

Volume of air taken in per stroke (or cycle), v_s = stroke volume, $v_s \times$ volumetric efficiency, η_v

$= 0{\cdot}15 \times 0{\cdot}7773 =$ **0·1166 m^3/cycle or stroke**

(b) Using eqn. (11.28), work required per cycle,

$$W = \frac{n}{n-1} p_1 v_a \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{Joule}$$

$$= \frac{1{\cdot}25}{1{\cdot}25 - 1} \times (100 \times 10^3) \times 0{\cdot}1166 \left[\left(\frac{750}{100} \right)^{\frac{1{\cdot}25-1}{1{\cdot}25}} - 1 \right]$$

= 28,920 Joule per cycle.

∴ Work required in Joules per second

$$= W \times \frac{N}{60} \text{ (where, } N = \text{no. of cycles per min.} = 2 \times 100 = 200)$$

$$= 28{,}920 \times \frac{200}{60} = 96{,}400 \text{ J/sec. or } 96{,}400 \text{ W}$$

∴ Indicated power of the compressor = 96,400 W or **96·4 kW.**

Problem-15 : *A single-cylinder air compressor compresses air from 1 bar to 7 bar. The clearance volume is 2 litres and law of compression and expansion is* $pv^{1.2}$ = *constant. If the volumetric efficiency of the compressor is 80 per cent, determine : (i) the stroke volume and (ii) the cylinder dimensions. Assume stroke of the piston equal to diameter of the cylinder.*

Here p_1 = 1 bar, p_2 = 7 bar, v_c = 2 litres, n = 1·2 and η_v = 80 per cent.

Using eqn. (11.31), Volumetric efficiency, $\eta_v = 1 - \frac{v_c}{v_s}\left[\left(\frac{p_2}{p_1} \right)^{\frac{1}{n}} - 1 \right]$

Substituting the values, wet get, $0{\cdot}8 = 1 - \frac{2}{v_s}\left[\left(\frac{7}{1} \right)^{\frac{1}{1{\cdot}2}} - 1 \right]$

i.e. $0{\cdot}8 = 1 - \frac{2}{v_s}[4{\cdot}06]$ or $\frac{2}{v_s} = \frac{1 - 0{\cdot}8}{4{\cdot}06} = \frac{0{\cdot}2}{4{\cdot}06}$

∴ $v_s = \frac{2 \times 4{\cdot}06}{0{\cdot}2}$ = **40·6 litres** = $40{\cdot}6 \times 10^3$ cm.

Now, $v_s = \frac{\pi}{4} \times d^2 \times d \qquad (\because l = d)$

i.e. $40{\cdot}6 \times 10^3 = \frac{\pi}{4} d^3$

∴ $d = \sqrt[3]{\frac{40{\cdot}6 \times 10^3 \times 4}{\pi}} = \sqrt[3]{51{,}693{\cdot}5} = 37{\cdot}25$ cm.

∴ Cylinder diameter, d = **37·25 cm,** and piston stroke, $l = d$ = **37·25 cm.**

Problem-16 : *A single-acting, single-cylinder air compressor is to deliver 15 kg of air per minute at 7 bar from suction conditions of 1 bar and 30°C. The clearance volume is 5% of the stroke volume and the law of compression and expansion is* $pv^{1.2}$ = *constant.*

The compressor is direct coupled to a four-cylinder, four-stroke petrol engine which runs at 1,400 r.p.m. with a brake mean effective pressure (b.m.e.p.) of 7 bar. If the stroke and bore are equal for both engine and compressor and mechanical efficiency of the compressor is 85 per cent, calculate the required cylinder dimensions of the compressor and engine.

Take R = 0·287 kJ/kg K for air.

Referring to fig. 11-25, and considering polytropic expansion 3-4,

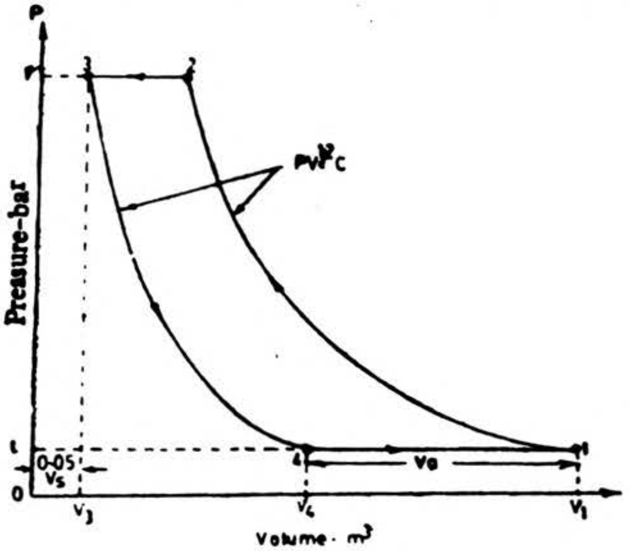


Fig. 11-25. Single-stage air compressor with clearance.

$$p_3 (v_3)^n = p_4 (v_4)^n$$

$$\therefore \; v_4 = v_3 \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}}$$

$$= 0{\cdot}05 \, v_s \times \left(\frac{7}{1}\right)^{\frac{1}{1{\cdot}2}}$$

$$= 0{\cdot}253 \; v_s$$

$$v_a = v_1 - v_4$$

$$= 1{\cdot}05 \; v_s - 0{\cdot}253 v_s$$

$$= 0{\cdot}797 v_s$$

Now, $pv_a = mRT$

(where v_a is the volume/min.)

$$\therefore \; v_a = \frac{mRT}{p} = \frac{15 \times 0{\cdot}287 \times (30 + 273)}{1 \times 10^2} = 13{\cdot}044 \text{ m}^3\text{/min.} = \frac{13{\cdot}044}{1{,}400} \text{ m}^3\text{/stroke.}$$

$$\therefore \; 0{\cdot}797 v_s = \frac{13{\cdot}044}{1{,}400} \qquad \therefore \; v_s = 0{\cdot}0117 \text{ m}^3 \text{ or } 11{,}700 \text{ cm}^3$$

Now, $v_s = \frac{\pi}{4} d^2 \times d = 11{,}700 \text{ cm}^3 \quad (\because l = d)$

$\therefore \; d^3 = 14{,}897 \qquad \therefore \; d = \sqrt[3]{14{,}897} = \mathbf{24{\cdot}61 \text{ cm}}$

$\therefore$ Piston stroke, $l = d =$ **24·61 cm** (compressor)

Using eqn. (11.30), W.D. per sec. $= \frac{n}{n-1} \, mRT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]$ J

(where m is the mass of air delivered/sec.)

$$= \frac{1{\cdot}2}{0{\cdot}2} \times \frac{15}{60} \times (0{\cdot}287 \times 10^3) \times 303 \left[\left(\frac{7}{1}\right)^{\frac{0{\cdot}2}{1{\cdot}2}} - 1 \right]$$

$$= 49{,}959 \text{ J/sec. or } 49{,}959 \text{ W}$$

Indicated power of the compressor = 49·959 kW

Brake power of the compressor

$$= \frac{\text{Indicated power}}{\text{Mechanical efficiency}} = \frac{49{\cdot}959}{0{\cdot}85} = 58{\cdot}775 \text{ kW (input power to compressor)}$$

But, brake power of engine,

$= p_m \times l \times a \times$ no. of working cycles of the *engine* per sec. × no. of cylinders

where p_m = brake mean effective pressure in pa,

l = length of engine piston stroke in metre, and

a = area of piston in m^2.

i.e. $(58{\cdot}775 \times 10^3) = (7 \times 10^5) \times d \times 0{\cdot}7854\ d^2 \times \frac{1{,}400}{120} \times 4$

$\therefore\ d^3 = 0{\cdot}002291 \quad \therefore\ d = 0{\cdot}1318$ m or **13·18 cm**, and l = **13·18 cm.**

Problem-17 : *A single-cylinder, single-acting air compressor is required to deliver 7 m^3 of free air per minute at a mean piston speed of 200 m per minute. The air is to be compressed from an initial pressure of 1 bar to a delivery pressure of 7·5 bar and the index of compression and expansion may be assumed to be 1·25. Assuming the stroke of piston to be 1·5 times the bore of the cylinder, clearance volume to be 1/18th of the swept volume per stroke, and suction pressure and temperature to be equal to the atmospheric air pressure and temperature, find the volumetric efficiency, bore, stroke and speed of the compressor.*

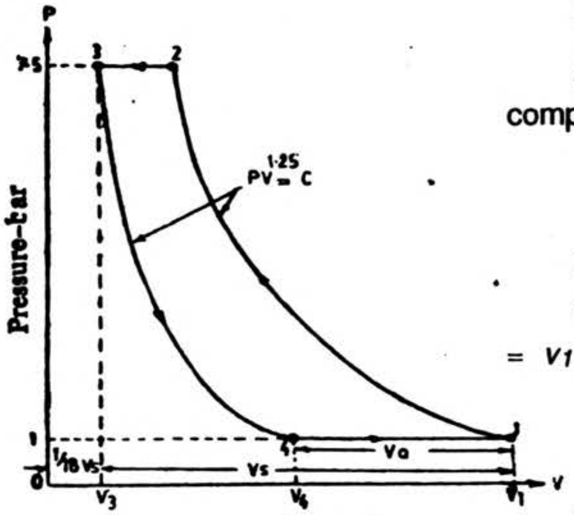


Fig. 11·26. Single-stage air compressor with clearance.

Referring to fig. 11-26.

Since the index n is the same for both compression and expansion,

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

But, $T_3 = T_2 \qquad \therefore\ T_4 = T_1$

Volume of free air drawn in per stroke V_a $= v_1 - v_4$

Now, $p_3 v_3^n = p_4 v_4^n$

$$\therefore\ v_4 = v_3\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = \frac{v_s}{18}\left(\frac{7{\cdot}5}{1}\right)^{\frac{1}{1{\cdot}25}}$$

$$= 0{\cdot}2782\ v_s$$

(where, free air pressure or atmospheric pressure is taken as 1 bar)

Now, $v_1 = v_s + \frac{1}{18} v_s = 1{\cdot}0555\ v_s$

$\therefore\ v_a = v_1 - v_4 = 1{\cdot}0555\ v_s - 0{\cdot}2782\ v_s = 0{\cdot}7773\ v_s$

$$\therefore \text{ Volumetric eff., } \eta_v = \frac{\text{Volume of free air drawn in per stroke, } v_a}{\text{Stroke volume, } v_s}$$

$$= \frac{0{\cdot}7773\ v_s}{v_s} = 0{\cdot}7773 \text{ or } 77{\cdot}73\%$$

Now, $v_s = \frac{\pi}{4} d^2 \times l = \frac{\pi}{4} d^2 \times 1{\cdot}5\, d = 1{\cdot}18\, d^3$

Also mean piston speed per min. = 200 m/min. = $2l$ × r.p.m.

$\therefore$ Speed of compressor, r.p.m. $= \frac{200}{2l} = \frac{200}{2 \times 1{\cdot}5d} = \frac{200}{3d}$

Volume of air delivered in m^3 per minute, $v_a = (v_1 - v_4) \times$ r.pm. $= 7$

$$\text{i.e. } 0{\cdot}7773\ v_s \times \text{r.p.m.} = 7$$

$$\text{i.e. } 0{\cdot}7773\ (1{\cdot}18 d^3) \times \frac{200}{3d} = 7$$

$\therefore$ **d = 0·3382 m or 33·82 cm.** Thus, stroke, $l = 1{\cdot}5 \times 0{\cdot}3382 =$ **0·5073 m or 50·73 cm.**

Speed of the compressor, r.p.m. $= \frac{200}{3d} = \frac{200}{3 \times 0{\cdot}3382} =$ **197 r.p.m.**

Problem-18 : *The following particulars refer to a two-stage, single-acting air compressor :*

Capacity, 5 m^3 per minute measured under free air conditions of 15°C and 1·01325 bar; Delivery pressure, 17 bar; Pressure during suction stroke, 0·98 bar; Temperature of air at the start of compression in each stage, 30°C; Clearance volume of low-pressure cylinder is 6% of the swept volume; Index of compression and expansion, 1·25 throughout; Speed, 125 r.p.m.

Assuming that the intercooler pressure is so chosen that theoretically the work is shared equally between the two cylinders, find : (a) the indicated power, and (b) the dimensions of the low-pressure cylinder if the bore is equal to the stroke. Take R = 287 J/kg K for air.

(a) Referring to fig.11-27,

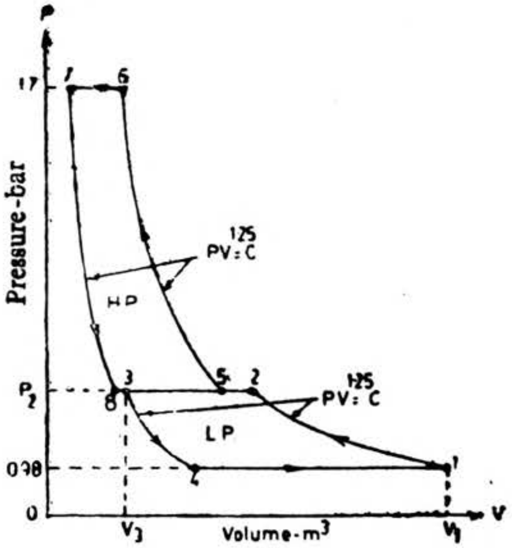


Fig. 11-27. Two-stage air compressor with clearance.

Mass of air dealt with,

$$m = \frac{pv}{RT} = \frac{(1{\cdot}01325 \times 10^5) \times 5}{287 \times 288}$$

$$= 6{\cdot}129 \text{ kg per minute.}$$

From eqn. (11.11) for work to be shared equally between the two cylinders, intercooler pressure is given by

$$p_2 = \sqrt{p_1 p_3} = \sqrt{0{\cdot}98 \times 17} = 4{\cdot}08 \text{ bar.}$$

$$\text{Now, } \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

$$\therefore\ T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 303 \left(\frac{4{\cdot}08}{0{\cdot}98}\right)^{\frac{0{\cdot}25}{1{\cdot}25}}$$

$$= 303 \times 1{\cdot}33 = 403 \text{ K.}$$

Work done in L.P. cylinder per *kg of air* is given by

$$W = \frac{n}{n-1} R\,(T_2 - T_1) = \frac{1{\cdot}25}{0{\cdot}25} \times 287\,(403 - 303) = 1{,}43{,}500 \text{ J.}$$

Since, the work is equally shared between the two cylinders,

The total work done per kg of air = 1,43,500 × 2 = 2,87,000 J

Work required in Joule per sec. = work done per kg of air × mass of air per sec.

$$= 2{,}87{,}000 \times \frac{6{\cdot}129}{60} = 29{,}320 \text{ J/sec. or } 29{,}320 \text{ W}$$

∴ Indicated power = **29·32 kW.**

(b) Let stroke volume of L.P. cylinder, $v_1 - v_3 = 100$ m^3, then $v_3 = 6$ m^3 and $v_1 = 106$ m^3 (fig. 11-27).

Mass of air dealt with per stroke, for a single-acting compressor with 100 m^3 stroke volume, $m = m_1 - m_3$.

$$m_1 = \frac{p_1 v_1}{RT_1} = \frac{0{\cdot}98 \times 10^5 \times 106}{287 \times 303} = 119{\cdot}46 \text{ kg (mass of air compressed)}$$

$$m_3 = \frac{p_3 v_3}{RT_3} = \frac{4{\cdot}08 \times 10^5 \times 6}{287 \times 403} = 21{\cdot}17 \text{ kg (mass of air left in the cylinder)}$$

∴ $m = m_1 - m_3 = 119{\cdot}46 - 21{\cdot}17 = 98{\cdot}29$ kg per 100 m^3 stroke volume and assuming that the temperature remains constant during delivery, i.e. from point 2 to 3,

$$\text{Actual stroke volume per minute} = \frac{100}{98{\cdot}29} \times 6{\cdot}129 = 6{\cdot}236 \text{ m}^3\text{/min.}$$

Now, stroke volume per min. $= \frac{\pi}{4} d^2 \times d \times r.p.m.$

i.e. $6{\cdot}236 = \frac{\pi}{4} d^2 \times d \times 125$ (the bore being equal to the stroke)

∴ $d^3 = 0{\cdot}0636$ ∴ $d =$ **0·3991 m** or **39·91 cm**, and $l = d =$ **39·91 cm.**

Problem-19 : *A two-stage air compressor delivering air at 17·5 bar has a clearance volume of 4% of the swept volume. The atmospheric conditions are 1·01 bar and 18°C, and at the start of compression the pressure in the cylinder is 1 bar. The temperature at the start of compression in each stage is 30°C, and the intercooler pressure is 4·04 bar. The law of compression and expansion for both stages is $pv^{1.25}$ = constant.*

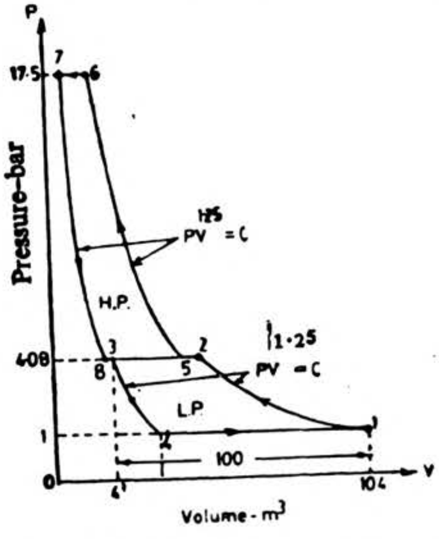


Fig. 11-28. Two-stage air compression with clearance.

Find : (a) the volumetric efficiency referred to atmospheric conditions,

(b) the work required per kg of air delivered by the compressor, and

(c) the isothermal efficiency referred to isothermal compression from atmospheric temperature and pressure. (R = 0·287 kJ/kg K for air).

Referring to fig. 11-28,

Let the stroke volume, $v_1 - v_3 = 100$ units,

then $v_3 = 4$ units and $v_1 = 104$ units.

$T_1 = T_5 = (30 + 273) = 303$ K

(∵ cooling is perfect)

(a) Considering polytropic compression 1-2 in L.P. cylinder.

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

$$\therefore\ T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 303 \times \left(\frac{4{\cdot}04}{1}\right)^{\frac{1{\cdot}25-1}{1{\cdot}25}} = 400{\cdot}5\text{K}$$

Now, $p_1 v_1^n = p_2 v_2^n$ or $\frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$

$$\therefore\ v_2 = \frac{v_1}{\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}} = \frac{104}{\left(\frac{4{\cdot}04}{1}\right)^{\frac{1}{1{\cdot}25}}} = \frac{104}{3{\cdot}056} = 34{\cdot}03 \text{ units}$$

$\therefore$ Volume of air delivered $= v_2 - v_3 = 34{\cdot}03 - 4 = 30{\cdot}03$ units measured at 4·04 bar and 400·5 K,

also $\frac{pv}{T}$ (measured conditions) $= \frac{p_2 v_2}{T_2}$ (free air conditions)

$\therefore$ Volume of free air (at atmospheric conditions i.e. at 1·01 bar and 291 K) delivered

$$= 30{\cdot}03 \times \frac{4{\cdot}04}{1{\cdot}01} \times \frac{291}{400{\cdot}5} = 87{\cdot}28 \text{ units.}$$

$$\left[\begin{array}{l}\text{Volumetric efficiency referred}\\ \text{to atmospheric conditions}\end{array}\right] = \frac{\text{Volume of free air delivered per stroke}}{\text{Swept volume of L.P. cylinder per stroke}}$$

$$= \frac{87{\cdot}28}{100} = 0{\cdot}8728 \text{ or } 87{\cdot}28\%$$

(b) Work done per *kg of air* in L.P. cylinder

$$= \frac{n}{n-1} R(T_2 - T_1) = \frac{1{\cdot}25}{1{\cdot}25-1} \times 0{\cdot}287\,(400{\cdot}5 - 303) = 139{\cdot}91 \text{ kJ}$$

Considering polytropic compression 5-6 in H.P. cylinder, $\frac{T_6}{T_5} = \left(\frac{p_6}{p_5}\right)^{\frac{n-1}{n}}$

$$\therefore\ T_6 = T_5\left(\frac{p_6}{p_5}\right)^{\frac{n-1}{n}} = 303\left(\frac{17{\cdot}5}{4{\cdot}04}\right)^{\frac{0{\cdot}25}{1{\cdot}25}} = 303 \times 1{\cdot}341 = 406{\cdot}3 \text{ K}$$

Work done per *kg of air* in H.P. cylinder

$$= \frac{n}{n-1} R\,(T_6 - T_5) = \frac{1{\cdot}25}{1{\cdot}25-1} \times 0{\cdot}287\,(406{\cdot}3 - 303) = 148{\cdot}24 \text{ kJ}$$

Therefore, total work required per kg of air

= work required in L.P. cylinder + work required in H.P. cylinder

= 139·91 + 148·24 = **288·15 kJ per kg of air.**

(c) Isothermal work done per kg of air for the same range of pressure and compression from atmospheric pressure and temperature,

$$W = R \times T_{atm} \times \log_e \left(\frac{p_6}{p_{atm}} \right)$$

$$= 0{\cdot}287 \times 291 \times \log_e \left(\frac{17{\cdot}5}{1{\cdot}01} \right) = 0{\cdot}287 \times 291 \times 2{\cdot}852 = 238{\cdot}19 \text{ kJ per kg of air.}$$

$$\therefore \text{ Isothermal efficiency} = \frac{\text{Work done when compression is isothermal per kg of air}}{\text{Actual work done on the air per kg of air}}$$

$$= \frac{238{\cdot}19}{288{\cdot}15} = \mathbf{0{\cdot}8266 \text{ or } 82{\cdot}66\%}$$

11.8 Actual Indicator Diagram of a Single-Stage Air Compressor

Actual air compressor differs from the ideal compressor in many respects, because the cylinder, the valves and the piston, store and release energy in the form of heat and further because air is throttled through the valves. An indicator diagram of an actual air compressor is shown in fig. 11-29 superimposed upon an ideal one (shown dotted) for the same compressor. The air pressure is to be raised from atmospheric pressure p_1 to a receiver pressure p_2.

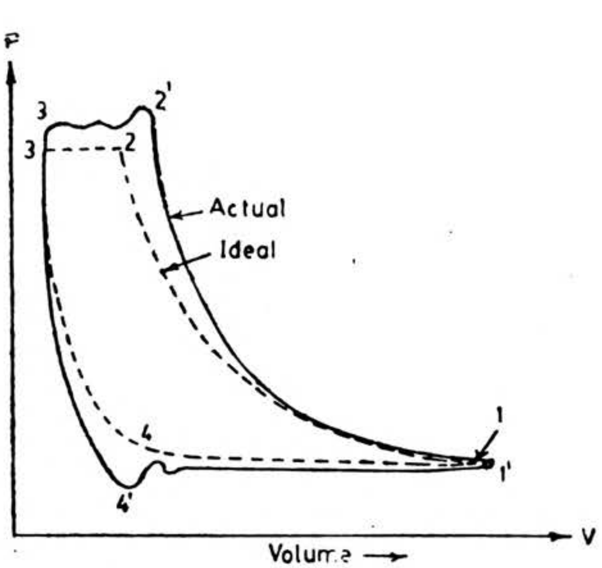


Fig. 11 29. Actual indicator diagram of single-stage air compressor.

The ideal indicator diagram is drawn for isothermal compression of air in the cylinder and for isothermal expansion of the clearance air. In actual compressor, isothermal compression is not obtained, instead polytropic compression is obtained, i.e. the compression falls between the adiabatic (isentropic) and the isothermal, but it is closer to adiabatic as indicated by compression line 1' – 2' in fig. 11-29. The value of index n varies from 1·25 to 1·35. In actual compressor, the clearance air will become cooler than the cylinder walls with which it is in contact hence it will receive heat during the latter part of the expansion process. The expansion line in actual compressor lies between an adiabatic and an isothermal, but generally it is closer to isothermal. The actual expansion line obtained will have shape and position similar to line 3' – 4' instead of line 3 – 4.

In ideal air compressor, the intake valve will open as soon as the clearance air pressure has decreased to atmospheric but actually the intake valve does not open until the pressure drops, a little below atmospheric. After the valve is open and air is in motion, there are generally several oscillations of the valve and the air column as indicated by the wavy portion on the suction line 4' – 1'. The discharge valve does not open till the pressure attained is slightly above that in the receiver; and after opening, it behaves much like the intake valve for similar reason. The discharge line actually obtained will have a shape and position similar to line 2' – 3' instead of line 2 – 3.

There is always some leakage past the valves, piston rings and piston rod packings in the actual compressor; and greater the leakage the larger will be the work required for any given mass of air delivered.

The net result of the actual compressor has been to deliver a smaller mass of *air than that handled by the ideal compressor and to require an expenditure of work in excess of that of the ideal one as shown by greater area.*

11.9 Reciprocating Compressed Air Motor

Air motor is in effect a reversed air compressor. The compressed air to be used in an air motor is taken from the compressor reservoir (receiver). The most common form of compressed air motor is the cylinder and double-acting piston type. The air is admitted into the motor cylinder through a mechanically operated inlet valve and drives the piston forward but after a portion of the stroke of the piston has been performed, the air supply is cut-off and the stroke is completed under decreasing pressure as the air expands in the cylinder. After the expansion stroke is completed, the air which has done the work is allowed to escape into atmospheric through a mechanically operated discharge valve. The return stroke is performed by compressed air acting on the other side of the piston. A motor of this type works like a reciprocating double-acting steam engine.

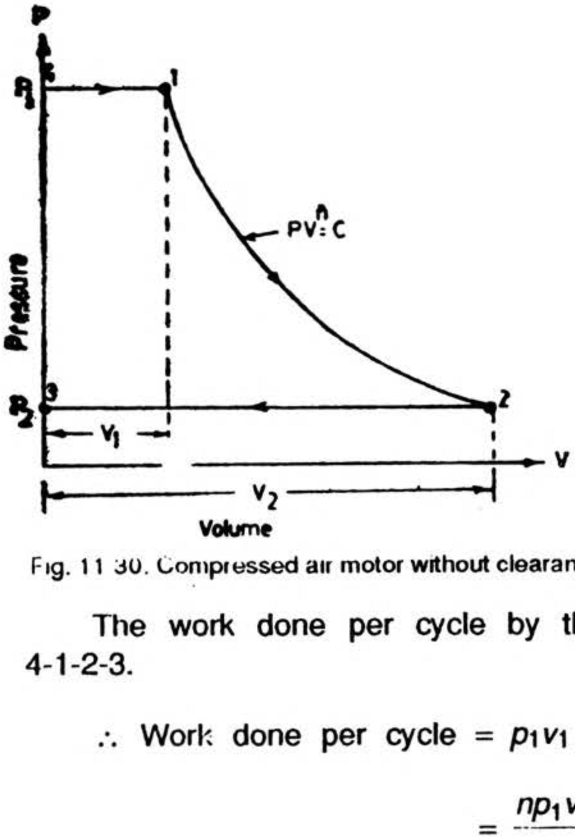


Fig. 11-30. Compressed air motor without clearance.

The important application of air motor is the use in mines where use of electric motor is dangerous. There will be a fall in the pressure of air due to friction in the pipe, the fall being greater, the greater the distance of the air motor from the compressor reservoir (receiver).

Figure 11-30 shows the pressure-volume diagram for a compressed air motor (air engine) in which clearance is neglected. Air is admitted at high pressure p_1 from 4-1, cut off takes place at point 1. From 1 to 2 air expands from pressure p_1 to atmospheric pressure p_2 and expansive work is done. The law of expansion is polytropic, i.e. $pv^n = C$. Exhaust takes place from 2 to 3.

The work done per cycle by the air motor with no clearance is given by area 4-1-2-3.

$$\therefore \text{ Work done per cycle } = p_1v_1 + \frac{p_1v_1 - p_2v_2}{n-1} - p_2v_2$$

$$= \frac{np_1v_1 - p_1v_1 + p_1v_1 - p_2v_2 - np_2v_2 + p_2v_2}{n-1}$$

$$= \frac{n}{n-1}\left[p_1v_1 - p_2v_2\right]$$

$$= \frac{n}{n-1}\,p_2v_2\left[\frac{p_1v_1}{p_2v_2} - 1\right] \text{ Joule}$$

But for polytropic expansion $1 - 2$,

$$p_1{v_1}^n = p_2{v_2}^n \quad \therefore \quad \frac{v_1}{v_2} = \left(\frac{p_1}{p_2}\right)^{-\frac{1}{n}}$$

Substituting the value of $\frac{v_1}{v_2}$ in the above equation, we get, work done by air,

$$W = \frac{n}{n-1} p_2 v_2 \left[\left(\frac{p_1}{p_2} \right)^{\frac{n-1}{n}} - 1 \right] \text{ Joule per cycle} \qquad .. (11.32)$$

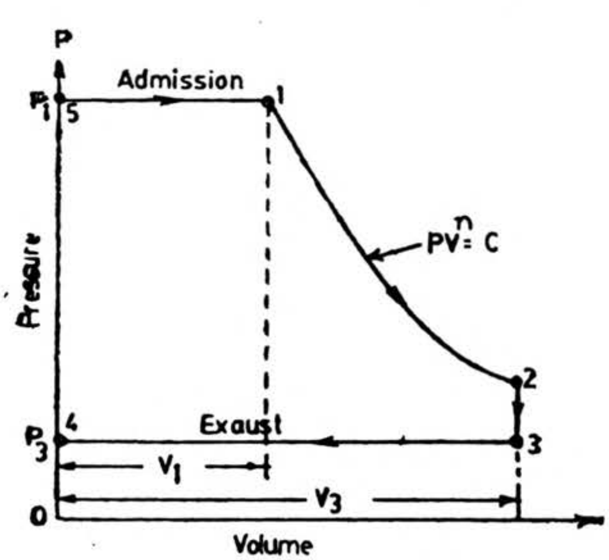


Fig. 11-31.

The fig. 11-31 shows the pressure-volume diagram in which air is admitted at high pressure p_1 from 5 to 1, cut-off takes place at point 1, expansive work is done from 1 to 2 and release takes place (before the atmospheric pressure reached) from 2 to 3 and then exhaust takes place from 3 to 4.

The work done per cycle by the air motor, with no clearance, is given by area 5-1-2-3-4.

Work done by air motor per cycle

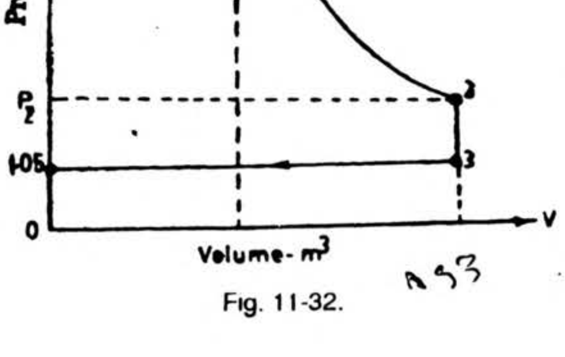
= area 5-1-2-3-4

$$= p_1 v_1 + \frac{p_1 v_1 - p_2 v_2}{n-1} - p_3 v_3 \qquad .. (11.33)$$

where, pressures are measured in Pa and volumes in m^3.

Problem-20 : *Air at a pressure of 9 bar enters the cylinder of a compressed air motor; the supply is cut-off at ¼th stroke and the air expands according to the law $pv^{1.3}$ = constant. Neglecting the effect of clearance, find the temperature of air at the end of expansion when the temperature of the compressed air is 25°C, and find the volume for a single-acting air motor to develop indicated power of 5 kW at a speed of 260 r.p.m. when the exhaust is at 1·05 bar. Also find the air motor cylinder dimensions assuming bore = stroke.*

Assume R for air = 0·287 kJ/kg K.



A 53

Fig. 11-32.

Referring to fig. 11-32, and considering polytropic expansion 1-2,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1} \right)^{n-1} = (r)^{n-1}$$

$$\therefore \ T_2 = \frac{T_1}{(r)^{n-1}} = \frac{25 + 273}{(4)^{0.3}} = 197 \text{ K}$$

i.e. t_2 = 197 − 273 = **−76°C,** temperature of air at the end of expansion.

It may be noted that the temperature of air at the end of expansion is −76°C which is much below the freezing point. When the temperature falls below the freezing point, the moisture in the air is frozen and, the snow formed may seriously interfere with the working of the discharge (exhaust) valve of the air motor.

The most satisfactory method of preventing too low a temperature at the end of expansion is to heat the air (before the air enters the motor cylinder) at constant pressure by passing it through a suitable pre-heater. By pre-heating the air, not only the freezing effect is prevented but the volume of air is increased, the volume being proportional to the absolute temperature and consequently a large proportion of the heat expended (used up) in the heater is converted into work in the motor cylinder, i.e. the work done in the motor cylinder, for the consumption of same mass of air increases, when the air is pre-heated.

Now, $p_1 v_1^n = p_2 v_2^n \quad \therefore \ p_2 = \dfrac{p_1}{\left(\dfrac{v_2}{v_1}\right)^n} = \dfrac{p_1}{(r)^n} = \dfrac{9}{(4)^{1.3}} = 1.485$ bar

Using eqn. (11.33), Work done/stroke $= p_1 v_1 + \dfrac{p_1 v_1 - p_2 v_2}{n-1} - p_3 v_3$

$$= 10^5 v_2 \left[\frac{p_1 v_1}{v_2} + \frac{\dfrac{p_1 v_1}{v_2} - p_2}{n-1} - p_3 \frac{v_3}{v_2} \right]$$

$$= 10^5 v_2 \left[\frac{9}{4} + \frac{\dfrac{9}{4} - 1.485}{0.3} - 1.05 \right] = 3{,}75{,}000 \ v_2 \ \text{ J per stroke}$$

Indicated power (work done in J/sec or W) = Work done per stroke × $\dfrac{N}{60}$

i.e. $5 \times 10^3 = 3{,}75{,}000 \times v_2 \times \dfrac{260}{60}$ (for a single-acting motor)

$\therefore \ v_2 = 0.003077 \text{ m}^3.$

But, $v_2 = \dfrac{\pi}{4} d^2 \times l \quad \therefore \ 0.003077 = \dfrac{\pi}{4} \times d^3 \quad (\because \ l = d)$

$$\therefore \ d^3 = \frac{0.003077}{0.7854} = 0.0039177$$

$\therefore$ Dia. of cylinder, $d = \sqrt[3]{0.0039177} \approx$ **0.1576 m** or **15.76 cm** and stroke, $l = d$ = **15.76 cm**.

11.10 Classification of Air Compressors

The compressors can be mainly classified as, depending on the type of mechanism, the nature of their motion, the flow of fluid, etc. Fig. 11-33 shows the classification of compressors.

In a positive displacement compressor, displacement of a fixed amount of fluid (positively contained during its passage through the machine), takes place. In that type, work changes are effected by the displacement of a boundary, i.e., by pressure and volume changes. All reciprocating compressors are positive displacement type, while rotary compressors may be positive displacement or non-positive displacement type.

In reciprocating positive displacement compressors, the displacement of a boundary reduces the volume and increases the pressure. In *rotary positive displacement* compressors the increase in pressure is by backflow of fluid as in Root's blower, or both by variation in volume and by backflow as in vane type blower.

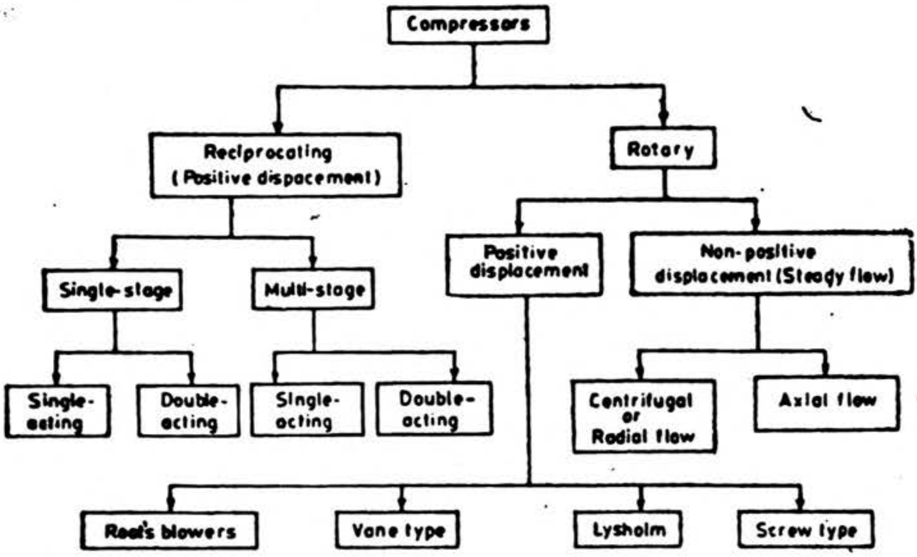


Fig. 11 33.

In *non-positive displacement or steady flow* compressors, the fluid is continuously in a steady-flow through the machine, undergoing changes in pressure primarily by means of dynamic effects. In this type of compressor, work changes are effected by means of velocity changes. Therefore, operating speed affects directly the operating efficiency of a non-positive displacement compressor; while operating speed, within reasonable limits, has little effect on the efficiency of a positive displacement compressor.

Reciprocating compressors have the following *advantages* over non-positive displacement rotary compressors :

- Capacity of delivering higher pressures,
- Greater flexibility in capacity and pressure range,
- Somewhat higher efficiency for pressure ratios > 2, and
- Less susceptibility to change in gas composition and density.

Reciprocating compressors are suitable for delivering small quantities of air at high pressure. The maximum free air delivery is limited to 300 m^3/min but the pressure may be as high as 1,000 bar. It is a low speed machine due to the inertia of the reciprocating parts.

Because of large quantities of air at relative low pressure (3 to 10 bar) required in gas turbine plants, reciprocating compressors, though they have above mentioned advantages are not employed.

Positive displacement rotary or displacement blowers are also not suitable for gas turbine plants as they, in general, are noisy, subject to wear, and also of limited capacity compared with the non-positive displacement or steady flow rotary compressors (centrifugal and axial flow compressors).

Non-positive displacement rotary or steady flow compressors, *Centrifugal* and *axial flow* compressors, offer the following advantages :

.. Handle large quantities of air or gas, at relatively lower pressure,

.. Being high speed machines, they are small in size,

.. Provide uniform delivery without requiring a large receiver,

.. Deliver clean air as there are fewer sliding parts requiring lubrication,

.. Present no balancing problems,

.. Lower maintenançe expenses, and

.. Less operating attention.

11.11 Rotary Air Compressors

Depending on their pressure ratio, $r_p = \frac{p_2}{p_1}$, rotary air compressors may be classified as follows :

Fans (r_p = 1 to 1·1) – The main purpose of a fan is to move air or to circulate air through the ducts of an air conditioning system and to supply low-pressure air blast. As the fluid passing through the unit does not suffer appreciable change in density, it can be treated as incompressible like water. Principally, fans are axial-flow compressors without casing around the impeller periphery.

Blowers (r_p = 1·1 to 4) *and compressors* ($r_p > 3$) – If compressibility effect is considerable, it is to be taken into account while designing the unit and the unit is known as *blower* or a *compressor* proper depending on the pressure ratio. Blower may be positive displacement rotary compressor (Root's blower, lysholm, vane type, screw type, etc.) or non-positive displacement rotary compressor (centrifugal compressor).

The purposes of a blower is to furnish forced draught to furnaces, besemer convertors, cupolas, etc.

The purpose of compressor (centrifugal or axial flow) is to supply compressed air in large quantity, such as in gas turbine plants.

11.12 Non-positive Displacement Rotary or Steady Flow Compressors

Centrifugal and *axial flow* compressors come under this heading. In all forms of turbo-machinery, whether expanders (called turbines or centrifugal and axial flow compressors), work changes are effected by means of velocity changes instead of by displacement changes of a boundary, such as with a piston. If the air enters at the centre axially and flows radially outward towards the circumference, the compressor is known as *centrifugal compressor.* On the other hand if the air flows parallel to the shaft, it is called *axial flow compressor.*

11.12.1 Centrifugal Air Compressors : A centrifugal compressor, mainly, consists of impeller and diffuser. The impeller consists of an impeller disc and impeller vanes, attached on the impeller disc radially, forming radial diverging passages as shown in fig. 11-34(a). The impeller rotates with the shaft at high speed and air is drawn into the impeller eye in an axial direction. The air then flows radially outwards through the impeller passages due to centrifugal force, and kinetic energy in imparted to the air with some static pressure rise as shown in fig. 11-34(b). The remainder of the pressure rise is obtained in the diffuser. The diffuser which is stationary, consists of a number of fixed diverging passages. The air leaves the impeller tip with high velocity and enters the diffuser. The diffuser, in its fixed diverging passages, reduces the high velocity. Thus, by the diffusion process or deceleration of air in the diffuser, kinetic energy is converted into pressure energy.

The flow from the diffuser is collected in a spiral passage from which it is discharged from the compressor.

Single-stage centrifugal compressor can develop a pressure ratio as high as 4 : 1. Pressure ratio as high as 10 : 1 can be developed with the help of multi-stage centrifugal compressors.

The impeller may be single or double sided. The impeller illustrated in fig. 11-34(a) is single sided one. The single sided impeller sucks in air from one side only. A double

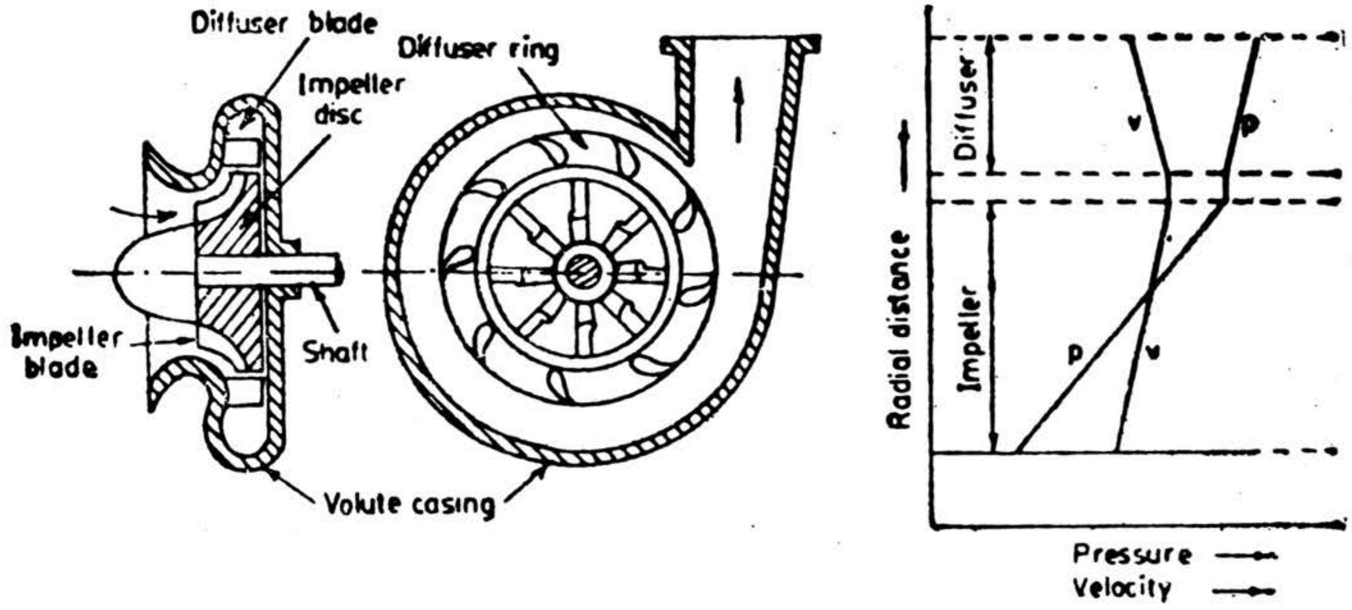


Fig. 11-34.

(b)

sided impeller consists of vanes on both sides and air is drawn in from both sides. A double sided impeller is used often when larger flows are to be handled.

The centrifugal compressor has the following *advantages* as compared with axial flow compressor :

(i) It is more rugged.

(ii) It is simpler.

(iii) It is less expensive.

(iv) It is smaller in length.

(v) It attains higher pressure ratio per stage.

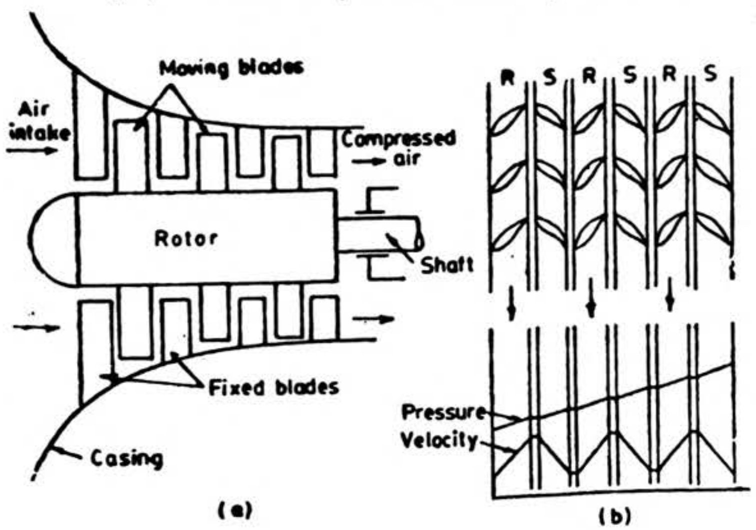
(vi) It is not liable to loss of performance by the effects of deposits left on the surfaces of flow passages of air when working in a contaminated atmosphere.

(vii) It is able to operate efficiently over a wider range of mass flow at any particular speed.

However, it suffers from *disadvantages* of large frontal area, lower maximum efficiency, and less adaptability to multi-staging.

Centrifugal compressor is mainly used in, (i) Oil refinery, (ii) Petrochemical plants, (iii) Natural gas transmission system, (iv) Refrigeration plants, (v) Supercharging of Petrol and Diesel engines, etc.

11.12.2 Axial Flow Air Compressors : Axial flow compressor consists of a rotor and a casing or stator. Blades mounted on the rotor



(a)

(b)

Fig. 11-35.

are moving blades (rotor blades) and blades fixed on the inner face of the casing are known as stationary blades (stator blades). Air enters the blades axially and also leaves them in the axial direction as shown in fig. 11-35(a).

An axial flow compressor is designed according to the reaction principle (diffusion in both rotor and stator). Thus, the rise in pressure through the stage (rotor and stator) is in general, attributable to both blade rows, moving as well as fixed. As a measure of the extent to which the rotor itself contributes to this pressure rise, the term "degree of reaction" is used. Thus, degree of reaction is defined as the ratio of pressure rise in rotor to the total pressure rise.

A single-stage compressor (one row of moving blades and one row of fixed blades) does not give appreciable pressure ratio. Axial flow compressor are mostly multi-stage compressors. Pressure ratio which can be produced per stage of an axial flow compressor is 1 : 2.

Figure 11-35(a) represents a three-stage, axial flow compressor. The moving blades, receive the air and increase its velocity, and also act as a diffuser to increase the pressure, while the fixed blades continue the diffuser action. Figure 11-35(b) shows the increase in pressure of air during its passage through rotor blades (R) and stator blades (S).

Axial flow type compressors are given preference over the centrifugal type in the application of aircraft and industrial gas turbine power plants. This is because axial flow compressor has a higher efficiency, less frontal area, and is capable of producing higher pressure ratio on a single shaft by increasing the number of stages. However, the axial flow compressors run at lower speeds, their weight is greater, have higher starting torque, are sensitive to any deposit formation on blades, and are complicated as compared with centrifugal compressors.

Tutorial - 11

1. (a) Estimate the amount of work required for compression of one kg of air for single-stage compression.

   (b) What are the uses of high pressure air?

   (c) A single-stage, single-acting air compressor deals with 85 m^3 of free air per hour at 1·01325 bar and 15°C. The pressure and temperature in the cylinder during the suction stroke remains constant at 1 bar and 40°C respectively. The index of compression is 1·22 and the delivery pressure is 5·5 bar. If the mechanical efficiency is 85%, find, neglecting clearance volume, the power required to drive the compressor.

   [6·107 kW]

2. A single-acting, single-cylinder, air compressor has a cylinder diameter 18 cm and a stroke of 25 cm. Air is drawn into the cylinder at 102 kPa (102 kN/m^2), 288 K. It is then compressed isentropically to 632 kPa (632 kN/m^2).

   Find : (a) the power required to drive the compressor if its speed is 120 r.p.m; and (b) the mass of air compressed per hour. Take $\gamma = 1{\cdot}4$ and $R = 0{\cdot}287$ kJ/kg K for air. Neglect clearance effects.

   [3·112 kW; (b) 56·54 kg/hr.]

3. It is desired to compress 15 m^3 of air per minute at 100 kPa (100 kN/m^2) and 297 K to 700 kPa (700 kN/m^2) in a single-stage, single-acting air compressor. Calculate the power required to drive the compressor if the compression is :

   (a) Isothermal,

   (b) Polytropic with index $n = 1{\cdot}3$, and

   (c) Isentropic.

   Take $\gamma = 1{\cdot}4$ for air. Neglect clearance effects.

   [(a) 48·648 kW; (b) 61·533 kW; (c) 65·1 kW]

4. (a) A single-stage, single-acting air compressor is belt driven from an electric motor at 400 r.p.m. The cylinder diameter is 15 cm and stroke is 18 cm. The air is compressed from 1·05 bar to 7·5 bar and the law of compression is $pv^{1.3}$ = constant. Find the power of the motor, if the transmission efficiency is 97% and the mechanical efficiency of the compressor is 90%. Neglect the clearance effects.

[6·354 kW]

(b) A single-cylinder, double-acting air compressor is of 24 kW. The law of compression is $pv^{1.2}$ = constant. The air is compressed from 1 bar to 8 bar. The compressor runs at 200 r.p.m. and average piston speed may be taken as 160 m/min. Find the dimensions of the cylinder required. Neglect the effect of clearance.

[d = 21·47 cm; l = 0·4 m]

5. (a) Define volumetric efficiency of a compressor.

(b) Find the diameter and stroke of a single-stage, double-acting air compressor from the following data : Capacity = 20 m^3 of free air per minute at atmospheric pressure and temperature; Delivery pressure = 7·5 bar; Inlet pressure = 1 bar; Inlet temperature = 32°C; Atmospheric pressure = 1·01325 bar; Atmospheric temperature = 15°C; Speed = 300 r.p.m.; Index of compression = 1·3; Stroke : bore = 0·9 : 1.

Also find the indicated power of the compressor. Neglect clearance effects.

[d = 36·99 cm; l = 33·291 cm; 91·753 kW]

6. A two-stage, single-acting air compressor draws air at 1·05 bar and temperature of 16°C. The air is compressed in the L.P. cylinder to a pressure of 5·25 bar. After compression, the air is cooled in the intercooler at constant pressure to a temperature of 30°C, before being taken to H.P. cylinder from which it is delivered at 20 bar. The compression is carried out each cylinder according to the law $pv^{1.35} = C$. Find the indicated work required in kJ to compress one kg of air. Take R = 0·287 kJ/kg K for air. Neglect clearance.

[304·92 kJ]

7. What are the conditions for obtaining maximum efficiency in the case of a two-stage reciprocating air compressor ?

A two-stage, single-acting air compressor deals with 5·75 m^3 of air per minute under atmospheric conditions. It delivers air at 40 bar. The compressor runs at 300 r.p.m. The stroke is equal to L.P. cylinder diameter. Assuming complete intercooling and compression in each stage according to the law $pv^{1.35} = C$ and mechanical efficiency of 85 per cent, calculate the cylinder diameters and minimum power required to drive the compressor. Assume atmospheric conditions as 1·01325 bar and 15°C. Neglect clearance.

[$d_{L.P.}$ = 29 cm; $d_{H.P.}$ = 11·56 cm; 53·758 kW]

8. A two-stage, single-acting air compressor takes in 3 m^3 of air per minute at atmospheric pressure of 1 bar abd temperature of 288 K. It delivers air at 35 bar (3·5 MPa). The compression is carried out in each cylinder according to the law $pv^{1.25} = C$. Assuming complete intercooling and mechanical efficiency of 80 per cent, calculate the minimum power required to drive the compressor and the heat carried away from the intercooler per minute by cooling water. If the inlet and outlet temperatures of the cooling water are 15°C and 40°C, calculate the amount of cooling water required per minute. Also compare the volumes of the two cylinders. Neglect clearance effects. Take R = 0·287 kJ/kg K and k_p = 1·0035 kJ/kg K for air.

[26·688 kW; 447·93 kJ/min.; 4·28 kg/min.; 5·916]

9. What is the object of compressing air in stages?

Derive an expression for the minimum work required to compress and deliver 1 kg of air in a two-stage compressor. State carefully the assumptions made.

10. What are the advantages of multi-stage compression ?

Determine the minimum driving power required for three-stage compression from 1·05 bar to 42 bar if the delivery is 4·5 m^3 of free air per min. Compression in each stage is carried out according to law $pv^{1.3}$ = constant. Suction temperature is 21°C and mechanical efficiency is 80%. Take free air conditions as 1·01325 bar and 15°C. The air compressor is single-acting. Neglect clearance.

[41·346 kW]

11. A three-stage, single-acting air compressor with perfect intercooling deals with 23 m^3 of air per minute at 1·01325 bar and 288 K. The L.P. cylinder suction pressure and temperature are 1 bar and 305 K and the final delivery pressure is 25 bar. If the stage pressures are in geometric progression and index of compression in each stage is 1·35, find the power required to drive compressor. Also find the heat rejected to the intercoolers per minute and isothermal efficiency of the compressor. Neglect clearance. Take R = 0·287 kJ/kg K and K_p = 1·0035 kJ/Kg K for air.

[152·3 kW; 2,760 kJ/min; 86·86%]

12. What is meant by the term "perfect cooling" ?

A three-stage, single-acting compressor has perfect intercooling. The pressure and temperature at the end of suction stroke in the L.P. cylinder is 1·01325 bar and 15°C respectively. The delivery pressure is 70 bar. Compression in each stage is carried out according to $pv^{1\cdot2} = C$, and 8·5 m^3 of free air (1·01325 bar and 15°C) are delivered per min. If the work done is minimum, calculate :

(i) the L.P. and I.P. delivery pressures,

(ii) the ratio of cylinder volume, and

(iii) the indicated power of the compressor.

Neglect clearance

[(i)4·158 bar, 17·059 bar; (ii) 16·835 : 4·103 : 1; (iii) 68·73 kW]

13. A three-stage, single-acting air compressor works between 1·05 bar and 42 bar. For one cubic metre of air taken in, calculate:

(i) the indicated work required in kJ for maximum efficiency,

(ii) the isothermal work required in kJ between the same pressure limits,

(iii) the indicated work required if the compressor were of one-stage only,

(iv) the isothermal efficiency of the compressor, and

(v) the percentage saving in work required due to using three-stages instead of one.

In parts (i) and (iii), assume the index of compression, $n = 1{\cdot}25$.

Neglect clearance.

[(i) 438·64 kJ; (ii) 387·34 kJ; (iii) 572·78 kJ; (iv) 88·3%; (v) 23·42%]

14. (a) Explain the effect of clearance on the performance of air compressor.

(b) Explain that volumetric efficiency depends on clearance volume and pressure ratio for a single-stage compressor.

The clearance volume in a single-stage, single-acting air compressor is 5% of the swept volume. Air is drawn in at constant pressure of 1 bar and temperature of 43°C. Compression and expansion follow the law $pv^{1\cdot25}$ = constant. The delivery pressure is 6·5 bar and atmospheric pressure and temperature are 1·01325 bar and 15°C respectively. Estimate.

(a) the volumetric efficiency.

(b) the volumetric efficiency referred to atmospheric conditions, and

(c) the work required per kg of air.

Take $R = 0{\cdot}287$ kJ/kg K for air.

[(a) 82·65%; (b) 74·34%; (c) 205·87 kJ/kg]

15. A single-acting, single-cylinder air compressor compresses air from 1 bar to 6·5 bar. Compression and expansion follow the law $pv^{1\cdot25}$ = constant. The clearance volume is 1 litre. If the volumetric efficiency of the compressor is 80%, calculate the stroke volume. If the ratio of diameter of cylinder to stroke of piston is 1·5, calculate diameter of the cylinder and stroke of the piston.

[0·0173 m^3; 24·52 cm; d = 36·78 cm]

16. A single-acting, single-cylinder air compressor runs at 100 r.p.m. The air is compressed from 1 to 8 bar. The stroke volume is 0·125 m^3 and the law of compression and expansion is $pv^{1\cdot3} = C$. If the clearance volume is 5% of the stroke volume, calculate :

(i) the volumetric efficiency,

(ii) the volume of air taken in per minute, and

(iii) the indicated power of the compressor.

[(i) 80·245%; (ii) 10·031 m^3/min.; (iii) 44·55 kW]

17. In an air compressor, show that cylinder clearance does not affect the theoretical work required to compress and deliver one kg of air, provided that delivery and suction pressures remain constant, and that the indices of compression and expansion have the same value.

A single-stage, single-acting air compressor is required to deliver 6 m^3 of free air per minute at a mean piston speed of 165 m per min. The air is to be compressed from an initial pressure of 1·05 bar to delivery pressure of 7 bar and index of compression and expansion is assumed to be 1·3. Assuming stroke of piston to be 1·25 times the bore of the cylinder, clearance volume to be 1/15th of the swept volume per stroke and suction pressure and temperature to be equal to atmospheric pressure and temperature, find : (i) the volumetric efficiency, (ii) the speed, and (iii) the bore and stroke.

[(i) 78%; (ii) 191·64 r.p.m.; (iii) d = 34·47 cm, l = 43·08 cm]

18. A single-stage, double-acting air compressor has a stroke volume of 0·06 m^3 and a clearance volume of 0·003 m^3. Find its volumetric efficiency referred to atmospheric conditions (1·01325 bar and 15°C) and the mass of air delivered per hour when the speed is 200 r.p.m. The suction pressure is 0·92 bar and suction temperature is 50°C and the delivery pressure is 6 bar. Also determine the indicated power of the compressor. Assume compression and expansion law as $pv^{1.35}$ = constant. Take R = 0·287 kJ/kg K for air.

[68·77%; 1,213·9 kg/hr; 75·476 kW]

19. A single-acting, single-stage air compressor is required to compress 5 kg of air per minute from 0·95 bar and 30°C to a pressure of 7·6 bar. The clearance volume is 5% of the stroke volume and the index of both the expansion and compression curves is 1·25. If the stroke and bore are equal and compressor runs at 120 r.p.m., find the size of the cylinder.

Take R = 0·287 kJ/kg K for air. [$d = l$ = 39·53 cm]

20. A two-stage, single-acting air compressor has to deal with 3 m^3 of air per minute under atmospheric conditions (1·01325 bar and 15°C) at 220 r.p.m. and delivers at 85 bar. Assuming complete intercooling between the stages, find the minimum power required to drive the compressor, the cylinder diameters and the common stroke. Assume a piston speed of 165 m per min. mechanical efficiency of compressor as 80% and volumetric efficiency of 85% for each stage. Compression in each cylinder is carried out according to the law $pv^{1.3}$ = constant. Neglect clearance.

[36·553 kW; $d_{L.P.}$ = 23·34 cm; $d_{H.P.}$ = 7·712 cm; l = 37·5 cm]

21. In a two-stage, single-acting air compressor, the delivery pressure is 17·5 bar and the suction pressure is 1 bar. The temperature at the start of compression in each stage is 30°C and the index of compression in each stage is 1·25. The clearance volume of the low-pressure cylinder is 5 per cent of the swept volume and the diameter of low-pressure cylinder is 0·8 of the stroke. The mass of air delivered by the compressor is 5 kg/min. and the intercooler pressure has the ideal value. Find :

(a) the bore of the L.P. cylinder if the speed of the compressor is 110 r.p.m., and

(b) the indicated power of the compressor.

The expansion of the clearance air may be assumed to follow the same law as that for compression.

[(a) 35·59 cm; (b) 23·988 kW]

22. The following particulars apply to a two-stage, single-acting air compressor: Stroke = 25·4 cm; Low-pressure cylinder diameter = 29·2 cm; Final pressure = 25 bar; Intermediate pressure = 5 bar; Temperature of air leaving the intercooler = 35°C.

If the volume of air drawn in the compressor and measured at 1 bar and 15°C, is 80% of the low-pressure cylinders swept volume, find the power expended (used) in compressing the air when running at 250 r.p.m. Assume law of compression as $pv^{1.3}$ = C in each stage. [22·88 kW]

23. Plot actual and theoretical indicator diagrams of a single-stage reciprocating air compressor. Discuss why they differ.

24. Describe briefly the working of an air motor (compressed air engine), Where it is used ?

Air at pressure of 8·5 bar enters the cylinder of an air motor; the supply is cut-off at quarter stroke and expands according to the law $pv^{1.35}$ = constant. Neglecting the effect of clearance, find the temperature at the end of expansion when the initial temperature of air is 30°C. Calculate also the cylinder volume for a single-acting air motor to develop 3·7 kW power at a speed of 250 r.p.m., when exhaust is at 1·05 bar. Take R = 0·287 kJ/kg K for air. [–86·5°C; 2,604·9 cm^3]

25. What are rotary compressors? Differentiate between blowers and compressors. What are axial flow compressors ?

26. (a) How are compressors classified ?

(b) Distinguish between :

(i) Reciprocating and rotary compressors, and

(ii) Positive displacement and non-positive displacement compressors.

27. (a) Differentiate between a fan, a blower, and a compressor.

(b) What is the difference between centrifugal and axial flow compressors ?

28. Sketch and describe the operation of a single-stage centrifugal compressor ?

29. (a) What are the advantages and disadvantages of centrifugal compressors ?

(b) Sketch and describe the operation of an axial flow air compressor.

(c) What are the advantages and disadvantages of axial flow compressors as compared with centrifugal compressors ?

12

LUBRICATION SYSTEMS

12.1 Introduction

Lubrication may be defined as supplying lubricant between two surfaces that are in contact in relative motion to one another. Lubrication may fulfil one or more of the following objectives :

- minimizes wear between moving parts and power loss due to friction,
- cools the surfaces in contact by carrying away heat generated by friction,
- seals a space adjoining the surfaces–such as a piston moving in a cylinder,
- absorbs shocks between bearings and other engine parts, thus reducing vibration and noise, and
- acts as a cleaning agent between surfaces in contact.

Of all these objectives the most important one is the first one mentioned above. Lubrication decreases friction by introducing a film of oil between the sliding metal surfaces. If the film of oil is heavy enough to separate the two surfaces so that they will not touch one another, then wear will be eliminated; otherwise it will be only reduced. The thickness of the oil film depends upon the viscosity of the oil. Viscosity of the oil in turn depends upon properties of the oil and oil temperature. A heavy oil that will not flow freely is said to be more viscous, while a thin free-flowing oil is said to have a low viscosity. The oils used for lubrication purposes are : mineral oils, vegetable oils, and animal oils.

12.2 Classification of Lubricants

Lubricants used for engines may be divided into three classes, namely : (i) solid lubricants, (ii) semi-lubricants or greases, and (iii) liquid lubricants or oils. The lubricants which are most commonly used for engines are the liquid oils.

The **solid lubricants** most commonly used are : graphite, talc and mica – all of which are mineral substances mined from earth. The solid lubricants are mixed with semi-solid lubricants and the mixture is supplied to the bearings. Sometimes solid lubricants are supplied separately to the bearings which are lubricated with oil. Solid lubricants have the property of filling the depressions in the bearing surfaces and cannot be easily squeezed (forced out) from the bearing. Solid lubricants are used on bearings and parts of machinery that operate at low pressures and low speeds.

Semi-solid lubricants are those which will not flow at room temperature. Most semi-solid lubricants are made from mineral oils and fatty oils. Semi-solid lubricants are commonly known as "greases".

The thickness of the grease depends upon the percentage of soap used. Graphite or power mica is sometimes added to grease, and it is claimed that these increase the lubricating qualities for certain purposes. Cheap greases contain clay, wax, resin and other substances which thicken the greases but have no lubricating value.

Greases are used on heavy, slow moving machinery, gear teeth, wheel bearings of railroads cars, trucks, automobiles, revolving parts of machinery where oil cannot be applied successfully, and parts that can not be easily lubricated or can be lubricated only at long intervals.

Liquid lubricants are mainly of two classes of oils used for lubricating purposes :

— Minerals oils, and

— Fatty oils (vegetable and animal).

All *minerals oils* are produced from crude petroleum by fractional distillation. These are the most commonly used as engine and machine lubricants. They are made in a wide variety of grades, ranging from an extremely thin oil suitable for lubricating fine mechanical movements, to a thick heavy oil almost in the class of grease. Mineral oils do not decompose or oxidise, and are free from gumming property. However, for most purposes its viscosity is not sufficient. As they do not possess any oiliness, they have sometimes to be compounded with a small amount of fatty oil.

The specific gravity of mineral lubricating oil varies from about 0·80 to 0·94. The boiling point is high, usually above 350°C in the case of refined distilled oils. The flash point ranges from 150°C to 350°C. The cold-test or setting point varies and depends partly upon the proportion of solid hydro-carbons present. Russian oils have setting point, sometimes below –15°C; most American oils solidify below –5°C, while Scottish oils set at about 0°C.

Fatty oils are either vegetable or animal fixed oils. They are likely to decompose in the presence of heat and produce acid that attacks metals. They are subjected to oxidation and gumming tendency. They have two distinct advantages over the mineral oils : (i) their viscosity is not lowered much on warming, and (ii) their adhesiveness is greater.

In specific gravity they range from about 0·88 to 0·97 at 15°C. Some of the best known animal oils are *lard oil* and *sperm oil.* Lard oil is obtained by separation of solid fat from a lard (pig fat) by pressure. It is used alone or compounded with mineral oil as a lubricant for cutting tools. Sperm oil is obtained from sperm whale (large fish like marine animal). It possesses low viscosity among fatty oils and does not gum. It is a good lubricant for delicate mechanisms, such as watches and mathematical instruments.

Some of the common vegetable oils are : *Olive oil, cotton-seed oil, linseed oil, castor oil and palm oil.* They are sometimes compounded with mineral oils but are seldom used alone, as they decompose quickly and have a tendency to become gummy.

12.3 Desirable Qualities of a Lubricant

A good lubricant should have the following qualities (properties) :

— *Maximum fluidity :* It should have sufficient thickness to prevent wearing surface coming into contact. All oils should have sufficient body to prevent their being squeezed out from between the surfaces they are lubricating. Flowing quality is represented by viscosity.

— *Maximum capacity for absorbing and carrying away heat :* Specific heat should be high.

— *Low coefficient of friction :* Very little power should be needed to overcome the fluid friction.

- *Fairly high flash point and a low freezing point :* Flash point is the temperature at which an oil will flash into flame for a moment only when brought in contact with flame. High flash point is required when oils are to be subjected to high temperatures, as in steam engines, air compressors and internal combustion engines. Freezing point is also called *cold pour point.* A low pour point is important where oils are to be exposed to very low temperature, as in machinery operating in cold countries and in the lubricating of refrigerating plants.
- *Freedom from corrosive acids with no tendency to gum up or oxide.*
- *Good wearing qualities :* It should retain its viscosity and adhesive capacity for long periods.
- *Demulsibility :* Oils when not properly refined, form emulsions due to continued agitation of the oil and the water. Such an emulsion may prevent the oil to flow freely. It also gives rise to sludge (thick muddy substance) formation which may choke up the whole lubricating system. Demulsibility is the property of the oil to separate itself quickly from an emulsion (milky liquid with oily particles suspended in it).
- *Cleanliness :* The absence of water and sediments is essential for lubricating oil. Water is not a lubricating fluid and it promotes corrosion while dirt and small foreign particles of insoluble matter cause more wear of engine parts.

12.4 Properties and Testing of Lubricants

Properties of lubricating oils may be tested by chemical analysis, by physical tests and by trial in service. Following are the important tests to be carried out :

- Viscosity,
- Specific gravity,
- Flash point,
- Fire point,
- Pour point,
- Emulsification,
- Precipitation number,
- Corrosion,
- Water content,
- Wick feed test,
- Specific heat,
- Carbon residue,
- Neutralization number, and
- Saponification number, and
- Accelerated service tests.

Viscosity is a measure of fluid friction or the resistance which particles or molecules of an oil will offer to one another when the main body of the oil is in motion. A high viscosity oil is thick and flows slowly. A low viscosity oil is thin and flows readily. Viscosity is measured by observing the time taken for 60 c.c. of oil to flow through standardised orifice (Saybolt viscosimeter). Viscosity to some degree is necessary for all lubricants. A high viscosity oil is only necessary for heavily loaded slow moving journals. In fast moving machinery it causes large loss of power.

It is customary to determine the viscosity of oil at 38°C, 50°C or 100°C. Viscosity is important because it indicates heat produced, rate of oil flow and the resistance oil offers to a shaft turning in its bearing. Viscosity index is the time taken for a definite quantity of oil to fall a certain level in a capillary tube compared with a standard oil. The types of viscosimeters in use are :

- Saybolt viscosimeter (used in U.S.A.),
- Redwood viscosimeter (used in U.K.), and

— Engler viscosimeter (used in Germany).

Specific Gravity is defined as the ratio of mass of oil compared to the mass of an equal volume of water, both measured at 15°C.

API (American Petroleum Institute) gravity is the comparative mass of oil to water as read on arbitrary scale. This is the scale commonly used by the petroleum industry. To find API gravity from the specific gravity, we use the formula

$$\text{API gravity} = \frac{141{\cdot}5}{\text{specific gravity}} - 131{\cdot}5$$

Specific gravity of water is 1 and API number of degree is 10. All oils have API gravities greater than 10, the degrees being given in whole numbers instead of the decimal fraction representing their specific gravities on the specific scale. Whole numbers are much more easily remembered than three or four places decimal fractions.

Lubricating oil is sold by bulk (volume), and litre is a volume unit. In hotter countries for the same mass, volume is more and in colder countries it is less; hence utility of specific gravity is required for commercial purposes. The specific gravity is found by using an instrument called a *hydrometer.* It is graduated in specific gravity and API degrees.

When oil is heated in a cup it vapories so rapidly at a certain temperature that it will ignite momentarily when a flame is brought close to it. This temperature is called the **flash point.** In this case fire only takes place and then extinguishes but does not burn continuously. When oil vaporises fast enough to keep on burning, it has reached its **fire point** which will be higher than the flash point. At normal temperature there must not be any tendency to form explosive mixtures of oil and vapour, and the rate of evaporation should be as slow as possible. To ensure this, the flash point must be well above the working temperature; otherwise oil consumption increases and fumes are given out which may be poisonous. If the fire takes place, damage may be done to the factory.

As oil is cooled, it resists pouring more and more until finally it refuses to pour. In making tests for pour point, oil temperature is lowered suddenly in steps of 3°C and at each 3°C interval the test jar is removed from the bath and tilted for three seconds. As soon as the oil ceases to flow when tilted, the test jar is placed horizontal for five seconds. If no movement takes place, **pour (cold) point** is 10°C above this *non-flow point.*

For **Emulsification test,** oil thoroughly mixed with water forms an *emulsion* which generally tends to separate on standing and which has little lubricating value. Some oils contain sulphonates, certain acids, alcohols and finely divided earthly metals which retard separation.

In case it is not possible to prevent the access of water to lubricating oil as in the steam engine cylinder, it is essential that oil and water shall easily separate after having been vigorously shaken together. The ability of the oil to separate easily is called *demulsibility.*

There are two tests for demulsibility :

- *Steam emulsion test :* The number of seconds required for an oil to completely separate from an emulsion when steam is passed through the oil in a standard manner is the steam emulsion number.

— *Water emulsion test :* An emulsion is formed by churning for 5 minutes 27 c.c. of oil and 53 c.c. of distilled water at either 55°C or 85°C. If complete separation is obtained in one minute, oil is said to have perfect demulsibility.

Precipitation number is an indicating of the amount of asphalt or tar present in the oil. It is determined by diluting 10 c.c. of oil with 90 c.c. of special Petroleum Naptha. Deposits are separated by centrifuging. The volume of separated layer in c.c. is the precipitation number. It is useful to see how much deposit of the residues is retained in the bearings. It may be taken as an indication of the time interval of cleaning the bearings. The precipitates may clog (choke up) the holes in the bearings.

In **Corrosive test** a clean strip of pure polished copper is placed in a sample of oil held at 100°C for 3 hours and is then washed in sulphur free acetone (liquid) and compared with fresh copper strip. It gives an indication of corrosive effect of the oil under test.

In **Water test** the presence of water can be detected by heating a sample of oil to a few degrees above 100°C. If cloudiness in oil is due to water, the oil will now become transparent. Water in oil retards the capillary action of the wick feed as it wets the wick (we may be paying money for water instead of oil). If water is there, it will settle down at the bottom. Boiling point of oil is higher than that of water at atmospheric conditions.

Wick feed test requires 14 days and involves measuring the quantity of oil lifted by capillary action through a wool wick. The results are interpreted by comparison of such tests. The amount of oil lifted depends upon the diameter of wick, level of oil, constitution of oil, and atmospheric conditions.

Specific heat test is important when oil is used to cool a bearing or a piston and higher specific heat is desirable.

Carbon residue test involves destructive distillation in an apparatus designed to exclude air, to permit control of heating rate and to eliminate condensation of distillates. The non-volatile residue is termed carbon residue. Many bearings, such as internal combustion engine bearing, operate at high temperature to cause chemical changes in the lubricant and so produce the deposits that clog (choke up) oil passages and cause wear.

Neutralization number indicates acidity of oil which is the mass in milligrams of KOH (Potassium Hydroxide) required to neutralize one gram of oil. While acids are harmful under all conditions, nevertheless their presence may lead to corrosion and formation of sludge, emulsion and deposits.

Saponification number test is applied to only a compounded oil or to a suspected mineral oil to find out whether it is compounded with animal or vegetable fatty material. The number of milligrams of KOH required to saponify one gram of oil is the saponification number.

Accelerated service test is for comparing service characteristics of oil. This test is carried out using oil testing machines – usually a motor driven journal and a plain bearing arranged so that a specified load can be applied in a specified lubricating system. One machine is loaded until seizure occurs, others operate at constant load, and the quality of the lubricant is determined either by testing the oil to determine changes in viscosity, neutralisation number and carbon residue, or by visual examination of bearing conditions.

The test used in practice by the operating engineer, is the performance of the oil in actual use on some engines or machines. Oil purchased from reputed oil companies are usually of fairly uniform quality, but a certain grade of oil supplied by some one company is sometimes better than the same grade supplied by other companies. The operating engineer compares various oils and determines which is best for this particular purpose by testing each oil in turn on same engine or machine, under the same conditions and for the same length of time, noting during these practical tests, the quantity of oil used, temperatures of bearings, effects on rubbing surfaces and other worthwhile points of comparison.

12.5 Methods of Engine Lubrication

The various methods of lubrication in general use in different types of engines are as follows :

The **drip system,** where oil is fed to the machine parts, drop by drop, from an oil cup. It is not particularly efficient but is often the most convenient way of lubricating the external parts of engines and machines. Valve gear pins, rocker shafts, main bearings of small engines, crank pins, crosshead pins, line shafts bearings, and many other machine and engine parts are lubricated in this fashion.

The **ring oiling system,** where the bearing is lubricated by oil brought up from an oil bath by revolving rings. This is commonly used on the shaft bearings of motors, generators, small engines and turbines.

The **Splash or bath system,** where the parts to be lubricated, dip into oil or actually run in an oil bath. The splash method is used in the crankcase of numerous types of steam and gas engines, for crank pin, main bearings and crosshead lubrication. Many enclosed parts of machine, such as gears, run in an oil bath.

The grease cup method of lubrication is used on heavy slow-moving parts, rocker arms and reciprocating parts with a jerky motion, and parts that are not readily accessible and can be lubricated only at fairly long intervals.

The **forced or pressure system,** where oil is forced into bearings by a pressure pump. This is used on large engines and turbine bearings or on any heavy machine where oil under pressure must be used to produce and maintain an oil film between the rubbing surfaces. It is of course the most efficient system of lubrication though more expensive to operate, as an oil pump is required.

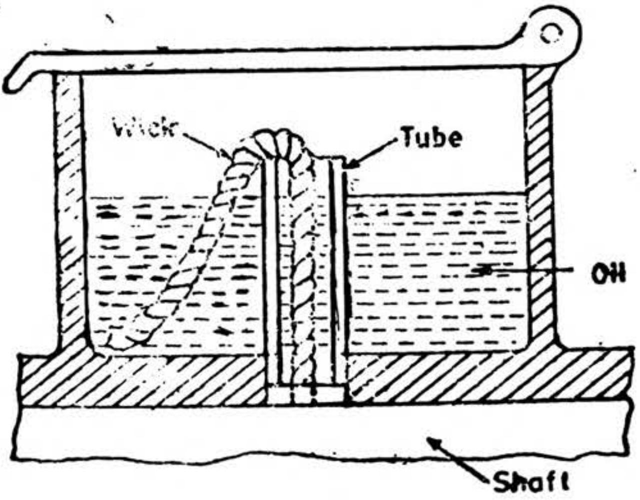


Fig. 12.1. Wick oiler.

The **wick oiler** shown in fig. 12-1 is rather an old fashioned method of lubrication and is used for automatic lubrication of carnkshaft bearings. It consists of an oil cup on the top of the bearing having a short piece of tube in a central oil hole and a piece of cotton or wool wick in this tube. The oil is carried over by wick capillary action and is fed by drops on the shaft. To facilitate the handling of the wick, a piece of twisted wire is attached to it.

The **sight-feed oil cup** shown in fig. 12-2, is the most common method of

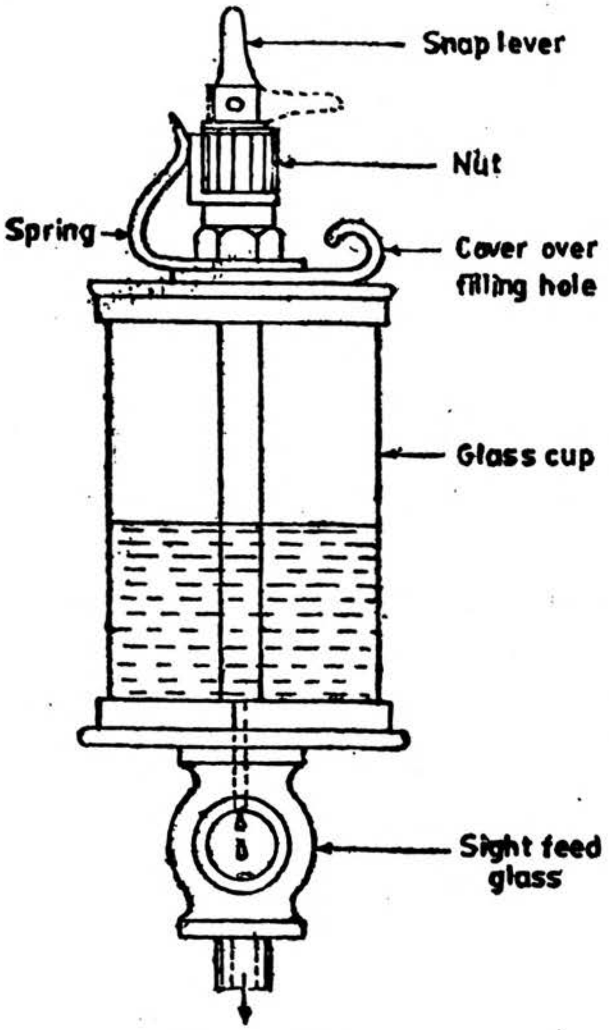


Fig. 12-2. Sight feed oil cup.

regulated drip lubrication. It consists of a glass cup with metal base and top and central tube. The cup is filled through a hole in the top and the flow of oil is regulated by a needle valve in the central tube. The needle valve opening is adjusted to a desired amount by means of the milled nut on the top. Once set, the spring prevents the milled nut from turning. The glass body shows the amount of oil in the cup at any time and rate of feed can be observed through the sight-feed glass at the bottom. The needle valve can be closed by pulling over the snap lever to the position shown dotted in fig. 12-2.

Ring oiler for shaft bearing is shown in fig. 12-3. This is a common method of lubricating main bearings on small high speed engines, generators and motors. Rings running loosely on the shaft, carry oil from an oil bath upto the top of shaft. This system works on the principle of adhesion. The oil brought up on the top of the shaft by the brass oil rings, spreads over the bearings surface and drains back down into the oil bath. Chains are sometimes used instead of rings.

Splash lubrication system as applied to small horizontal engines is shown in fig. 12-4.

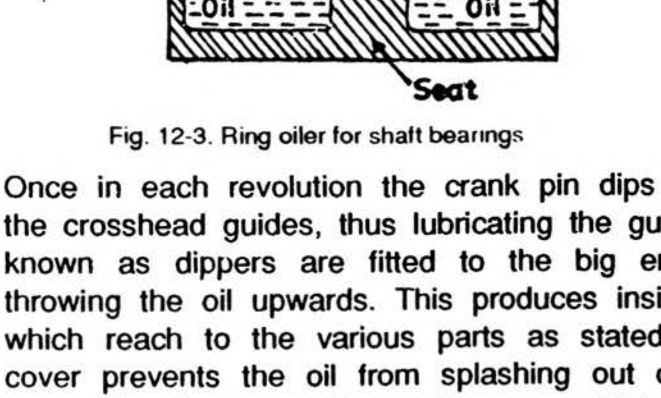


Fig. 12-3. Ring oiler for shaft bearings

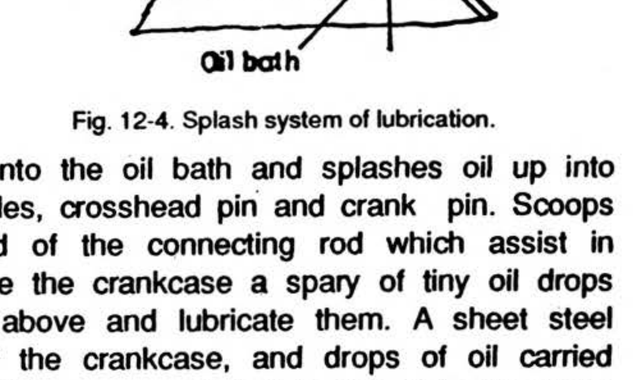
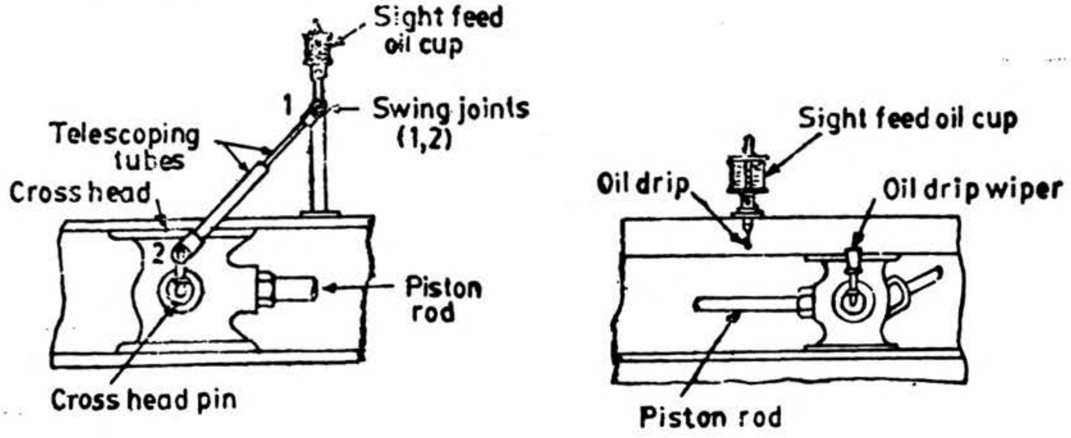


Fig. 12-4. Splash system of lubrication.

Once in each revolution the crank pin dips into the oil bath and splashes oil up into the crosshead guides, thus lubricating the guides, crosshead pin and crank pin. Scoops known as dippers are fitted to the big end of the connecting rod which assist in throwing the oil upwards. This produces inside the crankcase a spary of tiny oil drops which reach to the various parts as stated above and lubricate them. A sheet steel cover prevents the oil from splashing out of the crankcase, and drops of oil carried around by the connecting rod are caught in this cover and led to the main bearings. This method is employed for lubricating steam engines, I.C. engines and air compressors.



(a) Telescopic oiler

(b) Wiper oiler

Fig. 12-5.

A telescopic oiler for lubricating steam engine crosshead pin is shown in fig. 12-5(a). The telescopic part consists of two tubes, one of which slides into the other. The telescoping tubes are connected by swing joints to a short tube attached to the crosshead pin and a long tube bolted to the engine frame. A sight feed oil cup is attached to the top end of the tube and the oil fed from this cup passes through the telescopic tube to the crosshead pin. The telescopic tubes accommodate themselves to the various positions of the crosshead, by sliding in and out of each other as the crosshead slides to and fro.

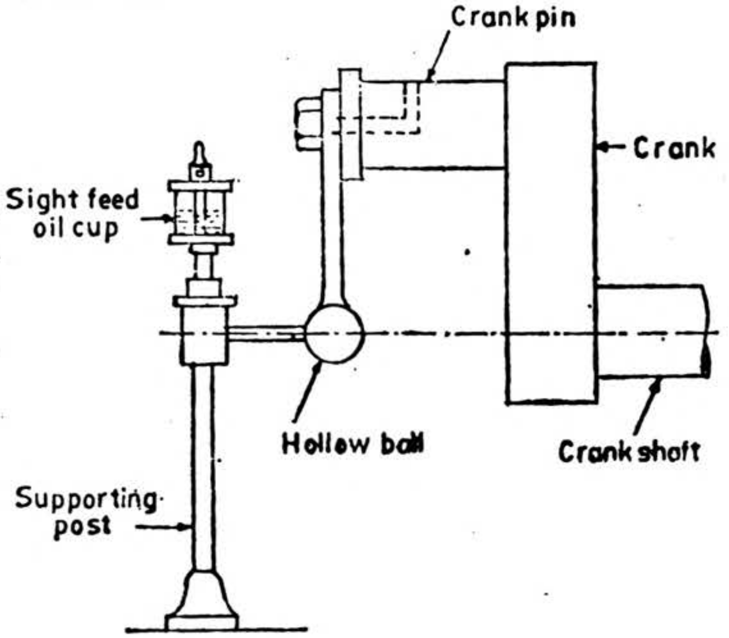


Fig. 12-6. Banjo type crank pin oiler for side-crank engine.

The wiper oiler fig. [12-5(b)] is used to lubricate reciprocating part such as steam engine crosshead. A sight-feed oil cup is attached to a bracket which is bolted to the upper crosshead guide. The oil fed from this cup drops on to a flat wick, which is wiped by a thin piece of metal set on top of another oil cup attached to the crosshead pin. The oil from the wiper trickles into the cup and from the cup into a hole in the centre of the crosshead pin, and through oil holes in the pin to the crosshead pin bearing.

Centrifugal oiler is used for lubricating crankpins on side-crank engines. It is made in two forms-the Banjo type shown in fig. 12-6 and weighted type (not shown).

In the first form, short pipe or banjo having a hollow ball at one end, is firmly attached to the crank by means of a hollow cap screw, care being taken to centre the ball exactly in line with the centre of the crankshaft. Oil from the sight-feed oil cup, supported on the stationary post, passes down the short horizontal pipe, drips into the hollow ball, and flows from there by centrifugal force to the crankpin.

A simple type of grease cup consists of a base which is screwed into the oil hole and a threaded cup which screws over the base. The cup is filled with grease and screwed on the base for a few threads. The cup is then screwed down occasionally by hand to force the grease into the bearing.

Forced feed lubrication is a central automatic oiling system for lubricating a number of parts like main bearings, crossheads, crosshead guides, etc. of steam and I.C. engines, and bearings of turbines. The oil is fed under pressure by mechanically operated pump to the various parts of the engine through a distributing pipe. Types of oil pumps used are : valveless oil pump, gear oil pump, eccentric driven oil pump, etc. In this system a good amount of oil is maintained in the engine crankcase which is a necessity for the heavy demands of modern service conditions.

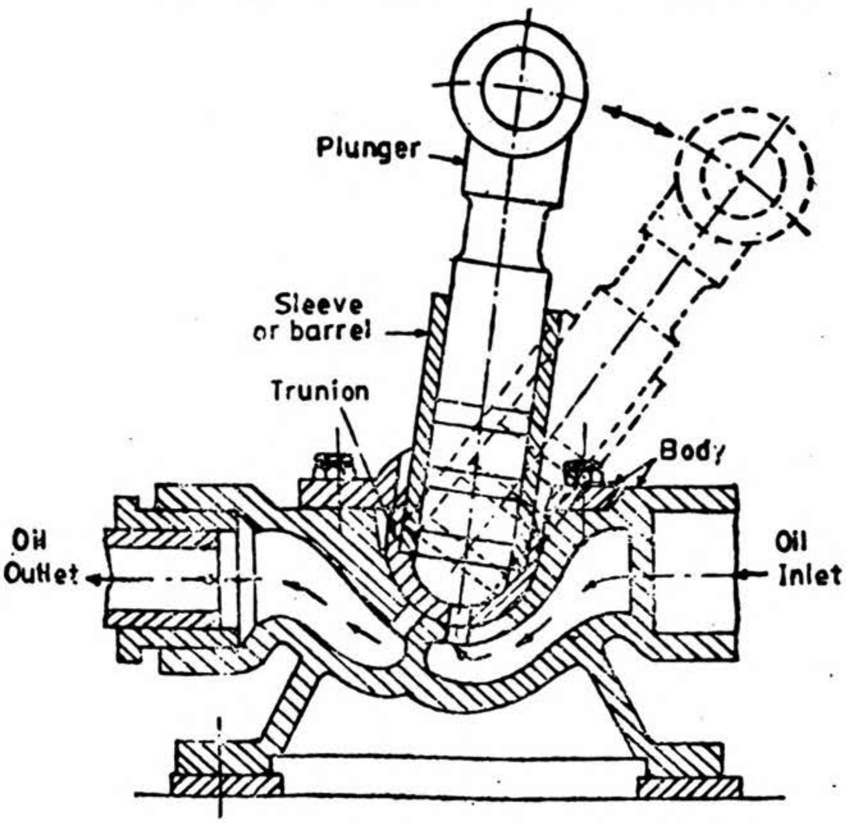


Fig. 12-7. Valveless lubrication oil pump.

A valveless lubrication oil pump may be used for lubrication of steam engines and I.C. engines. This pump mainly consists of a plunger, a sleeve or barrel, a trunion and pump body as shown in fig. 12-7. The pump is worked by the eccentric, fitted on the crankshaft.

The plunger works in a sleeve or barrel fitted in a circular disc of metal which is known as a trunion. The trunion is free to revolve in the body. As the crankshaft revolves the plunger not only reciprocates but also oscillates the trunion from side to side, uncovering the delivery and suction ports alternately.

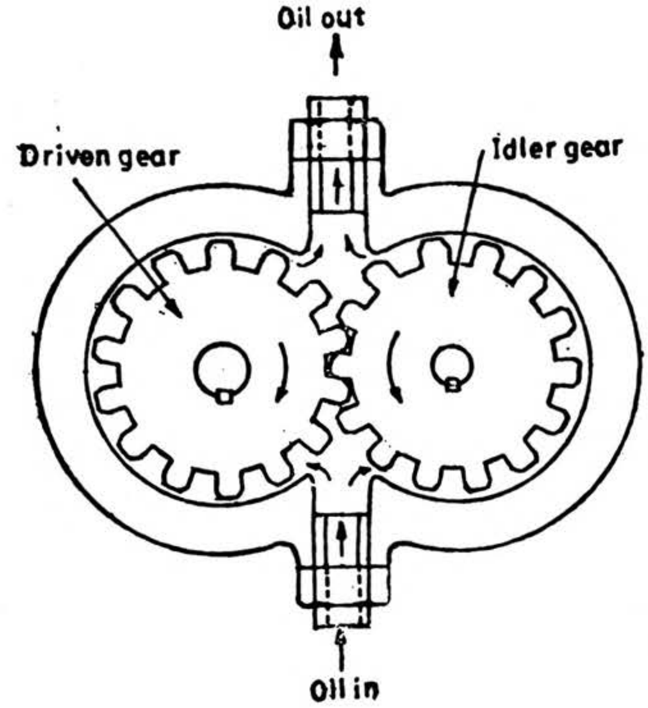


Fig. 12-8. Gear oil pump.

The Gear oil pump, in its simplest form has two gear wheels meshing together in a tightly enveloping housing as shown in fig. 12-8. One gear is driven and second is idler. Oil entrance is on one side and pressurised oil delivery is on the other side of the housing. Rotation of gears induces a fairly even flow of lubricant. Gear oil pump is by far the most popular of the various forms of lubricating oil pumps.

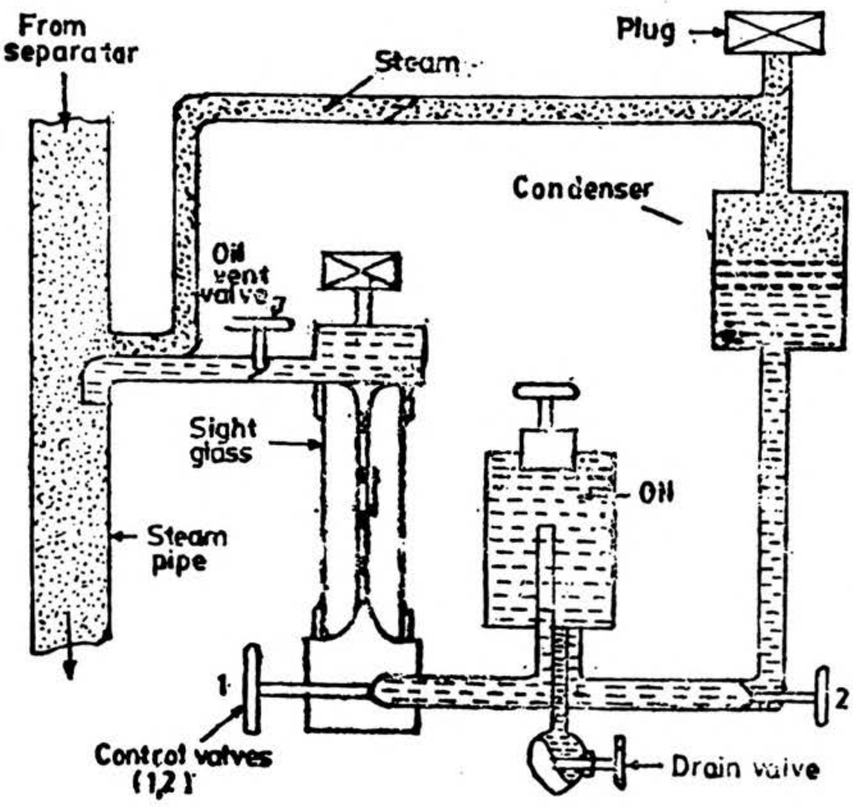


Fig 12-9. Hydrostatic sight feed lubricator.

A hydrostatic sight-feed lubricator is used for the lubrication of steam engine cylinder. The common form shown in fig. 12-9, is widely used because of its simplicity. It works on displacement principle. Some of the steam taken from the supply pipe is condensed to water in the *condenser* and this water trickles down the central pipe into the body of the lubricator where it displaces the oil. The oil floats on the top of the water and passes down the oil feed to the bottom of the sight-feed glass. It then passes through the control valve into the glass, floating up through the water in the glass, a drop at a time, and along the delivery pipe into steam main.

As the oil feeds out, its place is taken by condensate from the condenser so that the body of the lubricator is always full of oil and water. When all the oil has been used, the lubricator will be entirely filled with water. To fill this lubricator, close the valve on the top of the body and the valve on the horizontal delivery pipe, open the drain valve to let the water drain out, take out the filling plug on top, close the drain valve, fill the lubricator with oil, replace the filling plug, open the top valve and a delivery valve, and adjust the feed by means of the control valve. The top connection must be atleast 45 cm in length, as it is the small hydrostatic head from this pipe that forces the oil into the steam pipe line.

Tutorial - 12

1. (a) What is lubrication ? What is accomplished by lubrication ?
   (b) What substances are used as lubricants and where ?
   (c) Explain the necessity of engine lubrication.
2. (i) Give the desirable qualities of a lubricant.
   (ii) Explain the difference between viscosity and oiliness.
   (iii) What do you understand by 'Viscosity' and 'Flash point' of a lubricating oil ?
   (iv) What do you understand by "Saponification number", "Neutralization number" and "Precipitation number" of a lubricating oil ?
3. Describe in short various tests performed in deciding the qualities of a lubricating oil.
4. What do you understand by "Oiliness", "Coefficient of friction" and "Viscosity" of a lubricating oil ?
5. What are the various tests to be conducted before selecting a suitable oil for lubricating an engine ? Describe them in brief.
6. Explain the necessity of engine lubrication, and give the desirable properties of a lubricant for crankshaft bearings. Describe the various methods of lubrication of crankshaft bearings of an engine.

7. Explain the operation of a hydrostatic lubricator. Where such a lubricator is used ?
8. Name four methods of lubricating engine bearings and state which is the best and why ?
9. Describe various methods commonly employed for lubricating various parts of steam engines and I.C. engines.
10. Describe with neat sketches the following types of lubricators and state where they are used :
    (i) Sight-feed lubricator, (ii) Wick oiler, (iii) Telescopic oiler, (iv) Ring oiler, and (v) Centrifugal oiler.
11. What do you understand by Splash lubrication and Forced feed lubrication ?
12. Describe the methods of lubricating the cylinder and main bearings of a horizontal steam engine. In what respect does the lubricating oil for cylinder of such an engine differs from that used for the main bearings ?
13. How stuffing boxes of a steam engine are lubricated ?
14. How will you lubricate the following parts of a single-cylinder, 10 kW, horizontal, Diesel engine running at 400 r.p.m.
    (i) Cylinder, (ii) Piston pin or Wrist pin bearing, (iii) Crankpin bearing, (iv) Main bearings, and (v) Camshaft bearings.
15. What is forced feed lubrication ? Explain the construction and operation of a valveless lubricating oil pump.
16. Explain the construction and operation of a gear pump. Where is such a pump used ?

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

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m³/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
.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

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_f	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

STEAM TABLES

PROPERTIES OF DRY AND SATURATED STEAM

(Pressure Table)

Pressure bar p	Sat. temp. °C t_s	Specific vol. Sat. steam m³/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
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

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
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

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(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

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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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