HEAT ENGINES

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PREFACE

In recent years a number of changes have taken place in the presentation of the subject of Heat Engines and as this book continues to be well received, it has been thought advisable to send out a Third Edition which has been carefully revised.

Although the greatest alteration which has been made is the introduction of the Fahrenheit temperature system in all steam calculations and data, the opportunity has been taken to re-write some portions, to make a few additions to the text and to add a number of more recent examination questions to the examples. The calculations in the text and the steam tables at the end of the book have also been brought into line with the latest Callendar steam tables.

The authors are gratified that the book has achieved such a wide field of usefulness. They consider that it is now admirably adapted to the needs of those students who, having gained a knowledge of elementary physics of about intermediate degree standard, or taken an elementary course of engineering science, are now desirous of following a one-year course in Heat Engines such as is required for part one of a Final Degree Examination in Engineering or the ordinary stage of the National Diploma or Certificate in Engineering. By a simple, lucid yet accurate treatment of the fundamental principles of the subject it facilitates the approach to the study of the many more advanced and specialised textbooks dealing with the various branches of Heat Engines theory and practice.

The authors repeat their thanks to Messrs. Babcock and Wilcox, Ltd., Messrs Hick, Hargreaves and Co., Ltd., Metropolitan-Vickers Electrical Co., Ltd., Messrs. The National Gas Engine Co., Ltd., and Messrs. C. A. Parsons and Co., Ltd., for the use of blocks and diagrams; to the Union of Educational Institutions, the Union of Lancashire and Cheshire Institutes,

and the Northern Counties Technical Education Council, for permission to print recent examination questions and to change others to Fahrenheit units where desirable; to His Majesty's Stationery Office for permission to reproduce the entropy chart, and to Messrs. Edward Arnold and Co., for their consent to the use of Callendar's Steam Tables.

S. H. MOORFIELD.

H. H. WINSTANLEY.

July 1947.

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HEAT ENGINES

CHAPTER I

WORK

1. The work of this chapter should be thoroughly understood before passing on to the study of the subject of Heat Engines.

All heat engines are built for the definite purpose of utilizing heat in the doing of work, such as hauling a load by a locomotive, lifting weights by colliery winding engines, operating ship winches, excavators, etc., lifting and forcing water by pumps, driving shafts in mills and factories. A thorough understanding of the methods of calculating the sizes of engine cylinders cannot be obtained until the student has quite clear ideas of Work and Power.

2. Definition. Work is a result of the application of effort or force. A further definition often given is as follows: Work is the result of expenditure of energy.

This is the best definition for our purpose because we must later consider heat as a form of energy. The necessary condition that work may be done is that movement shall take place. An effort or force may be applied to a body, but no work is done unless the body moves at the point where the force is applied. The amount of movement may be large or small.

3. Measurement of Work. The engineer measures the work done by or on any of his productions by observing and measuring the force acting, and also measuring the distance moved in the direction of action of the force.

For example, let F = a uniform force, applied whilst a distance

S is traversed. Then a measure of the work done is obtained by multiplying F and S.

Thus, work done = F × S.

The units of the work are important and depend upon the units chosen for F and S.

When F is in tons and S inches, the work is in inch-tons.

When F is in pounds and S inches, the work is in inch-pounds.

When F is in pounds and S feet, the work is in foot-pounds.

The last-named unit is most commonly used. Other units, not needed for our present purpose, are often used.

EXAMPLE. The area of a steam engine piston is 30 square inches, and the uniform effective pressure upon the piston is 40 lb. per square inch. The piston moves 4 inches. Find the work done during the movement.

$$\begin{aligned}\text{Force on piston} &= \text{Intensity of pressure} \times \text{area of piston}\\ &= 40 \text{ lb. per sq. in.} \times 30 \text{ sq. in.}\\ &= 1{,}200 \text{ lb. force.}\end{aligned}$$

$$\begin{aligned}\text{Work done} &= \text{Force acting} \times \text{distance moved}\\ &= 1{,}200 \text{ lb.} \times 4 \text{ inches}\\ &= 4{,}800 \text{ inch-lb.}\end{aligned}$$

If we divide this result by 12, thus converting the inches to feet, we shall get

$$\text{Work done} = \frac{4{,}800 \text{ inch-lb.}}{12 \text{ inches per ft.}} = 400 \text{ ft.-lb.}$$

Alternatively, we may proceed as follows, changing the unit of distance:

$$4 \text{ inches} = \tfrac{4}{12} \text{ ft.} = \tfrac{1}{3} \text{ ft.}$$

$$\text{Then work done} = 1{,}200 \text{ lb.} \times \tfrac{1}{3} \text{ ft.} = 400 \text{ ft.-lb.}$$

4. Work done during Change of Volume. In heat-engine work, the student is constantly being called upon to recognize $P\,dV$ and $P(V_2-V_1)$ as quantities of work done, and for this purpose the units must be very carefully considered.

Pressure in this book denotes intensity of pressure. The word pressure itself is often used to denote force, so that the student must be careful to get correct ideas as to pressure, or intensity of pressure.

Pressure intensity, or (briefly) pressure, is the load per unit of area, e.g. pressure in lb. per square inch is the force in lb. on one square inch; pressure in lb. per square foot is the force in lb. on one square foot.

EXAMPLE. The pressure of the atmosphere is 14·7 lb. per square inch. What is the pressure in lb. per square foot?

Let P = pressure in lb. per square foot;
p = pressure in lb. per square inch.

Since there are 144 square inches in one square foot, and on each square inch a force of 14·7 lb., then pressure in lb. per sq ft.

$$= 144 \times 14{\cdot}7$$
$$= 2{,}116{\cdot}8$$

or

$$P = 144p.$$

EXAMPLE. A mercury U-tube pressure gauge gives a reading of 3 inches difference in level when denoting the pressure of air in a vessel. What is the pressure difference represented in lb. per square inch, if mercury is 13·6 times as heavy as water, and water weighs 62·4 lb. per cubic foot?

This type of example is very important.

First notice that the area of the tube is not given.

The pressure at level XX (fig. 1) is the same in both limbs.

On the left, it is the pressure in the vessel. On the right, it is the atmospheric pressure, plus the pressure due to the weight of a 3-inch column of mercury.

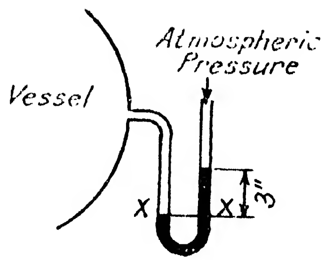


FIG. 1.

Consider the weight of mercury covering an area of 1 sq. ft. to a depth of 3 inches.

Then weight of mercury $= 1 \text{ ft.} \times 1 \text{ ft.} \times \frac{3}{12} \text{ ft.} \times 13{\cdot}6 \times 62{\cdot}4$
$= \frac{1}{4} \times 13{\cdot}6 \times 62{\cdot}4$
$= 212{\cdot}16$ lb.

Hence the pressure in the vessel is 212·16 lb. per square foot above atmospheric pressure.

Again, 212·16 lb. per sq. ft.

$$= \frac{212{\cdot}16}{144} \text{ lb. per sq. inch}$$

= 1·473 lb. per sq. inch above atmospheric pressure.

Note that the area of the tube is not involved because we are concerned with the *intensity* of pressure which is independent of this area.

Consider work done when volume changes at uniform pressure.

Imagine a cylinder, with its axis horizontal, fitted with a piston and supplied with gas at constant pressure through the opening at B (fig. 2). Let the piston move, and hence, the volume between piston and cover change as the gas is supplied.

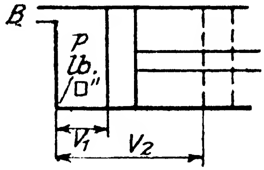


FIG. 2.

Let a = area of piston surface in contact with gas in square inches.

p = intensity of pressure of gas in lb. per sq. in.

Let L = linear horizontal movement of piston in *feet*.

Now the force on piston $= p \times a$ lb.

Work done $= p \times a \times$ L ft.-lb.

Again, let A = area of piston in square feet.

P = pressure of gas in lb. per square foot.

We then have $p = \frac{P}{144}$ and $a = 144$ A

$$\therefore \text{ work done } = p \times a \times L \text{ ft.-lb.}$$
$$= \frac{P}{144} \times 144\,A \times L \text{ ft.-lb.}$$
$$= PAL \text{ ft.-lb.}$$

The units of this equation are lb. per square foot for pressure, square feet for area, and feet for distance.

The product of P and A is force, and L is distance.

∴ the above (PA) × L = force × distance.

But, again, the product of A and L is the change of volume or the volume "swept" by the piston.

$$\therefore PAL \text{ becomes } P \times AL \text{ ft.-lb.}$$
$$= P \times \text{volume change, ft.-lb.}$$
$$= P \times (V_2 - V_1) \text{ ft.-lb.}$$

EXAMPLE. Four cubic feet of water at 700 lb. per square inch are supplied to a hydraulic engine. Find the work done during the supply.

$$\text{Work} = P \times \text{change in volume}$$
$$= 144p \times \text{change in volume}$$
$$= (144 \times 700) \times 4$$

(Brackets are used to keep the pressure distinct.)

$$= 403{,}200 \text{ ft.-lb.}$$

EXAMPLE. Three cubic feet of steam at a uniform pressure of 100 lb. per square inch are supplied to the cylinder of an engine. Find the work done during the supply. Neglect clearance.

Here let $V_1 = 0$ and let $V_2 = 3$ cubic feet.

Then change in volume $= (V_2 - V_1) = 3 - 0 = 3$ cubic feet.

$$\begin{aligned}\text{Work done} &= P(V_2 - V_1)\\ &= 144p(V_2 - V_1)\\ &= 144 \times 100 \times 3 \text{ ft.-lb.}\\ &= 43{,}200 \text{ ft.-lb.}\end{aligned}$$

It will be seen from the above example, that when the cylinder contains no steam initially, and a volume V cubic feet is admitted at a pressure P lb. per square foot,

Work done during admission = PV ft.-lb.

In considering examples such as the above, relating to engine cylinders, it is very helpful to draw the Pressure-Volume diagram.

5. The Pressure Volume Diagram as a Work Diagram. When a gas, a mixture of gases, or a vapour is operating in an engine cylinder, it is known as the working substance. The pressure-volume diagram is a diagram in which the pressure of the working substance is plotted vertically and its volume plotted horizontally. Lines drawn upon the field enclosed by the rectangular co-ordinates then have definite meaning, and represent the behaviour of the working substance in the cylinder of an engine. By means of such lines it is possible at a glance to convey to the mind a large amount of precise information. The practice of drawing or sketching the lines fixes the problem, and enables the initial and final states of the substance to be kept quite distinct. In fig. 3, O is the origin, and the volume scale is set out towards the right. The pressure scale is set up on the perpendicular line from O. Thus the diagram will be on the right of the pressure scale and

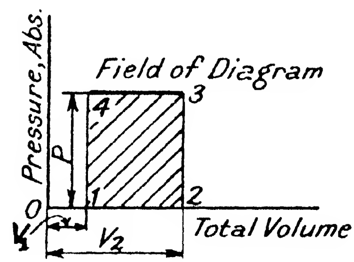


FIG. 3.

above the volume scale. If a point be chosen anywhere in the field of the diagram, two dimensions are fixed at once, one denoting the pressure, and the other denoting the volume of the substance for which the diagram is drawn. These dimensions are P (pressure) and V (total volume) in fig. 3. The position of this point establishes the " state " of some given quantity of substance as regards pressure and volume. The amount of the substance will be dealt with after discussing the characteristic equation for gases in article 28.

The " state " point is regarded as movable, and its path from the initial state to some final state can be traced out by a line known as a pressure-volume line. The pressure-volume line is used to denote a " process " in the changing state of a gas in an engine cylinder.

The state point in fig. 3 may now be considered to have arrived at (3) by moving horizontally from (4) to (3), and the area of the enclosed rectangle (1, 2, 3, 4) is proportional to the product $P \times (V_2 - V_1)$. This shows that an area in the field of the diagram can be made to represent a quantity of work done, as the state point moves from (4) to (3).

This is a very important process line, and could represent the supply of steam or gas to a cylinder at constant pressure, from volume V_1 until the quantity supplied occupied volume V_2.

The quantity $(V_2 - V_1)$ is called the *volume swept* by the piston, and is the product of the area of the piston and the distance it moves during the increase of volume.

The work done during the process is shown by the shaded area, and the magnitude of that work is obtained by multiplying the shaded area by the " work scale " in the appropriate units.

6. Work Scale of a Pressure-Volume Diagram. Definition. The work scale of a pressure-volume diagram is the number of foot-lb. represented by one square inch of the diagram, when inches are used as the units in plotting and foot-lb. as the unit of work. Other units may be employed, but it is advisable to use one set of units throughout, until proficiency in the subject itself is attained.

The work scale is obtained as follows:

Let p_s = pressure scale in lb. per sq. in. per inch of vertical measurement.

V_s = volume scale in cu. ft. per inch of horizontal measurement.

Then 1 square inch = 1 vertical inch × 1 horizontal inch.

Now writing the equivalents of these we have

$$\text{Foot-lb. per sq. in.} = (p_s \times 144) \times V_s = 144\, p_s\, V_s \text{ ft.-lb.}$$

Since $P_s = 144p_s$, we have

$$1 \text{ square inch} = P_sV_s \text{ ft.-lb.,}$$

where P_s is the scale of absolute pressure in lb. per sq. ft. per inch and V_s is the scale of volume in cubic feet per inch.

EXAMPLE. A pressure volume diagram is plotted to a scale of 1 inch = 25 lb. per sq. in. abs. and 1 inch = 2 cu. ft. Find the work scale.

1 square inch represents P_sV_s ft.-lb.

∴ " " " " $144 \times 25 \times 2 = 7{,}200$ ft.-lb.

EXAMPLE. The area under a certain line on a pressure-volume diagram is 3·13 square inches. The scales of pressure and volume are 30 lb. per square inch per inch, and 2·5 cubic feet per inch, respectively. Find the work represented by the given area.

Work scale $= 144p_sV_s$ ft.-lb. per sq. in.

$= 144 \times 30 \times 2{\cdot}5 = 10{,}800$ ft.-lb. per sq. in.

Work represented by area $= 3{\cdot}13 \times 10{,}800 = 33{,}804$ ft.-lb.

7. Work done when both Pressure and Volume Change. Returning to the state point of the substance, consider the point to move along the curved path (1) to (2), fig. 4. At some point where the pressure is P lb. per sq. ft., let an increase in volume of δV cu. ft. be taken.

If, during the increase in volume, the pressure remained constant, the work done would be $P \times \delta V$ ft.-lb., and $P \times \delta V$ would also be proportional very nearly to the area of the strip shown. By making δV smaller the error would be less. If the whole area under the curve be divided into strips, each strip could be made to represent the work done during a slight increase in volume as nearly as we choose by making δV

sufficiently small. Adding all these strips together, the total amount of work represented by them may be made as near the actual amount of work done as we choose, when the volume increases from (1) to (2).

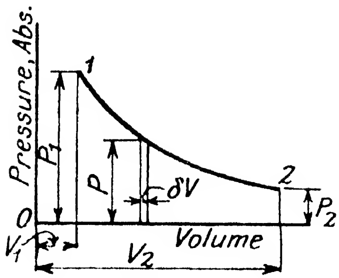


FIG. 4.

Also, by choosing smaller values of δV, the sum of the areas of all strips approaches more nearly to the actual area under the curve. Hence, when δV is infinitely small, and we have an infinite number of strips, the sum of all PδV products equals the work done, and the sum of all strips equals the area under the curve. Thus the area under the curve, to the proper work scale, equals the work done during the process.

It will be shown later that when the law connecting pressure and volume is known, this work can be calculated.

The indicator diagram from an engine cylinder is a pressure-volume diagram, and it is advisable to treat it as such, and not as a pressure-stroke diagram. As pressure-volume lines will be largely used to illustrate what is happening in engine cylinders, an appreciation of their value, and knowledge of their use, will be of great service in future consideration of the important engine cycles.

EXAMPLES I

1. A draught gauge in the form of a **U**-tube containing water shows a difference of level in the two limbs of 0·75 inch. Calculate the difference in pressure in lb. per sq. ft. indicated by this reading. Express the answer also in lb. per sq. in. One cu. ft. of water weighs 62·4 lb.

2. A mercury **U**-tube gauge is used to test the pressure of air in a pipe. The reading of the gauge is 3·5 inches difference of level. Find the absolute pressure of the air in lb. per sq. in. Assume the outside atmospheric pressure to be 14·7 lb. per sq. in.

3. The ram of a boiler feed pump is 4 inches diameter and the stroke is 8 inches. Find the work done per stroke by this pump when forcing water into a boiler whose pressure gauge reads 150 lb. per sq. in.

4. How much work is required to pump 1 cubic foot of water into a boiler the gauge pressure being 200 lb. per sq. in.?

5. A quantity of gas is heated at constant pressure and under these conditions the volume changes from 3·2 cu. ft. to 3·8 cu. ft. The pressure is 180 lb. per sq. in. absolute. Find the total work done during the heat supply period.

6. A pressure volume diagram for a gas is set down to a scale of 1″ = 200 lb. per sq. in. and 1″ = 0·4 cu. ft. The total area under the expansion line on this diagram is 2·31 sq. in. Find the work done during the expansion.

7. The area of a gas engine indicator card is 0·68 sq. in. The volume swept by the piston is 0·23 cu. ft. and the diagram is set down to a pressure scale of 300 lb. per sq. in. per inch. Find the work per cycle, the length of the diagram being 2·25 inches.

8. What do you understand by the "work scale" of a diagram? An ideal indicator diagram for a gas engine is plotted to a scale of 1″ = 0·3 cu. ft. and 1″ = 75 lb. per sq. in. The area of the enclosed figure is 2·5 sq. in., and the length of the diagram is $5\frac{1}{2}$ inches. What is the work scale of the diagram? Find from this the hypothetical mean effective pressure.

W. & D.M.T.C.

CHAPTER II

POWER AND ENERGY

8. Power is the Rate of Doing Work. This statement introduces the idea of time. The amount of work done in unit time is a measure of the power. If one engine does twice as much work as another in one minute, then it has twice the power. Hence

$$\text{Power} = \frac{\text{Work done}}{\text{Time}}.$$

If an engine, working at a uniform rate, does 270,000 ft.-lb. of work in 3 minutes,

$$\text{then work done per minute} = \frac{270{,}000}{3}$$

$$= 90{,}000 \text{ ft.-lb. per min.}$$

The power of this engine may be said to be 90,000 ft.-lb. per minute.

9. Unit of Power. The unit of power which has been chosen as convenient for practical engineering purposes is 33,000 ft.-lb. *per minute.*

Note well that it is not merely 33,000 ft.-lb.

This unit is called one *horse power.*

The horse power of the above engine is

$$\frac{90{,}000}{33{,}000} = 2{\cdot}73 \text{ horse power.}$$

One horse power, being 33,000 ft.-lb. per minute, is also 550 ft.-lb. per second, or $33{,}000 \times 60$ ft.-lb. per hour = 1,980,000 ft.-lb. per hour.

Although these numbers are different, the *rate* is the same, and all are one horse power.

10. Useful Horse Power. The power delivered at the coupling, or driving pulley, of an engine, or at the brake when the engine is under test, is called the Brake Horse Power. This is less than the horse power developed in the engine cylinder. This is because work has been absorbed at various points in the working parts of the engine, that is, a proportion of the power has been expended in overcoming engine resistances. It should be clearly understood, therefore, that the number of brake horse power units is less than that produced in the cylinder, but this is not because a brake horse power is less than any other. The value of the horse power is exactly the same, namely, 33,000 ft.-lb. per minute.

EXAMPLE. The work produced in an engine cylinder in 7 minutes is 287,000 ft.-lb., and the work done against the engine brake in 10 minutes is 280,000 ft.-lb. Find the horse power produced in the cylinder, and that delivered to the brake.

Cylinder horse power, or Indicated horse power

$$= \frac{\text{work done per minute}}{33,000}$$

$$= \frac{287,000}{7} \div 33,000 = \frac{41,000 \text{ ft.-lb. per min.}}{33,000}$$

$$= 1 \cdot 24 \text{ H.P.}$$

$$\text{Work done at brake} = \frac{280,000}{10} \text{ ft.-lb. per min.}$$

$$= 28,000 \quad \text{,,} \quad \text{,,} \quad \text{,,}$$

$$\text{Brake horse power} = \frac{28,000}{33,000} = 0 \cdot 848 \text{ H.P.}$$

Note that the value of the horse power is the same, whether in the cylinder or at the brake. It is the number of units of power which is less at the brake.

11. Energy. This may be defined as the capacity for doing work. It may be regarded as a store of something upon which we may draw, when suitable conditions are created, for the purpose of doing work. These conditions depend upon the nature of the source of energy, and they will be discussed as they affect our subject.

The many forms of energy known to science include (1) mechanical energy, (2) thermal or heat energy, (3) electrical energy, (4) chemical energy. Our concern is chiefly with the relationships existing between mechanical and thermal energy and the transformation of the latter into the former. It is generally possible to change one of the above forms of energy into another form.

The transformation of thermal into mechanical energy is the fundamental function of all Heat Engines, and its study is of vital importance to the engineer.

Transformation of mechanical energy into thermal energy in an engine is not of service. Heat thus produced is said to be degraded and is generally useless.

12. Conservation of Energy. Energy appears to be indestructible, and observation leads to the general conclusion which is known as the "Principle of the Conservation of Energy." It may be stated as follows: "Energy cannot be created or destroyed." This principle is essential in establishing many very important results in the theory of heat engines. It must be remembered that energy transformation is not energy loss. Clear recognition of this fact will avoid difficulty later.

13. Conversion of Energy. Because energy is convertible, it is necessary to know something of equivalent amounts of energy in its different forms. We therefore have to use conversion factors, to change an amount of energy, expressed in one kind of unit, into the corresponding amount, expressed in some other unit. Thus if energy is given in electrical units, we must use a conversion factor to change it into mechanical units.

The factor with which we are mainly concerned is the one for converting heat units into units of mechanical energy, usually foot-lb. This particular factor is known as Joule's Mechanical Equivalent of Heat, and has been obtained as the result of careful experimental measurement.

The value of the equivalent is—

One British Thermal Unit = 778 ft.-lb. or

One Centigrade Heat Unit = 1,400 ft.-lb.

The British Thermal Unit may be defined as the mean value of the quantity of heat required to raise the temperature of 1 lb. of water through 1° F.

This mean value is for the range of temperature from 32° F. to 212° F.

The Centigrade Heat Unit may be defined as the mean value of the quantity of heat required to raise the temperature of 1 lb. of water through 1° C.

Example. It is found by experiment that 1 lb. of a certain coal will give out 14,400 British Thermal Units (B.Th.U.) when completely burned. How much energy is this equal to in ft.-lb., and how far would this energy raise a weight of 1 ton vertically, supposing the whole of it could be applied to this purpose?

Heat energy available in coal per lb. = 14,400 B.Th.U.

Equivalent mechanical energy available in coal per lb.

$$= 14{,}400 \times 778 \text{ ft.-lb.}$$
$$= 11{,}200{,}000 \text{ ft.-lb.}$$

If the weight of 2,240 lb. is raised "h" feet

$$2{,}240h = 11{,}200{,}000 \text{ ft.-lb.}$$

$$h = \frac{11{,}200{,}000 \text{ ft.-lb.}}{2{,}240 \text{ lb.}} = 5{,}000 \text{ feet.}$$

We shall see later that only a fraction of this energy could be expended in doing mechanical work.

Example. The thermal energy stored in 1 lb. of petrol is 18,000 B.Th.U. A petrol engine consumes 15 lb. of petrol per hour, and converts into mechanical energy 20 per cent. of that supplied. What is the power of the engine?

Heat energy supplied to engine per hour $= 18{,}000 \times 15$

$= 270{,}000$ B.Th.U.

$$\text{Heat energy supplied per min.} = \frac{270{,}000}{60} = 4{,}500 \text{ B.Th.U.}$$

Mechanical energy supplied per min. $= 4{,}500 \times 778$

$= 3{,}501{,}000$ ft.-lb.

Since only 20 per cent. of energy is converted, we have

$$\text{Work done by petrol engine per min.} = \frac{3{,}501{,}000 \times 20}{100}$$

$$= 700{,}200 \text{ ft.-lb.}$$

$$\text{H.P. of engine} = \frac{700{,}200}{33{,}000} = 21{\cdot}2 \text{ H.P.}$$

Example. An engine of 200 I.H.P. is supplied with steam, each pound of which carries 1,152 B.Th.U. to the engine. If 12 per cent. of the heat supplied is converted into work on the pistons, how much steam is required by this engine per hour?

Work to be done on engine piston per min. $= 200 \times 33{,}000$ ft.-lb.

,, ,, ,, ,, ,, ,, hour $= 200 \times 33{,}000 \times 60$ ft.-lb.

$$\text{Heat equivalent of this work} = \frac{200 \times 33{,}000 \times 60}{778} \text{ B.Th.U. per hr.}$$

$$= 200 \times 2{,}545 \text{ B.Th.U. per hr.}$$

$$= 509{,}000 \quad ,, \quad ,, \quad ,,$$

Heat carried to engine per lb. of steam $= 1{,}152$ B.Th.U.

Heat converted into work per lb. of steam $= \frac{12}{100} \times 1{,}152$ B.Th.U.

$= 138{\cdot}24$,,

Weight of steam required per hour

$$= \frac{509{,}000}{138{\cdot}24} = 3{,}682 \text{ lb. per hr.}$$

Examples II

1. The area of an indicator diagram is 2·46 sq. in. The pressure scale is 1″ = 64 lb. per sq. in., and the volume scale 1″ = 0·5 cu. ft. The engine completes 220 of these cycles per minute. Calculate its indicated horse power.

2. The draw-bar pull of a locomotive when travelling at 30 miles per hour is 12,000 lb. Find the horse power expended by the engine upon the train.

3. A pump lifts 10,000 gallons of water per minute through a vertical height of 200 feet. Calculate the horse power thus expended. For the purpose of pumping, coal whose calorific value is 7,800 C.H.U. per lb. is used. Only 10 per cent. of this heat is effectively used by the plant. Find the weight of coal burned per hour. 1 gallon of water weighs 10 lb.

4. An alternator generates 15,000 kilowatts. The overall efficiency of boilers, engines and dynamo may be taken as 16 per cent. What weight of coal, whose calorific value is 12,000 B.Th.U., will be burned per hour? 1 kilowatt = $1\frac{1}{3}$ horse power.

5. Express 2,545 B.Th.U. per hour in horse power.

6. A fan delivers air at a water-gauge pressure of 3 inches. The volume of air passing through the fan is 200,000 cu. ft. per minute. Find the horse power required to drive the fan.

7. One pound weight of steam contains 1,200 B.Th.U. Of this quantity, 55·8 B.Th.U. are changed into kinetic energy. Find the velocity of the steam in feet per second.

8. The area of an indicator diagram taken from a double-acting steam engine is 2·1 sq. in., and its length is 3·5 inches. The strength of the spring used is 60 lb. per inch of height. The diameter of the cylinder is 18 inches, and the piston stroke is 26 inches. Speed, 150 r.p.m. Find the indicated horse power of the engine and, if the mechanical efficiency is 78 per cent., the brake horse power. U.L.C.I.

9. A double acting steam engine has a cylinder diameter of 6 inches and the stroke of the piston is 10 inches. The mean effective pressure of the steam in the cylinder is 40 lb. per sq. in. At what speed in revolutions per minute should the engine run in order to develop 20 horse power.
U.L.C.I.

10. At what level is the " feed check " valve generally fitted to a boiler and why?

Find the horse power required to drive a boiler feed pump which delivers 800 gallons of water per hour against a boiler pressure of 185 lb. per sq. in. by gauge. Efficiency of pump, 75 per cent. U.L.C.I.

CHAPTER III

HEAT AND ITS MEASUREMENT

14. Heat as Energy. Heat has been described as a form of energy, and must, therefore, be capable of doing work. In order that work may be done motion is essential, and this motion must be associated with applied force.

According to the kinetic theory of heat, heat is a measure of the molecular activity within a substance; it is said to be equal to the kinetic energy of the molecules of which the heated substance consists.

15. Specific Heat. The quantity of heat, which is taken in by, or removed from a body, depends upon three things:

(1) The mass of substance in the particular body.

(2) A property of the substance known as its Specific Heat.

(3) The change of temperature which takes place.

To assist the student in forming a conception as to how these three points affect the problem, we may consider a quantity of water being heated by a gas burner, or some other agency, which ensures the transmission of heat to the water at a uniform rate. A definite time will be needed to raise the water to boiling point, but if the mass of water be doubled, or increased in any other ratio, then the time of heating will be increased in the same ratio. This is because the heat added must be in proportion to the mass. The change of temperature or temperature range is the same in both cases. Again, if we take a mass of water, and apply the heat to raise its temperature from 50° F. to 80° F., and note the time taken, we should find on replacing the water by the same mass of some other liquid and raising the temperature of this liquid from 50° F. to 80° F., that the time would be different—usually less. This is due to the fact that different substances have different capacities for absorbing heat or different specific heats. If the quantity of heat taken in by the water is a quantity 1, say, and the other substance has absorbed a quantity C, then the ratio $\frac{C}{1} = C$ is called the specific heat of the substance. The specific heat of water is taken as unity in general.

Finally, if with the uniform flow of heat, we heat a mass of water from 50° F. to 80° F., thus giving a temperature range of 30°, and then, taking an equal quantity of water, we heat it up from 50° F. to 110° F., thus giving a range of 60°, we find the time doubled because the temperature range is doubled. Thus we see that the quantity of heat is proportional to the increase in temperature.

All the above reasoning would apply equally well if the substances were being cooled, and thus giving out, instead of absorbing heat at a definite rate.

16. Heat Calculation. In consideration of the preceding article, the following equation should hold:

Heat added = mass of substance × specific heat × temperature rise,

and all experiment tends to prove it true.

We have seen that specific heat may be regarded as a ratio, but it is sometimes convenient to treat it as a quantity of heat.

Definition. The specific heat of a substance is the *ratio of the quantity of heat* required to raise a definite mass of that *substance* through a given temperature range to the *quantity of heat required* to raise the same mass of *water* through the same temperature range.

Let C = specific heat of substance which may be a gas, liquid, or solid.

C_w = specific heat of water.

W = a given mass of water and also mass of substance.

$T_1 - T_2$ = rise in absolute temperature.

Then specific heat of substance $= \frac{C \times W \times (T_1 - T_2)}{C_w \times W \times (T_1 - T_2)}$

$= C$, when $C_w = 1$

EXAMPLE. The specific heat of air at constant pressure is 0·238. Find the amount of heat necessary to raise the temperature of $1\frac{1}{2}$ lb. of air from 200° C. absolute to 250° C. absolute.

$$\begin{aligned}\text{Heat added} &= \text{mass} \times \text{sp. ht.} \times \text{temperature rise}\\ &= 1\tfrac{1}{2} \times \cdot 238 \times (250 - 200)\\ &= 1\tfrac{1}{2} \times \cdot 238 \times 50 = 17\cdot 85 \text{ C.H.U.}\end{aligned}$$

The absolute temperatures are in centigrade degrees, and the student may note that the temperature rise would be exactly the same, if the temperatures were on the ordinary centigrade scale.

Now consider specific heat as a heat quantity instead of a ratio.

Definition. The specific heat of a substance is the quantity of heat required to raise one pound of that substance through one degree.

Thus 1 pound of iron requires 0·1146 centigrade heat unit to raise its temperature through one degree centigrade, or it requires 0·1146 British Thermal Unit to raise its temperature through one degree Fahrenheit. Hence the specific heat of iron = 0·1146 heat units per lb. per degree. Since one centigrade heat unit is required to raise 1 lb. of water through one degree centigrade, and one British Thermal Unit to raise 1 lb. of water through one degree Fahrenheit, we see that both definitions of specific heat give the same numerical value.

A table of specific heats of substances commonly met with in engineering is appended.

TABLE I

Water	1·0	Copper	0·0965
Ice	0·504	Brass	0·0940
Aluminium	0·2143	Zinc	0·0956
Cast iron	0·125	Gunmetal	0·0952
Cast steel	0·117	Nickel	0·1086
Mild steel	0·1158	Mercury (liquid)	0·0333
Wrought iron	0·1146	Coal	0·2412
Tin	0·0562	Petroleum	0·4340

17. Water Equivalent. It is often necessary to find the mass of water which has the same heat capacity as a given mass of some other substance. Here the idea of specific heat as a ratio has a direct application.

The water equivalent of a body may be defined as *the mass of water which requires the same quantity of heat to raise its temperature one degree as would be required to raise the temperature of the body one degree.*

EXAMPLE. Find the water equivalent of an iron container which weighs 20 lb. Specific heat of iron = 0·1146.

$$\text{Water equivalent of container} = 20 \times 0{\cdot}1146 = 2{\cdot}292 \text{ lb.}$$

That means that 2·292 lb. of water would absorb the same amount of heat as 20 lb. of iron, when the temperature rise is the same, or that 20 lb. of iron is the *equivalent in heat capacity* of 2·292 lb. of water.

EXAMPLE. An empty copper vessel weighs 2·19 lb. and 8·5 lb. of water are poured into it. The temperature of the water and vessel is then raised 20° F. Find the heat supplied to the system. Specific heat of copper = 0·0965.

Water equivalent of copper vessel = 0·0965 × 2·19 = 0·211 lb.
Total water equivalent of system = 8·5 + 0·211 = 8·711 lb.
Heat required to raise temperature 20° F. = 8·711 × 20 = 174·22 B.Th.U.

EXAMPLE. The parts of a simple calorimeter for testing fuels consist of 4·6 lb. of glass and 1·2 lb. of steel. Water weighing 2,560 grams is poured in. Find the heat required to raise the temperature of the whole system 3½° F., if the specific heat of steel = 0·1158 and of glass = 0·198.

Water equivalent of glass = 4·6 × 0·198 = 0·912 lb.
" " " steel = 1·2 × 0·1158 = 0·139 lb.

$$2{,}560 \text{ grams of water} = \frac{2{,}560}{454} \text{ lb. of water} = 5{\cdot}64 \text{ lb.}$$

Total water equivalent of system = 6·691 lb.

Heat required to raise calorimeter and water through 3½° F.
= 6·691 × 3·5 = 23·4 B.Th.U.

18. Calorimetry. The above examples indicate the method of using specific heats in the calculation of quantities of heat, and in heat measurement.

The measurement of heat quantities is a portion of science known as calorimetry, and whenever the term is met with, it should at once convey the idea of measuring heat quantity.

19. Thermometry. The measurement of heat will be seen to involve the measuring of a temperature change, and this requires the use of either a thermometer or a pyrometer. The student is supposed to be familiar with the ordinary type of glass thermometer containing mercury, or some other liquid. Such thermometers can only measure moderate temperatures. Some form of pyrometer is generally used for measuring high temperatures, such as the temperatures of molten metals or of furnaces, and such temperature measurement is generally known as *pyrometry*.

The most convenient pyrometers for use by engineers are based upon some electrical phenomenon. A full discussion of these is outside the scope of this book, and for fuller information the student may refer to books on electricity. We may, however, point out that electrical pyrometers depend, either upon the measuring of the increase in the resistance of a platinum wire when it is heated, or upon the measurement of a small electric current generated when the junction of two dissimilar metals is heated. As the latter can be measured by a form of voltmeter,

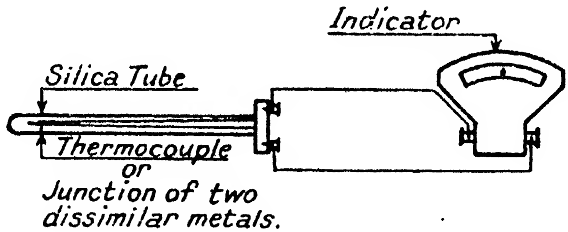


FIG. 5.

which is calibrated directly in temperatures, it requires no electrical knowledge to read it, and thus is very convenient for use in either engineering practice, or the engineering laboratory.

20. Specific Heat of a Gas. Hitherto we have dealt with specific heats in a general way, and have made no special reference to any peculiar property of the substance concerned. In treating solids and liquids, for all practical purposes nothing need be added, but the treatment of specific heats in relation to gases and vapours needs extending.

A vapour may be regarded as a gas near its temperature of liquefaction, and for the present, they may be treated alike.

Typical vapours are steam and ammonia, and typical gases are air, oxygen, nitrogen.

In converting heat into mechanical energy, we find it necessary to use some working substance, and so far, gases and vapours have been found to be the most convenient. This is because their expansion or increase in volume is so marked in contrast to that of solids and liquids. An exact discussion of the expansion of gases will be found in the next chapter.

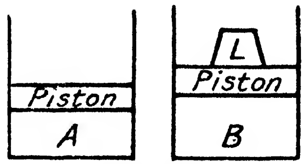


FIG. 6.

Let us now consider a quantity of gas enclosed in the space A, fig. 6. Let the piston shown be considered fixed so that the volume of space A cannot change nor can any gas escape. Suppose the gas in A to be heated from an initial temperature T_1 to a final temperature T_2; the pressure will vary but the volume cannot; hence the gas in A is heated at *constant volume*. The former equation for the heat added will hold here, namely:

Heat added = weight of gas $\times$ specific heat at constant volume $\times$ rise in temperature.

Let W = weight of gas, let C_v = sp. ht. at constant volume.

Then, heat added = $W \times C_v \times (T_2 - T_1)$ heat units.

Now suppose the piston in B to be movable, and let there be a load L upon it. This will ensure that the pressure in B remains constant.

On heating the gas in B, expansion will now occur, and again we will suppose that the initial temperature is T_1, and the final temperature T_2.

In this case the volume changes and the piston will rise, thus doing work on the weight L. Note that the rise in temperature is the same as before $(T_2—T_1)$, but in addition work has been done, hence, it is found that more heat has been passed into the gas if the weight of gas is W as before. Our equation for the heat added, therefore, can only be satisfied if the specific heat is greater than in the case of the cylinder A. We have, then,

Heat added = $W \times$ sp. ht. at constant pressure $\times (T_2 - T_1)$

or putting C_p = sp. ht. at constant pressure

Heat added = $W \times C_p \times (T_2 - T_1)$

Experiment proves our expectation that C_p is greater than C_v.

It is essential to grasp the following three points so as to facilitate the work later.

(1) A gas has two specific heats.

(2) The conditions under which a gas is heated have an important bearing on the quantity of heat required to produce a certain rise in temperature.

(3) That, otherwise, the method of calculating heat added to a gas is the same as that for a solid or liquid.

For the purposes of this work, we shall assume that the specific heats of any gas are constant throughout the whole temperature range. This is not strictly true of the gases met with in heat engines, but in dealing with the questions we shall consider, the results will not be materially affected. We have seen that C_p is greater than C_v, because in heating a gas at constant pressure, work is done on an external body in addition to raising the temperature. As the doing of work necessitates expenditure of energy, some of the heat *supplied* to the gas has not remained in it, or has not *entered* permanently into it. We may thus distinguish between heat *supplied* to a gas and heat *entering* a gas.

EXAMPLE. Heat is supplied to 5 lb. of air first at constant pressure and then at constant volume, so as to raise the temperature from 500° F. abs. to 575° F. abs. Find the difference in heat supplied in the two cases. $C_p = 0{\cdot}237$; $C_v = 0{\cdot}169$.

Express the difference in ft.-lb.

Heat supplied at constant pressure =
$5 \times 0{\cdot}237 \times (575 - 500) = 5 \times 0{\cdot}237 \times 75 = 88{\cdot}87$ B.Th.U.

Heat supplied at constant volume =
$5 \times 0{\cdot}169 \times (575 - 500) = 5 \times 0{\cdot}169 \times 75 = 63{\cdot}37$ B.Th.U.

Difference $= 88{\cdot}87 - 63{\cdot}37 = 25{\cdot}5$ B.Th.U.

Expressed in ft.-lb., $25{\cdot}5$ B.Th.U. $= 25{\cdot}5 \times 778$ ft.-lb.
$= 19{,}837$ ft.-lb.

We shall see later that this is the work which would be done on an external body.

The student may note here that when a portion of the atmosphere is heated, such as the air in an ordinary room, it is heated at constant pressure, and therefore the value C_p must be used.

Examples III

1. A block of copper weighing 5 lb. is taken from a furnace and dropped into a copper calorimeter weighing 0·75 lb. and containing 6·5 lb. of water. Initial temperature of calorimeter and water was 57° F. and after the copper had been added it rose to 136° F. Find the temperature of the furnace.

2. Mercury at a temperature of 140° F. and weighing 1·5 lb. is poured into 2·4 lb. of water at 59° F. Find the final temperature of the mixture.

3. A copper vessel weighs 2·8 lb. and contains 4 lb. of petroleum and 8·56 lb. of aluminium. Find the water equivalent of the whole. How much heat is needed to raise the temperature 32·83° F.?

4. Cooling water is supplied to a condenser at 51·8° F. and leaves it at 88·7° F. The supply is 1,250 lb. per minute. Find the heat carried away in ft.-lb. per minute and express it in horse power.

5. The air supplied to boiler furnaces is sometimes pre-heated. In a similar case, the boiler-house temperature was 64·4° F. and the heated air was at a temperature of 383° F. The weight of air supplied per lb. of coal is 18·4 lb. Find the quantity of heat supplied by the air to the furnace per lb. of coal burnt, if the specific heat of air at constant pressure is 0·238.

6. An oil-fired boiler takes 21·6 lb. of air per lb. of oil. The products of combustion may be taken as equal in weight to oil and air. The oil-supply and air-supply are both fed at boiler room temperature, which is 60·8° F. If the temperature of the flue gases is 500° F. and their specific heat is 0·24, calculate the quantity of heat carried away per lb. of oil consumed.

CHAPTER IV

THERMAL PROPERTIES OF PERFECT GASES

21. First Law of Thermodynamics. This law states a principle with which the student has already gained some familiarity and may be expressed as follows. Heat and mechanical energy are mutually convertible.

The conversion factor is called Joule's Mechanical Equivalent, and is such that one centigrade heat unit = 1,400 ft.-lb., or one British Thermal unit = 778 ft.-lb.

22. Principle of the Heat Engine. Mention has already been made (see art. 5) of a working substance. Engineers always use a gas or vapour as the working substance in a heat engine, nothing else having been found as yet to be practicable. Moreover, we have not yet discovered any practical method of changing large quantities of heat into work without the agency of a working substance.

This working substance receives and gives up heat, changing its pressure, volume and temperature to suit the circumstances, but it should be clearly understood that none of the working substance is destroyed or reduced in quantity.

The study of heat engines is largely directed to calculations relating to the changing pressure, volume, and temperature of the working substance. These quantities are determined when we know the *state* of the gas, and it is vital to know what relationships exist between the state of a gas, and its heat content.

We are, on this account, deeply concerned with the *properties* of the *working substance.*

23. State of a Substance. By the state, we mean the condition of *pressure*, *volume*, *temperature* and *heat content* at any instant

Unless otherwise stated, the amount of substance considered in calculations relating to the working substance is generally 1 lb., and most tables of properties, and standard diagrams, are prepared for 1 lb. of substance. Omission of a weight generally means assuming the weight to be unity, and failure to recognize this sometimes causes the student some difficulty.

Knowledge of the state of a substance will fix the properties of unit weight which are necessary to determine the amount of work done by 1 lb., or by 1 cu. ft.

Again, the state leads to a knowledge of the energy available in unit weight of the substance. For example, with the aid of steam tables, the state of steam is known when its pressure and temperature, or pressure and dryness fraction are known, and from this the volume and energy content can be deduced.

The state of a perfect gas is defined when we know the pressure, volume of 1 lb., and the temperature. From this information we can determine the available energy, when we know certain figures which are properly called gas constants.

We may summarize by stating the following facts:

(1) The heat engine uses heat and converts it into mechanical work.

(2) The working substance is not destroyed, though its state, and, therefore, its heat content changes.

(3) The heat content gives valuable information, which can only be extracted when we know the thermal properties of the working substance.

24. Laws of Perfect Gases. A perfect gas is one which obeys Boyle's Law and to which Joule's law of energy is applicable. Both these laws are dealt with below.

25. Boyle's Law. The volume of a given mass of a perfect gas varies inversely as the absolute pressure *when the temperature is constant.*

Let P = the absolute pressure of the gas.
V = volume occupied when pressure = P.

Then the above law states that $V \propto \frac{1}{P}$.

This may be written as an equation by introducing a constant C.

$$\text{Then } V = \frac{C}{P} \text{ or } PV = C.$$

This shows that the product of the absolute pressure and volume of a *given quantity* of gas is constant, when the temperature does not change.

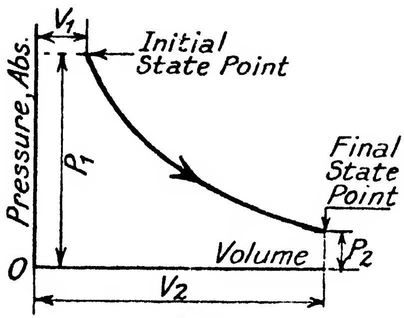


FIG. 7.

Let a quantity of gas at pressure P_1 and volume V_1 (see fig. 7) change its pressure and volume in a cylinder without change of temperature.

Let P_2 and V_2 be the final pressure and volume respectively.

Then anywhere on the curve $PV = C$.

$$\therefore P_1V_1 = C \text{ and } P_2V_2 = C.$$

$$\therefore P_1V_1 = P_2V_2$$

This is a useful working form of Boyle's Law as it avoids the necessity for calculating C.

EXAMPLE. Four cubic feet of gas at a pressure of 120 lb. per square inch absolute expand at constant temperature until the volume is 20 cubic feet. Find the final pressure.

Since temperature is constant the expansion is according to Boyle's Law, and fig. 7 may represent the change.

Initial state is P_1V_1. Final state is P_2V_2.

$$P_1V_1 = P_2V_2.$$

$$\therefore P_2 = P_1 \frac{V_1}{V_2}$$

$\left(\text{It may be noted that } P_2 = P_1 \div \text{a volume ratio, } \frac{V_2}{V_1}\right).$

$$\therefore P_2 = 120 \text{ lb. per sq. inch} \times \frac{4 \text{ cu. ft.}}{20 \text{ cu. ft.}}$$

$$= 24 \text{ lb. per sq. in. absolute.}$$

26. Charles's Law of Gases. Equal volumes of different gases expand equally for equal increases of temperature, provided that the *pressure remains constant* during the expansion.

It is found by experiment that the change in volume per degree centigrade of temperature change is $\frac{1}{273}$ of the volume of the gas at 0° C. Thus 5 cu. ft. of any gas at 0° C. would expand to $5 + (\frac{5}{273} \times$ temp. rise) cubic feet when the temperature increased to some value above 0°C., the pressure remaining constant.

When using the Fahrenheit scale of temperature the figure corresponding to 273 is $273 \times 1{\cdot}8 = 491{\cdot}6$, or 492 approximately. Hence the volume of a gas changes by $\frac{1}{491{\cdot}6}$ of its volume at 32° F. per degree Fahrenheit of temperature change if the pressure is constant.

EXAMPLE. Find the volume occupied when a given quantity of gas exists at a temperature of 400° C., if the volume of this quantity of gas at 0° C. is 20 cu. ft., the pressure being the same at both temperatures.

Since the pressure is the same throughout Charles's Law is applicable.

$$\text{Increase in volume} = \tfrac{1}{273} \times 20 \text{ cu. ft. per degree C.}$$

$$\text{Total increase in volume} = (\tfrac{1}{273} \times 20 \times 400) \text{ cu. ft.}$$

$$= \frac{8{,}000}{273} = 29{\cdot}3 \text{ cu. ft.}$$

$\therefore$ Final volume $= 20 + 29{\cdot}3 = 49{\cdot}3$ cu. ft.

Alternative View of Charles's Law. The above method of using Charles's Law is not the most convenient.

Imagine 273 cu. ft. of gas at 0° C., and at a pressure which remains constant whilst the temperature is reduced. Let the temperature be $-\ t$° C. and the volume at this temperature V. Then, if we assume that Charles's Law holds until the temperature reaches its lowest limit, we should get the following values of V and $-\ t$° C.

V	$-t$	T
273 cu. ft.	0° C.	273°
272 ,,	− 1° C.	272°
270 ,,	− 3° C.	270°
200 ,,	− 73° C.	200°
100 ,,	− 173° C.	100°
10 ,,	− 263° C.	10°
0 ,,	− 273° C.	0°

(It will be noticed that the volume has disappeared. In practice this cannot occur, for the gas would become liquid before − 273° C. was reached.)

If we constructed a new scale of temperature with its zero at $-273°$ C., we should get the values under T in the third column. Temperatures on this new scale are known as *absolute* temperatures. All that has been done is that 273 has been added to each value of $-t°$ C.

We now see that volumes and temperatures rise together in the same proportion and that

$$\frac{V}{T} = 1.$$

If the volume is made some multiple of 273, say 273 C, where C is a constant having any positive value whatsoever,

$$\text{then } \frac{V}{T} = C \text{ or } V = CT.$$

Thus the law of Charles may be stated as follows: The total volume of a given quantity of gas varies directly as the absolute temperature when the pressure is kept constant.

Stated algebraically the law is $V \propto T$ or $V = CT$ when pressure does not vary.

The useful form in the working of examples is

$$\frac{V}{T} = C \text{ or } \frac{V_1}{T_1} = \frac{V_2}{T_2}.$$

EXAMPLE. Five cubic feet of gas at 40° C. receive heat at constant pressure so that the final temperature is 200° C. Find the final volume, and write down the increase during the supply of heat.

Here Charles's Law applies, because the pressure is constant.

The pressure-volume diagram for this change is shown in fig. 8.

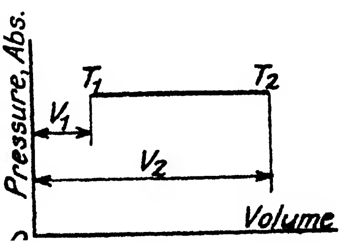


FIG. 8.

Since temperatures are not absolute, 273 must be added.

$$\therefore T_1 = 40 + 273 = 313° \text{ abs.}$$
$$\text{and } T_2 = 200 + 273 = 473° \text{ ,,}$$

Now
$$\frac{V_2}{T_2} = \frac{V_1}{T_1}$$

$$\therefore V_2 = \frac{V_1}{T_1}T_2 = \frac{5 \times 473}{313} \text{ cu. ft.}$$

$$= \frac{2{,}365}{313} = 7{\cdot}556 \text{ cu. ft.}$$

$\therefore$ Increase in volume $= 7{\cdot}556 - 5 = 2{\cdot}556$ cu. ft.

Examples such as the above may be solved when the temperatures are in Fahrenheit degrees. On the Fahrenheit scale, the absolute temperature corresponding to 32° F. is 491·6. Hence the absolute temperature corresponding to 0° F. is (491·6 — 32) = 459.6 (or 460 approximately). Consequently, in order to change a Fahrenheit temperature to absolute temperature 459·6 (or 460) must be added to it.

Example. Five cubic feet of a gas at 104° F. receive heat at constant pressure so that the final temperature is 392° F. Find the final volume, and write down the increase during the supply of heat.

$$T_1 = 104 + 460 = 564° \text{ abs.} \quad T_2 = 392 + 460 = 852° \text{ abs.}$$

$$\therefore V_2 = \frac{V_1}{T_1}T_2 = \frac{5 \times 852}{564} \text{ cu. ft.} = \frac{4260}{564} \text{ cu. ft.} = 7{\cdot}556 \text{ cu. ft.}$$

$\therefore$ Increase in volume $= 7{\cdot}556 - 5 = 2{\cdot}556$ cu. ft.

Example. The temperature of the charge of gas at the beginning of fuel admission in a Diesel engine cylinder is 350° C., and the volume of the gas charge at the end of the heat supply is 1·9 times the volume at the beginning. If the heat is supplied at constant pressure, find the temperature at the end of the heat supply period.

This again is Charles's Law. This application to an engine problem shows that these fundamental laws have their practical applications later. The P — V diagram is shown in fig. 8.

$$\frac{V_2}{V_1} = 1{\cdot}9 \text{ and } T_1 = 350 + 273 = 623° \text{ abs.}$$

$$\frac{T_2}{V_2} = \frac{T_1}{V_1}$$

$$\therefore T_2 = \frac{V_2}{V_1}T_1 = 1{\cdot}9 \times 623$$

$$= 1183{\cdot}7° \text{ absolute}$$

$$= (1183{\cdot}7 - 273)° \text{ C.}$$

$$= 910{\cdot}7° \text{ C.}$$

The working forms of Boyle's Law and Charles's Law are worthy of note for comparison.

Boyle's Law		*Charles's Law*	
$P_1V_1 = P_2V_2$ or $\frac{P_1}{P_2} = \frac{V_2}{V_1}$	Temperature Constant	$\frac{T_1}{T_2} = \frac{V_1}{V_2}$ or $\frac{V_2}{T_2} = \frac{V_1}{T_1}$	Pressure Constant

27. Combination of the Laws of Boyle and Charles. The pressure, volume and temperature of a gas may all change at once. In this case, because pressure changes, Charles's Law will not apply and, because the temperature changes, Boyle's Law will not apply. We require some principle by which to treat this important case.

This change of state may be regarded as taking place in two stages.

(*a*) By a change according to Boyle's Law, followed by
(*b*) a change according to Charles's Law.

Let us consider a given quantity of a perfect gas at pressure P_1, volume V_1, and temperature T_1, in a cylinder A (fig. 9); the same gas is found later to be in the state P_2, V_2, T_2, as in cylinder C.

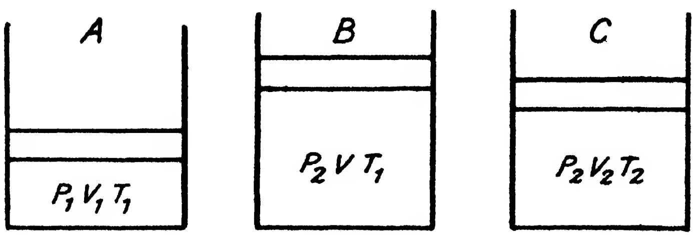


FIG. 9.

We may imagine an intermediate state as having existed—shown at B. Then, because the temperature is the same in A and B, the change from A to B follows Boyle's Law.

Hence
$$P_1V_1 = P_2V$$
$$\therefore V = \frac{P_1}{P_2}V_1 \quad \cdots \cdots \quad (1)$$

In the change from B to C the pressure remains the same whilst the temperature changes; hence Charles's Law applies.

$$\therefore \frac{V}{V_2} = \frac{T_1}{T_2}$$
$$\therefore V = \frac{T_1}{T_2}V_2 \quad \cdots \cdots \quad (2)$$

In equations (1) and (2), the volume V is the volume in B and hence is the same quantity for both equations.

$$\therefore \frac{P_1}{P_2}V_1 = \frac{T_1}{T_2}V_2 \quad \cdots \quad (3)$$
$$\therefore \frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \quad \cdots \cdots \quad (4)$$

It is obvious, by similar reasoning, that if this quantity of gas underwent a further change to the state P_3, V_3, T_3, the equation (4) could be added to as below.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3}$$

This result may be expressed thus: The product of the pressure and volume of a quantity of gas divided by its absolute temperature is a constant and this again may be written algebraically

$$\frac{PV}{T} = K \text{ or } PV = KT \quad \cdots \cdots \quad (5)$$

where K is a constant.

In considering Boyle's and Charles's Laws we have said nothing about the weight of the gas concerned.

The weight is important in many calculations because it is needed to calculate heat quantities. The weight depends upon the density, that is, upon the specific volume of a gas.

Definition. The density of a gas or vapour is the weight of unit volume at a given temperature and pressure.

Usually we shall state it as the weight of one cubic foot.

Definition. The specific volume of a gas or vapour is the volume of unit weight at some stated temperature and pressure. We shall generally state it in cubic feet per lb.

28. Characteristic Equation of a Gas. We shall now show that the constant K in equation (5) includes the weight of the gas.

Let V_s = the specific volume of a particular gas, whilst P and T are its pressure and temperature.

Then $V = wV_s$ where w is the weight of gas used.

Then we have
$$PV = KT$$
$$\therefore PwV_s = KT$$
$$\therefore PV_s = \frac{K}{w}T \quad \cdots \quad (6)$$

$\frac{K}{w}$ is a new constant. If, then, we deal with 1 lb. of gas which is represented by V_s, the value $\frac{K}{w}$ will always be the same for any given kind of gas.

$$\text{Let } \frac{K}{w} = R, \text{ then } K = wR$$

and equation (5) becomes
$$PV = wRT \quad \cdots \quad (7)$$
and we may also write equation (7) in the form
$$PV_s = RT \quad \cdots \quad (8)$$

Both equations (7) and (8) are very important ones, whilst equation (8) is known as the *Characteristic Equation* of a gas, and the value of R is called the *Characteristic Constant* for the gas.

29. Units of the Characteristic Equation. In using this equation, careful attention needs to be given to the units, and the student is recommended to make himself familiar with one system, confining himself to that system throughout.

The following are the most convenient units for our purpose.

P = absolute pressure in lb. per sq. ft.

T = absolute temperature in centigrade or Fahrenheit degrees.

V_s = volume of 1 lb. wt. of gas in cu. ft.

We may interpret the meaning of R in the following way.

Let 1 lb. of gas be enclosed in a cylinder fitted with a piston, and let it be heated at constant pressure so that its temperature is increased by one degree from some value T, to (T + 1). The volume will increase from, say, V_1 to V_2, moving the piston against the uniform pressure P.

Now $PV_1 = RT$

and $PV_2 = R(T + 1)$

$\therefore$ subtracting $P(V_2 - V_1) = R(T + 1) - RT = R$.

We have seen (art. 4) that $P(V_2 - V_1)$ is work done. Hence, we see that R is the work done against external resistance when 1 lb. of gas is heated through one degree at constant pressure. The constant, R, thus stands for a quantity of energy, and, with the units adopted in this article, will be in ft.-lb.

If one pound of a gas is heated at constant pressure through a temperature range $(T_2 - T_1)$, the external energy given to the gas is $R(T_2 - T_1)$.

The numerical value of R will vary according to the temperature scale used. If the Fahrenheit degree is used, then, since it represents a smaller temperature rise, the work done per degree will be less and hence R will have a smaller value.

When air is the gas, the value of R is 96 ft.-lb. per centigrade degree, and 53·2 ft.-lb. per Fahrenheit degree.

EXAMPLE. The pressure, volume and temperature of a quantity of gas are respectively 100 lb. per sq. in. abs., 3 cu. ft., and 302° F. ordinary temperature. A change of state results in the following, volume 9 cu. ft., temperature 50° F. What is the pressure?

Note that no weight is given, hence we use equation (4).

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\therefore\ P_2 = P_1 \times \frac{V_1}{V_2} \times \frac{T_2}{T_1}$$

$$= (100 \times 144) \times \frac{3}{9} \times \frac{510}{762}$$

$$= 3{,}210 \text{ lb. per sq. ft.}$$

$$= 22{\cdot}3 \text{ lb. per sq. in.}$$

EXAMPLE. The state of a quantity of gas enclosed in a gas engine cylinder at the beginning of compression is volume 1·8 cu. ft., temperature 80° C., and pressure 14 lb. per sq. in. What will be the temperature after the volume is reduced to 0·5 cu. ft., and the pressure raised to 105 lb. per sq. in.?

Fig. 10 indicates the change on the P–V diagram.

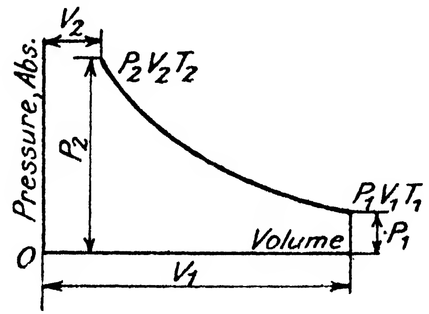


FIG. 10.

Equation (4) is again used.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$\begin{aligned} T_2 &= T_1 \times \frac{P_2}{P_1} \times \frac{V_2}{V_1} \\ &= 353 \times \frac{105 \times 144}{14 \times 144} \times \frac{0{\cdot}5}{1{\cdot}8} \\ &= \frac{353 \times 7{\cdot}5}{3{\cdot}6} = 736^\circ \text{ C. absolute} \\ &= 736 - 273 = 463^\circ \text{ C. ordinary.} \end{aligned}$$

The student may note that the pressure ratio and the volume ratio are the important factors in this example, and the units chosen are immaterial.

EXAMPLE. The characteristic equation of a certain gas is PV = 98T, T being in centigrade degrees. Find the weight of this gas which occupies 200 cu. ft. at 100 lb. per sq. in. abs. and 200° C. (ordinary temperature).

Equation (7) gives the desired result.

$$PV = wRT$$

$$\therefore\ w = \frac{PV}{RT} = \frac{100 \times 144 \times 200}{98 \times 473}$$

$$= 62{\cdot}2 \text{ lb.}$$

We may proceed thus:

Vol. of 1 lb. of gas at 100 lb. per sq. in. and 473° abs.

$$= V_s = \frac{RT}{P}$$

$$\therefore\ V_s = \frac{98 \times 473}{100 \times 144} = 3{\cdot}22 \text{ cu. ft.}$$

$$\therefore \text{ Density of gas} = \text{wt. per cu. ft.} = \frac{1}{3{\cdot}22} \text{ lb. per cu. ft.}$$

$$\therefore \text{ Weight} = \text{volume} \times \text{density} = \frac{200}{3{\cdot}22} = 62{\cdot}2 \text{ lb.}$$

EXAMPLE. Three pounds of gas at 120 lb. per sq. in. and 176° F. are placed in a cylinder. What is the volume of the cylinder if the characteristic constant is 53·9?

$$PV_s = RT$$

$$\begin{aligned}\therefore \text{ vol. of 1 lb.} &= \frac{RT}{P} = \frac{53{\cdot}9 \times 636}{120 \times 144}\\ &= 1{\cdot}98 \text{ cu. ft.}\\ \therefore \text{ Volume occupied by 3 lb.} &= 1{\cdot}98 \times 3\\ &= 5{\cdot}94 \text{ cu. ft.}\\ &= \text{volume of cylinder.}\end{aligned}$$

EXAMPLES IV

1. Air, occupying 12·4 cu ft. at 14·7 lb. per sq. in. pressure, is compressed into a space of 2·6 cu. ft. without change of temperature. What is the pressure?

2. A room measures 30 ft. × 25 × 12 ft. The temperature of the air in the room is raised from 42·6° F. to 60·6° F. What volume of air at the higher temperature passes out of the room?

3. A pound of hydrogen occupies 178·2 cu. ft. at 0° C. and 14·7 lb. per sq. in. pressure. What is the volume at 15° C. and 14·5 lb. per sq. in. pressure?

4. A gas at 0° C. and 14·7 lb. per sq. in. is said to be at normal temperature and pressure (N.T.P.). Reduce 56 cu. ft. of carbon dioxide at 30° C. and 18 lb. per sq. in. to its volume at normal temperature and pressure.

5. A volume of air, 12·39 cu. ft. at 32° F. and 14·7 lb. per sq. in. is raised to a temperature of 68° F. without change of pressure. Find the new volume, and the work done by the air during the expansion in foot-lb. What is the work done per degree rise of temperature in this case?

6. The characteristic constant of hydrogen is 1,382 ft.-lb. per degree centigrade. A quantity of hydrogen occupies 89·1 cu. ft. at 29·8 lb. per sq. in. and 0° C. What weight of gas is present?

7. The cylinder of an air compressor is 24 in. dia. and 4 ft. long. At the end of the suction stroke it is filled with air at 13·5 lb. per sq. in pressure, and at a temperature of 71·6° F. When 3 ft. of the compression stroke have been completed the pressure is 60 lb. per sq. in. Find the temperature of the air.

8. The specific volume of oxygen at N.T.P. is 11·21 cu. ft. What weight of oxygen is contained in a cylinder 5 ft. long by 5 in. diameter when the pressure is 1,400 lb. per sq. in. and the temperature 15° C.?

9. Calculate the volume of 15 lb. weight of air at 100 lb. per sq. in. absolute pressure and at a temperature of 25° C.

If this air is now heated to 50° C. at constant volume, what will be its new pressure? $R = 96$ N.C.T.E.C.

10. What is the law connecting the pressure, volume and temperature of 1 lb. of air, given that at atmospheric pressure (14·7 lb. per sq. in.) and at 32° F. the volume of 1 lb. of air is 12·39 cu. ft. Four cubic feet of air at a pressure of 30 lb. per sq. in. absolute and at a temperature of 95° F. are heated at constant volume. Calculate the number of units of heat added if the final temperature is 248° F.

Specific heat of air at constant volume is 0·169. U.E.I.

11. State separately and distinctly Boyle's and Charles's Laws. Give the rule relating the pressure, volume and temperature of a gas, which is derived from these laws, stating exactly the meaning of each letter, and the unit in which it is expressed.

If at 0° C., and standard atmospheric pressure of 14·7 lb./sq. in. the weight of dry air is 0·0807 lb. per cu. ft., obtain the value of the constant in the equation connecting the pressure, volume and temperature of a gas.

If one ounce of air at a certain pressure, volume 88 cu in., and temperature 13° C. be compressed to volume 48 cu. in. and the temperature raised to 39° C., find the ratio of the final to the initial pressure. U.E.I.

12. During a test of a gas engine working on the "Otto Cycle," 8,449 cu. ft. of mixture (air and gas) at atmospheric pressure and at a temperature of 13·8° C. entered the cylinder per hour. The engine was running at 165·8 rev. per min. and there was an impulse stroke every cycle; diameter of cylinder 14 in. and piston stroke 22 in.

Determine the increase of temperature of the charge which took place during the suction stroke, assuming that "wire drawing" is negligible. U.L.C.I.

13. The clearance volume of a gas engine is 750 cu. in., and immediately after explosion the temperature and pressure of the gases are found to be 1,120° C. and 220 lb. per sq. in. abs. respectively. When the piston has swept a volume of 2,250 cu. in., the pressure is shown by the indicator diagram to be 48 lb. per sq. in. abs. What is the temperature of the gases in the cylinder at this point?

CHAPTER V

HEAT CHANGES IN GASES

30. Introductory. In a former chapter, the quantity of heat absorbed or rejected by a gas under definitely prescribed conditions (constant pressure or constant volume) when its temperature changes, has been considered. Separately, in a succeeding chapter, the way in which the pressure, volume and temperature of a quantity of gas are related, no matter how the changes in these properties are produced, has been treated. These two aspects are now to be combined. Gases and vapours are used as working substances because they can readily absorb or yield up heat, and because the change in heat content produces changes in pressure and volume which are appreciable in amount. It should be understood, however, that the heat in the gas is the dominating factor, and that the pressure and volume changes are the results, which we find it convenient to use when the work is done in a cylinder. Consideration of the steam turbine shows that it is not necessary to obtain energy from a gas or vapour merely through its pressure, for in this case, change of momentum in the steam supplies the necessary propelling force.

The student is here warned against the common error of supposing that the pressure determines the energy contained in a working substance; that quantity depends upon the heat in the substance.

31. Effects of Heat Supply to a Gas. When heat is supplied to a given weight of a gas, several effects are possible. The following three cases cover these possibilities and the third is capable of further sub-division.

CASE I. The temperature may rise without change in volume.

The pressure will rise but no work is done since there is no movement.

CASE II. The temperature may rise and the volume increase while the pressure remains constant. Work is done at the expense of some of the heat which is being added.

CASE III. The temperature, volume, and pressure may all change during the supply of heat. This is the most general case and requires extended treatment.

Cases I and II are now considered in detail.

32. Heating a Gas at Constant Volume. This case has been mentioned in article 20. Let us consider 1 lb. of gas. Let the initial state be denoted by the suffix (2) and final state by suffix (1). Let T be absolute temperature.

Then heat supplied $= 1 \times C_v \times (T_1 - T_3)$

$= C_v\,(T_1 - T_2)$ heat units per lb.

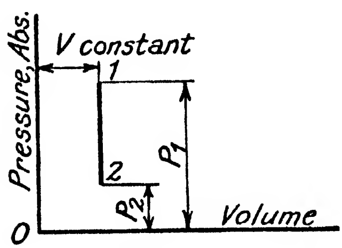


FIG. 11.

The PV diagram, fig. 11, shows this change of state. Note that there is no area under the "curve" and no external work is done. The heat supplied all enters into the gas and we may write

Heat supplied

$= $ heat entering gas.

$= C_v(T_1 - T_2)$.

33. Heating a Gas at Constant Pressure. In the cylinders *a* and *b*, fig. 12, we have 1 lb. of gas. The initial state is as in (*a*) and the final state as in (*b*). Note that P is the pressure in both cases, hence the piston must be free to move. As the temperature increases the volume increases. In these circumstances

Heat supplied to 1 lb. of gas $= 1 \times$ sp. ht. at constant pressure $\times$ temperature rise $= C_p \times (T_1 - T_2)$.

The change in the state of the gas is shown in the P – V diagram, fig. 13.

Owing to the increase in volume work has been done against external resistance, and the increase in volume is the direct result of supplying heat. Work would not have been done if heat had not been added.

Here we have the vital principle of all heat engines, viz. that work is done because heat is expended.

If we know the initial and final volumes V_2 and V_1 then we know that

External work done $= P(V_1 - V_2)$ ft.-lb.

$$= \frac{P}{J}(V_1 - V_2) \text{ heat units.}$$

where J = Joule's mechanical equivalent.

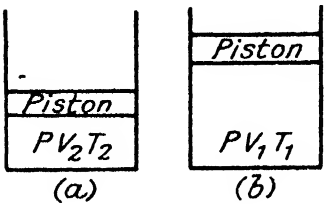


FIG. 12.

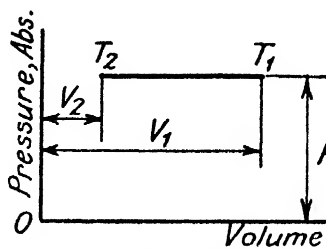


FIG. 13.

Here we may distinguish between the heat supplied to a gas and the heat entering a gas.

The heat has all been received by the gas, but a portion has been transformed into mechanical work, and has not entered permanently into the gas. The heat which is retained in the gas is spoken of as Internal Energy; the remaining portion of the heat which has done work against external resistance or pressure is called External Energy.

In this case we may say that

Increase in Internal Energy

= Heat supplied — Heat transformed into work.

$$= C_p(T_1 - T_2) - \frac{P}{J}(V_1 - V_2)$$

EXAMPLE. How much heat will be necessary to raise the temperature of 3 lb. of gas from 200° F. to 500° F., the volume remaining constant during the heat supply. Sp. ht. at constant volume = 0·17.

Heat required = 3 lb. × 0·17 × (500° — 200°)

= 3 × 0°17 × 300 = 153 B.Th.U.

34. Energy of Gases. The internal energy of a quantity of gas is the heat in it at a stated temperature, reckoned from some temperature chosen as a base.

Experiments made by Dr. Joule led him to the conclusion that the internal energy of a gas was dependent on temperature alone. It does not depend on pressure and volume. The external energy, however, does depend on pressure and volume.

Joule's Energy Law for Gases. "The internal energy of a given quantity of a gas depends entirely on temperature."

Generally we are not concerned with total internal energy but rather with change of internal energy. In calculations we may choose any suitable temperature as a base. In some cases we choose 100° C. as base, whilst in others, particularly with permanent gases, we calculate internal energies with reference to the absolute zero of temperature.

Thus if 1 lb. of gas has an absolute temperature of T_2 and it is raised to T_1, the increase in internal energy

$$= C_v(T_1 - T_2).$$

This is true no matter what happens to the volume, for Joule's Law tells us that the internal energy depends on temperature alone.

The external work done by a gas is the energy expended by the gas on outside matter in expanding against the constant pressure under which the gas exists. For doing this work heat is required, which must be supplied from some outside source.

If we start from some temperature as a base, and raise the temperature to some definite value at constant pressure, we shall

(1) put internal energy into the gas,

(2) expend energy in doing external work.

The first will be the internal energy gained by the gas and the second will be the increase of external energy of the gas, and the sum of these two quantities will be the total heat energy supplied.

If T_2 be our starting or base temperature, and T_1 the final temperature, we shall have

$$\text{Internal energy increase} = C_v(T_1 - T_2)$$

$$\text{Total heat supplied} = C_p(T_1 - T_2)$$

$$\therefore\ C_p(T_1 - T_2) = C_v(T_1 - T_2) + \text{External work done.}$$

EXAMPLE. How much heat must be supplied to 3 lb. of gas to raise its temperature from 200° F. to 500° F. at constant pressure? Find also the external work done during the supply of heat. Specific heat at constant pressure $= C_p = 0{\cdot}24$. $C_v = 0°17$.

$$\text{Heat required} = \text{Weight} \times \text{sp. ht.} \times \text{temperature rise}$$
$$= 3 \times 0{\cdot}24 \times (500 - 200)$$
$$= 216 \text{ B.Th.U.}$$

Heat entering gas as internal energy

$$= \text{Weight} \times \text{sp. ht. at const. vol.} \times \text{temp. rise}$$
$$= 3 \times 0{\cdot}17 \times (500 - 200)$$
$$= 153 \text{ B.Th.U.}$$

$$\text{External work done} = (216 - 153) \text{ B.Th U.}$$
$$= 63 \text{ B.Th.U.}$$

This should be compared with the last example.

EXAMPLE. The characteristic constant for a gas is 98 ft.-lb. per lb. per degree centigrade, and 2·2 lb. of this gas receive heat at constant pressure of 120 lb. per sq. in. abs., the temperature rising from 30° C. to 200° C. (ordinary temperatures) as a result of the heat supply. If $C_v = 0{\cdot}17$ for this gas, find (*a*) the increase in internal energy, (*b*) increase in external energy, (*c*) increase in total energy.

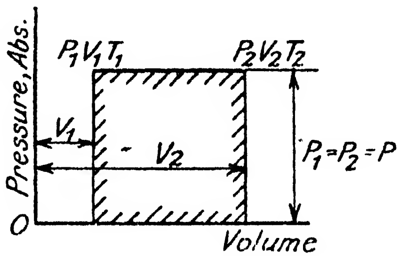


FIG. 14.

Fig. 14 shows the P–V diagram.

$$\text{Increase in internal energy} = \text{wt.} \times C_v\,(T_2 - T_1)$$
$$= 2{\cdot}2 \times 0{\cdot}17 \times (473 - 303)$$
$$= 2{\cdot}2 \times 0{\cdot}17 \times 170 = 63{\cdot}6 \text{ C.H.U.}$$

To find V_1.

$$P_1V_{s1} = RT_1$$
$$\therefore V_{s1} = \frac{RT_1}{P_1} = \frac{98 \times 303}{144 \times 120}$$
$$= 1{\cdot}718 \text{ cu. ft.} = \text{specific volume.}$$
$$\therefore V_1 = 1{\cdot}718 \times 2{\cdot}2 = 3{\cdot}78 \text{ cu. ft.}$$

V_2 may be found by using Charles's Law.

$$\frac{V_2}{T_2} = \frac{V_1}{T_1}$$
$$\therefore V_2 = \frac{V_1T_2}{T_1}$$
$$= \frac{3{\cdot}78 \times 473}{303} \text{ cu. ft.} = 5{\cdot}9 \text{ cu. ft.}$$

$$\text{Increase in external energy} = P(V_2 - V_1)$$
$$= 144 \times 120(5{\cdot}9 - 3{\cdot}78)$$
$$= 17{,}280 \times 2{\cdot}12$$
$$= 36{,}634 \text{ ft.-lb.} = 26{\cdot}17 \text{ C.H.U.}$$

Total energy increase $= 26{\cdot}17 + 63{\cdot}6 = 89{\cdot}77$ C.H.U.
Note that the total energy increase is the heat supplied.
We may thus proceed to calculate the specific heat at constant pressure.

$$\text{Total heat supplied} = 89{\cdot}77 \text{ C.H.U.}$$
$$= w \times C_p \times (T_2 - T_1)$$
$$= 2{\cdot}2 \times (473 - 303) \times C_p$$
$$= 2{\cdot}2 \times 170 \times C_p$$
$$\therefore C_p = \frac{89{\cdot}77}{2{\cdot}2 \times 170} = 0{\cdot}24.$$

This example illustrates the general equation which may now be written
Heat supplied = External work done + Gain in Internal Energy.

Let us now write the internal energy equation below the total energy equation.

$$\begin{array}{lrl} \text{Total energy} = & 89{\cdot}77 & = w \times C_p(T_2 - T_1) \\ \text{Internal energy} = & 63{\cdot}6 & = w \times C_v(T_2 - T_1) \\ \text{subtracting} & 26{\cdot}17 & = w(C_p - C_v)(T_2 - T_1) \\ \text{from which} & C_p - C_v & = \dfrac{26{\cdot}17}{w(T_2 - T_1)} = \dfrac{26{\cdot}17}{2{\cdot}2 \times 170} \\ & & = 0{\cdot}07 \text{ C.H.U.} \\ & & = 98 \text{ ft.-lb. per degree.} \end{array}$$

From this result it may be observed that the difference of the two specific heats is the same as the characteristic constant, when all units are in ft.-lb. or heat units.

35. Gas Energy Equation. Summarizing the foregoing results we may say that on heating a gas we may

(1) raise the temperature and therefore increase the internal energy,
(2) raise the temperature and cause external work to be done,
(3) cause external work to be done whilst the temperature remains constant.

All these results are covered by the gas energy equation given below.

Heat supplied to gas = External work done by gas + increase in internal energy of gas.

This equation may be written in a negative form for heat abstracted as follows :

Heat abstracted from gas = Work done on gas + Reduction in internal energy of gas.

In general applications of this equation any of the quantities may be negative relative to the others.

Let H = heat added to a gas and — H = heat abstracted,
W = work done by gas and — W = work done on gas,
E = gain of internal energy and — E = loss of internal energy.

Then the equation can be written to cover all cases

$$\pm H = \pm W \pm E.$$

It is, of course, necessary in numerical work to give each term its appropriate sign. Thus, when volume is increasing, work is being done by the gas and W is positive ; when volume is decreasing, work is being done on the gas and W is negative ; when temperature is rising, internal energy is increasing and E is positive ; when temperature is falling, internal energy is decreasing and E is negative. The proper sign can only be inserted when we know the circumstances in each case.

The student should note that this equation is based upon, and is consistent with, the Principle of the Conservation of Energy.

36. Relationship between Gas Constants. The last numerical example showed some relationship between specific heats and the characteristic constant. The general treatment is given below.

In the reasoning which follows 1 lb. of gas is assumed.

Let T_2 = initial absolute temperature.
T_1 = final ,, ,,
J = Joule's mechanical equivalent of heat.
R = characteristic constant in ft.-lb. per degree.
C_p = specific heat at constant pressure in heat units.

C_v = specific heat at constant volume in heat units.

V_2 = initial volume in cu. ft.

V_1 = final ,, ,, ,, ,,

P = pressure in lb. per sq. ft.

Now consider that heat is supplied to one pound of gas at constant pressure, so that work is done by the gas and internal energy is gained. Then

Heat supplied = work done + gain in internal energy.

$$C_p(T_1 - T_2) = \frac{P}{J}(V_1 - V_2) + \text{gain in internal energy.}$$

We have seen, article 34, that gain in internal energy

$$= C_v(T_1 - T_2)$$

$$\therefore\ C_p(T_1 - T_2) = \frac{P}{J}(V_1 - V_2) + C_v(T_1 - T_2)$$

(Note that J is introduced because all units must be either in ft.-lb. or in heat units.)

$$\therefore\ C_v(T_1 - T_2) = C_p(T_1 - T_2) - \frac{P}{J}(V_1 - V_2).$$

Now from the characteristic equation $P_1V_1 = RT_1$ and $P_2V_2 = RT_2$

and since the pressure is constant $P_1 = P_2 = P$

$$\therefore\ PV_1 = RT_1 \text{ and } PV_2 = RT_2$$

$$\therefore\ PV_1 - PV_2 = RT_1 - RT_2$$

$$\therefore\ P(V_1 - V_2) = R(T_1 - T_2)$$

$$\therefore\ \frac{P}{J}(V_1 - V_2) = \frac{R}{J}(T_1 - T_2).$$

Hence substituting

$$C_v(T_1 - T_2) = C_p(T_1 - T_2) - \frac{R}{I}(T_1 - T_2)$$

$$\therefore\ C_v = C_p \quad \frac{R}{J}$$

$$\therefore\ C_p - C_v = \frac{R}{I} \text{ or } (C_p - C_v)J = R.$$

EXAMPLE. The values of C_p and C_v for a gas are 0·2375 and 0·1691 respectively. Find the density of this gas at 15° C. and 14·7 lb. per sq. in. absolute.

$$\text{Density} = \frac{1}{\text{specific volume}}$$

$$PV_s = RT \text{ (article 28)}$$

$$\therefore \text{Specific volume} = \frac{RT}{P}.$$

$$\therefore \text{Density} = \frac{P}{RT} \text{ lb. per cu. ft.}$$

$$= \frac{14{\cdot}7 \times 144}{(0{\cdot}2375 - 0{\cdot}1691)1{,}400 \times 288}$$

$$= \frac{14{\cdot}7 \times 144}{95{\cdot}76 \times 288}$$

$$= 0{\cdot}077 \text{ lb. per cu. ft.}$$

EXAMPLE. A quantity of gas occupies 5 cu. ft. at 180 lb. per sq. in. and 212° F. Find the change in internal energy if the temperature is increased to 572° F. $C_p = 0{\cdot}24$ and $C_v = 0{\cdot}17$.

$$R = (C_p - C_v)J = (0{\cdot}24 - 0{\cdot}17) \times 778$$

$$= 0{\cdot}07 \times 778 = 54{\cdot}4$$

$$PV = wRT \text{ (article 28)}$$

$$\therefore w = \text{wt. of gas in lb.} = \frac{PV}{RT} = \frac{144 \times 180 \times 5}{54{\cdot}4 \times 672}$$

$$= 3{\cdot}55 \text{ lb.}$$

$$\text{Change in internal energy} = C_v(T_1 - T_2) \times \text{wt.}$$

$$= 0{\cdot}17 \times (1032 - 672) \times 3{\cdot}55$$

$$= 0{\cdot}17 \times 360 \times 3{\cdot}55$$

$$= 217{\cdot}3 \text{ B.Th.U.}$$

EXAMPLE. A quantity of gas occupying 3 cu. ft. at 120 lb. per sq. in. receives heat at constant pressure until the volume increases to 6 cu. ft. The initial temperature is 200° C. and $C_p = 0{\cdot}237$ and $C_v = 0{\cdot}169$. Find the change in internal energy and the external work done during the heat supply.

(There are alternative methods of solving this problem which are instructive.)

(i) $R = (C_p - C_v)J = (0{\cdot}237 - 0{\cdot}169)1{,}400 = 95{\cdot}2$ ft.-lb.

$$\text{Weight of gas} = \frac{\text{volume occupied}}{\text{specific volume}}$$

$$\text{Specific volume} = \frac{RT}{P} = \frac{95{\cdot}2 \times 473}{144 \times 120} = 2{\cdot}6 \text{ cu. ft.}$$

$$\text{Weight of gas} = \frac{3}{2{\cdot}6} = 1{\cdot}153 \text{ lb.}$$

Final temperature $(T_1) = T_2 \times \frac{6}{3} = 473 \times 2 = 946°$ C. abs.

$$\begin{aligned}
\text{Change in Internal Energy} &= w \times C_v(T_1 - T_2) \\
&= 1{\cdot}153 \times 0{\cdot}169 \times (946 - 473) \\
&= 1{\cdot}153 \times 0{\cdot}169 \times 473 \\
&= 92{\cdot}4 \text{ C.H.U.}
\end{aligned}$$

$$\begin{aligned}
\text{External work} &= \text{Heat supplied} - \text{Internal energy gain} \\
&= wC_p(T_1 - T_2) - 92{\cdot}4 \\
&= (1{\cdot}153 \times 0{\cdot}237 \times 473) - 92{\cdot}4 \\
&= 129{\cdot}5 - 92{\cdot}4 = 37{\cdot}1 \text{ C.H.U.}
\end{aligned}$$

(ii) The external work done may also be found as follows:—

$$\begin{aligned}
\text{External work} &= P(V_1 - V_2) \\
&= 120 \times 144 \times (6 - 3) \\
&= 120 \times 144 \times 3 \\
&= 51{,}840 \text{ ft.-lb.} \\
&= 37 \text{ C.H.U.}
\end{aligned}$$

(iii) Again $PV_1 = wRT_1$ and $PV_2 = wRT_2$.

$$\begin{aligned}
\therefore PV_1 - PV_2 &= wRT_1 - wRT_2 \\
&= w \times R \times (T_1 - T_2) \\
&= 1{\cdot}153 \times 95{\cdot}2 \times 473 \\
&= 51{,}840 \text{ ft.-lb.} = 37 \text{ C.H.U.}
\end{aligned}$$

Examples V

1. Six pounds of gas at temperature 200° C. and pressure 120 lb. per sq. in. receive heat at constant pressure until the final temperature is 600° C. If $C_p = 0{\cdot}24$ and $C_v = 0{\cdot}17$, find the external work done during the heat supply.

2. Five pounds weight of gas with specific heats as above receive heat at constant volume and the temperature rises from 150° C. to 400° C. How much heat is supplied? The same weight of gas is now heated at constant pressure from 150° C. to 400° C. How much heat is supplied in this case? What becomes of the difference in the two cases?

3. A cylinder contains 2 lb. of air occupying a volume of 18 cu. ft. at a temperature of 11° C. Later it has a volume of 5 cu. ft. and a temperature of 85° C. Find the increase in internal energy if $C_v = 0{\cdot}17$.

4. A gas has its temperature raised at constant pressure of 60 lb. per sq. in. absolute from 10° C. to 400° C. Its initial volume is 10 cu. ft. Find the external work done.

5. The values of C_p and C_v for a gas are 0·239 and 0·169 respectively. Find the specific volume at 0° C. and 14·7 lb. per sq. in. absolute of this gas. What is its density?

6. A quantity of gas occupying 10 cu. ft. at 180 lb. per sq. in. absolute receives heat at constant pressure until the volume increases to 16 cu. ft. The initial temperature is 150° C. If $C_p = 0{\cdot}239$ and $C_v = 0{\cdot}169$, find the change in internal energy and the external work done during the heat supply.

7. A gas is heated at constant pressure from a temperature of 100° C. to 400° C. Find the external work done per lb. of gas if R = 98.

8. 2 lb. of gas at 15° C. and 30 lb. per sq. in. occupy a volume of 8·68 cu. ft. If the specific heat of the gas at constant volume is 0·153, find the specific heat at constant pressure. N.C.T.E.C.

9. Four cubic feet of air at a pressure of 300 lb. per sq. in. absolute and a temperature of 180° C. expand at constant pressure to a volume of 12 cu. ft. Find the temperature at the end of the expansion and the heat absorbed. (Gas constant R = 96 ft.-lb. per lb. per degree C.; specific heat of air at constant volume = 0·17). N.C.T.E.C.

10. State, clearly, what is meant by the "characteristic constant" for a gas. The volume of a receiver, containing a gas whose specific heats at constant pressure and constant volume are 0·238 and 0·169 respectively, is 15 cu. ft. The temperature and pressure in the receiver are 15° C. and 30 lb. per sq. in. absolute. What will be the pressure when 2 lb. of gas have been pumped into the receiver and the temperature is 15° C.?

U.L.C.I.

CHAPTER VI

EXPANSION AND COMPRESSION OF GASES

37. When a gas undergoes expansion or increases in volume, work is done by the gas. When a gas undergoes compression or decreases in volume, work is done on the gas. This may be illustrated by means of a cylinder fitted with a piston which allows no leakage and which is loaded with a number of weights as shown in fig. 15.

The number of weights on the piston determines the pressure of the gas. Now remove the weights one by one. Reduction in pressure and, therefore, expansion occur. Movement of the remaining weights on the piston has resulted and hence work has been done in lifting the weights.

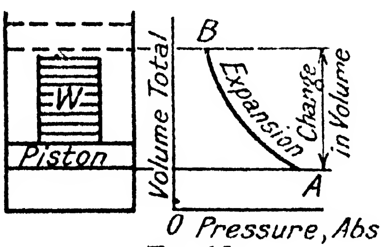


FIG. 15.

The curve AB shows the general outline of the curves of pressure and volume representing such processes. If the piston were started opposite B, and weights were added the gas would be compressed, and the descending weights would do work on the gas. In these changes

$$\text{Ratio of expansion} = \frac{\text{Volume at end of expansion}}{\text{Volume at beginning of expansion}}.$$

$$\text{Ratio of compression} = \frac{\text{Volume at beginning of compression}}{\text{Volume at end of compression}}.$$

Note that both ratios give a value greater than unity since the larger quantity is in the numerator.

38. Laws of Expansion and Compression. The relationship between the pressure and volume of a given quantity of gas can generally be written

$$PV^n = C,$$

where n and C are constants. This equation is known as the law of expansion or compression. The value of n is important and is controlled by the nature of the gas, and the circumstances under which the change takes place. Generally the value of n must be found experimentally. Some typical values of n are given below.

(1) Perfect gas expanding at constant temperature $n = 1$.

(2) Perfect gas expanding without receiving or emitting heat $n = \frac{C_p}{C_v}$. Usually denoted by γ, $\therefore \frac{C_p}{C_v} = \gamma$.

(3) Dry steam expanding, remaining dry throughout $n = \frac{16}{15}$.

(4) Dry steam expanding, without receiving or emitting heat $n = 1{\cdot}135$.

When n has a value greater than unity, the curves of expansion and compression become steeper. Fig. 16 shows approximately

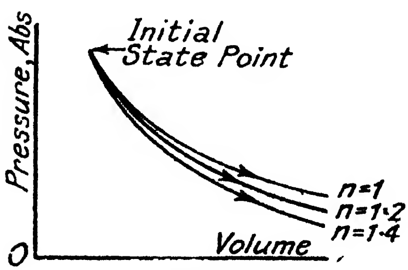


FIG. 16.

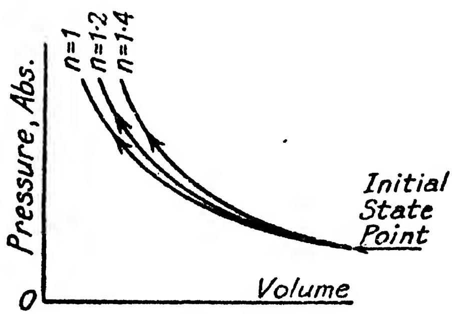


FIG. 17.

what occurs during expansion, whilst fig. 17 shows the changes occurring during compression.

39. Working Form of Equation. The working form of the general law of expansion or compression is obtained by writing it for the initial and final states.

Let suffix (1) denote initial state of gas (see fig. 18).

" " (2) " final " " "

Then $P_1V_1^n = C$, also $P_2V_2^n = C$.

$$\therefore P_1V_1^n = P_2V_2^n.$$

Note that an expression such as V_1^n is not a volume but a mere number.

EXAMPLE. The pressure of a quantity of gas enclosed in a cylinder is 100 lb. per sq. in. by gauge, and the volume occupied is 3 cu. ft. The gas expands according to the law $PV^{1 \cdot 3} = C$ until the volume is 12 cu. ft. Find the final pressure.

Draw and dimension the P–V diagram as in fig. 19.

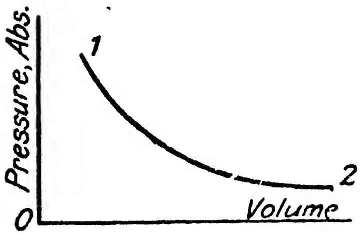


FIG. 18.

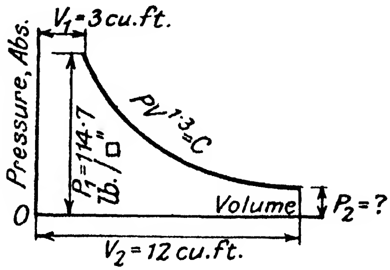


FIG. 19.

Pressure given is by gauge, but the pressure used in the equation is always absolute.

$$\therefore P_1 = 100 + 14 \cdot 7 = 114 \cdot 7 \text{ lb. per sq. in.}$$

$$P_1V_1^{1 \cdot 3} = P_2V_2^{1 \cdot 3}$$

$$\therefore P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1 \cdot 3}$$

$$\therefore P_2 = 114 \cdot 7 \times \left(\frac{3}{12}\right)^{1 \cdot 3} = 114 \cdot 7 \times \frac{1}{4^{1 \cdot 3}}$$

Note that the figure 4 is the expansion ratio, a mere number.

$$1 \cdot 3 \times \log 4 = 1 \cdot 3 \times \cdot 6021 = 0 \cdot 7827$$

$$\therefore 4^{1 \cdot 3} = 6 \cdot 063$$

$$\therefore P_2 = \frac{114 \cdot 7}{6 \cdot 063} = 18 \cdot 9 \text{ lb. per sq. in. abs.}$$

$$= 18 \cdot 9 - 14 \cdot 7 = 4 \cdot 2 \text{ lb. per sq. in. gauge.}$$

EXAMPLE. The absolute pressure of a quantity of gas is 120 lb. per sq. in., and the initial volume is 1·5 cu. ft. The gas is expanded according to the law $PV^{1 \cdot 17} = C$ until the pressure is 15 lb. per sq. in. absolute. Find the final volume.

Fig. 20 shows the P–V curve for the operation.

$$P_1V_1^{1\cdot17} = P_2V_2^{1\cdot17}$$

$$\therefore V_2^{1\cdot17} = \frac{P_1}{P_2}V_1^{1\cdot17}$$

$$\therefore V_2 = \left(\frac{P_1}{P_2}\right)^{\frac{1}{1\cdot17}}V_1$$

$$= \left(\frac{120}{15}\right)^{0\cdot854} \times 1\cdot5$$

$$= 8^{0\cdot854} \times 1\cdot5.$$

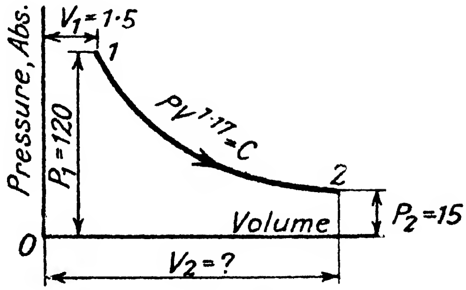


FIG. 20.

Note that 8 is the pressure ratio.

$$0\cdot854 \log 8 = 0\cdot854 \times 0\cdot9031 = 0\cdot7713$$

$$\therefore 8^{0\cdot854} = 5\cdot906$$

$$\therefore V_2 = 5\cdot906 \times 1\cdot5 = 8\cdot86 \text{ cu. ft.}$$

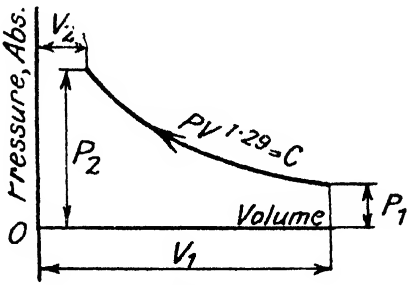


FIG. 21.

EXAMPLE. The " charge " in a gas engine cylinder is compressed through a volume ratio of 4½. Find the final pressure if the initial pressure is 13 lb. per sq. in. absolute and the law of compression is $PV^{1\cdot29} = C$.

$$P_1V_1^{1\cdot29} = P_2V_2^{1\cdot29} \text{ (fig. 21).}$$

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1\cdot29}$$

$$= 13 \times 4\cdot5^{1\cdot29}$$

$$1\cdot29 \log 4\cdot5 = 1\cdot29 \times 0\cdot6532 = 0\cdot8426$$

$$\therefore 4\cdot5^{1\cdot29} = 6\cdot96$$

$$\therefore P_2 = 13 \times 6\cdot96 = 90\cdot48 \text{ lb. per sq. in. abs.}$$

The above examples show the method of using the given law. In all cases the volume ratio or pressure ratio has been used. The lesser quantity has been made unity and this simplifies the logarithmic work.

The whole equation may be put in logarithmic form thus—

$$\log P_1 + n \log V_1 = \log P_2 + n \log V_2.$$

By transposing we may find an expression for n.

$$n \log V_1 - n \log V_2 = \log P_2 - \log P_1$$
$$n(\log V_1 - \log V_2) = \log P_2 - \log P_1$$
$$\therefore n = \frac{\log P_2 - \log P_1}{\log V_1 - \log V_2}$$

EXAMPLE. The following measurements were made of two points A and B on the expansion curve of a gas engine indicator diagram.

Vertical distance of A above atmospheric line	= 1·44 in.
,, ,, ,, B ,, ,, ,,	= 0·33 in.
Horizontal distance of A from zero volume line	= 1·775 in.
,, ,, ,, B ,, ,, ,, ,,	= 4·445 in.

The clearance volume was 140 cubic inches and the swept volume 400 cubic inches. The length of the indicator diagram was 3·7 inches and the

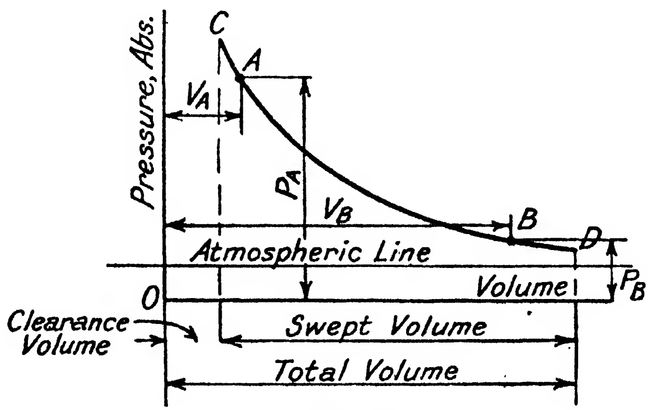


FIG. 22.

scale of the spring was 120 lb. per sq. in per in. Find the value of n in the law of expansion. Sketch and dimension the expansion line to illustrate the problem.

We require to find the pressures and volumes at A and B (fig. 22).

CD is the whole expansion line.

Pressure $P_A = (1{\cdot}44 \times 120) + 14{\cdot}7 = 187{\cdot}5$ lb. per sq. in.

Pressure $P_B = (0{\cdot}33 \times 120) + 14{\cdot}7 = 54{\cdot}3$ lb. per sq. in.

$$\text{Volume at A} = V_A = 1{\cdot}775'' \times \frac{400}{3{\cdot}7} = 192 \text{ cu. in.}$$
$$= 0{\cdot}111 \text{ cu. ft.}$$

$$\text{Volume at B} = V_B = 4{\cdot}445'' \times \frac{400}{3{\cdot}7} = 481 \text{ cu. in.}$$
$$= 0{\cdot}278 \text{ cu. ft.}$$

$$\begin{aligned} \log P_A + n \log V_A &= \log P_B + n \log V_B \\ n(\log V_A - \log V_B) &= \log P_B - \log P_A \\ n(\log 0{\cdot}111 - \log 0{\cdot}278) &= \log 54{\cdot}3 - \log 187{\cdot}5 \\ n(\bar{1}{\cdot}0453 - \bar{1}{\cdot}4440) &= 1{\cdot}7348 - 2{\cdot}2729 \\ n &= \frac{-0{\cdot}5381}{-0{\cdot}3987} = 1{\cdot}35. \end{aligned}$$

40. Work done during Expansion or Compression. The object of this article is to establish expressions for the work done when expansion or compression occurs according to the law $PV^n = C$. The student who has not yet done sufficient integral calculus to understand the proofs may note the results and examples, and return to the proofs later.

Two cases arise (1) when $n = 1$ (unity).
(2) when n is not unity.

Case I. When $PV = C$.

Fig. 23 shows the curve of expansion from volume V_1 to volume V_2 according to the law $PV = C$. We require to find the work done during the expansion.

Let us consider a point where the volume is V and the pressure P. Let a very small increase δV in the volume occur. The pressure P will decrease a very small amount, hence, (very approximately).

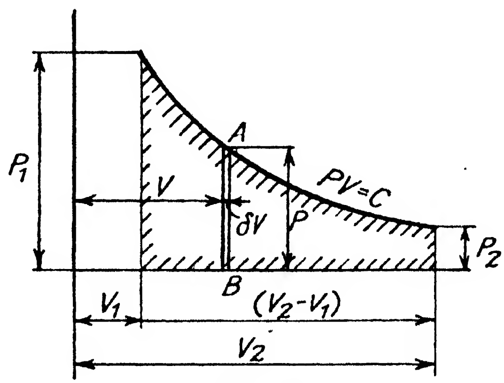


FIG. 23.

Work done during increase δV, in volume, $= P.\delta V$.
When δV becomes infinitely small we may adopt the calculus notation and write

$$dW = P.dV \text{ where } W = \text{work done.}$$

$\therefore$ Total work done during expansion

$$= W = \int_{V_1}^{V_2} P.dV.$$

but $$P = \frac{C}{V}$$

$$\therefore \quad W = C\int_{V_1}^{V_2}\frac{dV}{V} = C\Big[\log_e V\Big]_{V_1}^{V_2}$$

$$= C(\log_e V_2 - \log_e V_1)$$

But $C = PV$

$$\therefore \quad W = PV(\log_e V_2 - \log_e V_1)$$

$$= PV\log_e\frac{V_2}{V_1}$$

$$\frac{V_2}{V_1} = \text{expansion ratio} = r$$

$$\therefore \text{ Work done} = PV \log_e r.$$

If we are dealing with 1 lb. of gas PV becomes PV_s which equals RT, hence

Work done $= RT \log_e r$ per lb. of gas.

These results apply to compression also.

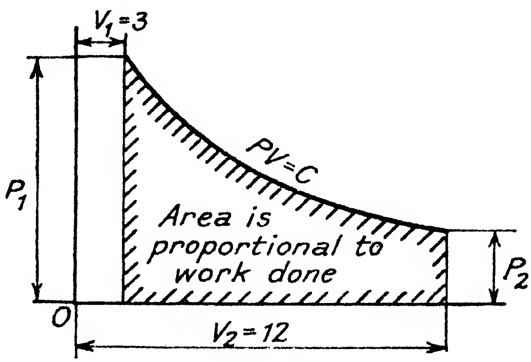


FIG. 24.

EXAMPLE. A quantity of gas, occupying 3 cu. ft. at a pressure of 120 lb. per sq. in. absolute, expands until the volume is 12 cu. ft. according to the law PV = C. Find the work done during expansion.

$$P_1V_1 = P_2V_2 = PV \text{ (fig. 24).}$$

Thus we can take any product of P and V which we happen to know.

$$\begin{aligned}\text{Work done} &= P_1V_1 \log_e r \\ &= 120 \times 144 \times 3 \times \log_e \frac{12}{3} \\ &= 120 \times 144 \times 3 \times \log_e 4 \\ &= 120 \times 144 \times 3 \times 2{\cdot}3026 \times \log_{10} 4 \\ &= 120 \times 144 \times 3 \times 2{\cdot}3026 \times 0{\cdot}6021 \\ &= 72{,}000 \text{ ft.-lb.}\end{aligned}$$

EXAMPLE. The characteristic constant for a gas is 53·2 ft.-lb. per degree Fahrenheit. One pound of this gas expands at constant temperature of 392° F. through a volume ratio of 6. Find the work done.

$$\begin{aligned}\text{Work done per lb.} &= PV_s \log_e r \\ &= RT \log_e r \\ &= 53{\cdot}2 \times 852 \times 2{\cdot}3026 \times \log_{10} 6 \\ &= 53{\cdot}2 \times 852 \times 2{\cdot}3026 \times 0{\cdot}7782 \\ &= 81{,}500 \text{ ft.-lb.}\end{aligned}$$

CASE II. When n is not unity. $PV^n = C$ (fig. 25). With the same notation and reasoning as in Case I we get

$$dW = P.dV$$

$$\therefore \text{ Work done} = W = \int_{V_1}^{V_2} P.dV$$

$$PV^n = C \therefore P = \frac{C}{V^n} = CV^{-n}$$

$$\therefore W = C\int_{V_1}^{V_2} V^{-n}dV = C\left[\frac{V^{-n+1}}{-n+1}\right]_{V_1}^{V_2}$$

$$= C\left(\frac{V_2^{1-n} - V_1^{1-n}}{1-n}\right)$$

$$\text{but } C = P_1V_1^n = P_2V_2^n$$

$$\therefore W = \frac{P_2V_2^n.V_2^{1-n} - P_1V_1^n.V_1^{1-n}}{1-n}$$

$$= \frac{P_2V_2 - P_1V_1}{1-n}$$

$$= \frac{P_1V_1 - P_2V_2}{n-1}.$$

This equation will give a positive result if P_1 and V_1 refer to initial state and expansion occurs. It will give a negative result if P_1 and V_1 refer to initial state and compression occurs. If P_1 refers to the higher pressure and V_1 the smaller volume it will give a positive result in all cases. The negative result in connection with compression means that work is done on the gas and not by the gas. Pressures must be absolute and if they

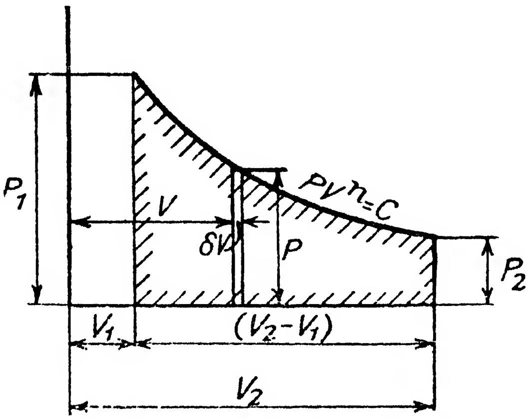


FIG. 25.

are in lb. per square foot and volumes in cubic feet the work done will be in ft.-lb.

The student is warned against using this formula when $n = 1$, for then the work done would be indeterminate.

EXAMPLE. Three cubic feet of gas expand according to the law $PV^{1\cdot3} = C$ from a pressure of 100 lb. per sq. in. absolute to a pressure of 20 lb. per sq. in absolute. Find the final volume and the work done.

$$P_1V_1^{1\cdot3} = P_2V_2^{1\cdot3}$$

$$\therefore V_2 = \left(\frac{P_1}{P_2}\right)^{\frac{1}{1\cdot3}} V_1$$

$$= \left(\frac{100}{20}\right)^{0\cdot77} \times 3 = 5^{0\cdot77} \times 3$$

$$= 3\cdot453 \times 3 = 10\cdot359 \text{ cu. ft.}$$

$$\text{Work done} = \frac{P_1V_1 - P_2V_2}{n-1}$$

$$= \frac{(144 \times 100 \times 3) - (144 \times 20 \times 10\cdot359)}{1\cdot3 - 1}$$

$$= \frac{43{,}200 - 29{,}834}{0\cdot3} = \frac{13{,}366}{0\cdot3}$$

$$= 44{,}553 \text{ ft.-lb.}$$

41. Hyperbolic Expansion. When values of P and V are plotted from the equation PV = C, the curve obtained is known as a rectangular hyperbola, and expansion or compression according to this law is said to be hyperbolic. This must not be confused with Boyle's Law, for whilst expansion according to Boyle's Law is hyperbolic, there are possible hyperbolic expansions which do not accord with Boyle's Law.

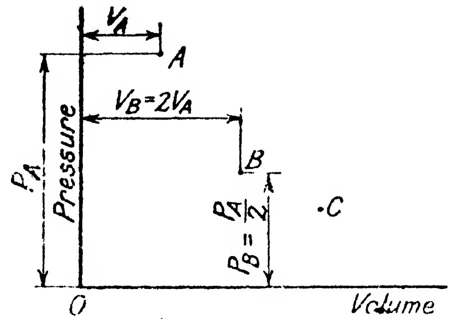


FIG. 26.

Any hyperbolic expansion curve may be readily plotted when one point on it is known. Take a point A, fig. 26, the co-ordinates of which are P_A and V_A. To find a new point. double the volume and halve the pressure. Thus $V_B = 2V_A$ and $P_B = \frac{1}{2}P_A$.

Repeat the process taking three times the volume and $\frac{1}{3}$ the pressure. In this way a large number of points may be obtained using the actual dimensions of P_A and V_A on the diagram.

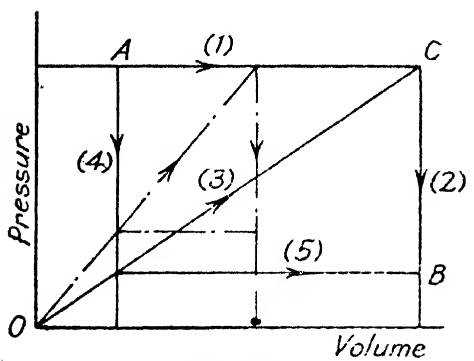


FIG. 27.

A graphical construction for finding the hyperbolic curve is given in fig. 27.

Let O be the point of zero volume and pressure, and let A be the point given on the curve.

Draw the horizontal AC where C represents a point immediately above the final volume. Drop a perpendicular from C. Join CO. Drop a perpendicular from A. From where this cuts the line CO, draw horizontally to B on CB. B is the required point on the hyperbola. Thus any number of points may be found.

42. Isothermal Expansion. An isothermal operation is one carried out at constant temperature, that is, in an isothermal expansion or compression the temperature remains the same throughout. Boyle's Law assumes constant temperature of the gas, and hence expansion according to Boyle's Law is isothermal. This is where some hyperbolic expansions differ from Boyle's Law, for hyperbolic expansion is not necessarily isothermal, for example, the hyperbolic expansion of steam is not isothermal.

Since temperature is unchanged it follows that in a perfect gas the internal energy is unchanged. Because, however, there is expansion or compression, work is done by or on the gas.

The gas energy equation tells us (Art. 35)

Heat added = work done + gain in internal energy.

When the temperature does not change the last term becomes zero (Art. 34).

Hence heat added = work done by gas.

This means that during isothermal expansion the work done, expressed in heat units, is exactly equal to the heat supplied to the gas. With isothermal compression we obtain similarly,

Heat abstracted = work done on gas.

Thus heat is given out equal to the work done on the gas. We see, then, that the heat is immediately transformed into mechanical work or, on compression, the work is transformed immediately into heat.

EXAMPLE. A quantity of gas expands isothermally and receives 15 C.H.U. or 27 B.Th.U. during the expansion. How much work is done?

$$\begin{aligned}\text{Work done} &= \text{heat supplied}\\ &= 15\text{ C.H.U.}\\ &= 15 \times 1{,}400\text{ ft.-lb.} = 21{,}000\text{ ft.-lb.}\end{aligned}$$

Alternatively:—

$$\begin{aligned}\text{Work done} &= 27\text{ B.Th.U.}\\ &= 27 \times 778\text{ ft.-lb.} = 21{,}000\text{ ft.-lb.}\end{aligned}$$

EXAMPLE. The amount of work expended in the isothermal compression of a quantity of gas is 70,000 ft.-lb. How much heat is given out from the gas during compression?

$$\begin{aligned}\text{Heat removed} &= \text{work done on gas}\\ &= \frac{70{,}000}{1{,}400} = 50\text{ C.H.U.}\end{aligned}$$

EXAMPLE. A quantity of gas is compressed isothermally from initial conditions of 20 cu. ft. and 14·7 lb. per sq. in. absolute, to a final pressure of 102·9 lb. per sq. in. absolute. Find the work done on the gas.

$$\text{Ratio of compression} = \frac{102 \cdot 9}{14 \cdot 7} = 7 = r$$

$$\begin{aligned} \text{Work done} &= \text{PV} \log_e r \\ &= 144 \times 14 \cdot 7 \times 20 \times \log_e 7 \\ &= 144 \times 14 \cdot 7 \times 20 \times 2 \cdot 3026 \times 0 \cdot 8451 \\ &= 82{,}850 \text{ ft.-lb.} \end{aligned}$$

This, in work units, is also the heat emitted by the gas.

43. Adiabatic Expansion. Another important special case of expansion or compression is that known as adiabatic. In this case the gas neither receives nor gives out heat. We may imagine the gas to be contained in some cylinder made of a material which is a perfect non-conductor of heat. The gas can then expand or be compressed, but no heat can either be given to or taken from the gas. Students very frequently misunderstand the meaning of adiabatic expansion or compression. Taking first the case of expansion; here the temperature falls and there is loss of internal energy, but no heat is lost in the form of heat. Heat, however, is transformed into work and so the gas has given up heat in the form of work.

The gas energy equation is again applied.

Heat added = work done + gain in internal energy. Since the operation is adiabatic no heat is added, hence the equation becomes

$$0 = \text{work done} + \text{gain in internal energy.}$$

∴ Work done = — (gain in internal energy). Negative gain is the same as loss.

∴ Work done by gas = loss of internal energy.

In the case of compression,

Work done on gas = gain in internal energy.

Hence the work done passes into the gas and increases its internal energy, thus raising its temperature. It has received no heat from any external heat supply, but it has transformed the work done upon it into heat.

EXAMPLE. The temperature of 1½ lb. of gas at the beginning of an adiabatic expansion is 700° C. absolute, and at the end is 300° C. absolute.

Find the work done during expansion. $C_v = 0{\cdot}169$.

$$\begin{aligned}\text{Change in internal energy} &= 1\tfrac{1}{2} \times 0{\cdot}169 \times (700 - 300)\\ &= 1\tfrac{1}{2} \times 0{\cdot}169 \times 400\\ &= 600 \times 0{\cdot}169\\ &= 101{\cdot}4 \text{ C.H.U.}\\ \therefore \text{Work done} &= 101{\cdot}4 \times 1{,}400 \text{ ft.-lb.}\\ &= 141{,}960 \text{ ft.-lb.}\end{aligned}$$

Note that if the given temperatures were in Fahrenheit degrees, the change in internal energy would be in B.Th.U. and the conversion factor would be 778 instead of 1,400.

EXAMPLE. A quantity of gas weighing 1¼ lb. expands from a pressure of 120 lb. per sq. in. absolute to a pressure of 15 lb. per sq. in. absolute, in such a way that the gas does not receive or reject any heat. The initial temperature is 200° C. Find the work done and the change in internal energy. $C_p = 0{\cdot}24$ and $C_v = 0{\cdot}17$.

$$\text{Initial specific volume} = \frac{RT}{P} \text{ and } R = J(C_p - C_v)$$

$$\begin{aligned}\therefore \text{Specific volume} &= \frac{1{,}400 \times (0{\cdot}24 - 0{\cdot}17) \times 473}{144 \times 120}\\ &= \frac{1{,}400 \times 0{\cdot}07 \times 473}{144 \times 120}\\ &= 2{\cdot}686 \text{ cu. ft. at 120 lb./sq. in. and } 200^\circ \text{ C.}\\ \therefore \text{Initial volume} &= 1{\cdot}25 \times 2{\cdot}686\\ &= 3{\cdot}36 \text{ cu. ft.}\end{aligned}$$

The value of n or γ for adiabatic change $= \dfrac{C_p}{C_v}$, see Article 38.

$$\therefore \gamma = \frac{0{\cdot}24}{0{\cdot}17} = 1{\cdot}41.$$

$$\text{Hence} \quad P_1V_1^{1{\cdot}41} = P_2V_2^{1{\cdot}41}$$

$$\begin{aligned}V_2 &= \left(\frac{P_1}{P_2}\right)^{\frac{1}{1{\cdot}41}} V_1 = \left(\frac{120}{15}\right)^{0{\cdot}71} \times 3{\cdot}36\\ &= 8^{0{\cdot}71} \times 3{\cdot}36 = 4{\cdot}377 \times 3{\cdot}36\\ &= 14{\cdot}7 \text{ cu. ft.}\end{aligned}$$

Work done during expansion

$$\begin{aligned}&= \frac{P_1V_1 - P_2V_2}{n - 1}\\ &= \frac{144 \times 120 \times 3{\cdot}36 - 144 \times 15 \times 14{\cdot}7}{1{\cdot}41 - 1}\\ &= \frac{58{,}200 - 31{,}880}{0{\cdot}41} = \frac{26{,}320}{0{\cdot}41}\\ &= 64{,}200 \text{ ft.-lb.}\\ &= 45{\cdot}8 \text{ C.H.U.}\end{aligned}$$

Final temperature is found from

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

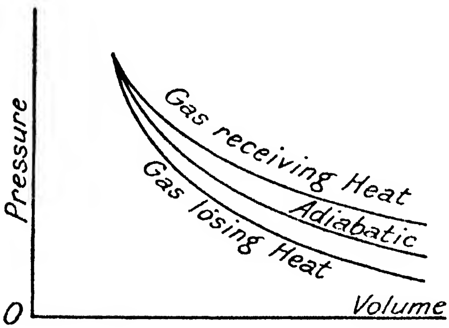
$$T_2 = \left(\frac{P_2}{P_1}\right)\left(\frac{V_2}{V_1}\right)T_1 = \frac{4 \cdot 385}{8} \times 473$$

$$= 259^\circ \text{ C. abs.}$$

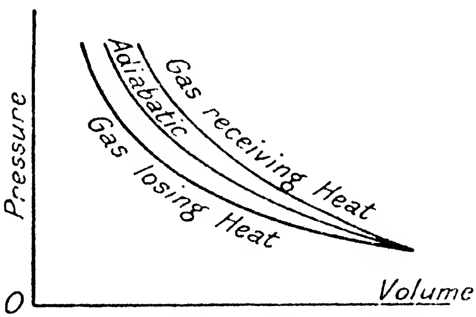
$\therefore$ Reduction in internal energy $= 1 \cdot 25 \times 0 \cdot 17(473 - 259)$
$= 1 \cdot 25 \times 0 \cdot 17 \times 214$
$= 45 \cdot 6$ C.H.U.

This quantity is equal to the work done above. The slight difference of 0·2 is due to slide rule error.

44. If a gas is expanding and receiving heat, the curve will be above the adiabatic, and if losing heat, below the adiabatic (fig. 28).



Expansion.
FIG. 28.



Compression.
FIG. 29.

If the gas is being compressed, the curve lies above the adiabatic when the gas receives heat, and below the adiabatic when heat is being rejected (fig. 29).

EXAMPLE. Five cubic feet of gas expand in a cylinder from a pressure of 200 lb. per sq. in. absolute, and temperature of 300° C. to a pressure of 25 lb. per sq. in. absolute, according to the law $PV^{1 \cdot 2} = C$. $C_p = 0 \cdot 238$ and $C_v = 0 \cdot 169$. Find the heat interchanged between the gas and the cylinder wall.

To find the final volume. $P_1V_1^{1 \cdot 2} = P_2V_2^{1 \cdot 2}$

$$V_2 = \left(\frac{P_1}{P_2}\right)^{0 \cdot 835} \times 5 = \left(\frac{200}{25}\right)^{0 \cdot 835} \times 5$$

$$= 8^{0 \cdot 835} \times 5 = 5 \cdot 676 \times 5$$

$$= 28 \cdot 38 \text{ cu. ft.}$$

$$\text{Work done} = \frac{P_1V_1 - P_2V_2}{n - 1}$$
$$= \frac{144 \times 200 \times 5 - 144 \times 25 \times 28{\cdot}38}{1{\cdot}2 - 1}$$
$$= \frac{144{,}000 - 102{,}000}{0{\cdot}2}$$
$$= 210{,}000 \text{ ft.-lb.} \div 150 \text{ C.H.U.}$$

Now find weight of gas.

$$PV = wRT$$
$$w = \frac{PV}{RT} = \frac{200 \times 144 \times 5}{1{,}400(0{\cdot}238 - 0{\cdot}169) \times 573}$$
$$= 2{\cdot}6 \text{ lb.}$$

To find the final temperature.

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$
$$T_2 = \frac{P_2V_2}{P_1V_1}.T_1$$
$$= \frac{25}{200} \times \frac{28{\cdot}38}{5} \times 573 = \frac{5{\cdot}676}{8} \times 573$$
$$= 407^\circ \text{ C. abs.}$$

The temperature is lower than at the commencement, hence, internal energy is less.

$$\text{Reduction in internal energy} = w \times C_v(T_1 - T_2)$$
$$= 2{\cdot}6 \times 0{\cdot}169 \times (573 - 407)$$
$$= 2{\cdot}6 \times 0{\cdot}169 \times 166$$
$$= 73{\cdot}0 \text{ C.H.U.}$$

$$\text{Heat added} = \text{work done by gas} + \text{gain in internal energy}$$
$$= 150 - 73$$
$$= 77 \text{ C.H.U.}$$

Referring to fig. 16, it will be seen that the curve for $n = 1{\cdot}2$ lies between the isothermal, where $n = 1$, and the adiabatic, where $n = 1{\cdot}41$.

In the isothermal case, heat would be added equal to the external work done. In the adiabatic case, no heat would be added.

The case of this example is that where some amount of he less than the external work must be added.

Examples VI

1. A quantity of gas, enclosed in a cylinder, is at 150 lb. per sq. in. by gauge and has a volume of 4 cu. ft. The gas expands according to the law $PV^{1{\cdot}25} = C$ until the volume is 10 cu. ft. Find the pressure after expansion.

2. Air at a pressure of 160 lb. per sq. in. absolute is confined in a space of 1·5 cu. ft. It expands according to the law $PV^{1 \cdot 4} = C$ until the pressure is 30 lb. per sq. in. Find the final volume.

3. The same quantity of gas is found at different times during an expansion in the following two states:—(*a*) Pressure 160 lb. per sq. in. by gauge, volume 2·4 cu. ft.; (*b*) Pressure 40 lb. per sq. in. by gauge, volume 7 cu. ft. Find the law of expansion.

4. 4 lb. of air expand according to the law $PV = 96T$ at an absolute temperature of 305° C. The expansion ratio is 4. Find the work done during expansion.

5. Steam in a cylinder is cut off at $\frac{1}{3}$ stroke. The clearance space is 0·25 cu. ft. and the swept volume is 3·5 cu. ft. Admission pressure, 80 lb. per sq. in. absolute. Assuming expansion according to the law $PV = C$, find the work done during the expansion.

6. Air at a pressure of 180 lb. per sq. in. absolute and occupying a volume of 12 cu. ft. expands according to the law $PV^{1 \cdot 4} = C$ until its volume is 32 cu. ft. Find the work done during expansion.

7. A quantity of air at 13·5 lb. per sq. in. absolute pressure and volume 14 cu. ft. is compressed according to the law $PV^{1 \cdot 2} = C$ until the pressure is 80 lb. per sq. in. absolute. Calculate the work done on the air during compression.

8. If 2 lb. of gas expand isothermally at a temperature of 14° C. with an expansion ratio of 5, determine the amount of heat in heat units which must be supplied to the gas. $R = 96$.

9. An internal combustion engine has a clearance volume of 774 cu. ins., diameter of cylinder 15 in., stroke 19 in. Compression takes place according to the law $pv^{1 \cdot 3} =$ constant.

The pressure at the commencement of compression is 14 lb. per sq. in. absolute and the temperature 100° C. Find the pressure and temperature at the end of compression. U.L.C.I.

10. A motor-car engine has a diameter of cylinder of $3\frac{3}{4}$ in. and a stroke of 5 in. The clearance volume is 14·4 cu. in. Determine the compression ratio of the engine. If the temperature of the mixture at the commencement of compression is 60° C., find the temperature at the end of the compression stroke, assuming compression follows the law $PV^{1 \cdot 3} =$ constant. Atmospheric pressure = 14·7 lb. per sq. in. U.L.C.I.

11. At the commencement of the compression stroke of a gas engine the pressure is 14·7 lb. per sq. in. and the temperature is 90° C. If the compression follows the law $pv^{1 \cdot 3} =$ constant, find the pressure and temperature of the gas when it is compressed to $\frac{1}{3}$ of its volume. U.L.C.I.

12. A gas-engine cylinder has a bore of $6\frac{1}{2}$ in. and a stroke of 12 in. The volume of the combustion chamber is 120 cu. in. At the beginning of the compression stroke the mixture enclosed has a temperature of 120° C. and a pressure of 13·5 lb. per sq. in. If the law connecting the pressure and volume during the compression stroke is $PV^{1 \cdot 4} =$ constant, find the pressure and temperature at the end of the compression stroke.

State the laws of gases on which your determinations depend. U.E.I.

13. One pound of a gas at a pressure of 15 lb. per sq. in. (absolute) and a temperature of 15° C. is compressed adiabatically to 400 lb. per sq. in. (absolute). Assuming that PV = 96T, and that the index for adiabatic compression is 1·408, calculate the final volume and temperature.

What is the ratio of compression and how much work has been done during the compression? U.L.C.I. (A.)

14. The characteristic equation of a certain gas is PV = 100T, and the specific heat at constant volume is 0·175. One pound of this gas at a pressure of 75 lb. per sq. in. (absolute) is expanded from a volume of 6 cu. ft. to a volume of 36 cu. ft. and the equation of the curve of expansion is $PV^{1\cdot31}$ = constant. Find the number of C.H.U. interchanged between the gas and the cylinder walls during the expansion, and the direction of the flow. U.L.C.I. (A.)

15. A gas had a pressure of 150 lb. per sq. in. absolute, its volume was 160 cu. ft., and it weighed 119 lb. After adiabatic expansion to 15 lb. per sq. in. absolute, its temperature was 15° C., and its volume was 1,000 cu. ft. How much work was done per lb. of gas during expansion and what was the temperature before expansion? U.L.C.I.

16. A gas container of 20 cu. ft. capacity contains gas at 500 lb. per sq. in. absolute and 70° F. (21·1° C.). The specific heats of this gas are Cp = 0·218 and Cv = 0·155. If 10 lb. of gas are discharged, determine (*a*) the volume occupied by the discharged gas at 100 lb. per sq. in absolute and 40° F. (4·4° C.); (*b*) the weight and pressure of the gas remaining in the container; (*c*) the index γ (in pv^{γ} = constant, for adiabatic operations) for this gas. U.L.C.I.

17. If 0·1 lb. of gas occupying 0·5 cu. ft. is expanded in a cylinder at a constant pressure of 160 lb. per sq. in. absolute until its volume is 1 cu. ft. and then expanded adiabatically to 5 cu. ft., find (*a*) the temperature of the gas at the end of each expansion; (*b*) the total work done; (*c*) the change in internal energy during the adiabatic expansion. The specific heat of the gas at constant pressure and constant volume may be taken as 0·254 and 0·185. U.L.C.I.

18. An oil-engine has a cylinder diameter of 15 in. and a stroke of 18 in. Assuming the pressure and temperature at the beginning of compression to be 15 lb. per sq. in. and 90° C., and that the compression is according to the law $PV^{\frac{4}{3}}$ = C., find the clearance volume in order that the temperature at the end of compression may be 600° C. What is the compression ratio and what is the compression pressure? U.L.C.I.

19. 10 cu. ft. of a gas at 15 lb. per sq. in. absolute and 20° C. are compressed, according to the law $PV^{1\cdot2}$ = constant, to a pressure of 150 lb. per sq. in. absolute. Determine for this gas (*a*) its weight; (*b*) the ft.-lb. of work done during the compression; (*c*) the change in internal energy, and (*d*) the heat rejected or absorbed (state which). Assume Cp = 0·249 and γ = 1·4.

CHAPTER VII

THE WORKING CYCLE

45. Meaning of Cycle. A cycle may be defined as a series of operations, occurring in a certain order, which restore the initial conditions. When the series is complete the cycle may be repeated, the processes being repeated in the same order.

The action of all engines is cyclic and may be represented by lines upon the field of a Pressure-Volume diagram, or Temperature-Entropy diagram (see Chapter XII).

The cycles may be those of imaginary perfect engines, or of actual engines. In the former case they are called ideal cycles and in the latter actual cycles. Examples of ideal cycles are the Carnot or Constant Temperature cycle, the Otto or Constant Volume cycle, and the Rankine cycle. The ideal cycles are used as bases of comparison for the performance of actual engines.

The cycles are often named in accordance with the conditions under which the heat is supplied ; for example, in the Constant-Temperature cycle the heat is received at constant temperature. In ideal cycles all accidental heat losses are prevented, and the working substance is assumed to behave like a perfect working substance, and to follow simple laws.

46. The Carnot Cycle. This cycle consists of four simple operations, namely, Isothermal Expansion, Adiabatic Expansion, Isothermal Compression, Abiabatic Compression, carried out in the order given. The adiabatic compression brings the substance back to the same state as that in which it existed at the beginning of the isothermal expansion.

The efficiency of the Carnot cycle is the maximum possible efficiency attainable by a perfect heat engine using a perfect working substance. It is the ultimate standard of comparison for all heat engines. No engine works upon this cycle, because the mean effective pressure is low, and it is impossible to achieve the conditions necessary to carry out the cycle.

The conditions of the Carnot cycle may be imagined to occur in the following manner.

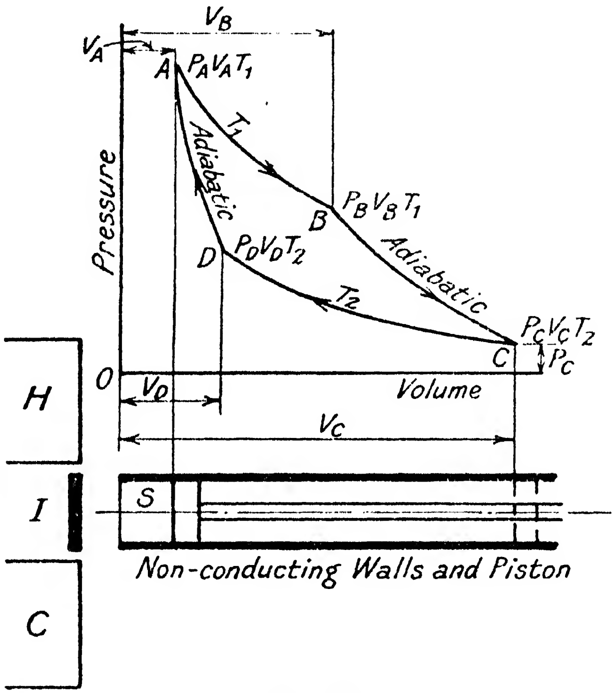


FIG. 30.

One pound weight of a perfect gas is placed in the cylinder at S (fig. 30). The cylinder material is supposed to be perfectly non-conducting, except at the end. A source of heat H is supposed to produce unlimited heat at constant temperature. A non-conducting cover, I, and a sump for heat are also required. The sump is of infinite capacity, its temperature thus remaining constant.

The temperature of H is T_1 and the temperature of the working substance is also T_1 when receiving heat from the source. The sump is at temperature T_2 and the working substance has that temperature when rejecting heat to it.

7. The following data were recorded during a boiler test:

Duration of trial	8 hours
Boiler pressure, in lb. per sq. in. absolute	183
Temperature of air in boiler house	88° F.
Temperature of gases in chimney	522° F.
Temperature of feed water	59·9° F.
Total feed water	102,600 lb.
Total coal	10,738 lb.
Contents of ash pans	148 lb.
Combustibles percentage of ash pans contents	42·7
Air used, per lb. of coal	17 lb.

Determine:

(*a*) The heat units used in generating steam from feed water per lb. of coal.

(*b*) The number of heat units passing away up the chimney per lb. of coal.

(*c*) The heat units represented by unburnt fuel per lb. of coal.

(*d*) The thermal efficiency per cent.

(*e*) By difference the heat units lost by radiation, imperfect combustion, etc., per lb. of coal.

Given—The calorific value of the coal 14,300 B.Th.U. and the specific heat of the products of combustion 0·24. U.L.C.I.

8. A steam plant consisting of a boiler, superheater and economizer generates steam at 200 lb. per sq. in. absolute. During a test 11,200 lb. of steam were generated per hour and 1,490 lb. of coal were burnt per hour, the calorific value of the coal being 12,100 B.Th.U. per lb. The temperatures of the feed water on entering and leaving the economizer were 91° F. and 264° F. The dryness fraction of the steam leaving the boiler was 0·97 and the temperature of the steam leaving the superheater was 613° F.

Determine the overall efficiency of the plant and the percentage of the available heat utilized in the economizer, boiler and superheater respectively. U.L.C.I.

9. State the principal sources of loss of efficiency in a boiler plant and the methods adopted to minimize these losses. A boiler generates 50,000 lb. of steam per hour at 250 lb. per sq. in. absolute, superheated to a temperature of 561° F., from water at 75° F. If the calorific value of the fuel is 13,000 B.Th.U. per lb. and the boiler efficiency is 78 per cent., what weight of fuel is burnt per hour ? U.L.C.I.

10. What do you understand by "equivalent evaporation from and at 212° F.," and what is the significance of this quantity ?

A coal fired boiler plant consumes 800 lb. of coal per hour of calorific value 14,000 B.Th.U. per lb. and evaporates 6,752 lb. of feed water per hour at a temperature of 112° F., into steam at a pressure of 160 lb. per sq. in. absolute and temperature 523·6° F. Determine (*a*) the efficiency of the boiler plant, and (*b*) the equivalent evaporation from and at 212° F. U.L.C.I.

CHAPTER X

THE STEAM ENGINE PLANT

71. Essentials of a Steam Plant. In most of the preceding chapters, we have been dealing with the fundamental properties of gases and vapours in relation to the action of heat upon them.

In the last chapter we have treated boilers for steam raising, since that subject followed naturally upon the question of steam generation.

It is proposed now to deal in a general way with the equipment which is used in connection with a heat engine, using water vapour or steam as the working substance.

The component parts of a modern plant are:

(1) the feed pump, (2) the steam generator or boiler, (3) the engine, and (4) the condenser.

In addition to these, there are auxiliaries which assist these parts in better performing their respective functions.

The engine mentioned in (3) above may be a reciprocating engine or a turbine.

The relative arrangement of the components is shown in fig. 41, and it will be noted it is divided into three sections:

(*a*) The high pressure section on the left of OX.

(*b*) The low pressure section between O′Y and O′Y′.

(*c*) The engine section, between OX and OY, where the work is done and where the pressure falls.

The condenser will generally need, as accessories, an extraction pump to remove condensed water from the condenser, an air ejector to remove air which would otherwise accumulate, and a circulating pump to maintain a supply of condensing water. In greater detail, then, the functions of the principal parts are as follows:

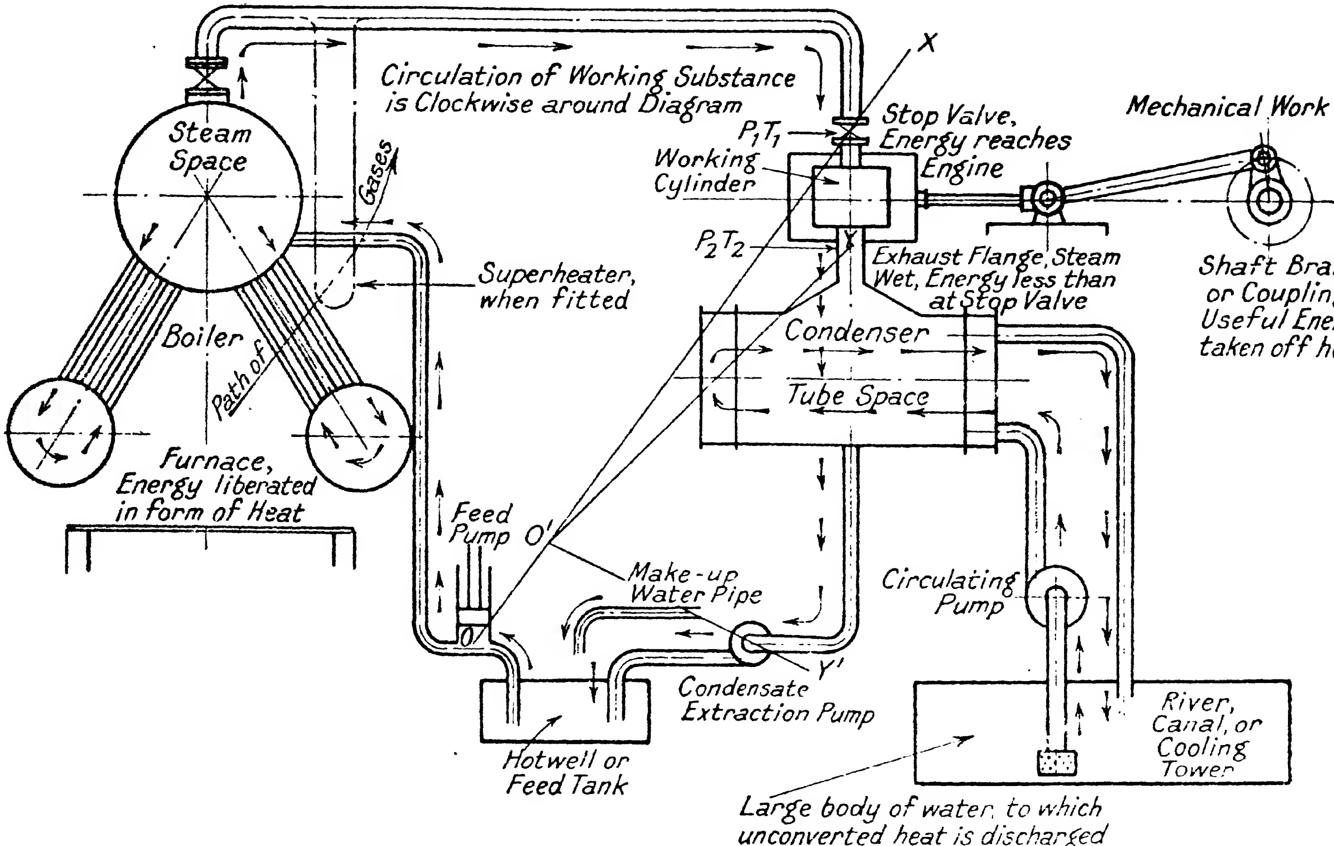

Circulation of Working Substance
is Clockwise around Diagram
Steam
Space
Boiler
Path of
Gases
Furnace,
Energy liberated
in form of Heat
Superheater,
when fitted
Feed
Pump
O'
O
Make-up
Water Pipe
Hotwell or
Feed Tank
Condensate
Extraction Pump
Y'
X
P_1T_1
P_2T_2
Working
Cylinder
Stop Valve,
Energy reaches
Engine
Exhaust Flange, Steam
Wet, Energy less than
at Stop Valve
Condenser
Tube Space
Mechanical Work
Shaft Brake
or Coupling,
Useful Energy
taken off here
Circulating
Pump
River,
Canal, or
Cooling
Tower
Large body of water, to which
unconverted heat is discharged

FIG. 41.

(1) The extraction pump and feed pump remove water from the low pressure section and force it into the boiler which is at high pressure.
(2) The boiler produces steam with its high energy content.
(3) The engine receives heat for the purpose of converting it into mechanical work.
(4) The condenser is to maintain a low temperature and low pressure, thus increasing the amount of work which may be obtained per pound of steam.
(5) The air ejector removes air from the condenser as fast as it enters.
(6) The hot-well receives hot water discharged from the condenser, and serves as feed-water tank.

To make allowance for losses due to blowing at the safety-valve and leakage into the atmosphere, it is necessary to add a small amount of water to the boiler supply. This is called the make-up feed, and is a very small quantity compared to the amount circulating in a well-controlled up-to-date plant. It is now the practice to close the feed system and de-aerate the make-up water before introducing it. This assists in reducing corrosion and in maintaining a high vacuum.

72. The Reciprocating Engine. At this stage the student is expected to be familiar with the general arrangement of a reciprocating steam engine, and the details are best studied in connection with Engineering Drawing and Design.

It is necessary to grasp, however, that it is in the cylinder of such an engine that the heat is converted into work. The rest of the engine is concerned mainly with changing the reciprocating motion of the piston into rotary motion, working the necessary valves, and controlling the speed by means of the governor.

The steam is distributed to the cylinder by means of the admission valves which admit steam during an appropriate interval, and the exhaust valves which permit it to escape at the end of the useful stroke. There must be one admission and one exhaust valve for each side of the piston in a double-acting engine. This is quite apparent in the case of drop valves, Corliss valves and similar types, but it is not so apparent in the case of the common slide valve which is really four valves in one casting. Each edge corresponds to one single-purpose valve.

In the uniflow engine the faces of the piston act as exhaust valves.

The student at this stage is recommended to take any available opportunity of studying technical journals and makers' catalogues, with a view to becoming familiar with different types of engines and their details.

73. Steam Movements. The events occurring for *one side of the piston* are as follows :

(1) Admission of steam by opening inlet valve.
(2) Cessation of steam supply, or "cut-off" by closing inlet valve.
(3) Release of steam by opening the exhaust valve.
(4) Compression of steam in the clearance space, after closing of the exhaust valve.

These events are arranged so that there is appreciable movement of the piston between events (2) and (3).

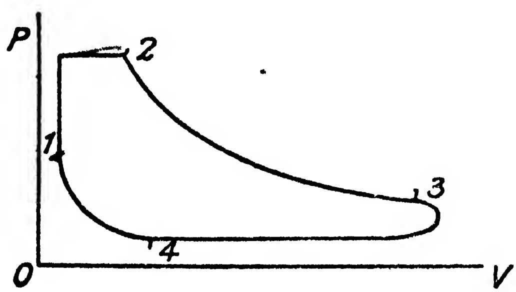


FIG. 42.

Here expansion occurs, when the steam gives up some of its internal energy. These events are shown by corresponding numbers on the diagram of fig. 42.

When the steam is exhausted it is passed to the atmosphere in the case of a non-condensing engine, or to the condenser in a condensing engine. In the former type, the whole of the indicator diagram lies above the atmospheric line as shown in fig. 43.

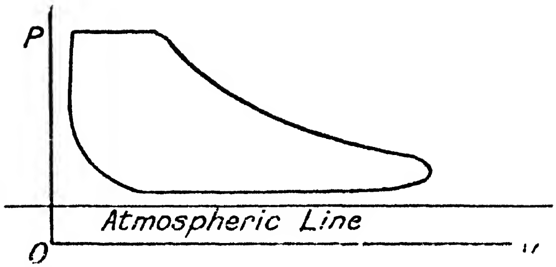


FIG. 43.

A condensing engine diagram will lie partly below the atmospheric line as indicated in fig. 44, and in the low-pressure cylinder of a compound engine, the whole diagram is sometimes below this line.

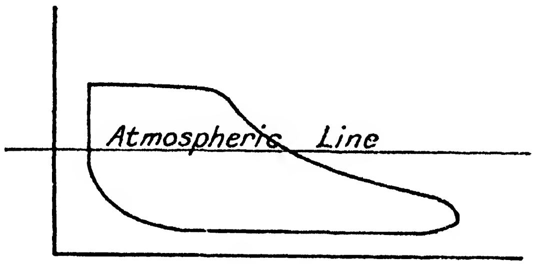


FIG. 44.

74. Condenser Function and Types. The function of the condenser is to maintain a low temperature. This is necessary to obtain the maximum possible energy from the steam and to secure high efficiency. This fact will be seen at once if the student has grasped the principle involved in Carnot's Cycle. The condenser is the cold body to which heat is rejected. Low pressure accompanies the low temperature, and thus all condensers maintain a vacuum under normal conditions. The lowest temperature ideally possible is that of the available water supply for condensing purposes. This is called the condensing water, or sometimes the circulating water. The condensed steam itself is called the *condensate*. The temperature of the condensate is higher on leaving the condenser than that of the circulating water at inlet. The condensate thus has a considerable liquid heat reckoned from 32° F. It is discharged into the hot-well, whence the feed pump transfers it to the boiler.

Total expansion ratio $= 4 \times 3 = 12$

$$\text{Referred hyp. M.E.P.} = \frac{p_1(1 + \log_e r)}{r} - p_B$$

$$= \frac{120(1 + \log_e 12)}{12} - 3$$

$$= 10(1 + 2{\cdot}3026 \times 1{\cdot}0792) - 3$$

$$= 34{\cdot}9 - 3 = 31{\cdot}9 \text{ lb. per sq. in.}$$

Probable referred M.E.P. $= 31{\cdot}9 \times 0{\cdot}65 = 20{\cdot}7$ lb. per sq. in.

Volume of L.P. cylinder $\times$ M.E.P. referred $\times$ strokes per min. $=$ Work done per min.

Vol. of L.P. $\times 20{\cdot}7 \times 144 \times 2 \times 75 = 300 \times 33{,}000$.

$$\therefore \text{Vol. of L.P.} = \frac{300 \times 33{,}000}{2 \times 75 \times 20{\cdot}7 \times 144}$$

$$= \frac{33{,}000}{20{\cdot}7 \times 72} = 22{\cdot}1 \text{ cu. ft.}$$

Let D = diameter in ft. of L.P. cylinder.

Then $0{\cdot}7854\text{D}^2 \times \text{D} = 22{\cdot}1$

$$\text{D}^3 = \frac{22{\cdot}1}{0{\cdot}7854} = 28{\cdot}15$$

$$\therefore \text{D} = 3{\cdot}05 \text{ ft.}$$

and Stroke $= 3{\cdot}05$ ft.

$$\text{Vol. of H.P. cylinder} = \frac{22{\cdot}1}{3} = 7{\cdot}36 \text{ cu. ft.}$$

If $d =$ diam. of H.P. cylinder

then $0{\cdot}7854d^2 \times \text{D} = 7{\cdot}36$

$$d^2 = \frac{7{\cdot}36}{0{\cdot}7854 \times 3{\cdot}05} = 3{\cdot}07$$

$$\therefore d = 1{\cdot}75 \text{ ft.}$$

Cylinder sizes are H.P. 1·75 ft. dia., 3·05 ft. stroke.
L.P. 3·05 ft. dia., 3·05 ft. stroke.

Examples XI

1. Using the method of Article 41, draw to scale a hypothetical indicator diagram for a steam engine whose stop-valve pressure is 120 lb. per sq. in. absolute, back pressure 17 lb. per sq. in., and cut-off $\frac{1}{4}$ stroke. Find the mean effective pressure on this diagram. Use a pressure scale of 50 lb. per sq. in. per in. height of diagram, and let 3 in. represent the full stroke.

2. By the same method as in example 1 draw a hypothetical indicator diagram for an expansion ratio of 12. Steam pressure is 180 lb. per sq. in. absolute and back pressure is 2 lb. per sq. in. Choose your own scale (not too small) and find the mean effective pressure.

3. Assume that example 2 applies to a cylinder of 20 in. diam. and 36 in. stroke. Take a diagram factor of 0·8 and 120 r.p.m. as the speed of the engine, double acting, and calculate the probable I.H.P.

4. Check the results of examples 1 and 2 by calculation.

5. A two-cylinder double-acting steam engine has to develop 280 I.H.P. at a speed of 160 r.p.m. The pressure is 180 lb. per sq. in. absolute, and the back pressure is 2 lb. per sq. in. absolute. Cut-off occurs at $\frac{3}{8}$ stroke and the diagram factor is 0·75. If the stroke is $1\frac{1}{2}$ times the diameter, find the cylinder sizes.

6. Assume the particulars in example 2 to apply to a compound double-acting engine of 75 I.H.P. with diagram factor 0·7, and speed 120 r.p.m. The cylinder ratio is 3 and the diameter and stroke of the low-pressure cylinder are equal. The strokes of both cylinders are equal. Calculate the cylinder sizes.

7. Determine the size of the cylinder for an engine which is required to develop 200 I.H.P. at 250 r.p.m., when the initial pressure is 180 lb. per sq. in. absolute and the back pressure is 16 lb. per sq. in. Assume a hyperbolic expansion curve, and a cut-off of 25 per cent. of the stroke. Assume a diagram factor of 0·8. Proportion of stroke to piston diameter = 4: 3. N.C.T.E.C.

8. Calculate the cylinder diameters and stroke of a compound engine to develop 600 I.H.P. under the following conditions:—Admission pressure 170 lb. per sq. in. absolute; back pressure 2 lb. per sq. in. absolute; revolutions per min. 120; mean piston speed 720 ft. per min.; number of expansions 15; diagram factor 0·85; cylinder ration 3·5. N.C.T.E.C.

9. Calculate the cylinder diameter and stroke of a four-stroke gas engine to develop 25 B.H.P. at 400 r.p.m.; assuming a mechanical efficiency of 80 per cent., mean effective pressure 80 lb. per sq. in., ratio of stroke to cylinder diameter 1·5. N.C.T.E.C.

10. Steam at an absolute pressure of 240 lb. per sq. in. is to be supplied to a triple-expansion engine having a total expansion ratio of 24. Assuming that the whole of the work is done in the low pressure cylinder, find the diameter of this cylinder in order that the engine may develop 3,000 horse power at a speed of 105 r.p.m. Length of stroke 3 ft., back pressure 3 lb. per sq. in. absolute, and diagram factor 0·6. U.L.C.I.

11. A double-acting single-cylinder steam engine is to develop 50 I.H.P. at a speed of 120 r.p.m. Steam pressure at admission 65 lb. per sq. in. absolute, cut-off 0·4 stroke. Stroke 18 in. Assuming a back pressure of 16 lb. per sq. in. absolute and a diagram factor of 0·85, find the diameter of the cylinder. What would you expect the B.H.P. to be? U.L.C.I.

12. Explain the term "diagram factor."

In a double-acting single-cylinder steam engine, the cylinder diameter is 15 in. and the stroke 20 in. Steam is admitted at an absolute pressure of 70 lb. per sq. in., and is cut off at 0·35 stroke, and the back pressure of 15·5 lb. per sq. in. absolute. The B.H.P. at a speed of 150 r.p.m. is 60, and the mechanical efficiency is 81 per cent. Find the diagram factor for the engine. U.L.C.I.

13. Write down the formula for the determination of the mean effective pressure obtained from the theoretical (steam engine) indicator diagram. State the meaning of each symbol used and the assumptions upon which the formula is based.

A single-cylinder double-acting steam engine develops 50 I.H.P. when running at 230 r.p.m. The cylinder is 9 ins. diam., and the stroke $13\frac{1}{2}$ in. The steam is supplied at 75 lb. per sq. in. gauge, and the condenser pressure is 5 lb. per sq. in. The cut-off takes place at 0·625 stroke. Determine the value of the diagram factor. U.E.I.

14. A two-cylinder compound double-acting steam engine is required to develop 180 I.H.P. in each cylinder. The average piston speed must not exceed 750 ft. per min. Given the following further data, find the stroke of the engine and the requisite diameter of the H.P. cylinder. Boiler pressure 80 lb. per sq. in. absolute. Pressure in L.P. steam chest 16 lb. per sq. in. absolute. Revolutions per min. = 125. Cut-off in H.P. cylinder at $\frac{1}{2}$ stroke. Diagram factor, 0·85. U.E.I.

15. If the ratio of volumes of the cylinders in a compound engine be $1:3\frac{1}{2}$, the boiler pressure 90 lb. per sq. in. by gauge, the terminal pressure 10 lb. per sq. in. absolute, and there is a drop in pressure of 5 lb. per sq. in. between the boiler and the engine, determine the point of cut-off in the H.P. cylinder. U.E.I.

16. A locomotive has two cylinders each 20 in. diam. by 2 ft. stroke. The initial pressure of the steam is 250 lb. per sq. in. absolute, and the back pressure is 20 lb. per sq. in. absolute. The clearance volume is 7 per cent. of the volume swept by the piston. Use a diagram factor of 0·85 and calculate the indicated horse power developed by the engine when cut-off occurs at $\frac{3}{8}$ stroke, and the driving wheels are making 120 r.p.m. U.E.I.

17. A locomotive has three double-acting cylinders, each 18 in. diameter and 26 in. stroke. The steam supply pressure is 210 lb. per sq. in., exhaust pressure 25 lb. per sq. in. and cut-off takes place at 0·375 of the stroke. Determine the I.H.P. developed by the locomotive when running at 150 r.p.m. Assume a diagram factor of 0·7.

If the steam consumption is 32,000 lb. per hour at initial temperature of 546° F., determine the indicated thermal efficiency of the engine. U.L.C.I.

(Assume the feed temperature 126·1° F.—Authors).

CHAPTER XII

ENTROPY AND ENTROPY DIAGRAMS

85. Entropy. Entropy is a very useful conception in heat engine theory. When understood it leads us to important results which, by other methods, can only be obtained much more laboriously.

Entropy cannot be readily defined in physical terms, and hence the student is advised to follow carefully what is said concerning entropy, and the examples given in this chapter, with confidence that his conception of it will become clear.

All heat is not equally valuable for converting into work. Heat that is put into a substance at a high temperature has a greater possibility of conversion into work than heat put into the substance at a lower temperature. Entropy is a function of a quantity of heat which indicates the possibility of converting that heat into work. When heat is added at a high temperature entropy increase is small; when heat is added at a low temperature entropy is greater. Thus, for maximum entropy, we have minimum availability for conversion into work. For minimum entropy, we have maximum availability for conversion into work.

Carnot's cycle shows us that the higher the temperature of supply of the heat, the greater will be the efficiency, and hence, the greater will be the work done. Entropy can indicate this availability of the heat.

If entropy be plotted horizontally, and absolute temperature be plotted vertically on a diagram during the addition of heat to a body, then we have the area of this diagram representing heat. For any point on the curve it follows that

Entropy change $\times$ Absolute temperature $=$ Heat change.

$$\text{Hence, Entropy change} = \frac{\text{Heat change}}{\text{Absolute temperature}}.$$

Another way of defining entropy is to say that it is that thermal property of a substance which remains constant when the substance expands or is compressed adiabatically in a cylinder.

86. Behaviour of Entropy. (1) Entropy increases when heat is supplied whether temperature changes or not.

(2) Entropy decreases when heat is removed whether temperature changes or not.

Entropy remains constant in all adiabatic frictionless processes.

Entropy increases if heat is lowered in temperature without work being done, as in a throttling process.

Entropy increases if heat is degraded or reduced in availability by friction.

The unit of Entropy is called the "Rank."

EXAMPLE. 50 B.Th.U. of heat are supplied to a substance at a constant temperature of 600° F. absolute. Find the increase in entropy.

$$\text{Increase in entropy} = \frac{50}{600}$$

$$= 0{\cdot}0833 \text{ Ranks.}$$

This gives a working form for the definition of entropy.

Entropy, like quantities of heat, must be reckoned from some specified datum of temperature, and for unit mass of substance.

We are not often concerned with absolute units of entropy, but we require to calculate changes in entropy very frequently.

Let ϕ_1 = final entropy. ✓

ϕ_2 = initial entropy.

H = heat supplied in B.Th.U. or C.H.U.

T = absolute temperature in °F. or °C. during the supply.

Then when T is constant $\phi_1 - \phi_2 = \text{H}$ ranks.

The number of units of entropy will be the same whether C.H.U. and ° C. are used on B.Th.U. and ° F., thus showing that the entropy depends only on the state of the heat supplied, as regards temperature. The actual quantity of heat and the actual temperature will be the same no matter in what units they are expressed.

EXAMPLE. Calculate the increase in entropy when 100 C.H.U. are supplied (*a*) at 500° C. absolute and (*b*) at 1,500° C. abs.

(*a*) $\phi_1 - \phi_2 = \frac{100}{500} = 0{\cdot}2$ ranks.

(*b*) $\phi_1 - \phi_2 = \frac{100}{1{,}500} = 0{\cdot}067$ ranks.

It is useful to note here the Carnot efficiency of this heat in the two cases, assuming the lowest temperature limit of the cycle to be 300° C. absolute in both cases.

$$\text{(a) Carnot efficiency} = \frac{500 - 300}{500} = 0{\cdot}4.$$

$$\text{(b)} \quad ,, \quad ,, \quad = \frac{1{,}500 - 300}{1{,}500} = 0{\cdot}8.$$

Thus work available in case (*a*) $= 0{\cdot}4 \times 100 = 40$ C.H.U.
and ,, ,, ,, ,, (*b*) $= 0{\cdot}8 \times 100 = 80$ C.H.U.

87. Case of Isothermal Expansion. With isothermal expansion or compression the temperature is constant, and the heat supplied or removed is equal to the work done by or on the gas.

$$\therefore \phi_1 - \phi_2 = \frac{RT \log_e r}{T}$$

for a perfect gas, R being in heat units per lb. in this case.

Isothermal expansion in a vapour, when in the presence of the liquid, is always evaporation and then

$$\phi_1 - \phi_2 = \frac{\text{Amount of latent heat supplied}}{\text{Absolute temperature of evaporation}}.$$

Thus entropy increase during evaporation of unit weight

$$= \frac{L}{T} \text{ ranks.}$$

The work done here is not equal to the heat supplied, because of the large amount of heat required to change the state from liquid to vapour.

The two cases of entropy dealt with above are very simple ones, since temperature has been constant. It is now necessary to consider the change in entropy, when the temperature is changing during the supply or removal of heat.

88. Change of Entropy. Let 1 lb. of substance be considered.

Let ϕ_1 = final entropy, ϕ_2 = initial entropy.

T_1 = final temperature, T_2 = initial temperature.

C = specific heat.

The value of C will depend upon the nature of the substance and its state. It will be different for different substances and for different states of any particular substance.

Let δH = a small increment of heat supplied whilst the temperature remains sensibly constant at T.

$\delta\phi$ = corresponding increment of entropy.

Then
$$\delta\phi = \frac{\delta H}{T}$$

but
$$\delta H = C.\delta T$$

$$\therefore\ \delta\phi = C\frac{\delta T}{T}$$

The total change of entropy from T_2 to T_1 becomes

$$\phi_1 - \phi_2 = C\int_{T_2}^{T_1} \frac{dT}{T}$$

$$= C\Big[\log_e T\Big]_{T_2}^{T_1} = C \log_e \frac{T_1}{T_2}.$$

The two forms of entropy expression, viz.

$$\frac{H}{T} \text{ and } C \log_e \frac{T_1}{T_2},$$

are the only forms met with and are very important.

89. Entropy of Steam. In this case, entropy is reackoned from 32° F. This means that the entropy in water at 32° F. is regarded as zero, but it is necessary to use absolute temperatures in entropy calculation.

Let T_s = absolute temperature of superheated steam.

T = ,, ,, ,, generation of steam.

ϕ_{ss} = entropy of superheated steam.

Entropy added to water in raising it from 32° F. (492° absolute) to T° absolute $= C \log_e \frac{T}{492}$.

C for water is unity.

Entropy added to steam during evaporation $= \frac{L}{T}$.

Entropy added to steam during superheating $= C_p \log_e \frac{T_s}{T}$ where C_p = specific heat at constant pressure. Thus the total entropy added in all three stages

$$= \phi_{ss} = C \log_e \frac{T}{492} + \frac{L}{T} + C_p \log_e \frac{T_s}{T}.$$

EXAMPLE. Calculate the entropy of 1 lb. of water at 122° F.

Zero entropy is at 32° F. or 492° absolute,

$$\therefore \phi_{w122} = C \log_e \frac{T}{492}$$
$$= 1 \times 2{\cdot}3026 \times \log_{10} \frac{582}{492}$$
$$= 2{\cdot}3026 \times \log_{10} 1{\cdot}182$$
$$= 2{\cdot}3026 \times 0{\cdot}073$$
$$= 0{\cdot}168 \text{ ranks.}$$

EXAMPLE. Calculate the change in entropy when water is heated from 50° C. to 150° C.

$$\phi_{w50} = C \log_e \frac{323}{273} = 0{\cdot}1672 \text{ ranks.}$$
$$\phi_{w150} - C \log_e \frac{423}{273} = 2{\cdot}3026 \times \log_{10} 1{\cdot}55$$
$$= 2{\cdot}3026 \times 0{\cdot}1903$$
$$= 0{\cdot}438 \text{ ranks.}$$

Education T – ϕ diagram is also shown in fig. 58, and is copied by permission of the Controller of His Majesty's Stationery Office. The base of the diagram is drawn at a temperature of — 460° F., or 0° F. absolute. The entropy of water is reckoned from 32° F., the ordinary freezing-point of water.

Consequently, all lines constructed on this diagram lie above this level of temperature, i.e. above the line AE of fig. 53. The point A is the starting point of the diagram and the line AB is the entropy-temperature line for water. It is called the *water line*. This line is the curve plotted from the equation $\phi_w = \log_e \frac{T}{492}$, assuming the specific heat of water to be constant and equal to unity. The highest point is fixed by the highest pressure for which the diagram is likely to be used.

The three points B_2, B_1 and B have values of temperature T_2, T_1 and T respectively, and these temperatures, when substituted in the above equation, give the corresponding entropies of water.

The steam line, CE, lies towards the right hand of the diagram, and slopes in the opposite direction to that of the water line. It can only be plotted when the water line has been set down. If evaporation be permitted to take place at temperature T, the latent heat at this temperature being L, the the length of the line BC will represent the evaporation entropy $\frac{L}{T}$, which is the change in entropy during evaporation at the temperature T. Similarly, the lengths B_1C_1 and B_2C_2 will represent $\frac{L_1}{T_1}$ and $\frac{L_2}{T_2}$ respectively.

The several lengths are set out from B, B_1 and B_2 and thus points C, C_1 and C_2 are obtained. A smooth curve drawn through these points is the *steam line*.

The diagram may now be divided into three parts or fields.

(1) The liquid or water field to the left of the water line.

(2) The saturation field between the water and steam lines.

(3) The superheat field on the right of the steam line.

91. Constant Pressure Lines in the Superheat Field. These lines are plotted with entropy and temperature as co-ordinates, the pressure being constant whilst the steam is superheated. The constant pressure superheat lines start from points on the steam line such as C, C_1 and C_2. The lines, such as CD, fig. 54, are plotted from the equation $\phi_s = C_p \log_e \frac{T_s}{T}$, using C as origin, and ϕ_s and $(T_s - T)$ as co-ordinates.

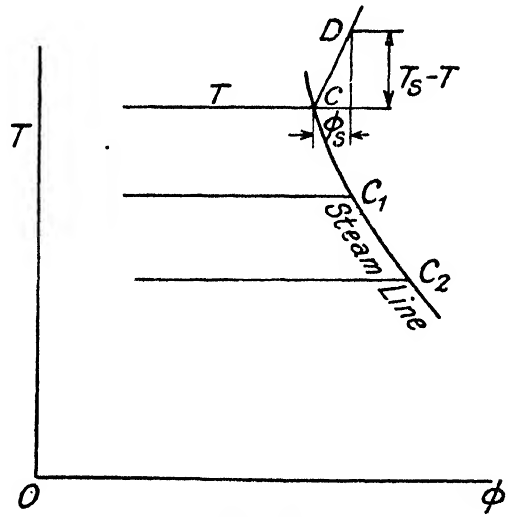


FIG. 54.

One point D is shown, and similar points can be found up to any degree of superheat likely to be needed. The whole line ABCD, in fig. 53, is a complete constant pressure line.

AB represents the supply of liquid heat.
BC „ „ „ „ latent heat.
CD „ „ „ „ additional heat for superheating.

92. Lines of Constant Dryness and Constant Superheat. If the evaporation is not complete, the steam will be wet and the entropy less than that of dry steam. The shortage of entropy is proportional to the shortage of latent heat. The actual evaporation entropy is now $\frac{xL}{T}$, where x is the dryness fraction. Suppose that F, in fig. 55, represents the state of the wet steam, then length BF is proportional to the actual evaporation entropy. Hence

$$\frac{BF}{BC} = \frac{\frac{xL}{T}}{\frac{L}{T}} = x.$$

Thus, the point F lies on the line BC such that

$$\frac{BF}{BC} = \frac{\text{amount of latent heat supplied}}{\text{latent heat of the steam (dry)}} = \frac{xL}{L} = x.$$

If points F_1 and F_2 divide the lines B_1C_1 and B_2C_2 in the same ratio $\frac{BF}{BC}$, then the points F, F_1 and F_2 are all state points for steam with the same dryness fraction, but at different pressures.

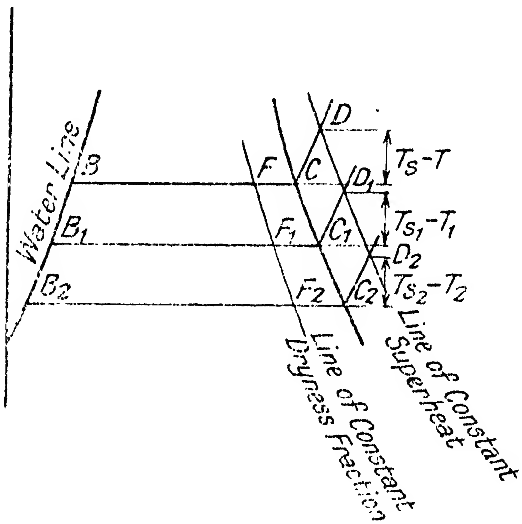


FIG. 55.

A smooth curve drawn through such points is a line of constant dryness.

The lines of constant superheat are plotted by locating points D, D_1 and D_2 on the constant pressure lines in the superheat field, starting from C, C_1 and C_2, by making $T_s - T = T_{s1} - T_1 = T_{s2} - T_2$. These lines are not often required, and consequently are not printed on the diagrams supplied at this stage of the subject.

93. Lines of Constant Volume. The curved lines AC, AC_1

and AC_2 shown in fig. 56 are lines of constant volume. (See also Art. 58).

Let the specific volumes at P, P_1 and P_2 be V, V_1 and V_2 respectively.

Then the volume represented by any point on the line AC must be V, that is, the volumes at C, H_1 and H_2 are equal to V.

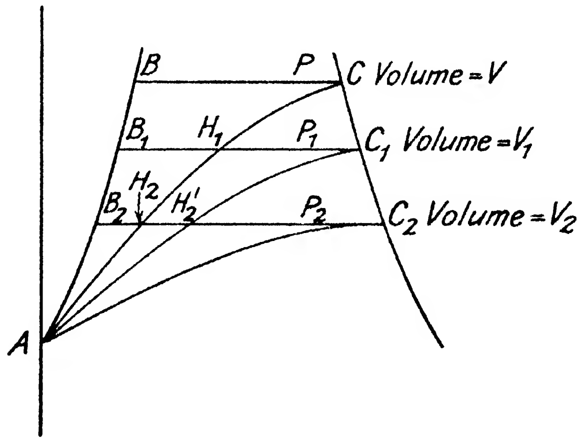


FIG. 56.

During the evaporation of 1 lb. of water, at a given moment, the proportion of the 1 lb. of water, which has become steam, is the same as the proportion of the latent heat which has been given to the water. Thus, if a point H_1 on B_1C_1 be taken, $\frac{B_1H_1}{B_1C_1}$ of the latent heat has been given to the water and thus $\frac{B_1H_1}{B_1C_1}$ of a pound of water has become steam. If, then, 1 lb. of steam at pressure P_1 occupies V_1 cu. ft., the volume of steam represented by H_1 is $\frac{B_1H_1}{B_1C_1} \times V_1$.

Make this equal to V, that is, let

$$V = \frac{B_1H_1}{B_1C_1} \times V_1$$

then

$$\frac{B_1H_1}{B_1C_1} = \frac{V}{V_1}$$

$\therefore$ if B_1C_1 represents V_1, B_1H_1 represents V.

H_2 may be chosen in a similar way to represent volume V at pressure P_2. Points thus located, when joined by a smooth curve, give a line of constant volume. Also for volume V_1, H_2' may be chosen on the pressure line P_2.

Note that the volume of the unevaporated water is neglected, but it is relatively very small. The completed diagram is shown in fig. 57.

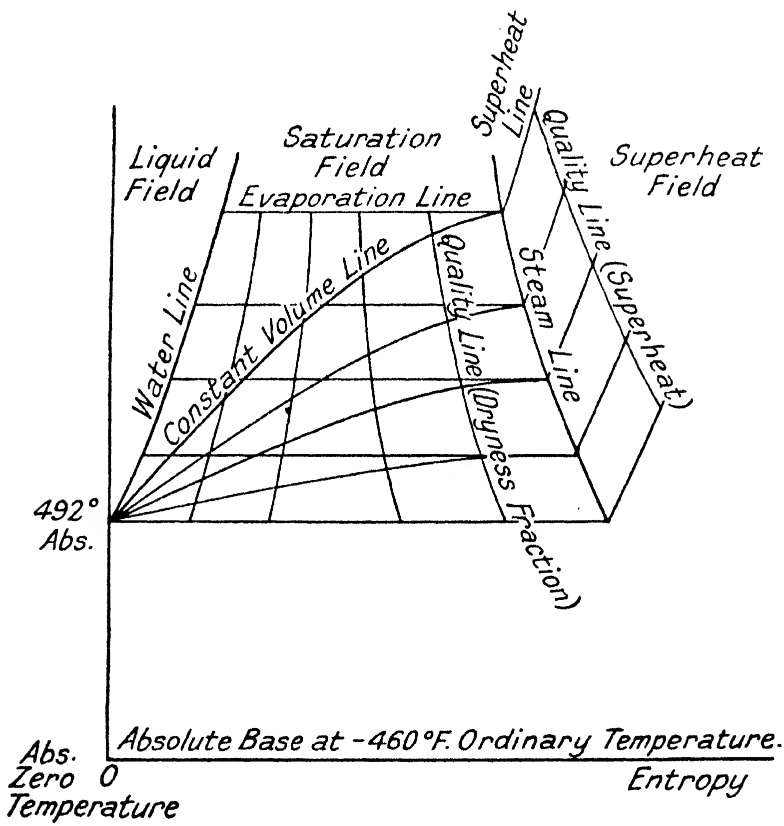


FIG. 57.

94. Scale of Diagram. Let

S_T = temperature scale in ° F. per inch vertical.

S_ϕ = entropy scale in Ranks per inch horizontal.

Then 1 square inch area $= S_T \times S_\phi$ heat units.

The area of the diagram represents energy in heat units, and work quantities may be found by measuring areas on this diagram, as in the case of the P – V diagram, but with this difference; in the P – V diagram the area must be multiplied by a work scale in foot-lb., whilst in the T – ϕ diagram the area must be multiplied by a heat scale in heat units.

The scales of the T – ϕ diagram generally in use are

entropy, 1 in. = 0·2 ranks.

temperature, 1 in. = 25° C. intervals or 45° F. intervals.

Thus 1 sq. in. = 0·2 × 25 = 5 C.H.U.

or „ „ „ = 0·2 × 45 = 9 B.Th.U.

EXAMPLE. The area of the diagram, representing a cycle plotted on the T-ϕ chart, is measured and found to be 14½ sq. in. Find the ideal work thus represented.

Work represented = 5 × 14·5 = 72·5 C.H.U.

or = 9 × 14·5 = 130·5 B.Th.U.

or = 5 × 14·5 × 1,400 = 101,500 ft.-lb.

EXAMPLE XII

1. One pound of water at 212° F. is changed into dry steam at 212° F. Calculate the increase in entropy.

2. Find the increase in entropy during the evaporation of 1 lb. of water at 356° F.

3. What is the increase in entropy during the evaporation of 1 lb. of water at a pressure of 200 lb. per sq. in. absolute?

4. Calculate the increase in entropy when 1 lb. of water at 59° F. is heated to a temperature of 356° F.

5. Determine the entropy, reckoned from 32° F., when 1 lb. of water is changed into steam at 356° F.

6. Steam has a pressure of 160 lb. per sq. in. absolute and 85 per cent. dry. If it was formed from feed water at 104° F., what was the entropy added to it per lb.?

7. Reckoned from 32° F., what is the entropy in 1 lb. of steam at 60 lb. per sq. in. absolute with 180 degrees Fahrenheit of superheat?

8. One pound of wet steam at 14·7 lb. per sq. in. has 1·6 ranks of entropy reckoned from 32° F. Find the dryness fraction and the volume of 1 lb. of the wet steam.

9. Starting from a pressure of 130 lb. per sq. in. absolute read off from an entropy chart the dryness fraction of 3 cu. ft. of steam at pressures descending by 20 lb. per sq. in. to 30 lb. per sq. in. What is the line thus followed on the entropy chart called?

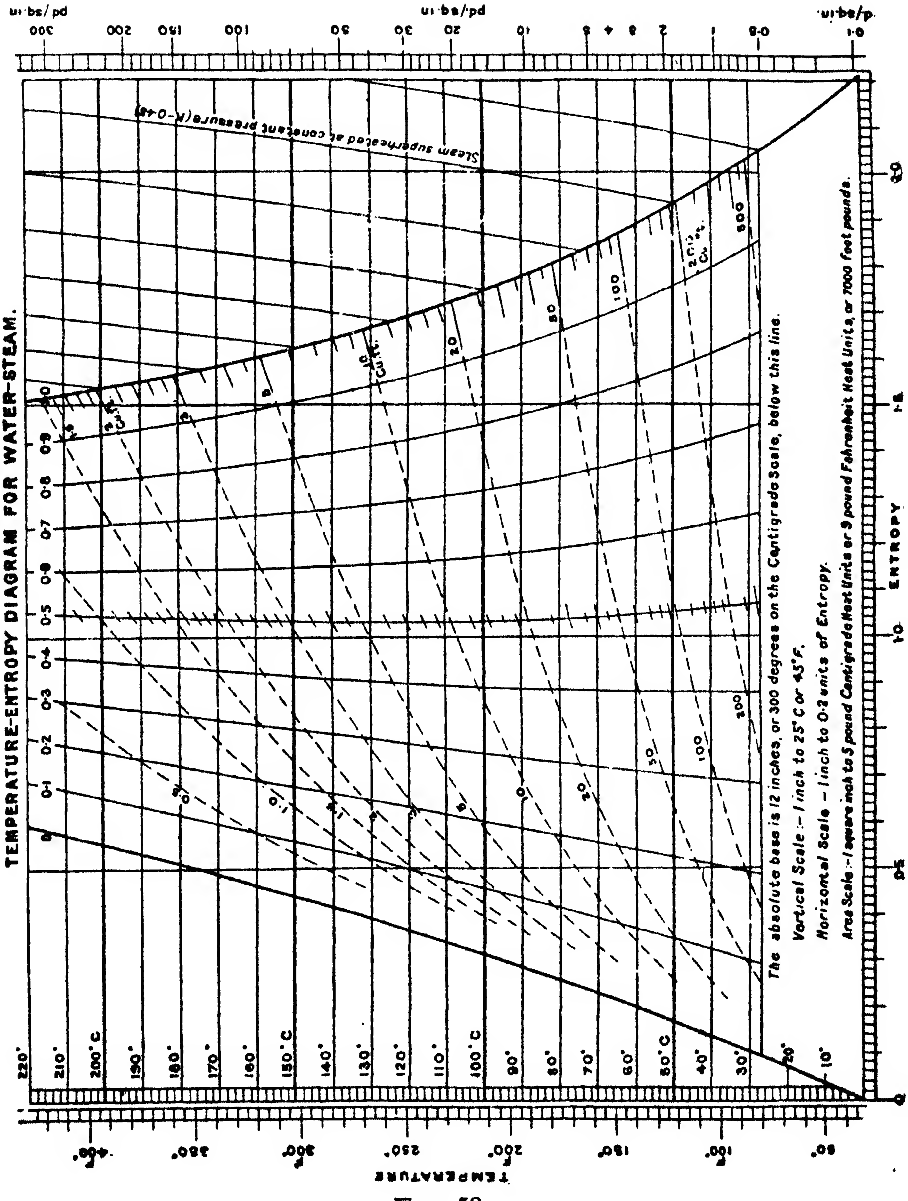

TEMPERATURE-ENTROPY DIAGRAM FOR WATER-STEAM.
Steam superheated at constant pressure (K=0·48)
ENTROPY
TEMPERATURE
The absolute base is 12 inches, or 300 degrees on the Centigrade Scale, below this line.
Vertical Scale:- 1 inch to 25° C or 45° F.
Horizontal Scale - 1 inch to 0·2 units of Entropy.
Area Scale:- 1 square inch to 5 pound Centigrade Heat Units or 9 pound Fahrenheit Heat Units, or 7000 foot pounds.

FIG. 58.

CHAPTER XIII

ENTROPY APPLICATIONS

95. Ideal Steam Cycle. This is known as the Rankine Cycle and is the standard of comparison for all steam engines. It is carried out as follows :

(1) One lb. of water is pumped into the cylinder or boiler at pressure P_1.

(2) Heat is supplied to raise the temperature of the water from feed temperature T_2 to boiling temperature T_1.

(3) Latent heat is now supplied at T_1 and evaporation takes place, shown by line B.D., fig. 59.

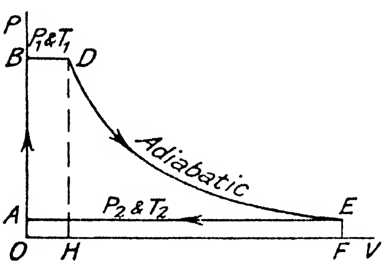


FIG. 59.

(4) Heat supply is discontinued and the steam expands adiabatically until the pressure falls to the back pressure. This is known as complete expansion to distinguish it from the expansion of the hypothetical diagram, which has a terminal pressure higher than the back pressure. This adiabatic expansion is represented by the line DE and at E the steam is wet.

(5) The wet steam is condensed at pressure P_2 until it is all water. The volume diminishes to that of 1 lb. of water and this stage is shown by EA.

The cycle is now complete.

The feed temperature in the ideal cycle is that of the condensing steam in stage (5).

The pressures P_1 and P_2 are the upper and lower limits respectively, and the corresponding temperatures are T_1 and T_2.

Comparing the Rankine Cycle with the Carnot Cycle for steam, it will be seen that the supply of heat to the water replaces the supply of energy to the wet steam by means of the adiabatic compression (see Art. 47) and that the isothermal compression is completed.

96. Work done per Pound of Steam from P – V Diagram.

Let V_1 = volume of 1 lb. of steam at P_1 shown at D.

V_2 = „ „ „ as shown at E.

n = index in the expansion law $PV^n = C$.

Work done in pumping and generation of steam is represented by area OBDH,

$$= P_1V_1 \text{ ft.-lb.}$$

Work done in adiabatic expansion is represented by area DEFH,

$$= \frac{P_1V_1 - P_2V_2}{n - 1} \text{ ft.-lb.}$$

Work done by piston in condensation and compression is represented by area EFOA $= P_2V_2$ ft.-lb.

Work done in cycle per lb. of steam

$$= P_1V_1 + \frac{P_1V_1 - P_2V_2}{n-1} - P_2V_2$$

$$= \frac{(n-1)P_1V_1 + P_1V_1 - P_2V_2 - (n-1)P_2V_2}{n-1}$$

$$= \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$= \frac{n}{n-1} \times P_1V_1\left(1 - \frac{P_2V_2}{P_1V_1}\right)$$

But $P_1V_1^n = P_2V_2^n$

$$\therefore \frac{V_2^n}{V_1^n} = \frac{P_1}{P_2} \qquad \therefore \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

$\therefore$ Work done per lb. of steam

$$= \frac{n}{n-1} \times P_1V_1\left(1 - \frac{P_2\,P_1^{\frac{1}{n}}}{P_1\,P_2^{\frac{1}{n}}}\right)$$

$$= \frac{n}{n-1} P_1 V_1 \left\{ 1 - \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \right\}$$

The value of n for adiabatic expansion of steam is 1·135, as given in Article 38.

If the above work is divided by V_1 we obtain the work done per cubic foot of steam used. Thus

Work done per cubic foot

$$= \frac{n}{n-1} P_1 \left\{ 1 - \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \right\}$$

$$= \frac{n}{n-1} \left(P_1 - P_1^{\frac{1}{n}} P_2^{\frac{n-1}{n}} \right) \text{ ft.-lb.}$$

97. The Rankine Cycle on the T − ϕ Diagram. The superheated steam cycle is dealt with here because it is now the most common. If dry steam is used the result is easily modified (see example worked in Chapter XIX p. 258).

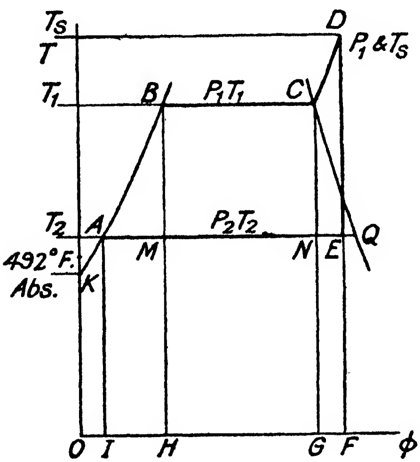


FIG. 60.

Starting with feed water at T_2 and remembering that the diagram is drawn for 1 lb. of steam we have the initial state point at A (fig. 60).

The liquid heat contained in this state is represented by the area OIAK, and in the ideal cycle the liquid heat is never less than this value.

The stages in the cycle are :

Stage (*i*). Liquid heat is supplied to raise the temperature from T_2 to T_1. The state point moves from A to B and the heat supplied is $h_1 - h_2$. The area ABHI represents this heat.

Stage (*ii*). Evaporation now takes place. Latent heat is supplied at the temperature T_1 and the state point moves from

B to C. This heat is represented by the area HBCG. To superheat the steam, further heat is added and is represented by the area CDFG.

The total energy at D is represented by the area of the diagram under the line KABCD, and the energy at A is represented by the area of the diagram under the short length of the water line KA ; both areas being measured down to the absolute temperature base.

Thus the area under ABCD, down to the absolute base, measures, to scale, the heat supplied, and is

$$(h_1 - h_2) + L_1 + C_p(T_s - T_1).$$

Stage (*iii*). The steam in condition D expands adiabatically in the engine cylinder to the pressure P_2. As no heat passes into or out of the steam during this process, the entropy is not changed, but the temperature falls to T_2, and hence, the process is represented by a vertical line on the diagram. The state point moves from D to E. The energy in the steam at E is represented by the area KAEFO and is less than the energy at D. The steam, too, is generally wet and has a dryness fraction $\frac{AE}{AQ}$, see Article 92.

Stage (*iv*). Condensation or Isothermal compression now occurs during which the steam is condensed at constant temperature T_2 and constant pressure P_2 until it is all liquid, the liquid also being at T_2. The heat removed from it is x_2L_2, where $x_2 = \frac{AE}{AQ}$.

The heat now remaining in the substance is that represented by the area OIAK, and is just the same as that contained at the commencement of the cycle.

$$\begin{aligned}\text{Work done in the cycle} &= \text{Heat supplied} - \text{Heat rejected}\\ &= [(h_1 - h_2) + L_1 + C_p(T_s - T_1)] - [x_2L_2]\\ &= [h_1 + L_1 + C_p(T_s - T_1)] - [h_2 + x_2L_2]\\ &= H_1' - H_2'\end{aligned}$$

where H_1' = total heat in steam in condition D.
and H_2' = " " " " E.

Note that neither H_1' nor H_2' can be taken directly from the tables. The steam table values must be corrected for superheat or wetness as the case may be.

If the steam is dry, having no superheat, when expansion begins, H_1' will be the total heat as given in the tables. H_2' will still need to have the wetness taken into account.

When the steam is wet at the beginning of the expansion, it will also be wet after expansion.

In this case, both H_1' and H_2' will require correction.

Total heat supplied in the cycle = heat at D — heat at A

$$= H_1' - h_2.$$

Thermal efficiency on Rankine cycle

$$= \frac{\text{work done in cycle}}{\text{heat supplied in cycle}}$$

$$= \frac{H_1' - H_2'}{H_1' - h_2}$$

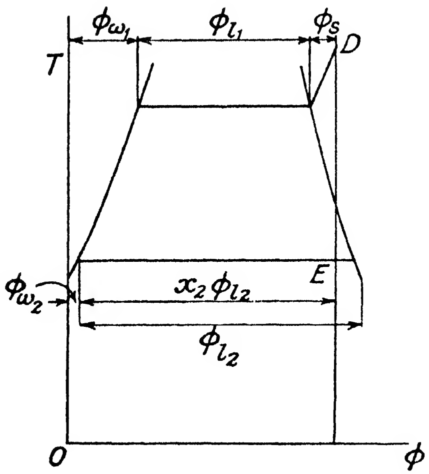


FIG. 61.

98. To Find the Dryness Fraction. The dryness fraction of the steam after adiabatic expansion (Stage iii, Article 97) may be found by drawing on an entropy chart the line DE, and finding the value, by measurement, of the ratio $\frac{AE}{AQ}$. In the absence of an entropy diagram the steam tables together with the adiabatic equation may be used. This equation is readily derived from the entropy chart.

Let ϕ_{l_1} = entropy of latent heat at T_1

ϕ_s = " " superheat at T_s

ϕ_{l_2} = " " latent heat at T_2

ϕ_{w_2} = " " water at beginning of the cycle.

Entropy at E = Entropy at D

$$x_2\phi_{l_2} + \phi_{w_2} = \phi_{w_1} + \phi_{l_1} + \phi_s$$

$$x_2 = \frac{\phi_{w_1} - \phi_{w_2} + \phi_{l_1} + \phi_s}{\phi_{l2}}$$

$$= \frac{\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + C_p \log_e \frac{T_s}{T_1}}{\frac{L_2}{T_2}}$$

$$= \frac{T_2}{L_2}\left[\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + C_p \log_e \frac{T_s}{T_1}\right]$$

When there is no superheat and no wetness, the last term disappears and

$$x_2 = \frac{T_2}{L_2}\left[\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1}\right]$$

When the steam at T_1 has dryness fraction $= x_1$

$$x_2 = \frac{T_2}{L_2}\left[\log_e \frac{T_1}{T_2} + \frac{x_1 L_1}{T_1}\right]$$

EXAMPLE. Find the work done per lb. of steam on the Rankine Cycle between pressure limits of 100 lb. per sq. in. and 14·7 lb. per sq. in. Take $n = 1{\cdot}135$. Also find the work done per cubic foot supplied.

Values from steam tables are:

p	t_s	V_s	L	h	H	ϕ_w	ϕ_s
100	327·8	4·43	889·7	298·5	1188·2	0·4742	1·6038
14·7	212	26·8	970·7	180	1150·7	0·3122	1·7574

$$V_2 = \left(\frac{P_1}{P_2}\right)^{\frac{1}{1\cdot 135}} V_1$$

$$= \left(\frac{100}{14\cdot 7}\right)^{0.881} \times 4{\cdot}43$$

$$= 6{\cdot}81^{0.881} \times 4{\cdot}43$$

$$= 5{\cdot}42 \times 4{\cdot}43 = 24{\cdot}01 \text{ cu. ft.}$$

$$\text{Work done} = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$= \frac{1 \cdot 135}{0 \cdot 135}(100 \times 144 \times 4 \cdot 43 - 14 \cdot 7 \times 144 \times 24 \cdot 01)$$
$$= 8 \cdot 42 \times 144(443 - 353)$$
$$= 8 \cdot 42 \times 144 \times 90 = 109{,}123 \text{ ft.-lb.}$$

$$\text{Work done per cu. ft. suppled} = \frac{109{,}123}{4 \cdot 43}$$
$$= 24{,}650 \text{ ft.-lb.}$$

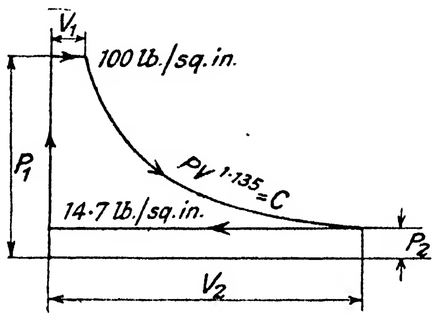


FIG. 62.

Alternative Method I.

Work done per cu. ft. of steam supplied

$$= \frac{n}{n-1}\left[P_1 - P_1^{\frac{1}{n}} P_2^{\frac{n-1}{n}}\right]$$
$$= 8 \cdot 42\left[14{,}400 - 14{,}400^{\frac{1}{1 \cdot 135}} \times 2{,}116^{\frac{0 \cdot 135}{1 \cdot 135}}\right]$$
$$= 8 \cdot 42(14{,}400 - 4{,}608 \times 2 \cdot 487)$$
$$= 8 \cdot 42\,(14{,}400 - 11{,}460)$$
$$= 8 \cdot 42 \times 2{,}940$$
$$= 24{,}750 \text{ ft.-lb.}$$

(This is close agreement for slide rule values.)

Work done per cubic foot of volume swept.

This is the final volume, since there is no clearance.

$$\text{Work done per cu. ft.} = \frac{109{,}123}{24 \cdot 01}$$
$$= 4{,}545 \text{ ft.-lb.}$$

Note that this result is numerically the same as the mean effective pressure expressed in lb. per square foot.

Alternative Method II.

By heat given up during cycle. Heat in 1 lb. of steam at C. (from tables) = 1188·2 B.Th.U.

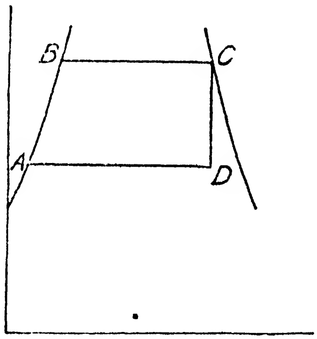


FIG. 63.

Final dryness fraction at D

$$x_D = \frac{T_2}{L_2}\left[\frac{x_1 L_1}{T_1} + \log_e \frac{T_1}{T_2}\right]$$

$$= \frac{672}{970{\cdot}7}\left[\frac{889{\cdot}7}{787{\cdot}8} + 2{\cdot}3026 \times \log_{10}\frac{787{\cdot}8}{672}\right]$$

$$= 0{\cdot}693\,(1{\cdot}129 + 2{\cdot}3026 \log 1{\cdot}17)$$

$$= 0{\cdot}693\,(1{\cdot}129 + 0{\cdot}157)$$

$$= 0{\cdot}693 \times 1{\cdot}286$$

$$= 0{\cdot}893.$$

Heat in 1 lb. of steam at D = $180 + 0{\cdot}893 \times 970{\cdot}7$
= $180 + 867$
= 1,047 B.Th.U.

Total work done = (1188·2 − 1,047) B.Th.U.
= 141·2
= (141·2 × 778) ft.-lb.
= 109,800 ft.-lb.

99. Rankine Cycle in terms of Temperature. An expression for the Rankine Cycle work and efficiency may be obtained by consideration of the $T - \phi$ diagram.

Consider heat supplied (See fig. 60).

$$\text{Area ABHI} = h_1 - h_2 = T_1 - T_2$$

$$\text{,, BCGH} = L_1 = \phi_{l_1} T_1$$

$$\text{,, CDFG} = C_p(T_s - T_1).$$

Consider heat rejected.

$$\text{Heat rejected} = x_2 L_2 = \text{area AEFI}$$

$$= T_2\left[\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + C_p \log_e \frac{T_s}{T_1}\right]$$

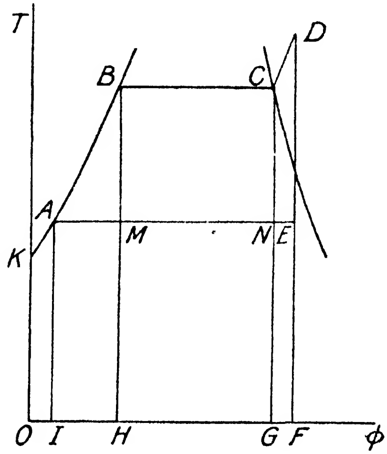


FIG. 64.

$\therefore$ Work done on the cycle

$$=T_1-T_2+L_1+C_p(T_s-T_1)-T_2\left(\log_e\frac{T_1}{T_2}+\frac{L_1}{T_1}+C_p\log_e\frac{T_s}{T_1}\right)$$

$$=T_1-T_2+\frac{T_1L_1}{T_1}-\frac{T_2L_1}{T_1}$$

$$+C_p(T_s-T_1)-T_2\left(\log_e\frac{T_1}{T_2}+C_p\log_e\frac{T_s}{T_1}\right)$$

$$=(T_1-T_2)\left(1+\frac{L_1}{T_1}\right)+C_p(T_s-T_1)-T_2\left(\log_e\frac{T_1}{T_2}+C_p\log_e\frac{T_s}{T_1}\right)$$

Dividing by the heat supplied gives the efficiency of the cycle.

This is a rather cumbersome expression and is only given here to enable the reader to recognize it as work done in heat units.

EXAMPLE. Find, from the T-ϕ diagram, the ideal work per lb. of dry steam and the Rankine efficiency between pressure limits of 120 lb. per sq. in. and 1 lb. per sq. in absolute. This example is intended to illustrate the use of the entropy diagram for finding work by areas.

Sketch and dimension a T-ϕ diagram as shown in fig. 65, taking the dimensions from a Board of Education chart.

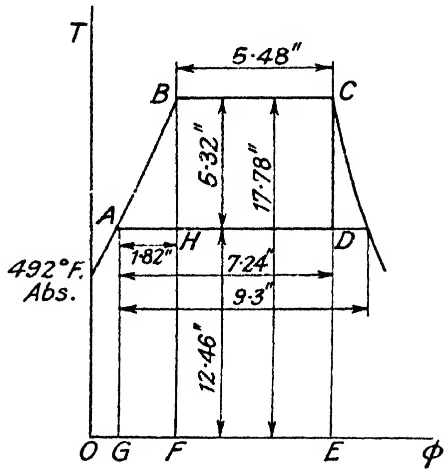


FIG. 65.

The work done is proportional to the area ABCD and in finding this area AB may be taken as a straight line.

For a more correct estimate a number of ordinates may be taken along AB.

$$\text{Area ABH} = \tfrac{1}{2} \times 1{\cdot}82 \times 5{\cdot}32 = 4{\cdot}84 \text{ sq. in.}$$
$$\text{,, BCDH} = 5{\cdot}32 \times 5{\cdot}48 = 29{\cdot}15 \text{ ,,}$$
$$\text{Total area} = 33{\cdot}99 \text{ ,,}$$

The scale of this diagram is such that 1 square inch represents 9 B.Th.U.

$$\therefore \text{Work done} = 33{\cdot}99 \times 9$$
$$= 305{\cdot}91 \text{ B.Th.U.}$$

Alternative Method.

Note that the work could be obtained very quickly by the use of the diagram and steam tables as follows.

$$\text{Final dryness at D} = \frac{7{\cdot}24}{9{\cdot}3} = 0{\cdot}779.$$

$$\text{Heat in steam at C (from 32° F.)} = 1{,}191{\cdot}4 \text{ B.Th.U.}$$
$$\text{,, ,, ,, D (,, ,,)} = 69{\cdot}7 + 0{\cdot}779 \times 1036{\cdot}1$$
$$= 69{\cdot}7 + 806{\cdot}9$$
$$= 876{\cdot}6 \text{ B.Th.U.}$$

$$\therefore \text{Ideal work} = 1{,}191{\cdot}4 - 876{\cdot}6$$
$$= 314{\cdot}8 \text{ B.Th.U.}$$

Rankine Efficiency.

Heat supplied is proportional to area ABCEG

$$= 33{\cdot}99 \text{ sq. in.} + (7{\cdot}24 \times 12{\cdot}46) \text{ sq. in.}$$
$$= 33{\cdot}99 + 90{\cdot}2$$
$$= 124{\cdot}19 \text{ sq. in.}$$
$$= 1{,}118 \text{ B.Th.U.}$$

$$\therefore \eta_R = \frac{305{\cdot}91}{1{,}118} = 0{\cdot}274 \text{ or } 27{\cdot}4 \text{ per cent.}$$

From the tables heat supplied $= 1{,}191{\cdot}4 - 69{\cdot}7 = 1{,}121{\cdot}7$ B.Th.U.

$$\therefore \eta_R = \frac{314{\cdot}8}{1{,}121{\cdot}7} = 0{\cdot}281 \text{ or } 28{\cdot}1 \text{ per cent.}$$

The steam tables used are more modern than the T-ϕ diagram. This accounts for the discrepancies.

100. Various cycles on P – V and T – ϕ diagrams compared.

Carnot Cycle (fig. 66) :

AB shows isothermal expansion.

BC ,, adiabatic ,,

CD ,, isothermal compression.

DA ,, adiabatic ,,

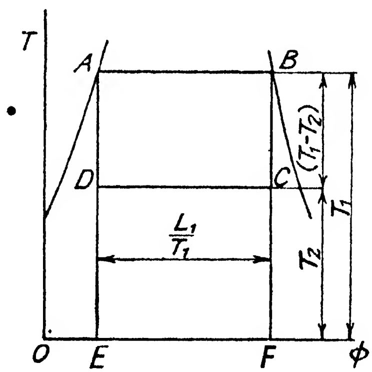


FIG. 66.

The diagram thus becomes a rectangle.

The work done is represented by area ABCD and heat supplied by ABFE.

$$\text{Efficiency} = \frac{\text{area ABCD}}{\text{area ABFE}} = \frac{T_1 - T_2}{T_1}$$

In actual steam engines it is impracticable to expand the steam completely as in the Rankine Cycle. Expansion is stopped at such a point as C and the steam exhausted. Assuming that the pressure falls to exhaust or back pressure whilst the piston is stationary, the P – V diagram becomes ABCDE, fig. 67. The line CD is evidently a constant volume line. Fig. 68 shows this modification on the T– ϕ diagram. CD is the constant volume line passing through C, the point where adiabatic expansion is stopped.

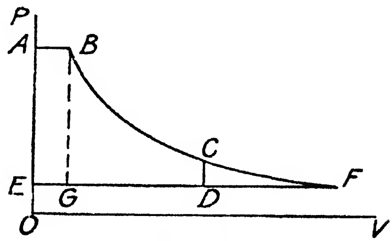


FIG. 67.

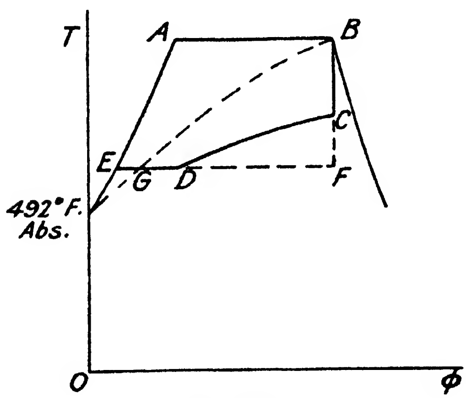


FIG. 68.

In both figures ABCDE represents the work and also CFD shows by how much this cycle falls short of the Rankine Cycle work.

When the steam is not expanded at all, the constant volume line follows the course BG. In this case, the area ABGE simply represents the external work done during formation of the steam.

101. Effect of Back Pressure. From fig. 69 it will be seen that, if the back pressure rises from P_3 to P_2 with P_1 remaining constant, a reduction in work will result.

If expansion was complete to pressure P_3, then increase in back pressure to P_2 causes loss at the widest part of the diagram. The reduction of work is proportional to the area shown shaded. This is the effect of decrease in the vacuum in a steam turbine. The steam turbine permits complete expansion. With a reciprocating engine, stopping its expansion at the terminal pressure P_T, the diagram follows a constant volume line from this pressure to the back pressure. The reduction in work, due to raising the back pressure, is then shown by the double shaded area.

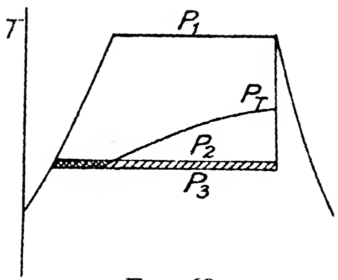


Fig. 69.

Fig. 69 shows (*a*) that the work done per lb. of steam in a reciprocating engine is less than in a turbine, (*b*) that a slight variation in condenser pressure is not very serious in the reciprocating engine, but becomes a very important factor in the efficient operation of a turbine.

Examples XIII

1. Steam is admitted to a cylinder at 160 lb. per sq. in. absolute and expanded down to the back pressure of 17 lb. per sq. in. at which it is exhausted. Find the work done per lb. of steam, assuming the law of expansion to be $PV^{1.135} = C$.
2. What would be the work done per lb. of steam if it were expanded to and exhausted at a back pressure of 2 lb. per sq. in.?
3. State clearly the processes of the Rankine Cycle.
4. An engine receives dry saturated steam at 180 lb. per sq. in. absolute, and rejects its exhaust to the condenser at 2 lb. per sq. in. It works on the Rankine Cycle. Calculate the work done per lb. of steam and the efficiency.
5. Steam at a pressure of 200 lb. per sq. in. absolute and a temperature of 500° F. passes through an engine working on the Rankine Cycle. The condenser pressure is 1·5 lb. per sq. in. Find the work done per lb. of steam and the efficiency on this cycle.
6. Steam at a pressure of 120 lb. per sq. in. absolute and 90 per cent. dry is expanded adiabatically to 16 lb. per sq. in. absolute. Find both from the entropy chart and by calculation the dryness fraction after expanding.
7. Steam at 100 lb. per sq. in. absolute with 90° F. of superheat is expanded adiabatically to 2 lb. per sq. in. absolute. Find the dryness fraction after expansion both by calculation and from the diagram. At what pressure will this steam be dry and saturated during the expansion?
8. On the temperature-entropy chart with which you are supplied mark with the letters A, B and C the points which show the following states of 1 lb. of water-steam:

A at 374° F. and 95 per cent. dry.

B at 100 lb. per sq. in. absolute and 60 per cent. dry.

C at 50 lb. per sq. in. absolute and superheated to 428° F.

If a pound of water-steam receives 5·4 British Thermal Units of heat, its temperature remaining constant at 302° F., what is its gain of entropy? Steam at 228° F. 95 per cent. dry expands adiabetically to 176° F What is the condition of the steam at this temperature? N.C.T.E.C.

9. Mark on the temperature-entropy chart a line showing the expansion with constant entropy of steam from 120 lb. per sq. in., 95 per cent. dry, to 5 lb. per sq. in. Measure off and write down the dryness and volume per lb. at the latter pressure. N.C.T.E.C.

CHAPTER XIV

VALVES AND VALVE GEAR

102. Eccentric and Eccentric-rod Motion. The admission of steam to an engine cylinder, and its release to exhaust at the proper moment, are of great importance in the efficient working of a steam engine. Many different types of valves have been used for this purpose, and a large number of these types still survive. The slide valve, with its modification the piston valve, is the most important arrangement used in reciprocating engines for controlling the movement of the steam, and only this type will be dealt with in this chapter. The knowledge gained in a study of the slide valve should make it possible for the student to understand, from a simple description, the working of almost any other type.

The slide valve is generally driven by a crank or eccentric, on the main shaft of the engine, through a connecting rod or eccentric rod. It is necessary to see what connection exists between the position of the crank or eccentric and the crosshead or valve.

Let the eccentricity or "throw" of the eccentric be OA (fig. 70), and let AB be the length of the eccentric rod. As OA rotates about O, B will move backwards and forwards along the direction OB through a distance equal to $2 \times OA = AA' = BB'$.

Take an intermediate position such as OC, then, if $CD = AB$, the distance moved by the valve attached to D will be BD. With centre D and radius DC draw the arc CM, then $AM = BD =$ the distance moved by the valve. Now the eccentric rod is usually very long compared with the throw of the eccentric, and the curvature in the arc CM is consequently small, so small, in fact, that it may generally be taken as coinciding with

8. Describe with the aid of sketches:

(*a*) Any type of reversing gear for a steam engine and explain its action; or

(*b*) The general outline of any form of water-tube boiler, showing in detail the jointing of a tube with the steam drum. N.C.T.E.C.

9. The travel of a slide valve is $3\frac{1}{2}$ in. and the lead $\frac{1}{4}$ in. At the crank end steam is cut off at 0·75 and released at 0·95 of the stroke. Find for the crank end (*a*) the angle of advance (*b*) the maximum opening for steam, (*c*) the exhaust lap. The connecting rod is 6 cranks long. N.C.T.E.C.

10. With the aid of sketches explain the terms steam lap, exhaust lap, and lead, as applied to a slide valve.

The travel of a slide valve is $3\frac{1}{2}$ in.; steam lap $\frac{3}{4}$ in.; exhaust lap $\frac{5}{8}$ in.; lead, $\frac{1}{8}$ in. Determine the angle of advance and the crank positions at admission, cut-off, release, and compression. N.C.T.E.C.

11. Explain, by reference to an outline diagram, the construction and action of Stephenson's link motion. N.C.T.E.C.

12. Explain carefully, with the aid of a sketch in each case, the effect on an indicator diagram of the following: (*a*) an increase of steam lap; (*b*) a decrease of exhaust lap; (*c*) an increase of lead. N.C.T.E.C.

13. A simple slide valve is required to give a maximum port opening of $1\frac{1}{2}$ in.; the angle of lead is $6°$ and the cut-off is to occur at $\frac{3}{4}$ stroke. Determine the travel of the valve necessary, also the outside lap and the angle of advance. If compression is to occur at 90 per cent. of the return stroke, what must be the inside lap?

(You may assume that the connecting rod is infinitely long.

U.E.I.

14. The slide valve of a vertical engine has a travel of 4 in. The lead at the crank end is 0·4 in., and the steam and exhaust laps at this end are 1 in. and 0·4 in. respectively.

The steam ports in the cylinder face are $1\frac{1}{2}$ in. wide in the direction of the travel. The length of the connecting-rod is four times that of the crank. Find from these data:

(*a*) The angle of advance.

(*b*) Where cut-off occurs on the up-stroke.

(*c*) Where release occurs on the up-stroke.

(*d*) During what fraction of a revolution exhaust is open.

(*e*) The maximum port-opening to steam at the crank end of the cylinder.

(*f*) The maximum port-opening to exhaust at the crank end of the cylinder.

And, by projection from your Valve Diagram, sketch the form of Indicator Diagram you would expect to get. U.E.I.

15. The travel of a simple slide valve is 4 in.; the steam ports in the cylinder face are each 1¼ in. wide in the direction of the travel, and the connecting rod is three times as long as the crank. Draw up, and complete, a table similar to the following:

Event.	*Head End*	*Crank End.*
Cut-off	0·7 stroke	0·7 stroke
Release	0·9 ,,	0·9 ,,
Outside lap		
Inside lap		
Maximum opening to steam		
Maximum opening to exhaust		
Lead	0·1 in.	
Angle of advance		

How have the equal cut-offs probably been effected? Is this satisfactory?
U.E.I.

The air may be warmed by bringing it in contact at some suitable point with the exhaust system or heat may be obtained from the circulating water.

EXAMPLES XVI

1. Why is the term " internal combustion " applied to certain types of heat engines?

Which of the following are internal combustion engines: gas engine, steam turbine, hot-air engine, petrol engine, reciprocating steam engine, Diesel engine, gas turbine?

2. Describe the operations of the four-stroke and two-stroke cycles as applied to a gas engine.

3. What are the relative advantages and disadvantages of the four-stroke and two-stroke cycles?

4. Explain carefully why it seems reasonable to compare the performance of a gas or petrol engine with the air standard cycle.

5. Why is a carburettor necessary to a petrol engine, whilst no similar device is used in conjunction with a gas engine?

6. The cylinder bore of a petrol engine with six cylinders is 70 mm. Calculate its H.P. according to the R.A.C. rating. What important details affect the actual horse power which the R.A.C. formula does not take into account?

7. Explain generally or, if you prefer it, in relation to some particular make, what is the function of a carburettor and how it carries out that function.

8. What are the most common methods of governing the speed of small gas engines? Illustrate your answer by means of neat sketches.

Sketch carefully a good representative indicator card for a gas engine of the four-stroke cycle type. Show on the diagram, by dotted lines, the variation you would expect:

(*a*) If the ignition were too early.

(*b*) If the mixture were too rich. U.E.I.

9. Sketch neatly and describe a modern type gas producer.

Explain how producer gas is made.

It is said that if the process merely oxidized carbon to carbon monoxide, and there were no heat losses, the efficiency of the producer would be at least 70 per cent. Explain this. U.E.I.

10. Describe the series of operations known as the Otto Cycle. Describe, with the aid of an outline sketch, one method of governing an internal combustion engine. N.C.T.E.C.

CHAPTER XVII

HEAVY OIL ENGINES

126. Introduction. Heavy oil engines constitute a very important class of internal combustion engines. These engines use residual oils, heavy mineral oil in the unrefined or crude state, and tar oils obtained from the distillation of solid fuels. On this account they are sometimes called crude oil engines. They differ from petrol and paraffin engines in the method of initiating combustion. In heavy oil engines the fuel is not induced during the suction stroke, but is forced in by means of a pump towards the end of the compression stroke. By this means, pre-ignition is made impossible and dangerous pressures are avoided. This permits of the compression in crude oil engines being much higher than in engines using the lighter fractions, and inducing the oil through a carburettor or vaporizer. Thus the thermal efficiency is higher.

127. Classification. An exact classification of oil engines is not a simple matter as the various types merge into each other. An approximate classification is as follows:

(1) Low compression engines, based on the Hornsby-Ackroyd engine.

(2) Medium compression engines, or cold-starting crude oil engines.

(3) High compression, or Diesel engines.

The high and medium compression engines have no ignition device of any kind, and are known as compression-ignition engines. The reason for this term is that the compression of the air on the compression stroke, which is approximately adiabatic, is carried on until the temperature of the air rises well above the ignition point of the fuel oil. Thus ignition devices, like sparking-plugs, magnetos, coils or lamps, are dispensed with.

128. Low Compression Engines. In the low compression type the heavy oil is heated in a vaporizer, by heat externally applied, before entering the actual cylinder. The oil may be injected either at the commencement of the suction stroke, or towards the end of the compression stroke. In the former case, the oil vapour and air are compressed into the vaporizer. The combined heating effect of the vaporizer and the compression raises the temperature to ignition point, and produces the combustion, no spark being required as with the light oil engine. The compression pressure in this type of engine is from 50 to 100 lb. per sq. in., depending upon the ignition point of the oil used. The combustion takes place approximately at constant volume.

When the oil is injected towards the end of the compression stroke, the compression pressure may be carried higher without risk of pre-ignition. If, however, the fuel valve leaks, dangerous pressures may be produced due to pre-ignition, but this applies to all pure compression-ignition engines, where the compression pressure is unassisted by hot surface, hot bulb, or lamp.

In the well-known Petter engines the vaporizer is modified to a hot bulb, which is an uncooled portion of the combustion chamber.

The hot-bulb engines are sometimes referred to as semi-Diesel engines ; an incorrect term since there is nothing similar about the engines except that they use the same type of fuel, and that air alone is compressed.

The vaporizer and hot-bulb types of engine follow the Ackroyd principles of low compression and constant volume heat supply. The Ackroyd engine made its appearance in 1889 and was intended to operate on the constant volume thermal cycle.

129. Medium Compression Engines. These engines may be called "sprayer" or airless injection engines, but there are also airless injection engines in the high compression class.

In the medium compression engine the oil is atomized by means of a fine nozzle sprayer, supplied with oil at high pressure from a pump of the ram type. With a sufficiently high hydraulic pressure, a fine bore orifice produces a high degree of atomization, and by suitably arranging the position of the nozzle in the combustion chamber, good penetration of the oil mist into the high temperature air, turbulence, and complete combustion are secured. Pressures in this type of engine at the end of compression are round about 350 to 450 lb. per sq. in. The oil pressure in the pump system is from 1,200 to 1,500 lb. per sq. in. Special arrangements of nozzles, pistons and passages to produce turbulence when ignition occurs, form the subjects of many patents, and constitute the difference between crude oil engines from different makers. These arrangements have considerable effect upon the running qualities and efficiency of the engine.

The combustion takes place in these engines partly at constant volume, and partly at constant pressure, with the result that a good mean effective pressure is secured without unduly high maximum pressure. Engines in which this feature is prominent are said to work on the principle of *Dual Combustion.*

130. High Compression Engines. Sometimes known as airless injection, or compression-ignition engines, these engines are of the sprayer type, but the compression pressure is higher—450 to 550 lb. per sq. in. The fuel supply is controlled to produce combustion approximately at constant pressure. No ignition device is fitted.

The general design of these engines is similar to that of the Diesel engine. This class of engine has made very rapid progress and appears certain to be the leading type in the future. The main difference between this engine and the Diesel, which is described more fully in Article 132, is that no air compressor is used. The fuel is injected by means of a ram pump accurately timed. It is a type of engine largely used for marine work and power stations, and, at present, is being largely used for mechanical road transport. It is thus becoming a competitor of the petrol engine. Its advantages as compared with the petrol engines are absence of fire risk and ignition system. In small units it is largely used for driving ship lighting sets.

131. Nomenclature. There is no uniformity or agreement as to the title by which heavy oil engines are to be known. Various terms are in use such as, semi-Diesel, solid injection, mechanical injection and airless injection.

The term semi-Diesel is applied indiscriminately to crude oil engines following the principles of Ackroyd Stuart's patents, and to high compression engines using pump injection. The latter approach more nearly to the true Diesel, both in mechanical construction, and thermo-dynamic cycle.

The term "solid" often gives a wrong impression, because the oil is not injected in a solid form nor in a solid stream. In fact, the very opposite is aimed at.

The terms "airless-injection, compression-ignition" or "airless-injection assisted compression-ignition" appear to comprehend the principal features of the main types, but these are cumbersome titles.

Another term coming into use is that of "cold-starting." A cold-starting oil engine is one which does not require a blow-lamp or heater to be applied to a portion of the combustion chamber before the engine can fire.

Cold-starting engines are usually of the medium-compression airless-injection class, but Messrs. Vickers-Petter manufacture an engine which is cold starting and has low compression. It operates on the two-stroke cycle.

132. The Diesel Engine. This engine is unique in many ways. It was the outcome of experiments by Dr. Rudolf Diesel, and was originally intended to use coal-dust in the cylinders and work on the Carnot Cycle. So great were the difficulties, that both these ideas were abandoned very early. The Carnot Cycle gives very low mean effective pressures, no matter what the maximum pressures may be, and consequently a massive engine would be needed for but low power. To overcome this difficulty, Diesel conceived the idea of complete adiabatic compression and combustion at constant pressure. The engine, thus constituted, quickly showed its possibilities and is now manufactured all over the world. The first engine made in this country was built in 1897 by Messrs. Mirlees Watson of Glasgow.

The essential features of the Diesel engine are :

(1) The oil fuel is ignited by compression alone.

(2) The fuel injection is by means of compressed air.

(2) The combustion is regulated so that there is no appreciable rise in pressure during injection.

(4) The combustion commences as soon as injection begins and continues so long as injection is carried on. The combustion begins immediately the oil reaches the compressed air in the working cylinder.

The compression pressure varies from 500 to 700 lb. per sq. in., and the blast air pressure from 800 to 1,000 lb. per sq. in. There is no explosive effect in the cycle, and this feature makes for quiet and smooth running.

The heat is liberated by combustion at constant pressure, and at exhaust opening it is removed at constant volume. Thus the thermal cycle has the essential features of constant pressure heat supply and constant volume heat rejection.

133. The Ideal Diesel Cycle. The cycle described is the four-stroke cycle and is carried out as follows :

(1) Air alone is drawn into the cylinder from A to B (fig. 86).

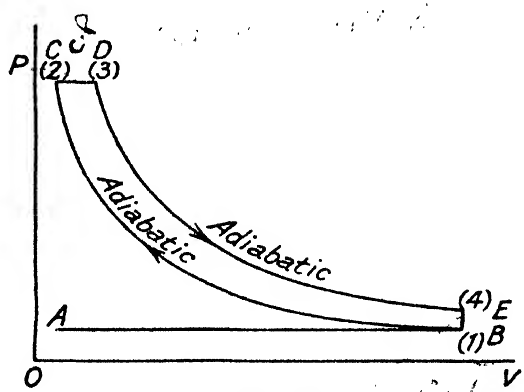


FIG. 86.

(2) Air alone is compressed adiabatically to high pressure shown by BC.

(3) At the end of compression, fuel oil is sprayed in by means of compressed air and ignites at once. It continues to burn at constant pressure until fuel supply is cut off at D.

(4) From D the gases resulting from combustion expand adiabatically to the end of the stroke, shown by DE.

(5) The pressure at E is now greater than that at B and

exhaust occurs E to B, which is equivalent to removing heat at constant volume.

The thermal cycle is complete at B and the mechanical cycle is completed in the fourth stroke B to A. The operations of the thermal cycle are thus :

(1) Adiabatic compression.
(2) Heat supply at constant pressure.
(3) Adiabatic expansion.
(4) Heat rejection at constant volume.

The efficiency of the ideal Diesel cycle may be obtained in exactly the same way as the Carnot Cycle efficiency, i.e. by applying the principle of the conservation of energy. Let the figures in fig. 86 become suffixes, and consider 1 lb. of ideal working substance.

Then heat received from C to D $= C_p(T_3 - T_2)$.
,, ,, rejected ,, E ,, B $= C_v(T_4 - T_1)$.

$\therefore$ Ideal work = Heat supplied — Heat rejected.

$\therefore$ Work done per lb. of substance in ideal cycle

$$= C_p(T_3 - T_2) - C_v(T_4 - T_1).$$

Hence the thermal efficiency $= \dfrac{\text{work done}}{\text{heat supplied}}$

$$= \frac{C_p(T_3 - T_2) - C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{C_v}{C_p}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{1}{\gamma}\left(\frac{T_4 - T_1}{T_3 - T_2}\right) \text{ since } \frac{C_p}{C_v} = \gamma.$$

It can be shown that this efficiency may be written in terms of the ratio of compression, and the cut-off ratio.

Let ρ = cut-off ratio $= \dfrac{V_3}{V_2}$.

γ = index of expansion and compression laws $PV^{\gamma} = C$.

r = ratio of compression $= \dfrac{V_1}{V_2} = \dfrac{V_4}{V_2}$.

Then $\eta_{\text{ideal}} = 1 - \frac{1}{r^{\gamma-1}} \cdot \frac{1}{\gamma}\left(\frac{\rho^{\gamma} - 1}{\rho - 1}\right).$

When this equation has $\gamma = 1{\cdot}41$ substituted, it gives the air standard efficiency of the Diesel cycle, which is less than that of the Otto Cycle.

134. Mechanical Considerations. The compression ratio in Diesel engines varies from 14 to 18. It has been mentioned that the blast air is from 800 to 1,000 lb. per sq. in. pressure, and this is provided by a two-stage or three-stage compressor. This compressor is often built on the engine frame, and serves to provide starting air as well as blast air. The blast air is that which injects and atomizes the oil. The atomization of the oi, is secured by smashing it through a series of perforated platesl separated from each other by distance pieces. The perforations are staggered, with the result that the fuel and blast-air form an oil fog, which thoroughly penetrates the high temperature compressed air with considerable turbulence. This brings about complete combustion. The amount of oil admitted is controlled by the fuel pump and by the time of opening of the needle fuel valve, and is always less than the amount which would require all the oxygen in the air charge.

The blast air, expanding through the needle fuel valve, is reduced in temperature by the expansion and therefore cools the compressed air. To maintain a self-ignition temperature in this type of engine, it is necessary to compress through a greater range than would be the case if this cooling did not occur. The air compressor adds to the cost and reduces the mechanical efficiency of the complete engine. It is the custom in this country to take the work expended on the air compressor into account, when calculating the mechanical efficiency of the engine. The air blast contributes a small amount of work to the piston during expansion. A fuel pump is used to pump the oil into the high pressure region occupied by the blast air. The oil rests by gravity upon the bottom of the injection valve chamber until it is blown in by the air. There is not hydraulic continuity in the oil supply system.

The arrangement of the Diesel engine is vertical and there are four valves in the cylinder head. This makes it complicated and considerable trouble arose in the early stages from cracked cylinder heads. With greater knowledge of temperature stresses, and flow and distribution of heat, these difficulties have been overcome. The four valves in the cylinder head are:

(1) The needle fuel valve.
(2) The air inlet, or suction valve.
(3) The exhaust valve.
(4) The air starting valve.

135. Valve Setting in Four-stroke and Two-stroke Engines. Admission of the oil starts from 1° to 2° before dead centre and continues for about 15° to 20° of crank angle movement in the working stroke. Typical valve setting diagrams for four-stroke and two-stroke Diesel engines are shown in figs. 87 and 88 respectively.

136. Cycles used in Heavy Oil Engines. Heavy-oil engines are built to work on the two-stroke or four-stroke cycle. They are also made for single-acting or double-acting operation. The Diesel engine is sometimes constructed to work double-acting, but more frequently is of the single-acting four-stroke type.

137. Methods of Starting. Small units of the low compression class may be started by hand as in the case of petrol engines. When this is done, oil of lower flash point than that of the normal fuel is often used until the cylinder is heated by a few cycles. With larger units in the low compression class, the starting is accomplished (1) by compressed air, (2) by blow-lamp, (3) by electric heater in the combustion chamber, (4) by special cartridge.

The blow-lamp is applied to the hot bulb or uncooled part of the cylinder, until the spot is hot enough to ignite the oil vapour. The engine is then " barred over " and when injection occurs the oil ignites and the engine commences to run.

This method is sometimes modified by using a lighter oil. Blow-lamps have often been a source of trouble, but improvements in design have obviated the difficulties.

The electric heater acts exactly as the blow-lamp, but the heat is supplied by electrical means.

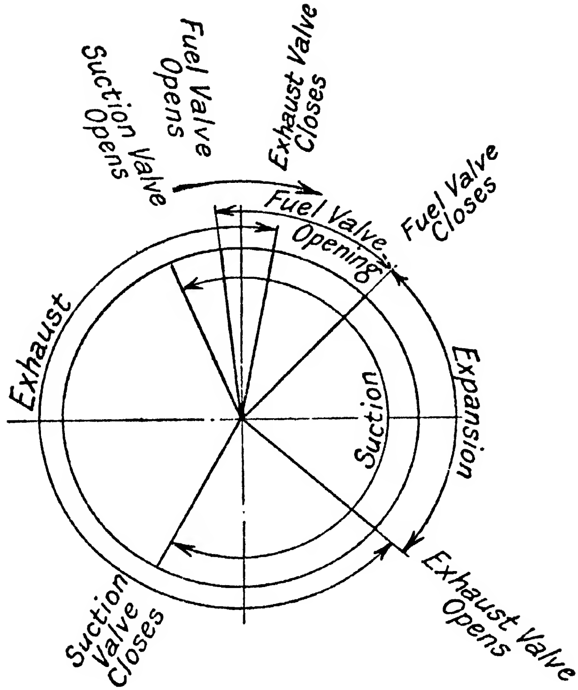


FIG. 87.—FOUR-STROKE DIAGRAM.

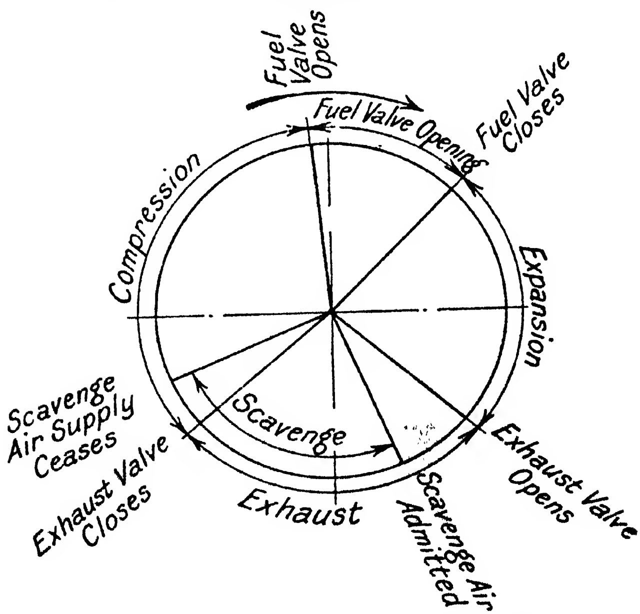


FIG. 88.—TWO-STROKE DIAGRAM.

The pressure is now very low, either atmospheric, or condenser pressure, and, if any water is left in the cylinder, evaporation becomes very vigorous. During exhaust stroke, evaporation will continue as long as moisture is available, with consequent cooling of the walls at a rapid rate. The amount of cooling during this period very largely decides the temperature of the walls which will meet the incoming steam at the beginning of the next stroke, and therefore, the amount of condensation which will occur. If there is not enough moisture at the beginning of the exhaust stroke for this evaporation to continue throughout it, then the cooling of the walls will cease when there is no more water to evaporate. If, however, there is ample moisture, this cooling may go on throughout the whole of the exhaust stroke, with consequent heavy condensation at the next admission of steam.

If, as we have just supposed, the steam supply is dry, it is very probable that there would not be sufficient moisture at the end of the expansion for the evaporation to continue throughout the whole of the exhaust stroke, unless there was a very big expansion ratio in the cylinder, when, of course, the amount of condensation during the expansion would be large.

If the steam supply is wet, all the above processes will operate in exactly the same way. The water in the wet steam will be additional to that produced in the way just described, and will be present at the beginning of the exhaust stroke. Evaporation on that stroke will thus be prolonged and may continue throughout the whole of it. Hence, in such a case, initial condensation becomes a maximum, and the amount of extra steam required to make up that condensation is a maximum. The steam used per stroke is increased without any more work being done in the cylinder, and hence the consumption rate is greater. In bad cases, this process builds up and may bring the engine to a standstill.

The reader should carefully note the following facts:

(1) If steam entering the cylinder is condensed and then re-evaporates on the exhaust stroke, the amount of heat drawn from the walls is less than that absorbed by the cylinder metal during condensation.

(2) The cooling of the cylinder walls is caused by the evaporation, during exhaust, of the water which entered as wetness in the steam, and that which condensed during expansion. This is because such water is present, without having given any heat to the cylinder metal. It is this second phenomenon which is the more serious.

151. Wall Temperature. The actual phenomena are very complex and the foregoing may be taken as a statement of the main principles involved.

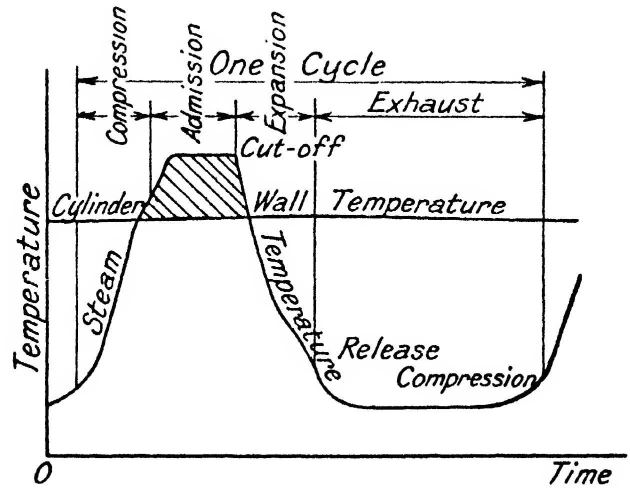


FIG. 95.

For any point on the wall surface a diagram of steam and wall temperatures may be drawn on a time or cycle base. The fluctuation of temperature of wall is very small compared with the fluctuation of steam temperature, and the study of the diagram will show that the period, during which the wall is hotter than the steam, is longer than that in which the steam is hotter than the wall. Thus, the water has ample time to evaporate. The shaded area is known as the condensation area, and, in suitable units, is a measure of the amount of steam which can condense on one square foot of surface at the chosen point. Such a diagram is due to the researches of Callendar and Nicholson in 1895, and is shown in fig. 95.

1·25 cu. ft. The engine takes in 10 cu. ft. of gas per min. having a calorific value of 275 C.H.U. per cu. ft. as used, and develops 41 I.H.P. Calculate the thermal efficiency and the efficiency ratio. Ratio of specific heats = 1·4.
N.C.T.E.C.

17. Explain why initial condensation occurs during admission to a reciprocating steam engine, and state what methods have been adopted to minimize this condensation. N.C.T.E.C.

18. Calculate the engine thermal efficiency in the case of a steam engine taking steam at 180 lb. per sq. in. absolute, 0·98 dry, if the temperature of the exhaust steam is 158° F. and the steam consumption is 15 lb. per horse power hour.

If the coal used has a calorific value of 14,400 B.Th.U. per lb. and the consumption is 1·8 lb. per horse power hour, determine the overall thermal efficiency of the plant. N.C.T.E.C.

19. The condensate from an engine is 4,410 lb. per hour when the I.H.P. is 210, and 6,520 lb. per hour when the I.H.P. is 362. Calculate the probable steam consumption when the I.H.P. is 280. State the law on which the solution depends. N.C.T.E.C.

20. In the constant volume cycle the temperatures at the beginning and end of the compression are 43° C. and 323° C. respectively. Determine the air standard efficiency and the compression ratio. Assume $\gamma = 1{\cdot}4$.

A petrol engine with the above compression ratio, develops 30 I.H.P. and consumes 1·63 gall. of fuel per hour. The specific gravity of the fuel is 0·78 and its calorific value is 10,500 C.H.U. per lb. Determine the indicated thermal efficiency and the relative efficiency of this engine. (1 gall. of water weighs 10 lb.) U.L.C.I.

21. Describe the sequence of events in the constant volume cycle. An engine having a compression ratio of 12·5, and working on this cycle, develops 22·7 horse power when the fuel consumption is 8·1 lb. per hour. The calorific value of the fuel is 10,600 C.H.U. per lb. Find the air standard efficiency and the relative efficiency. U.L.C.I.

22. The principal points on the sketch of the pressure-volume diagram for a Diesel cycle, commencing with the beginning of compression, are denoted by A, B, C and D. If temperatures at these points are $t_A = 100°$ C., $t_B = 697°$ C., $t_C = 2{,}152°$ C., $t_D = 987°$ C., what is the efficiency of the cycle, assuming the characteristic constant R = 94 ft.-lb. per lb. per ° C. and the index for adiabatic expansion $\gamma = 1{\cdot}38$? Determine the work done per lb. and the increase or decrease of the internal energy per lb. (*a*) from B to C, (*b*) from C to D. U.L.C.I.

23. Sketch the pressure volume diagrams for the constant volume, constant temperature and Diesel cycles. Mark the beginning of compression A and the principal points in consecutive order, B, C, etc.

For a certain constant volume cycle, the pressure and temperature at A are 14 lb. per sq. in. and 50° C. (122° F.)and the temperatures at B and C are 360° C. (680° F.) and 550° C. (1,022° F.). Assume $\gamma = 1{\cdot}4$ and find (*a*) the efficiency of the cycle, (*b*) the pressure at B, (*c*) the temperature at D.

(Author's hint:—For constant temperature cycle, see art. 47). U.L.C.I.

CHAPTER XIX

ENGINE TRIALS

156. Objects of Trials. Engine trials are carried out for the purpose of comparing actual results with theoretical or ideal performance. They are also carried out when the builders have entered into an agreement to guarantee a specified efficiency for the engine or turbine. The tests in this case are made to verify the guaranteed efficiency, in order that bonuses or penalties may be paid. The standard of attainment in these tests is often the consumption rate under specified pressure and vacuum conditions in the case of a steam plant, and with fuel of given calorific value in the case of gas, oil or petrol engines.

Tests have been the direct cause of, and incentive to, the improvement in engines and boilers throughout the period of their development. Many steam engines were built before tests were standardized, but the development of the steam turbine, petrol engine and oil engine has been largely due to the interest taken in the cost of engines, and particularly the cost of running them. This interest created a demand for authentic records of engine performance, which could only be satisfied by exhaustive trials carefully observed and calculated.

Heavy-oil engine development in this country has been greatly encouraged by the Royal Agricultural Society in offering prizes for the best and most efficient engines. Makers attach great value to these awards.

157. Results Sought. In a comprehensive trial of any heat engine, the following are the principal results sought.

(1) The power developed.
(2) The heat supplied per minute or per hour.
(3) The distribution of this heat. This appears in the heat balance or account.

(4) The efficiency of the engine compared to that of the ideal engine using the generally-accepted standard cycle.

These items may now be considered in detail and the equipment necessary will be mentioned.

158. Power Developed. The power may be measured (*a*) by indicator, or (*b*) by brake.

The indicator is used to take observations which are necessary to calculate the cylinder power. It can only be used for reciprocating engines. In steam turbines the horse-power, corresponding to the I.H.P. of the reciprocating engine, can only be calculated by working back from the measured B.H.P., using the results of tests on bearing losses, disc and drum losses, windage and friction.

The brake measures the work available for use external to the engine itself, and gives the useful power.

159. Heat supply. The heat supply to the engine is obtained in two stages; (*a*) the measurement of the amount of substance going to the engine in a given time, say the duration of the test; (*b*) the measurement of the heat carried by each unit of substance. This latter figure is obtained by calorimetry. Where oil, petrol or gas is being used, the calorific value is found during the period of the trial. In steam engine or turbine trials the calorimetry results are recorded in the steam tables and diagrams; hence, steam-quality tests only need be taken during the trial.

The weight of steam used during the trial is usually measured by condensing it, and measuring the condensate. This assumes that there is no leakage of steam at drains, glands, etc., between engine stop-valve and condenser. When precautions are taken to prevent or measure such leakage, the condensing method is the most reliable.

Another method now in use for measuring daily steam supply is the installation of a steam meter in the supply steam pipe. In many power stations, readings are taken half-hourly or hourly throughout the whole period of twenty-four hours.

In gas-engine trials the gas is measured by meter or by means of gas-holders of known capacity. The gas is then supplied at constant pressure and its amount can be accurately determined.

Two gas-holders are normally used, one being filled, while the other is being drawn upon by the engine. One only may be used if all the conditions are kept constant, and the gas rate is measured for short intervals of time. The engine runs from the main, except during the period of gas-flow measurement.

160. Heat Distribution. The determination of the distribution of the heat requires the measurement of water and gas quantities, and their temperatures.

In testing a gas engine an exhaust gas calorimeter may be fitted. In this calorimeter, the heat is absorbed by water which rises in temperature, and whose quantity may be measured.

Heat passing through the cylinder walls to the jacket is measured by the quantity of jacket water heated and the amount of temperature rise.

Thus in both these cases—

Heat removed per minute = weight of water per minute × temperature rise. The heat removed in the steam-engine condenser is measured in exactly the same way.

161. Efficiency Ratio. The efficiency ratio is obtained by calculating the indicated thermal efficiency and dividing it by the ideal efficiency for the type of engine under consideration.

162. Observations Made. The observations to be recorded in a thermal test of an engine are as follows:

For the indicated horse power—

(1) Scale of spring on indicator.

(2) Diameter of engine piston.

(3) Diameter of engine piston rod and tail rod, if any.

(4) Length of stroke.

(5) Whether single-acting or double-acting.

(6) Nature of cycle—two-stroke or four-stroke.

(7) Revolutions per minute, from which working strokes per minute may be calculated. (In the internal-combustion engine, an explosion counter may be needed.)

For the calculation for the I.H.P.—

Work done per minute = mean effective force × distance travelled by piston per minute whilst under action of force.

= M.E.P. × effective piston area × stroke length × working strokes per minute.

Effective area of one side of piston = area of piston − area of rod.

Thus, the work done per minute on one side of the piston may be calculated. If the engine is double-acting, the work done on the other side is added. It is not correct to multiply the work done on one side of the piston by two, in order to find the whole work per revolution, because the M.E.P.'s from each side often differ considerably, and, if the engine has no tail rod, the effective areas are different.

The algebraic treatment for the steam engine is now given.

Let D = diameter of cylinder in inches.

d = „ „ piston-rod in inches.

d_t = „ „ tail rod „ „

P_F = M.E.P. from piston-rod side of piston.

P_B = „ „ tail-rod „ „ „

L = length of stroke in feet.

N = revolutions per minute.

Then I.H.P.—

$$= \frac{P_F\left(\frac{\pi D^2}{4} - \frac{\pi d^2}{4}\right)L \times N + P_B\left(\frac{\pi D^2}{4} - \frac{\pi d_t^2}{4}\right)L \times N}{33{,}000}.$$

With a single-acting engine, with no tail rod and pressure on back of piston,

$$\text{I.H.P.} = \frac{P_B\left(\frac{\pi D^2}{4} - 0\right)L \times N}{33{,}000} = \frac{P_B \times \frac{\pi D^2}{4} \times L \times N}{33{,}000}.$$

With the *gas engine*, single-acting, using above notation.

$$\text{I.H.P.} = \frac{P_B \times \frac{\pi D^2}{4} \times L \times \frac{N}{2}}{33{,}000}$$ (for four-stroke, with no cycles cut out due to hit-and-miss governing).

$$\text{or I.H.P.} = \frac{P_B \times \frac{\pi D^2}{4} \times L \times N}{33{,}000}$$ (for two-stoke, with no cycles cut out).

The calculation for the double-acting gas engine is the same as for the steam engine. In all gas engine equations, where cycles are cut out, E must be substituted for N or $\frac{N}{2}$, where E = actual number of charges ignited per minute.

163. Brake Horse Power Observations. It is necessary to measure the (1) speed, (2) torque.

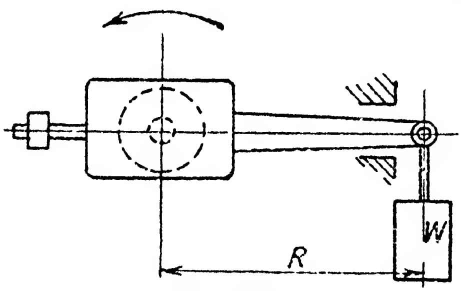


FIG. 100.

Brake horse power calculations are the same whether some lever type of brake (as the Prony or Froude), or rope type (Hirn) is used.

In both cases, work done per revolution
= resisting torque × angular movement in radians.

The torque in the lever type of brake (fig. 100),
= effective or equivalent weight × radius.
$= W_E R$.

W_E is not the weight actually on the lever, but allows for the weight of lever itself and its attachments.

In the rope brake shown in fig. 101, the radius is measured to the centre of the rope. The effective weight lifted is the difference between actual dead weight W and the reading of the counterbalance S.

$$\therefore W_E = W - S.$$

Assuming a rope brake to be in use,
Let W = weight lifted in lb.
S = counterweight in lb.
D = diameter of brake wheel in feet.
d = ,, ,, rope ,, ,,
R = effective radius in feet.

The $R = \frac{D + d}{2}$.

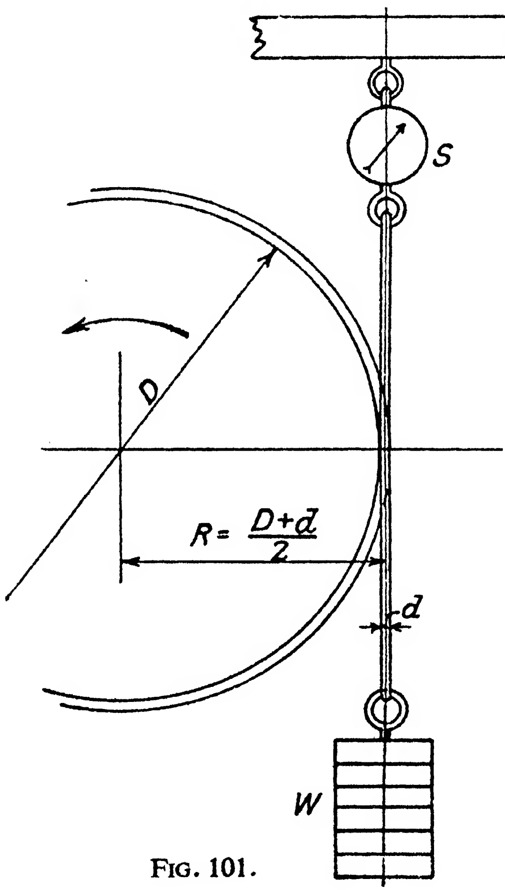


FIG. 101.

Work done per revolution = torque × angle turned in radians.

$$= (W - S)R \times 2\pi \text{ ft.-lb.}$$
$$= \pi(W - S)(D + d) \text{ ft.-lb.}$$
$$= 2\pi R(W - S) \text{ ft.-lb.}$$

Let N = no. of revs. per minute.

W_E = effective load = $W - S$.

Work done per min. = $2\pi RNW_E$

and Brake horse power $= \dfrac{2\pi RNW_E}{33{,}000}$.

(If T = the torque in lb. ft. and ω = the angular velocity in radians, per sec., then B.H.P. $= \dfrac{T\omega}{550}$, and the student is advised to become familiar with this form.)

164. Thermal Efficiency. The observations to be made for the determination of thermal efficiency are:

(*a*) For the steam engine:

(1) Weight of steam supplied.

(2) State of steam supply.

(3) Indicator diagrams.

(*b*) For the internal combustion engine:

(1) Weight or volume of fuel supply.

(2) Calorific value of fuel.

(3) Indicator diagrams.

It is necessary to measure the substance going to the engine in a given period of time when the working conditions are steady.

The Indicator. The chief instrument in use is the indicator. It is an instrument of precision and must be handled with care and intelligence to secure reasonably accurate indications.

The precautions to be taken are tabulated below:

(1) The driving cord or wire must move freely over guide pulleys.

(2) The cord must be of the right length to ensure that the drum does not strike its stop at the ends of its travel. Cord too long will fall slack, and cord too short will stretch or break.

(3) The correct spring must be chosen for the range of pressure required.

(4) The piston must not strike the stop at either the top or bottom of its stroke.

(5) All parts must be assembled so that there is no back-lash.

(6) The pencil must be applied to the paper by means of the small handle usually provided for the purpose. On no account must the fingers touch the pencil mechanism.

(7) The pencil must be applied to the paper as lightly as possible consistent with obtaining a line on the paper. This line may not be visible until the paper is removed. It is a mistake to allow the pencil to trace the diagram time after time.

(8) The indicator cocks must be opened fully. It is a common error to open them partially with the result that the cards are of no use.

(9) The atmosphere line must be drawn before opening the indicator cock.

The motion of the indicator piston is multiplied by a link-work, which draws a vertical straight line when the drum is not in motion. There are several of these straight line motions which the reader will study in the higher classes in Theory of Machines. The scale of the spring, which is stamped upon it, takes account of this multiplication of motion and also of the actual area of the indicator piston. For example, if the spring is actually compressed $\frac{1}{4}$ in. and the pencil moves $\frac{3}{4}$ in., the scale of the spring being 80 lb. per sq in. per inch, the pressure increase is $\frac{3}{4} \times 80 = 60$ lb. per sq. in. Thus each indicator must have its own springs to correspond with pencil movement multiplication and indicator piston area.

The reader should take the opportunity of examining the indicator and indicator rig, when he carries out his laboratory work.

165. Trials. Two trials will now be worked out in full. The first relates to a National Gas Engine and the second to a Cross Compound Condensing Steam Engine. All the observations were made by Third Year Senior Engineering Students in the Wigan and District Mining and Technical College.

166. Gas Engine Trial. The trial was carried out on the above engine operating on the four-stroke cycle and governed on the "hit and miss" principle. The brake was of the rope type, and the counterbalance force was measured by means of a spring balance.

Harding's counters were used, one to measure revolutions and another to record explosions.

The jacket cooling water was measured by means of a calibrated orifice plate in a horizontal plane. Centigrade thermometers were fitted in the supply pipe and in the discharge pipe of the water jacket. The gas supply was measured by means of a gas meter. The reading of the meter was not corrected for temperature or for pressure, since manometers and thermometers were not fitted to the gas supply pipe. When these data are observed, the correction can easily be made by applying the equations of Article 27.

If a determination of the calorific value of the gas is made, the same correction should be applied to the calorific value test result, so that the heat supply to the engine can be accurately obtained.

Three different loads were taken, all other conditions remaining constant. The loads on the brake were as follows:

Series A, 40 lb. weight.

Series B, 69 „ „

Series C, 97 „ „

The trial for series A was carried out for 15 minutes after the engine had run under this load for sufficient time to allow all parts to reach normal working temperatures, and to obtain uniform readings at all observation points.

Series B and C were run for 10 minutes after allowing a suitable interval for the engine to adjust itself to the change in load.

Duration of test = 10 minutes.
Indicator spring scale = 120 lb. per sq. in. per inch.
Engine dimensions as before.

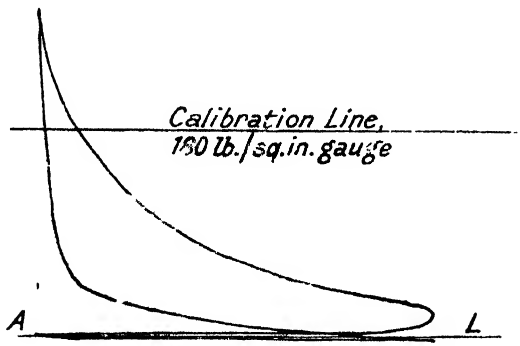


FIG. 103.

The chief results collected are:

Indicated horse power	= 6·33
Brake horse power	= 5·21
Consumption per I.H.P. per hour	= 37·7 cu. ft.
,, ,, B.H.P. ,, ,,	= 45·8 ,, ,,
Mechanical efficiency	= 82·3 per cent.
Indicated Thermal efficiency	= 15·71 per cent.
Brake Thermal efficiency	= 12·87 per cent.
Ideal efficiency	= 41·7 per cent.
Efficiency ratio	= 37·6 per cent.

Heat Balance.

Heat supplied per min.	=950 C.H.U.	100%
Heat at Brake = 122·1 ,, by Friction = 27·1 Indicated heat . . . . . . Jacket heat . . . . Exhaust and other losses . . .	 149·2 267·5 533·3	 15·71% 28·2% 56·09%
	950	100%

SERIES C.

Table VIII gives the observations logged in the engine room.

TABLE VIII

Call No.	Explosions Counter Readings.	Revolution Counter Readings.	Weight on Brake.	Spring Balance Reading.	Weight of Cooling Water.	Inlet Temperature.	Outlet Temperature.	Quantity of Gas/5 min.
1	557,070	149,444	97 lb.	13 lb.	13·6 lb./min.	8° C.	32° C.	0 cu. ft.
2	557,745	150,846	97 ,,	13 ,,	13·6 ,,	8° C.	34° C.	24·5 ,,
3	558,415	152,252	97 ,,	14 ,,	13·6 ,,	8° C.	35° C.	25·0 ,,

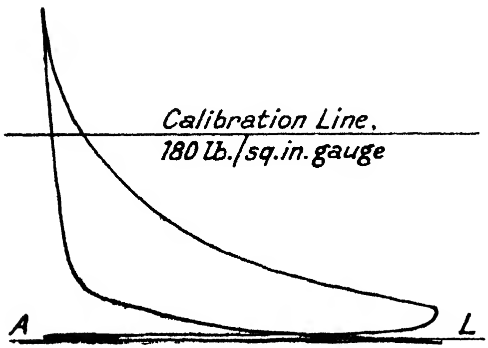


FIG. 104.

Duration of test = 10 minutes.
Indicator spring scale = 120 lb. per sq. in. per inch.
Engine dimensions as before.

The chief results collected are:

Indicated horse power	= 8·49
Brake horse power	= 6·9
Consumption per I.H.P. per hour	= 35 cu. ft.
,, ,, B.H.P. ,, ,,	= 43·1 cu. ft.
Mechanical efficiency	= 81·3 per cent.
Indicated Thermal efficiency	= 16·9 per cent.
Brake Thermal efficiency	= 13·72 per cent.
Ideal efficiency	= 41·7 per cent.
Efficiency ratio	= 40·8 per cent.

The student should compare the results of the three tests with one another, when he may draw useful conclusions concerning the theory in the earlier chapters.

Heat Balance.

Heat supplied per min.	1,182 C.H.U.	100%
Heat at Brake = 162·5 ,, by Friction = 37·5 Indicated heat . . . . . . Jacket heat . : . . . . . Exhaust and other losses . . .	 200 349·5 632·5	 16·9% 29·56% 53·54%
	1,182	100%

167. Steam Engine Trial. The engine was run as a cross compound condensing engine at constant speed and load.

The observations were:

(1) Indicator diagrams.

(2) Engine revolutions.

(3) Indicator spring scales.

(4) Brake weight and counterbalance.

(5) Stop valve pressure.

(6) Vacuum pressure.

(7) Weight of steam supplied (taken as weight of condensate).

(8) Steam and water temperatures.

Engine Data.

High Pressure cylinder diameter	= 7 in.	Stroke = 20 in.
,, ,, piston rod ,,	= 1¾ in.	No tail rod.
Low Pressure cylinder ,,	= 13 in.	Stroke = 20 in.
,, ,, piston rod ,,	= 1¾ in.	No tail rod.
Brake wheel ,,	= 8 ft.	
,, rope ,,	= 1 in.	

Calls were made at five minute intervals.

Duration of test, 25 minutes.

The observations logged in the engine room are recorded in Table IX.

A number of the engine thermometers are Centigrade and others Fahrenheit. The actual observations are recorded.

TABLE IX

Call No.	Engine Revolution Counter.	Brake Weight. Lb.	Spring Balance. Lb.	Stop Valve Pressure. Lb./Sq. In. Guage.	Compound Gauge. Lb./Sq. In.	Vacuum Gauge. In. Mercury.	Condensate Temperature. °C.	Weight of Condensate. Lb.	Quantity of Cooling Water. Gallons.	Inlet Temperature of Cooling Water. °F.	Outlet Temperature of Cooling Water. °F.	Exhaust Temperature. °F.
1	52,968	376	27·5	115	13	22·5	50	—	—	48	80	140
2	53,564	376	27·5	120	12	24·5	51	—	180	49	82	136
3	54,191	376	27·5	112	12	24·9	51·5	—	185	49	83	138
4	54,819	376	27·5	112	12	23·0	52	—	175	50	85	136
5	55,445	376	27·5	111	11·5	23·8	52	—	185	50	85	134
6	56,073	376	27·5	111	11·0	24·5	52	458	185	50	85	134

Calibrated indicator diagrams from the high pressure and low pressure cylinders are shown in figs. 105 and 106 respectively.

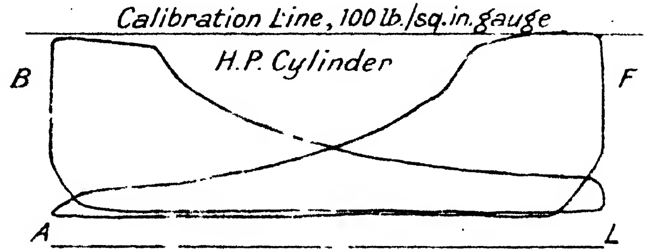


FIG. 105.

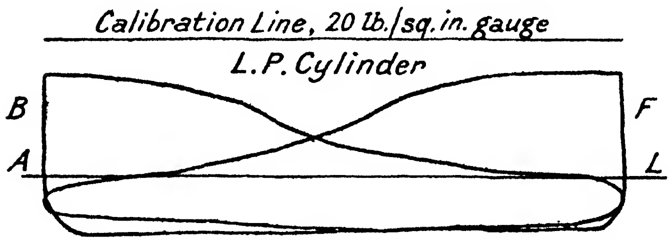


FIG. 106.

Indicator springs.
High pressure = 64 lb. per sq. in. per in.
Low ,, = 20 ,, ,, ,, ,, ,, ,,

Reduction of Table of Observations on Steam Engine Trial.

$$\text{Mean speed} = \frac{56{,}073 - 52{,}968}{25} = 124 \text{ revs. per min.}$$

Effective brake load = 376 − 27·5 = 348·5 lb.

Average stop-valve pressure = 113·5 lb. per sq. in gauge
= (113·5 + 14·7) lb. per sq. in. abs.
= 128·2 ,, ,, ,, ,,

Average vacuum = 23·86 in. mercury

∴ Pressure in condenser = 30 − 23·86 at 30 in. barometer
= 6·14 in. mercury

$$= \frac{6{\cdot}14 \times 14{\cdot}7}{30} \text{ lb. per sq. in. abs.}$$

= 3·01 ,, ,, ,, ,,

Corresponding temperature = 60·9° C. = 141·6° F.
Mean temperature of condensate = 51·41° C. = 124·5° F.

Weight of condensate per min. $= \dfrac{458}{25}$

$= 18{\cdot}32$ lb.

„ „ „ „ hour $= 1{,}100$ lb.

Total quantity of condensing water in 25 min.

$= 910$ gallons $= 9{,}100$ lb.

$\therefore$ Weight of condensing water per min. $= \dfrac{9{,}100}{25} = 364$ lb.

Mean inlet temperature of condensing water

$= 49{\cdot}33°$ F.

Mean outlet temperature of condensing water

$= 88{\cdot}33°$ F.

Temperature rise of cooling water $= 83{\cdot}33 - 49{\cdot}33 = 34°$ F.

* Mean exhaust temperature at inlet to condenser

$= 136{\cdot}33°$ F.

I.H.P. Calculations.

(1) High Pressure Cylinder.

Mean height of indicator diagram, back of piston $= 0{\cdot}618$ in.

„ „ „ „ „ front „ „ $= 0{\cdot}63$ in.

Average mean height $= 0{\cdot}624$ in.

„ „ effective pressure $= 0{\cdot}624 \times 64$

$= 39{\cdot}9$ lb. per sq. in.

Area of piston (7 in. dia.) $= 0{\cdot}7854 \times 7 \times 7 = 38{\cdot}5$ sq. in.

„ „ „ rod ($1\frac{3}{4}$ in. dia.) $= 0{\cdot}7854 \times 1{\cdot}75 \times 1{\cdot}75$

$= 2{\cdot}406$ sq. in.

Effective area, back of piston $= 38{\cdot}5$ sq. in.

„ „ front „ $= 36{\cdot}094$ sq. in.

Average effective area $= 37{\cdot}297$ sq. in.

Mean force on piston $= 37{\cdot}297 \times 39{\cdot}9 = 1{,}486$ lb.

Distance moved by piston per min. $= 2 \times \dfrac{20}{12} \times 124 = 412{\cdot}92$ ft.

$\therefore$ Work done per minute in this cylinder $= 1{,}486 \times 412{\cdot}92$

$= 615{,}000$ ft.-lb.

$\therefore$ Indicated horse power $= \dfrac{615{,}000}{33{,}000} = 18{\cdot}62.$

---

* This is less than the value corresponding to vacuum, pressure probably due to observational or instrumental errors, or undercooling effect in the steam.

(2) Low Pressure Cylinder.

Mean height of diagram, back of piston = 0·716 in.
" " " " front " = 0·724 in.
Average mean height = 0·72 in.
Average mean effective pressure = 0·72 × 20
= 14·4 lb. per sq. in.
Area of piston (13 in. dia.) = 0·7854 × 13 × 13 = 132·8 sq. in.
Area of piston rod = 2·406 sq. in.
Effective area (front) = 132·8 − 2·406 = 130·394 sq. in.
Average effective area = 131·597 sq. in.
Mean force on piston = 131·597 × 14·4 = 1,895 lb.
Distance moved by piston per minute = 412·92 feet.
∴ Work done per min. in low pressure cylr. = 1,895 × 412·92
= 784,000 ft.-lb.

∴ Indicated horse power low pressure cylr. $= \dfrac{784{,}000}{33{,}000}$
= 23·75

Total Indicated Horse Power
= I.H.P. (H.P. cyl.) + I.H.P. (L.P. cyl.)
= 18·62 + 23·75 = 42·37.

B.H.P. Calculations.

Effective radius of brake wheel = 4 ft. 0½ in. = 4·04 ft.
Effective brake load = 348·5 lb.
Work done against brake per rev. = 2 × π × 4·04 × 348·5
= 8,870 ft.-lb.
Work done against brake per min. = 8,870 × 124

∴ Brake horse power $= \dfrac{8{,}870 \times 124}{33{,}000}$
= 33·3 horse power.

Mechanical efficiency $= \dfrac{33{\cdot}3}{42{\cdot}37} = 78{\cdot}6$ per cent.

Steam consumption rate per I.H.P. per hour

$$= \frac{\text{Wt. of steam used per hour}}{\text{I.H.P.}}$$

$$= \frac{1{,}100}{42{\cdot}37} = 25{\cdot}9 \text{ lb.}$$

Steam consumption rate per B.H.P. per hour $= \dfrac{1{,}100}{33{\cdot}3} = 33{\cdot}08$ lb.

The consumption rates may also be obtained by dividing 458 lb. by the number of horse power hours worked during test.

Indicated horse power hours worked $= 42{\cdot}37 \times \dfrac{25}{60} = 17{\cdot}63.$

Brake horse power hours worked $= 33 \cdot 3 \times \frac{25}{60} = 13 \cdot 88.$

Steam consumption rate per I.H.P. per hour $= \frac{458}{17 \cdot 63}$

$= 25 \cdot 9$ lb.

" " " " B.H.P. " " $= \frac{458}{13 \cdot 88}$

$= 33 \cdot 08$ lb.

Indicated thermal efficiency

$$= \frac{\text{Heat equivalent of work done on piston}}{\text{Heat supplied to engine}}$$

Total heat at 128·2 lb. per sq. in. abs. and dry	= 1,192·5	B.Th.U.
Liquid heat at 3·01 " " " "	= 109·5	"
Heat supplied per lb. of steam	= 1,083·0	"

$\therefore$ Indicated thermal efficiency $= \frac{60 \times 33{,}000}{778 \times 25 \cdot 9 \times 1{,}083}$

$= \frac{2{,}545}{25 \cdot 9 \times 1{,}083}$

$= 0 \cdot 0904$ or $9 \cdot 04$ per cent.

Brake thermal efficiency $= \frac{2{,}545}{33 \cdot 08 \times 1{,}083}$

$= 0 \cdot 071 = 7 \cdot 1$ per cent.

Rankine Efficiency.

Using the equation of Article 99 to find the Rankine work and modifying it to allow for the fact that no superheater was fitted to the engine, we have Rankine work for dry steam per lb.

$$= (T_1 - T_2)\left(1 + \frac{L_1}{T_1}\right) - T_2 \log_e \frac{T_1}{T_2}$$

$$= (806 \cdot 2 - 601 \cdot 6)\left(1 + \frac{875}{806 \cdot 2}\right) - 601 \cdot 6 \times 2 \cdot 3026 \times \log 1 \cdot 327$$

$$= 204 \cdot 6 \times 2 \cdot 085 - 601 \cdot 6 \times 2 \cdot 3026 \times \log 1 \cdot 327$$

$$= 427 - 170 = 257 \text{ B.Th.U.}$$

$\therefore$ Rankine efficiency $= \frac{257}{1{,}083} = 0 \cdot 237$ or $23 \cdot 7$ per cent.

Efficiency ratio $= \frac{0 \cdot 0904}{0 \cdot 237}$

$= 0 \cdot 381$ or $38 \cdot 1$ per cent.

The chief results collected are:

Indicated horse power	= 42·37
Brake horse power	= 33·3
Mechanical efficiency	= 78·6 per cent.
Consumption rate per I.H.P. per hour	= 25·9 lb.
,, ,, ,, B.H.P. ,, ,,	= 33·08 lb.
Thermal efficiency	= 9·04 per cent.
Brake thermal efficiency	= 7·1 per cent.
Rankine efficiency ratio	= 38·1 per cent.

Heat account on a one-minute basis.

		B.Th.U.	per cent.
Heat supplied to engine in steam per min.	=	19,900	100
Indicated heat $= \dfrac{42{\cdot}37 \times 33{,}000}{778}$	=	1,794	9·04
Brake heat $= \dfrac{33{\cdot}3 \times 33{,}000}{778}$	=	1,414	
Friction heat	=	380	
Heat in condensing water = 364 × 34		12,376	62·3
Other losses		5,730	28·66
			100·00

EXAMPLES XIX

1. In a test of a gas engine the following observations were made:

Diameter of piston	= $6\frac{1}{2}$ in.
Stroke	= 12 in.
R.P.M.	= 324·6
Explosions per min.	= 143
Brake weight	= 84 lb.
Spring balance	= 4·25 lb.
M.E.P.	= 71·4 lb. per sq. in.
Gas per min.	= 4 cu. ft.
Calorific value of gas	= 240 C.H.U. per cu. ft.
Jacket water per mm.	= 12·75 lb.
Temperature rise in jacket water	= 27·5° C.
Compression ratio	= 3·85
Effective circumference of brake wheel	= 9·92 ft.

Find: (*a*) The I.H.P. (*b*) The B.H.P. (*c*) Mechanical Efficiency. (*d*) Consumption rate per B.H.P. and per I.H.P. per hour. (*e*) The thermal efficiency, brake thermal efficiency and efficiency ratio.

Also draw up the heat balance sheet on a basis of one minute.

2. The following observations were taken during a test of a small gas engine working on the four-stroke cycle and governed by the hit and miss method.

Cylinder diameter = $6\frac{1}{2}$ in. stroke = 1 ft.
Brake wheel diameter = 3·185 ft.

Area of mean indicator diagram positive loop)	= 1·084 sq. in.
,, ,, ,, ,, ,, (negative loop)	= 0·064 sq. in.
Length of diagram	= 3·2 in.
Scale of spring	= 240 lb. per in.
Duration of trial	= 30 mins.
Total revolutions	= 9,999
Calorific value of gas	= 240 C.H.U. per cu. ft.
Brake weight	= 38 lb.
Total explosions	= 2,871
Spring balance	= 2·5 lb.
Quantity of jacket water per min.	= 4·2 lb.
Temperature rise in jacket water	= 22° C.
Total gas	= 83·4 cu. ft.

Reduce the observations and write down—

(1) I.H.P. hours and B.H.P. hours worked.

(2) Indicated and Brake Thermal Efficiencies.

(3) Draw up a heat balance sheet on a minute basis, giving results as quantities of heat and also as percentages.

3. The following data were recorded during the test of a gas engine—I.H.P. 560, B.H.P. 447 r.p.m. 300, gas consumption (reduced to 15° C. and 30 in. of mercury) 32,500 cu. ft. per hour, lower calorific value of the gas per cu. ft. 71·2 C.H.U., jacket water used per hour 2,870 gallons, temperature of the jacket water at inlet and outlet respectively 18·1° C. and 54·3° C. Determine:

(*a*) The mechanical efficiency of the engine. (*b*) The thermal efficiency of the engine. (*c*) The percentage heat energy of the gas carried away by the jacket water. (*d*) Neglecting waste of heat by radiation, etc., find the percentage heat energy of the gas carried away by the exhaust gases. U.L.C.I.

4. The following data are the results of a test of a gas engine having a diameter of 8 in. and stroke of $15\frac{3}{4}$ in.

Duration of test, 1 hour; Revolutions per min. 280; Explosions per min. 110; M.E.P. from indicator diagrams 82·7 lb. per sq. in.; B.H.P. 14·4; Gas used during test 475 cu. ft.; Calorific value of gas per cu. ft. 270 C.H.U.; Weight of water passing through the jacket 700 lb.; Rise of temperature of jacket water 34° C.

Determine (*a*) the I.H.P. and (*b*) the thermal efficiency of the engine. Draw up a heat balance for the engine. U.L.C.I.

5. Describe, with the aid of sketches of apparatus used, the method of determining the mechanical efficiency of an engine. U.L.C.I.

6. A ship is driven by two sets of eight-cylinder internal combustion engines working on the four-stroke cycle with an impulse stroke every cycle. Diameter of cylinders 19·7 in.; stroke 26 in., r.p.m. 145·3, average mean effective pressure for all cylinders 74 lb. per sq. in. The thermal efficiency of the engines is 44 per cent.

Find the total I.H.P. driving the ship and the weight of oil used per I.H.P. per hour and per day's run of 24 hours. The calorific value of the oil is 9,890 C.H.U. per lb. U.L.C.I.

7. The mean effective pressure in an internal combustion engine working on the two-stroke cycle was 91·4 lb. per sq. in. The total fuel consumption per hour was 126·7 lb. and the calorific value of the fuel was 10,120 C.H.U. per lb. The B.H.P. was 327·3 and the speed 125 r.p.m. Diameter of cylinder 22 in., stroke 36 in. Determine the mechanical efficiency, the indicated thermal efficiency and the brake thermal efficiency. U.L.C.I.

8. The area of an indicator diagram taken from a double-acting steam engine is 2·1 sq. in. and its length is 3·5 in. The strength of spring used is 60 lb. per inch of height. The diameter of the cylinder is 18 in. and the piston stroke 26 in. Speed 150 revolutions per minute. Find the I.H.P. of the engine and also the B.H.P., the mechanical efficiency being 78 per cent. U.L.C.I.

9. Draw up a heat balance sheet for an oil engine from the following data, taking quantities in C.H.U. per hour.

Oil per hour 3·08 lb., calorific value 10,200 C.H.U. per lb., indicated horse power 5·8, brake horse power 4·4, cooling water per hour 385 lb., rise in temperature of cooling water 36° C. U.E.I.

10. The following particulars were obtained from a test of a four-stroke cycle gas engine:

Cylinder diameter $6\frac{1}{2}$ in., stroke 12 in., diameter of brake wheel $5\frac{1}{2}$ ft., effective brake load 60 lb., revolutions per min. 306, explosions per min. 128, area of indicator card 0·68 sq. in., length 2·25 in., spring, $\frac{1}{300}$ gas used per hour 453 cu. ft., calorific value 530 B.Th.U. per cu. ft.

Calculate the I.H.P., B.H.P., mechanical efficiency, gas consumption per B.H.P. per hour, and the brake thermal efficiency. U.E.I.

11. The results of a gas engine test gave the following: I.H.P. 7·6; B.H.P. 6·1; Jacket water 700 lb. per hour; Mean temperature rise 35° C.; gas used, 175 cu. ft. per hour; Calorific value of gas, 344 C.H.U. per cu. ft.

Draw up a heat balance sheet, taking quantities in C.H.U. per hour, and state the indicated and brake thermal efficiencies. U.E.I.

12. In a one-hour trial of a Diesel engine, the I.H.P. was 60·2 and the mechanical efficiency 70 per cent. The cooling water entered the jacket at 10° C., and left at 50° C., and 141 gallons were circulated. The calorific value of the fuel was 9,100 C.H.U. per lb. and 22·1 lb. were used. Calculate the brake thermal efficiency; the heat given to the jacket water; the heat carried off by the exhaust gases and otherwise; and draw up a heat balance sheet. U.E.I.

13. The following results were obtained during a one-hour test of an oil engine: I.H.P., 25·2; B.H.P. 20·6. The oil fuel was of specific gravity 0·75, lower calorific value 6,600 C.H.U. per lb. and the consumption for the run was 2·4 gallons. The weight of water passed through the jacket was 540 lb., the inlet and outlet temperatures being respectively 16° C. and 66° C. The exhaust gases on leaving the cylinder were conducted through

an exhaust calorimeter, and raised the temperature of 950 lb. of water from 16° C. to 70° C. Calculate the mechanical and thermal efficiencies of the engine; and draw up a heat balance sheet, showing the distribution of heat in C.H.U. per minute. U.E.I.

14. In a recently reported test of a pumping plant, the following results were obtained:

Boiler. Coal fired per hour, 688 lb.
Calorific value per lb., 15,000 B.Th.U.
Feed water per hour, 6,630 lb.
Temperature of feed water, 197·6° F.
Pressure of steam, 200 lb. per sq. in absolute.
Amount of superheat, 126° F.

Engine. Indicated horse power, 566.
Steam used per hour, 5,618 lb.

Find (*a*) the thermal efficiency of the boiler;
(*b*) the thermal efficiency of the engine;
(*c*) the thermal efficiency of the plant.

Note. There is a leakage of steam between the boiler and the engine and (*c*) must be calculated as the overall efficiency under the actual conditions of the test. Assume in (*b*) that the steam reaches the engine in the same condition as it leaves the boiler. N.C.T.E.C.

15. The following particulars were obtained during a test of a four-stroke-cycle gas engine:

Duration of test, 40 minutes. Average rev. per min., 204. Average explosions per min., 100. Mean effective pressure in cylinder, 96 lb. per sq. in. Net load on brake, 262 lb. Effective diameter of brake pulley, 6 feet. Diameter of piston, 12 in. Stroke, 1·5 ft. Gas used during period of test, 334 cu. ft. Calorific value of 1 cu. ft. of gas, 500 B.Th.U. Quantity of water passing through jacket, 738 lb. Rise in temperature of jacket water, 74° F.

Draw up heat balance and calculate:
(*a*) The indicated horse power.
(*b*) The brake horse power.
(*c*) The mechanical efficiency of the engine. N.C.T.E.C.

16. You are required to test a condensing steam engine developing about 20 I.H.P. in order to determine the mechanical and thermal efficiencies and to construct a heat balance sheet. Make a list of the observations which would be necessary and briefly describe how you would measure the necessary quantities. N.C.T.E.C.

17. The following observations were made during a test of a gas engine. Gas used, 396 cu. ft. per hour; calorific value of gas, 270 C.H.U. per cu. ft.; brake torque, 510 lb.-ft.; speed, 172 rev. per min.; explosions, 78 per min.; mean effective pressure, 70 lb. per sq. in.; cylinder diameter, 10 in.; stroke 18 in.

Calculate the indicated horse power, brake horse power, mechanical efficiency and brake thermal efficiency. N.C.T.E.C.

CHAPTER XX

SPEED CONTROL

168. It is essential that the speed of rotation of a heat engine should never be allowed to attain a value which would throw undue stresses upon its working parts. The speeds of steam locomotives, motor-cars and similar machines are controlled by the driver whose personal safety usually ensures that he will not allow excessive speed.

In many other cases, however, it is necessary that the speed be kept within limits, which are sometimes very narrow, as in the case of engines driving cotton spinning machinery and electrical alternators when running in parallel.

There are two causes of speed variation:

(1) the fluctuating torque on the crank shaft;

(2) alteration in the load which the engine has to take.

Each of these has to be corrected separately, and though absolute uniformity of speed is not attainable, the variations may be kept within narrowly defined limits.

The first of these types of fluctuation is dealt with by the flywheel entirely. These are known as cyclic variations because they occur with regularity. This part of the subject is not treated here, but will be found fully discussed in chapter IX of *Applied Mechanics*, by Mr. J. Boothroyd, B.Sc.

169. Governing. The governor is a device whose function is to check variations in speed when a change in the load occurs. If a steam engine is working against a load with a uniform rate of steam supply and the load decreases, the result will be an increase in speed. A decrease in speed will follow an increase in load. By decreasing the rate of steam supply in the first instance and increasing it in the second the governor will prevent a change of speed in excess of a certain percentage which has been decided upon.

A governor functions either (*a*) by throttling the main steam supply, (*b*) by altering the cut-off, or, (*c*) by both these means at the same time.

170. Centrifugal Governor. The principle generally made use of in governors is that of centrifugal force. Two weights placed diametrically opposite to each other are capable of moving outwards when speed is increased, and through a system of levers the throttle valve is partially closed or the cut-off is made earlier.

171. Watt Governor. The diagram (fig. 107) shows in outline the governor used by James Watt. It revolves about

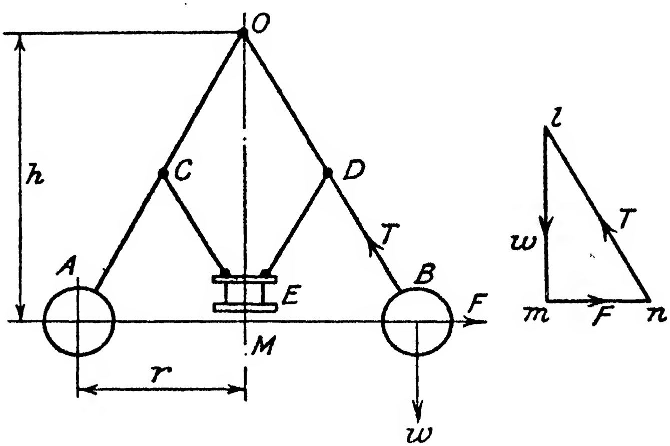


FIG. 107.

the axis OM which is caused to rotate by direct connection with the engine shaft. OA and OB are arms pivoted at O and carrying weights at A and B. Links attached at C and D operate a sleeve E, which transmits its movement to a bell crank lever connected to the steam control arrangement. Increased speed causes A and B to move outwards owing to increased centrifugal force, thus raising the sleeve. A decrease in speed is followed by an inward movement of A and B and a lowering of the sleeve.

The vertical height from the centre of one of the weights to O is called the height, *h*, of the governor, and this is obviously a ruling factor in the working of the governor, since it decides the position of the sleeve.

Let w = the weight of one of the balls in lb.

r = distance from centre of a weight to axis OM in ft.

F = centrifugal force on one weight in lb.

ω = angular velocity of rotation of weights in radians per second.

Consider the equilibrium of weight B.

Three forces act upon it, (a) gravitation, w lb., (b) centrifugal force, F, (c) tension in the arm OB, T lb.

The triangle lmn is the triangle of forces for the equilibrium of the forces acting on the ball. It is obviously similar to the triangle OBM.

$$\therefore \quad \frac{\mathrm{OM}}{\mathrm{BM}} = \frac{lm}{mn}$$

$$\therefore \quad \frac{h}{r} = \frac{w}{\mathrm{F}}$$

$$\therefore \quad h = \frac{wr}{\mathrm{F}}, \text{ but } \mathrm{F} = \frac{w}{g}\,\omega^2 r.$$

$$\therefore \quad h = wr \times \frac{g}{w\omega^2 r}$$

$$= \frac{g}{\omega^2}.$$

h and r are supposed to be in feet.

Example. Find the height of a Watt governor for speeds of 20, 60, and 90 revolutions per minute.

Note that the value of the weights and length of arms are of no importance in this calculation.

1st Case. $h = \dfrac{g}{\omega^2} = \dfrac{32{\cdot}2}{\left(\dfrac{2\pi \times 20}{60}\right)^2} = \dfrac{32{\cdot}2 \times 3 \times 3}{4\pi^2}$

$= 7{\cdot}34$ ft.

If the arms OA and OB were shorter than 7·34 ft., they would hang down against the axis.

2nd Case. $h = \dfrac{g}{\omega^2}$

$$h = \frac{32 \cdot 2}{\left(\dfrac{2\pi \times 60}{60}\right)^2}$$

$$= \frac{32 \cdot 2}{4\pi^2}$$

$$= 0 \cdot 816 \text{ ft. or } 9 \cdot 8 \text{ in.}$$

3rd Case. $h = \dfrac{g}{\omega^2}$

$$= \frac{32 \cdot 2}{\left(\dfrac{2\pi \times 90}{60}\right)^2}$$

$$= \frac{32 \cdot 2 \times 4}{4\pi^2 \times 9}$$

$$= 0 \cdot 363 \text{ ft. or } 4 \cdot 36 \text{ in.}$$

172. Porter Governor. The results of the above example show that at low speeds the governor would need unreasonably long arms, whilst at speeds which are now considered low, the variation in height, and consequently the movement of the sleeve, would be so small as to make governing difficult. This has given rise to other types, important amongst which is the Porter or loaded governor. In its normal form it is shown diagrammatically in fig. 108. A and B are rotating weights carried by arms OA and OB whilst arms BC and AD attach the weights to the sleeve.

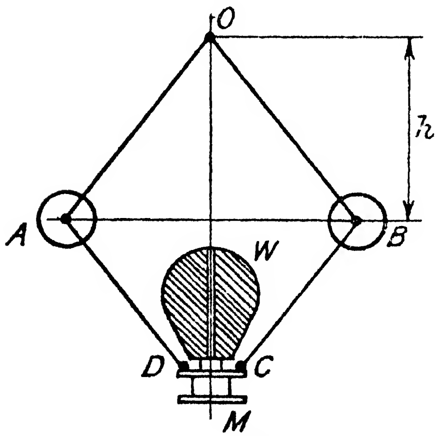


FIG. 108.

On the sleeve rests a weight, W, which is free to slide up and down the central axis with the sleeve. Usually it is arranged that the four arms OA, OB, BC and DA form a parallelogram.

The height of the governor is defined as before.

To find an expression for the height of this governor, produce OB to meet the horizontal through D in E, and consider the equilibrium of the link BD by taking moments about E. We take the particular case, which is the common one, where all arms are equal.

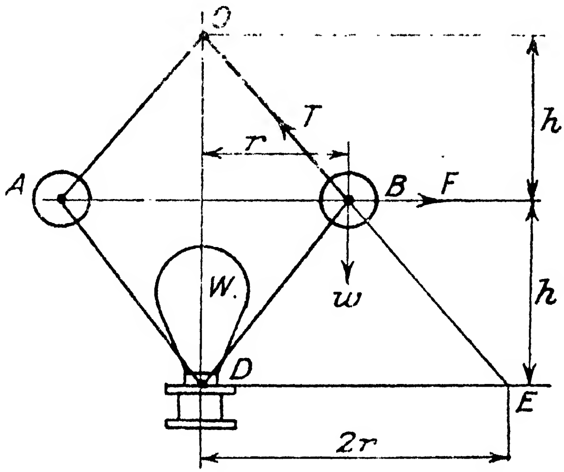


FIG. 109.

Let w = weight of each rotating weight in lb.

W = ,, ,, central load in lb.

h = height in feet.

F = centrifugal force on each weight in lb.

ω = angular velocity in radians per second.

The forces acting on BD are F, w, and T at B, with $\frac{W}{2}$ at D. For equilibrium, the moments of these forces about E must be zero. Hence, neglecting weights of all arms we have

$$Fh - wr - \frac{W}{2} \cdot 2r = 0$$

$$\therefore Fh = r(W + w)$$

$$\text{but } F = \frac{w}{g}\omega^2 r$$

$$\therefore h = r(W + w) \times \frac{g}{w\omega^2 r}$$

$$= \frac{g}{\omega^2} \cdot \frac{W + w}{w}.$$

If the arms are not equal, or if their weights are to be accounted for, the same method of attack gives the solution to the problem, but it leads to a formula too complicated to be of value. Such a problem is best worked from first principles as above.

EXAMPLE. The rotating weights of a Porter governor are each 5 lb. in weight. If the height of the governor is to be 10 in. when rotating at 200 r.p.m., find the weight of the load. What percentage increase in speed will lift the sleeve 1 in. if all the arms of the governor are equal?

$$\omega = \frac{2\pi \times 200}{60} \text{ radians per sec.}$$

$$= 20{\cdot}92$$

$$h = \frac{g}{\omega^2} \frac{W + w}{w}$$

$$\frac{10}{12} = \frac{32{\cdot}2}{20{\cdot}92^2} \cdot \frac{W + 5}{5}$$

$$W + 5 = \frac{10 \times 5 \times 20{\cdot}92 \times 20{\cdot}92}{12 \times 32{\cdot}2}$$

$$= 56{\cdot}5$$

$$\therefore W = 56{\cdot}5 - 5 = 51{\cdot}5 \text{ lb.}$$

To lift the sleeve 1 in. the height of the governor must become $9\frac{1}{2}$ in.

$$\therefore \frac{9{\cdot}5}{12} = \frac{g}{\omega^2} \cdot \frac{51{\cdot}5 + 5}{5}$$

$$\omega^2 = \frac{32{\cdot}2 \times 12}{9{\cdot}5} \cdot \frac{56{\cdot}5}{5}$$

$$= 459$$

$$\therefore \omega = 21{\cdot}44 \text{ radians per sec.}$$

$$\text{Rev. per min.} = \frac{21{\cdot}44 \times 60}{2\pi}$$

$$= 204{\cdot}6$$

$$\therefore \text{Increase in speed} = 204{\cdot}6 - 200 = 4{\cdot}6 \text{ r.p.m.}$$

$$\text{Increase per cent.} = \frac{4{\cdot}6}{2} = 2{\cdot}3 \text{ per cent.}$$

It will be seen that, since $\dfrac{W + w}{w}$ is under our control, by suitably choosing values for W and w, we may give h any reasonable value for a given speed. The great disadvantage, therefore, of the Watt governor is overcome.

173. Effect of Friction. The frictional resistance in the linkwork connected to a governor together with that in the governor itself, has a serious effect upon its operation. It will oppose the movement of the sleeve whether it is upward or downward.

In allowing for this friction, it is reduced to a force at the sleeve. This force can be found only by experiment and may be tested by a spring balance when the engine is stationary.

The central load in a Porter governor acts directly on the sleeve, hence the force due to friction may be added to or subtracted from the weight of the central load.

Let the force at the sleeve due to friction $= f$, then, when the governor is rising, the resistance at the sleeve will be $W + f$. If the governor is falling, then the downward movement of the sleeve will also be opposed by the friction. The effect will be to decrease the effective value of the central load, and the governor will behave as if the value of the central load were $W - f$.

The formula of the last article, therefore, may be written in the form

$$h = \frac{g}{\omega^2}\,\frac{W \pm f + w}{w}.$$

The positive sign is taken when the speed is increasing, and the negative when it is decreasing.

Example. In the example of the last article, find the limits of speed of the governor between the heights 10 in. and $9\frac{1}{2}$ in., if the friction is equivalent to 8 lb. at the sleeve.

Before the governor can rise to a smaller height than $9\frac{1}{2}$ in., the friction and weight must be overcome.

$\therefore$ we take $W + f$.

$$h = \frac{g}{\omega^2}\cdot\frac{W + f + w}{w}$$

$$\frac{9\cdot5}{12} = \frac{32\cdot2}{\omega^2} \times \frac{51\cdot5 + 8 + 5}{5}$$

$$\omega^2 = \frac{32{\cdot}2 \times 12 \times 64{\cdot}5}{9{\cdot}5 \times 5} = 524$$

$$\therefore \omega = 22{\cdot}9 \text{ rad. per sec.}$$

$$\text{Revs. per min.} = \frac{22{\cdot}9 \times 60}{2\pi} = 218{\cdot}7.$$

At the lower limit of speed, when the height = 10 in., before the sleeve can move downwards, the load must overcome the friction and its effective weight is thus reduced. Hence W − f must be taken.

$$h = \frac{g}{\omega^2} \cdot \frac{W - f + w}{w}$$

$$\therefore \frac{10}{12} = \frac{32{\cdot}2}{\omega^2} \cdot \frac{51{\cdot}5 - 8 + 5}{5}$$

$$\omega^2 = \frac{32{\cdot}2 \times 12 \times 48{\cdot}5}{10 \times 5} = 375$$

$$\omega = 19{\cdot}4 \text{ rad. per sec.}$$

$$\text{Rev. per min.} = \frac{19{\cdot}4 \times 60}{2\pi} = 185{\cdot}3.$$

This is a rather extreme case, but, when compared with the range of speeds in the last example, it shows how serious the effect of friction may be in reducing the sensitiveness of the governor.

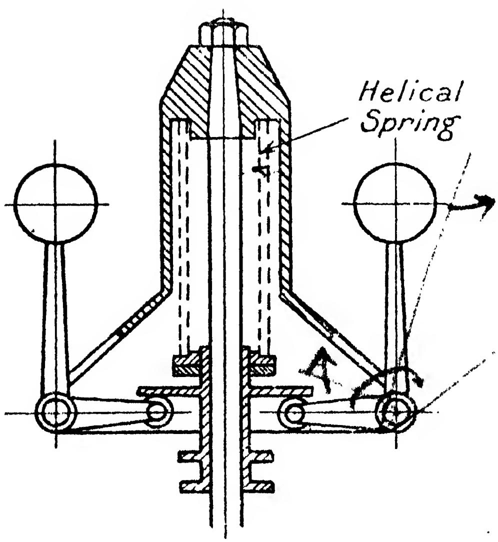


FIG. 110.

174. Hartnell or Spring-loaded Governor. In the governors just treated, gravity has been the force which centrifugal action had to overcome. The Hartnell governor and its many modifications use a spring in either compression or tension for this purpose. The sketch gives the main idea of a spring-loaded governor. Two bell crank levers carry a weight at one end whilst the other end operates on the sleeve. The pivots of the levers are carried by a bracket which rotates with the spindle. On the sleeve a spring in compression acts. When the rotational speed increases the spring is compressed further due to the outward movement of the weights. The spring has an initial compression given to it, so that the governor will only begin to function when the centrifugal force, acting on the levers, is sufficient to compress the spring further. That can be at any predetermined speed. There is so great a freedom of choice in weights, and stiffness of springs, that a governor on this principle can be made to meet almost any conditions. With modifications in design, it can be made to work with its axis horizontal, which is often an advantage.

The principle of this governor is very simple and no formula need be deduced to deal with it. The following example will illustrate the method of calculation.

EXAMPLE. The weights of a Hartnell governor are each 4 lb. The vertical members of the levers are 5 in. long and the horizontal members 6 in. long. The radius of the circular path of the weights is 7 in. The initial compression on the spring is 90 lb. and its stiffness is 40 lb. per in. Find the speed at which the governor will begin to function, and the speed when the radius of the path of the weights is $7\frac{1}{2}$ in.

very Good

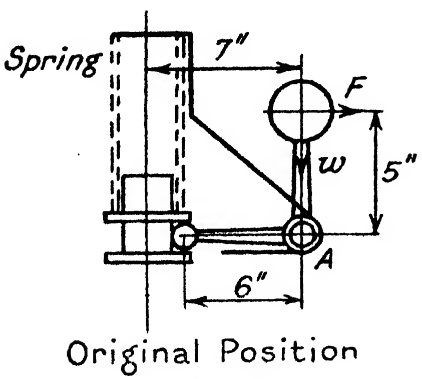


FIG. 111.

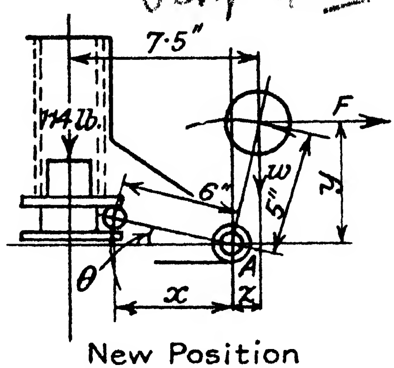


FIG. 112.

Fig. 111 shows the arrangement.

Only half the force of the spring acts on each lever.

Let F = centrifugal force on one weight.

Then, when the sleeve is just on the point of rising, we have, taking moments about A

$$F \times 5 = \frac{90}{2} \times 6$$

$$\text{but } F = \frac{w}{g}\omega^2 r = \frac{4 \times 7 \times \omega^2}{32{\cdot}2 \times 12}$$

$$\therefore \frac{4 \times 7 \times \omega^2 \times 5}{32{\cdot}2 \times 12} = \frac{90}{2} \times 6$$

$$\therefore \omega^2 = \frac{90 \times 6 \times 12 \times 32{\cdot}2}{2 \times 4 \times 7 \times 5}$$

$$= \frac{9 \times 6 \times 3 \times 32{\cdot}2}{7}$$

$$= 746$$

$$\omega = 27{\cdot}33 \text{ radians per sec.}$$

$$\text{Rev. per min.} = \frac{27{\cdot}33 \times 60}{2\pi} = 261.$$

The stiffness or rate of a spring is the force per inch of compression or elongation.

When the radius increases by $\frac{1}{2}$ in. the rise of the sleeve will be $\frac{1}{2} \times \frac{6}{5}$ = 0·6 in. for this example. Total force on spring with extra compression = 90 + 0·6 × 40.

$$= 114 \text{ lb.}$$

$$\therefore F \times y + w \times z = \frac{114}{2} \times x \text{ (see fig. 112)}$$

$$\frac{w}{g}\omega^2 r \times 5 \cos\theta + w \times \tfrac{1}{2} = \frac{114}{2} \times 6 \cos\theta$$

$$\frac{w}{g}\omega^2 r \times 5 + \frac{w}{2\cos\theta} = 57 \times 6$$

$$\frac{4 \times 7{\cdot}5 \times 5 \times \omega^2}{32{\cdot}2 \times 12} + \frac{4}{2\cos\theta} = 342.$$

Cos θ is very nearly equal to 1.

$$\therefore \frac{4 \times 7{\cdot}5 \times 5}{32{\cdot}2 \times 12}\omega^2 = 342 - 2 = 340$$

$$\omega^2 = \frac{340 \times 32{\cdot}2 \times 12}{4 \times 7{\cdot}5 \times 5}$$

$$= 876$$

$$\omega = 29{\cdot}6 \text{ radians per sec.}$$

$$\text{Rev. per min.} = \frac{29{\cdot}6 \times 60}{2\pi} = 282{\cdot}5.$$

Generally wz may be neglected.

Friction may be taken into account as in the case of the Porter governor by adding it to the total compression on the spring for a rising governor and subtracting it for a falling governor.

175. Range of Speeds. Any governor must be so designed that at the lowest speed in its range, it will give the engine full steam supply. At the highest point it should so reduce the steam supply, that the engine will not gain speed, even when all external load is removed. On account of this, flywheel control and governor control have to be properly regulated relatively to each other.

Suppose, for instance, a governor allows a maximum variation of 10 revs. per min. Let the flywheel allow the same cyclic variation. Then, in every cycle (usually one revolution), the governor sleeve would move from its lowest position to its highest position and back again. As this would happen during each revolution, whether load was great or small, the governor would be of no use. When these conditions exist, or even if they are approached, a phenomenon known as "hunting" arises.

It is necessary, therefore, that the flywheel should only permit cyclic variations which are considerably smaller than those permitted by the governor.

Examples XX

1. Why is it necessary in practice to control the speed of an engine? Distinguish between the function of a flywheel and a governor.

2. Give reasons why it is impossible to maintain absolutely uniform speed of rotation.

3. For what range of speeds will a Watt governor vary in height from 12 in. to 6 in.?
Why is the Watt governor not used in modern practice?

4. A Porter governor with equal arms has a load of 45 lb. and rotating weights each 4 lb.
What is its height at a speed of 240 r.p.m.?

5. In a spring-loaded governor, the compression on the spring is 80 lb. The ball-arms of the bell-crank levers are 5 in. long and the sleeve-arms 6 in. long. The radius of the path of the balls is 7 in. and the weight of each is 5 lb. Neglecting friction, at what speed must the governor rotate to satisfy the given conditions?

6. The governor of question (4) has friction equivalent to a force of 4 lb. at the sleeve. At what speed will it begin to fall from the given position and at what speed will it rise?

7. A spring-loaded governor has a compression in the spring of 60 lb. The spring compresses 1 in. for a force of 40 lb., and friction is equivalent to a force of 6 lb. at the sleeve. If the radius of the path of the balls is 7 in. when the spring has the above compression and if both ball-arms and sleeve-arms are 6 in. long, find the increase in speed for a rise of 1 in. in the sleeve. Assume that the lower limit of speed is as low as possible and that the balls weigh 8 lb. each.

8. Distinguish between the governing of a steam engine by (*a*) throttling and (*b*) variable expansion, and illustrate your answer by showing the effect of each method on an indicator diagram.

Give a sketch of a spring-loaded governor and explain its action.

U.L.C.I.

9. Obtain a formula for the height h of a Porter governor in terms of the central load W, the weight of each ball w, the frictional resistances f, and the angular velocity ω radians per second. Assume the pendulum arms and the suspension links of equal length and that they intersect on the main axis.

Using this formula, calculate the amount of central weight required so that the links, each 15 in. long, shall be inclined at 45° to the main axis when the speed is 150 r.p.m. You may assume the speed to be increasing. The weight of each ball is 4 lb. and the resistance on the sleeve due to moving the governor gear is 3 lb. U.E.I.

10. Prove that h, the height in feet of a loaded governor of the Porter type with equal arms and links, is given by the equation

$$h = \frac{W + w}{w} \cdot \frac{2{,}934}{N^2}$$

N = r.p.m., w = weight of each ball,
W = weight of central load.

If w = 4 lb., W = 56 lb., and the height does not alter when the speed increases suddenly from 250 to 255 r.p.m., calculate the force opposing the rising of the sleeve. U.E.I.

11. A Porter governor with equal arms runs normally at 180 r.p.m. If the height at this speed is $7\frac{1}{2}$ in., what is the ratio of the central load to the weight of one of the balls? If the frictional resistance on the sleeve be equal to 10 per cent. of the central load, what is the proportional increase of speed before the governor acts? If you use a formula, show, if you can, how it is derived. U.E.I.

CHAPTER XXI

STEAM TURBINES

176. Introductory. The steam turbine is the result of much experiment and research. Many attempts to produce a rotary engine had been made prior to 1880, at which time several inventors were endeavouring to perfect their respective machines. It was not until the year 1892 that the inventiveness, skill, and courage of the late Honourable C. A. Parsons produced a turbine which became a competitor with the reciprocating steam engine.

His first mechanically successful turbine was produced in 1884 and now occupies an honourable place in the Engineering Museum at South Kensington. This turbine developed 10 horse power and was used to drive a dynamo. Its steam consumption was so high that it could not hope to displace the high class reciprocating steam engines of that day. Such progress, however, was made in the next eight years that the turbine thereafter rapidly gained popularity, and in 1905, the famous *Mauretania* was engined with turbines to develop 75,000 horse power.

The production of electrical energy on the scale of modern demands would be impracticable without the steam turbine. The horse power developed by a single turbine set is now as much as 200,000, whilst there are one or two sets in the world designed to develop even higher powers than this. In the field of marine engineering, the turbine is supreme for high powered ships.

The turbine is very different from a piston engine, both in the manner in which the steam delivers its energy as work, and also in the mechanical construction and general appearance of the machine. If the controlling gear be excepted, there are no parts of a steam turbine moving with reciprocating motion. The moving parts are all rotating and are entirely enclosed. Consequently, there is no very evident sign to indicate that anything is taking place in the turbine. A slight vibrational hum, some radiated heat, and the speed indicator, are the only means of detecting the fact that heat is in process of transformation into work.

In the case of the piston engine, the pressure of steam as registered by a pressure gauge (known as the statical pressure) is the means of propulsion. Although the pistons may be moving at speeds as high as 1,000 feet per minute, and the steam is following up the pistons at this speed, the kinetic energy of the steam is negligible. The steam velocity is small.

In the steam turbine the statical pressure is not used directly for creating the propulsive force. It is used to create high steam velocities, in some cases as high as 4,000 *feet per second.* The propulsive force on the moving parts of the turbine is produced by changing the direction of this high velocity steam and thus changing its momentum. The force set up by this means on the working elements is a dynamic force (i.e. one set up by change of motion), as distinct from the static force exerted by steam on the sides of its containing vessel. There is, therefore, a double transformation of energy in the steam turbine, first from heat into kinetic energy, and second, from kinetic energy into force and thence into mechanical work. The piston engine does the transformation directly, namely from heat into mechanical work on the piston.

Notwithstanding these important differences between the reciprocating steam engine and the steam turbine, the calculations set out in Chapter XIII are equally applicable to both types of engine, and the term " all steam engines " in the second line of art. 95, includes steam turbines.

177. Advantages of the Turbine. The chief advantages of the turbine engine are as follows:—

I. Very large units can be constructed as indicated in the last article.

II. Since the turbine is a high speed machine it develops high power per unit volume occupied.

III. It can utilize high vacuum very advantageously.

IV. No internal lubrication is needed. Therefore condenser tubes can be kept free from oil and a high rate of heat transmission maintained in the condenser. The boiler feed water is also free from oil with resulting freedom from overheating of tubes. No special oil elimination treatment of the feed water is necessary to maintain high heat transmission rates in the boiler.

V. Heavy foundations are unnecessary as the machine can be accurately balanced.

VI. The high rotational speed of the turbine makes it very suitable for electrical power generation.

VII. Considerable overloads can be carried at the expense of slight reduction in overall efficiency.

VIII. The construction of a turbine readily lends itself to the process of cascade or series feed water heating, or to the provision of steam at reduced pressures for processes such as distilling, heating, humidifying, digesting, etc.

IX. The condenser inlet can be made large and the condenser itself can be placed very close to the last working blade ring. This enables the expansion of the steam to be carried down to condenser pressure and therefore the expansion in a steam turbine is complete, as in the Rankine Cycle.

X. The turbine can be operated against any reasonable back pressure and it can be arranged to use high pressure steam, low pressure steam, or both; and also steam exhausted from reciprocating engines.

178. The Simple Turbine. The essential features of a simple steam turbine are illustrated in fig. 113. The steam pipe from the boiler terminates in the main stop valve M. Close to this valve is placed a second valve G under the control of the turbine governor. The steam enters the nozzle chest C through the pipe P. The nozzle chest consists of a special steam pipe curved so that its centre line coincides with the mean path of motion of the working blade. Passages of special shape, known as nozzles and indicated at N, lead the steam from the nozzle chest to the working blades B. During its passage through the nozzles the steam falls in pressure and increases enormously in velocity. It is directed towards the blades by the nozzles which are inclined to the plane in which

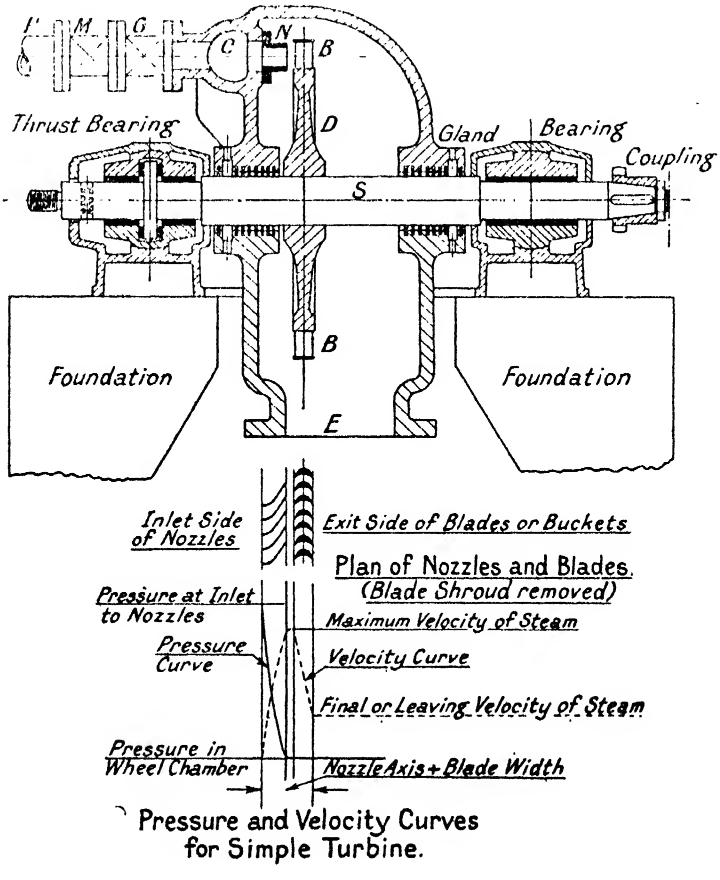


FIG. 113.

the blades move, as shown in the plan. The blades are shaped and arranged somewhat as indicated, with the result that the steam which leaves the nozzle enters the blade passages and literally blows the blades along. Special shapes of nozzles and of blades are used to increase the driving force of the steam upon the blades. The steam leaves the blade wheel on the right and flows downwards through the exhaust passage E into the condenser. The work done on the blades is transferred to the shaft S by means of the turbine wheel or disc D.

179. The Turbine Stage. The word stage used in this connexion means a pressure stage. A pressure stage is any portion of the turbine in which a pressure drop occurs followed by the development of mechanical work in a moving element or blade ring. It follows from this definition that fig. 113 illustrates a single pressure stage turbine. This simple turbine forms the basis of the calculations treated in this chapter.

180. General Description of the Steam Turbine. A complete steam turbine of the impulse type, by the Metropolitan-Vickers Electrical Co. Ltd., is illustrated in section in fig. 114. It consists of fourteen pressure stages, the first of which is compounded for velocity as in the Curtis turbine. The remaining thirteen stages are simple pressure stages as in the Rateau turbine, and this arrangement is known as the Rateau system. Each of the fourteen stages comprises a unit turbine and the complete machine illustrated is said to be pressure compounded. The last two stages embody the makers patent multi-exhaust construction, part of the steam going direct to the condenser after passing through the outer portion K, of the thirteenth stage working blade. The steam passing through the inner portion L, expands through the nozzles M and does work on the fourteenth set of blades I. This construction is adopted to avoid excessively long blades in the last stage and at the same time to take advantage of the highest vacuum. To make the working portions clearer, the first eight pressure stages have been repeated in fig. 115, and both figures are similarly lettered except that in the latter figure, N_2, N_3, N_4, etc., represent the successive nozzles after the first, whilst B_2, B_3, B_4, etc., represent the corresponding blades attached to their respective wheels W_2, W_3, W_4, etc., which rotate.

The steam is supplied to the first stage nozzles N, through the nozzle box S. The expansion in the first nozzles is usually sufficient to remove any superheat from the steam and thus avoid temperature stress distortion in the casing and wheels.

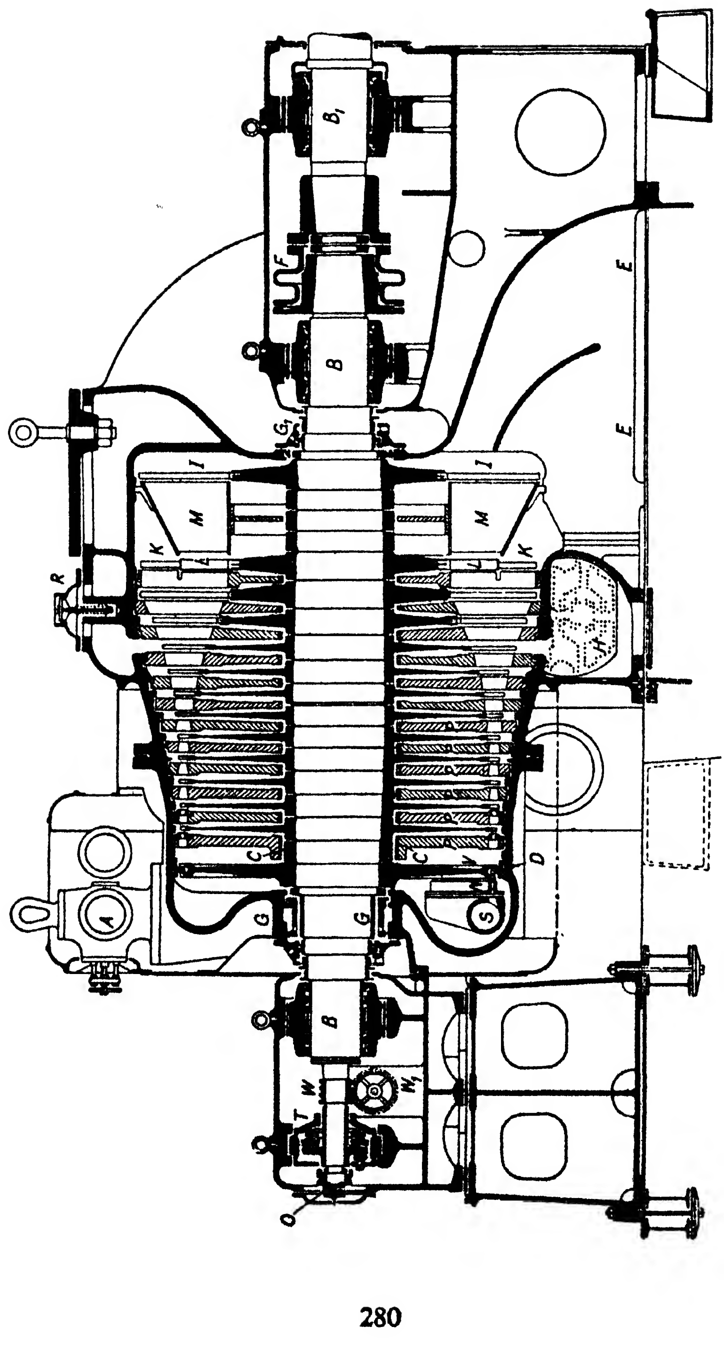


FIG. 114.

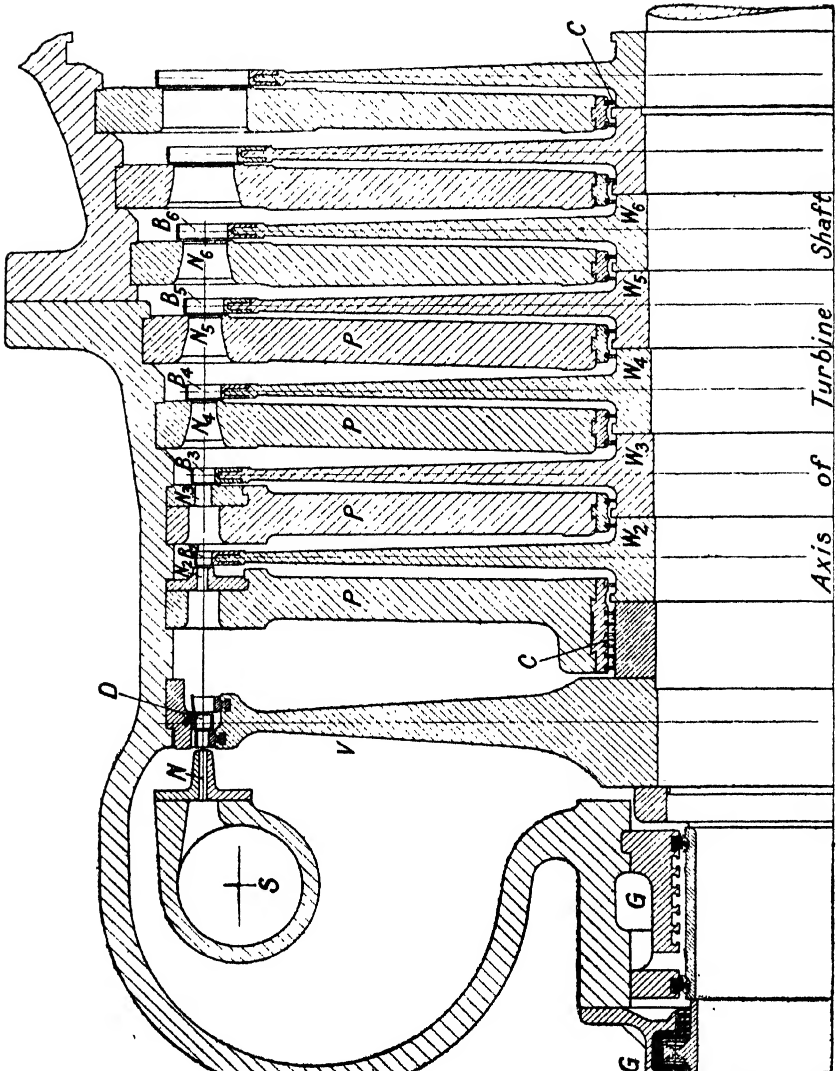
The velocity of the steam leaving the first nozzles N is higher than at any other point in the turbine and this enables two rings of moving blades to be used for the first pressure drop with a directing ring D, of blades between them.

The two moving rings are clearly shown on the first wheel V. This arrangement is known as velocity compounding.

The first nozzles do not extend round the full circumference of the wheel V. The second and third sets of nozzles are formed in steel blocks attached to the diaphragms P. These diaphragms consist of cast steel plates fitted into grooves formed in the casing. Casing and diaphragms are in halves in the horizontal plane containing the axis of the shaft. All stages after the third have the nozzles cast in the diaphragms with the exception of the last one M, in which the nozzles are built up. Usually the arc of admission to successive pressure stages increases until, in the last stages, the full circumference is utilized, although full circumferential admission in the early stages is frequently used to reduce windage losses.

Blade heights and nozzle heights increase in the successive stages in order to accommodate the increasing volume of steam. The increasing nozzle arc also assists in providing this accommodation. Overloads in this type of turbine can be met by increasing the number of first stage nozzles in action, or by supplying steam through the overload valve A to the chamber between the first and second stages. Between the tenth and eleventh stages, steam is extracted from the turbine to heat the feed water in the heater H. Also at this point a relief valve R is fitted, to warn the attendant of unusual conditions and to protect the casing from accumulation of pressure. The steam not extracted passes on through the multi-exhaust stages to the condenser and leaves the turbine at EE.

The pressure on the left-hand side of each diaphragm is greater than on the right. It is, therefore, necessary to prevent steam from passing in considerable quantity through the essential working clearances between the inner edges of the diaphragms and the outer surfaces of the wheel bosses. This is practically accomplished by means of labyrinths which



C
B6
N6
W6
B5
N5
W5
P
B4
N4
W4
B3
N3
W3
N2
B2
W2
C
D
N
V
S
G
G
Axis of Turbine Shaft

FIG. 115.

consist of rings with projecting fin edges as shown at C. The packing edges are thinned down to a few thousandths of an inch to prevent "seizing" if contact should occur. In the event of contact taking place, the heat generated is localized in the fin which is softened and gives way until a clearance is again established. The principle on which this packing operates is that of throttling the pressure down through a very restricted opening as explained in Articles 56 and 57.

The remaining features of the turbine illustrated in fig. 114 are common to large turbines and differ, in machines from different makers, in constructional detail only.

The two main bearings BB carry the rotor and are supplied with oil under pressure as is also the alternator bearing B_1.

The turbine and alternator shafts are connected by a flexible coupling at F, the flexibility being provided by the bellows shape of the coupling sleeve close to the letter.

At the points where the shaft passes out of the casing, steam and water sealed glands are fitted. The gland G prevents leakage of steam from the casing to atmosphere, and the gland G_1 prevents air from leaking into the condenser where it would interfere with condensation. At starting, both glands are steam sealed to prevent air from entering the casing, since the vacuum is created before the turbine is started. When the turbine has run up to speed, water is supplied to the glands and a forced vortex is created as a means of preventing leakage. This method has long been in use on Metropolitan-Vickers turbines.

A detail of a high pressure gland is shown in fig. 116, whilst a water gland is shown in fig. 117. The gland shown in fig. 116 is of the comb type and consists of nine rings P of special form supported at the outer circumference by springs F, and having axial projections overlapping similar radial projections on the packing sleeve Q. The high pressure steam acts at A. The space B is at low pressure and is connected either to drain or to condenser. The spaces B and C are separated by another packing ring known as the leak-off ring. Steam passing from B to C is sometimes allowed to leak through the passage E, into the atmosphere where it indicates, by its quantity, the state of the packing. In the case of turbines fitted with the water gland, no leakage to the atmosphere takes place. There is some evaporation due to the heat generated in the gland, but this can be piped to the condensing system. This evaporation is made up continuously.

It will be noted that each comb consists of five throttling orifices. The shaft is indicated at SS and D is a ring which throws oil and water off the shaft and also keeps them separate.

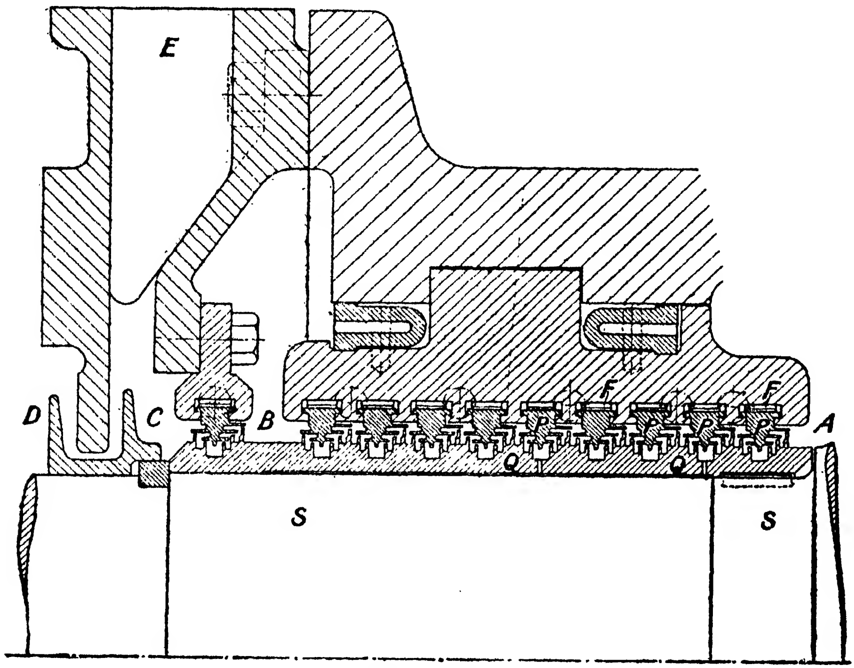


FIG. 116.

High Pressure Gland—12,500 KW. Turbine

In fig. 117 the packing rings R are simpler than those of fig. 116, and are similar to those employed in the diaphragm glands. The impeller I produces a water pressure which seals the gland by slightly different radial depths of water, that on the higher pressure side being less than that on the low pressure side.

A typical double thrust Michell bearing is illustrated in fig. 118. This bearing is used to register the rotor accurately in an axial direction and thus keep the distance between the outlet edges of the nozzles and the inlet edges of the blades the correct amount. This registration is accomplished by adjusting the thickness of the serrated rings S. The thrust collar is indicated at T and the bearing pads at BB. The pads are carried in the foundation rings F, and are pivoted in such a way as to ensure very effective and reliable lubrication. The governor and oil pump are driven from the turbine shaft by the worm W and wheel W_1 (fig. 114), whilst O is an emergency trip governor

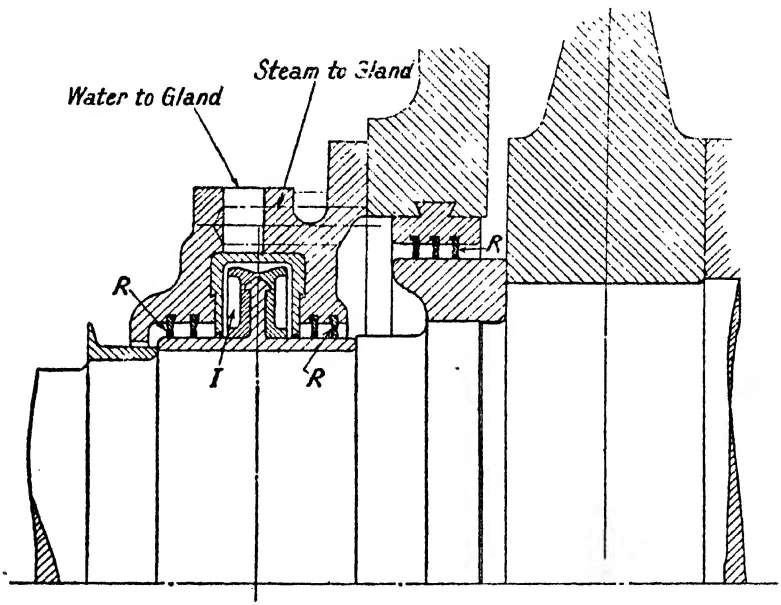


FIG. 117.

incorporated in the shaft and operating by centrifugal force. It is restrained by a light spring so long as the predetermined safe speed is not exceeded. The situation of the thrust block is indicated at T in fig. 114.

Examination of figs. 114 and 115 will reveal the large working clearances between the blades and casing, and between blades and nozzle diaphragms, which are characteristic of the impulse turbine. The pressure in successive wheel chambers gets less as the steam approaches the condenser, but the pressure on each side of any wheel is the same, holes being provided in the wheels to ensure this. The successive fall in pressure in each Rateau stage is arranged to produce the same leaving velocity at each set of nozzles.

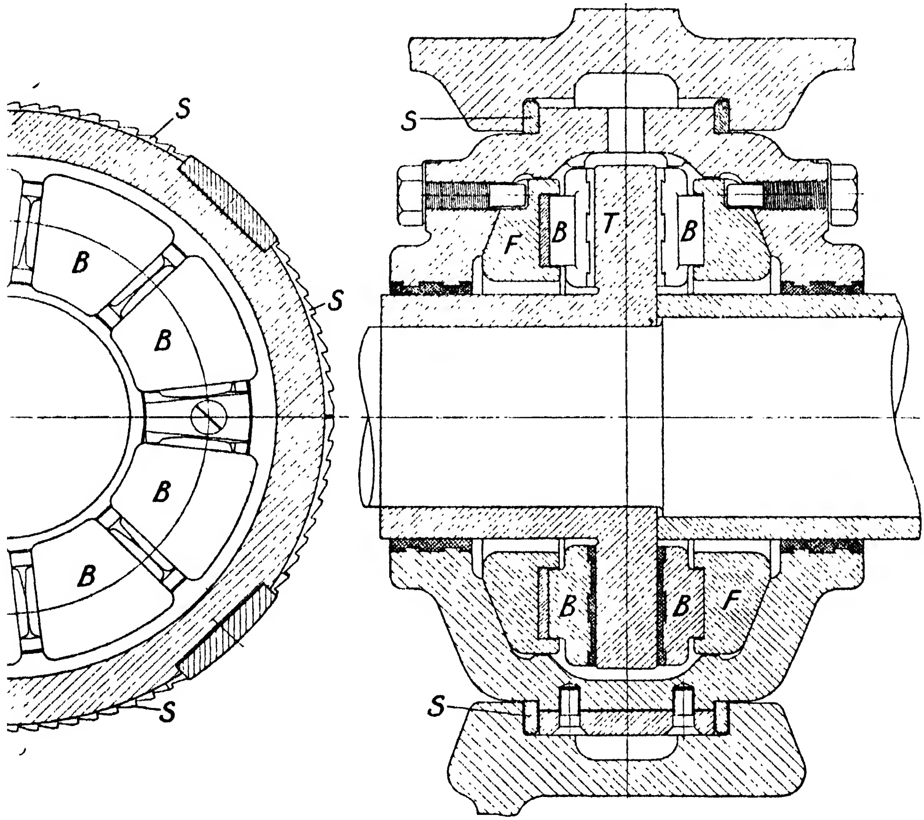
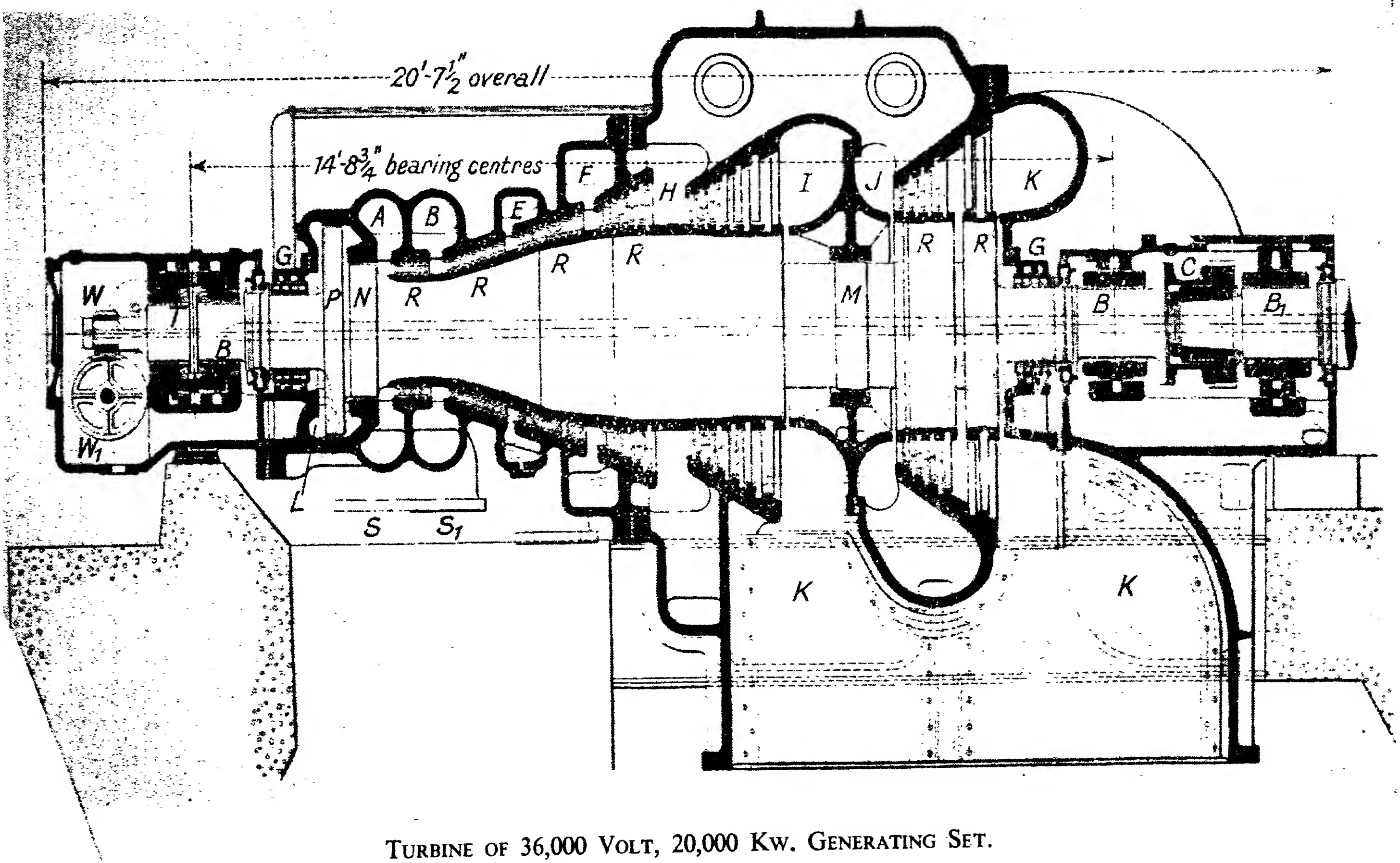


FIG. 118.

181. The Reaction Steam Turbine. A longitudinal section of a modern turbine of the well-known Parsons reaction type is shown in fig. 119.

This machine, by C. A. Parsons & Co. Ltd., is the direct descendant of the famous 1884 turbine and is remarkable for the comparatively small space occupied by a turbine of its output. It is capable of producing 20,000 kilowatts at full load and its thermal efficiency at its most economical load is 31 per cent.



TURBINE OF 36,000 VOLT, 20,000 KW. GENERATING SET.

In this machine the steam is admitted through the pipe S to the steam belt A, and enters the first group of blades round the full circumference of the blade as is usual with reaction turbines. This first group, between the belts A and B, consists of six pairs of blade rings on a parallel portion of the rotor R.

The second group between B and E consists of ten pairs on a conical portion of the rotor. When the turbine is called upon to take overloads, steam is supplied through the pipe S_1 and the belt B, to the gap between the first and second groups of blades. This supply is additional to the normal supply through S and A. The next two steam belts E and F, with their respective gaps between the blade groups, are points at which a portion of the steam is led away from the turbine for the purpose of heating boiler feed water or for other heating purposes.

The third group of blades between E and F consists of eleven pairs of rings. The fourth group between F and H, on a parallel portion of the rotor, contains six pairs which increase considerably in height to accommodate the increased volume of steam. The steam pressure at E in the first pass-out belt is 35 lb. per sq. inch, and at F in the second one, it is $6\frac{1}{2}$ lb. per sq. inch. At discharge from the fourth group of blades, shown at H, the steam remaining in the turbine divides, part of it passing outside the casing carrying the fixed blades of the fifth group. This latter group occurs between the points H and I. The steam which passes out at H enters the last group at J and expands through the five pairs of blade rings comprising this group into the exhaust passage K and thence downward to the condenser. The remaining steam expands through the six pairs forming the fifth group and also enters the condenser. The portion of the turbine between H and K is the low pressure portion and here the steam has a large specific volume.

The division of the low pressure steam into two parts as explained above, is known as the double flow arrangement. In the turbine illustrated, the steam travels in the same direction through each flow. In many cases the two portions of low pressure steam are arranged to flow in opposite directions so that the thrust set up in one flow is balanced by the thrust set up in the other. The arrangement shown in fig. 119 necessitates a labyrinth steam packing M, between the two low pressure expansions since the pressure at J is greater than that at I. The reason for the adoption of the double flow arrangement is the need for provision of large steam flow area at exhaust from the last moving ring without unduly long blades which would be mechanically weak.

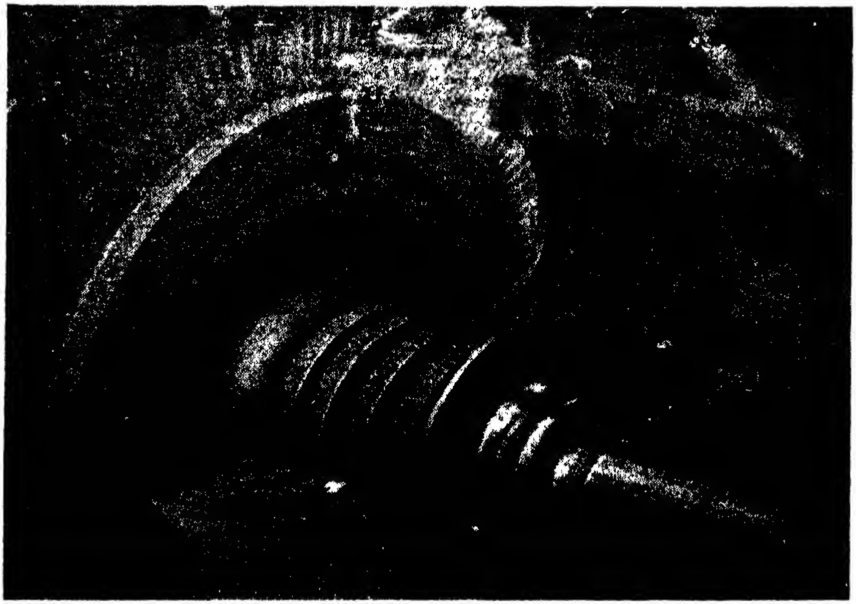


FIG. 120.

In the reaction turbine, blades attached to the casing fulfil the function of nozzles and alternate with blades attached to the rotor. Thus the first blades which the steam encounters are fixed whilst the last are moving. Both fixed and moving blades in each pair are identical, but reversed in direction. This can be followed by reference to figs. 121 and 133.

The pressure drop across any blade ring is small, and therefore the increase in steam velocity is also small compared with that which occurs in an impulse turbine nozzle. Thus there are more sets of blades in a reaction turbine than in an impulse turbine for the utilization of the same heat drop. The blade heights and the mean diameters of succeeding blade rings progressively increase along the turbine, and the absolute steam velocities also increase as the steam flows through the turbine from inlet to exhaust. As there is a pressure drop across each moving row of blades in a reaction turbine, it follows that there must be some end thrust due to the steam pressure on the blades and on the shaft. To counteract this thrust two dummy pistons N.P. are provided at the steam inlet end of the turbine. The piston N balances that part of the blading between A and E, while piston P balances the blading between E and H. The space L is connected to the low pressure port H.

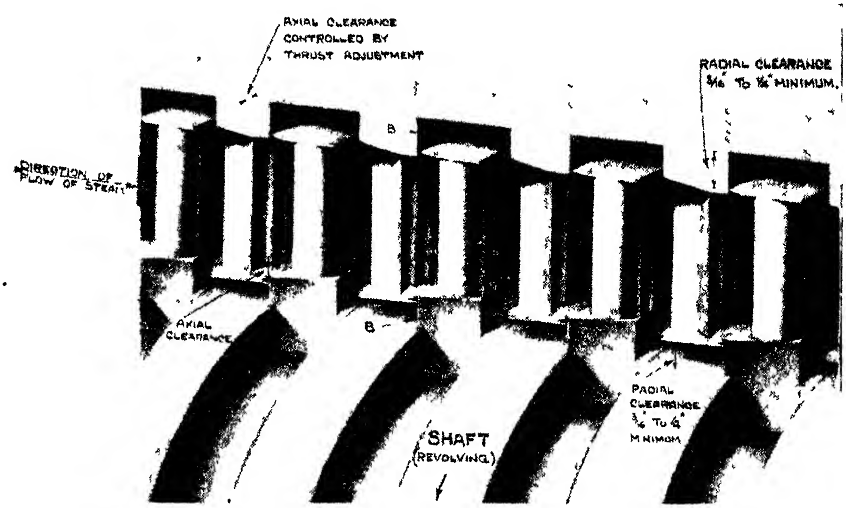


FIG. 121.

Any out of balance still remaining is taken up by the thrust block T. As in the impulse turbine, this block serves also to register the rotor accurately in its casing.

Leakage to or from the casing is prevented by steam sealed carbon glands G, one of which is illustrated in fig. 120.

This figure also shows the shaft bearing with its oil throwing rings and the end of the rotor R with its blading.

The flexible coupling C, of the Parsons claw type, with the end of the alternator shaft B_1 can be seen on the right in fig. 119, and the governor and oil pump driving worm W and wheel W_1 on the left.

c Reaction blading is very efficient, especially in low pressure sections. In the high pressure sections, where the steam is relatively dense and the blades short, the essential radial clearances were formerly the cause of considerable steam leakage. The whole of the steam did not pass through the blade channels, with the result that the efficiency was reduced. This leakage has been overcome by means of Parsons patent end tightened blades which are illustrated in fig. 121.

The projecting blade shrouds reduce the clearance between fixed and moving parts of the turbine and serve as labyrinth packings. The shrouds on the blades fixed to the cylinder have forked ends, the lower one almost touching the foundation ring B of the adjoining blades, whilst the upper one serves to guide the steam into the moving blades without shock.

The feather edges ensure freedom from damage if contact should be made between tightening strip and foundation ring. By the use of this blading, the radial clearance can be increased as shown in fig. 121, with consequent less risk of blade stripping and without the excessive leakage of steam which would otherwise accompany the use of larger radial clearances.

It must be pointed out that this type of turbine is really an impulse-reaction turbine since both impulse and reaction are operating. A pure reaction turbine would have moving nozzles which would move in a direction opposite to the steam flow.

A pure impulse machine would have no expansion or generation of kinetic energy in the moving blades. In the Parsons turbine, the blades fixed to the casing perform the same function as the nozzles of an impulse turbine and the blades fixed to the rotor perform the same function as the moving nozzles of the pure reaction turbine. Thus the working blades are also nozzles and this is the reason for the name reaction.

Usually the term nozzle is applied to fixed passages, but it is applicable to any part of a turbine where controlled expansion and generation of kinetic energy occur. The difference between impulse and reaction turbines is further explained in Articles 186 and 187.

182. The Steam Nozzle. It will be noted from Articles 178 and 180, that the portion of the elemental turbine in which the steam operates to do work consists of (*a*) the nozzle passage, (*b*) the working blade which moves. The function of the steam turbine nozzle is threefold, first it must expand the steam from the higher pressure at inlet to the lower pressure at exit. In doing this, the nozzle allows the heat energy in the steam to be converted into velocity or kinetic energy. Secondly, it must deliver the steam with a maximum of kinetic energy for which reason it needs to be smooth and of correct shape; thirdly it must direct the steam into the blade channels and for this purpose it is set at an angle to the path of motion of the blades.

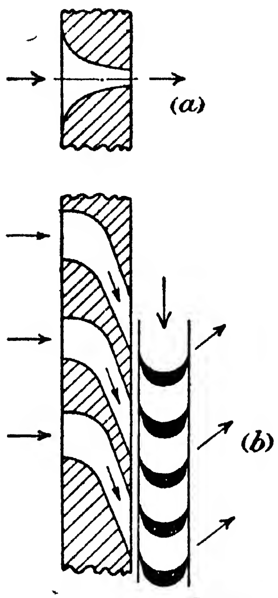


FIG. 122.

There are two principal types of nozzle used in turbine practice, (*a*) the convergent nozzle, (*b*) the convergent-divergent nozzle. The convergent nozzle is indicated conventionally at (*a*) in fig. 122, whilst (*b*) of the same figure shows how the nozzle is arranged in the turbine diaphragm.

The cross-sectional area of a convergent nozzle diminishes from inlet section to outlet section. In other words the outlet section is the least section of the nozzle. This type of nozzle is used when the ratio, $\dfrac{\text{absolute pressure at exit section}}{\text{absolute pressure at inlet section}}$ is not less than about 0·58. This ratio in any type of nozzle is known as the pressure ratio of the nozzle and the actual value of the limiting ratio for convergent nozzles depends on the value of the index n in the expansion law of the steam. For example, if the steam expands through the nozzle according to the law $PV^{1.135} = C$ then the ratio is 0·58, whilst if the law is $PV^{1.3} = C$ the ratio becomes 0·5457. The latter value is used in all cases except those in which the steam is initially wet. Thus with n equal to 1·3 the least pressure at the exit section of a convergent nozzle will be 0·5457 P_1, where P_1 is the pressure at inlet to the nozzle. This least pressure is known as the *critical pressure* of the nozzle.

It will be shown in more advanced work that when the exit pressure from a convergent nozzle reaches the value indicated by the above pressure ratio the nozzle is discharging its maximum weight of steam per unit time. The lower pressure referred to is that in the nozzle itself and this pressure is not necessarily that in the chamber into which the nozzle discharges.

EXAMPLE. The steam supply to a convergent nozzle is at 200 lb. per sq. inch absolute. Assume the expansion law to be $PV^{1\cdot3} = C$ and calculate the least theoretically correct exit pressure.

$$\frac{\text{Exit pressure}}{\text{Inlet pressure}} = 0{\cdot}5457 \text{ when } n = 1{\cdot}3.$$

$\therefore$ Exit pressure $= 0{\cdot}5457 \times 200 = 109{\cdot}1$ lb. per sq. inch absolute.

The pressure at exit could be any value higher than 109·1 lb. per square inch but the nozzle would not in this case discharge its maximum amount of steam.

EXAMPLE. The pressure in front of certain nozzles is 80 lb. per sq. inch absolute. The exit pressure is 48 lb. per sq. inch absolute. What type of nozzle is indicated?

$$\text{Pressure ratio} = \frac{48}{80} = 0{\cdot}6.$$

Since this is higher than 0·58 or 0·5457, the nozzles are convergent, irrespective of the value of n. If this ratio had fallen between 0·58 and 0·5457 it would have been necessary to state the value of n to be used.

The convergent-divergent nozzle is illustrated in fig. 123, (*a*) indicating, as before, the conventional nozzle and (*b*) the nozzle arranged in the turbine casing. This nozzle consists of the convergent nozzle of fig. 122 with a diverging portion added, and it is used when the pressure at exit is less than $0{\cdot}5457\ P_1$.

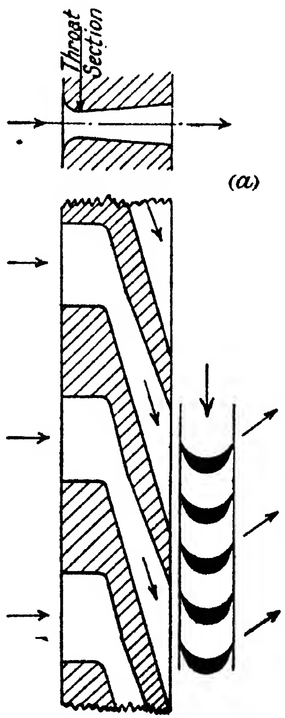


FIG. 123.

The object of the diverging portion is to expand the steam to a lower pressure *in the nozzle* than the critical pressure. The effect is to control the expansion down to lower pressures than can be done with the convergent nozzle. By this means, higher steam velocities can be attained with a convergent-divergent nozzle than with a convergent nozzle.

The least cross section of this combination nozzle is known as the *throat* section and it corresponds to the exit section of the convergent nozzle. Beyond this section the nozzle widens until the exit section is reached. The explanation is briefly as follows. During the expansion of the steam and its passage along the nozzle its pressure falls continuously, and expansion is complete as in the Rankine Cycle. The heat released produces a continuous increase in velocity and the fall in pressure produces increased volume in the steam. At first the volume increase is not at sufficient rate to keep pace with the velocity increase and thus the nozzle can be reduced in section. At the throat the limit of this condition is reached and thereafter the specific volume increases at a greater rate than the velocity. Hence this portion of the nozzle must diverge to accommodate the steam.

EXAMPLE. A nozzle is required to expand steam from 200 lb. per sq. inch absolute to 20 lb. per sq. inch absolute. What type of nozzle is needed?

Pressure ratio $= \frac{20}{200} = 0{\cdot}1$. Since this is less than 0·5457 the nozzle must be convergent-divergent.

EXAMPLE. A nozzle of the convergent-divergent type is required to expand steam from a pressure of 180 lb. per sq. inch dry, to 2 lb. per sq. inch. The expansion law is $PV^{1{\cdot}3} = C$. Find the pressure and specific volume of steam at the throat and the specific volume at exit. Both pressures are absolute.

Pressure at the throat $= 0{\cdot}5457 \times 180$

$= 98{\cdot}2$ lb. per sq. inch absolute.

From steam tables the specific volume at 180 lb. per sq. inch absolute $= 2{\cdot}534$ cu. ft.

$P_1V_1^{1{\cdot}3} = P_TV_T^{1{\cdot}3}$. Suffix 1 referring to inlet and T to throat.

$$\therefore V_T = \left(\frac{P_1}{P_T}\right)^{\frac{1}{1\cdot3}} V_1$$

$$= \left(\frac{180}{98\cdot2}\right)^{\frac{1}{1\cdot3}} \times 2\cdot534$$

$$= 1\cdot83^{0\cdot77} \times 2\cdot534$$

$$= 1\cdot592 \times 2\cdot534$$

$= 4\cdot03$ cubic feet per lb. at the throat.

$P_1V_1^{1\cdot3} = P_2V_2^{1\cdot3}$. Suffix 2 refers to exit.

$$\therefore V_2 = \left(\frac{P_1}{P_2}\right)^{\frac{1}{1\cdot3}} \times 2\cdot534$$

$$= \left(\frac{180}{2}\right)^{0\cdot77} \times 2\cdot534$$

$$= 90^{0\cdot77} \times 2\cdot534$$

$$= 31\cdot97 \times 2\cdot534$$

$= 81\cdot02$ cubic feet per lb. at exit.

183. Calculation of Steam Velocity. The calculation of the velocity with which steam leaves a nozzle is based on the law of the conservation of energy. The steam expands freely through the nozzle from a pressure P_1, to a pressure P_2. There is no piston upon which the steam can do work, hence it does work upon itself and sets itself in motion. The heat energy rendered available by the expansion between the two pressure limits is converted into kinetic energy. The amount of energy available in the ideal case for each pound of steam, is the Rankine Cycle work (see pages 150 *et seq*). For convenience this quantity is called the *heat drop* or *available heat.*

Since the heat available per pound of steam is operating to increase the kinetic energy of one pound weight of steam we have, in the ideal case,

Heat drop = Increase in kinetic energy of one pound weight.

$$\text{That is } H_R = \frac{V^2}{2gJ}.$$

Where H_R = Rankine heat drop in B.Th.U. per pound.
V = Velocity of steam in feet per second.
$g = 32\cdot2$ pounds per engineers unit of mass.
J = Joule's mechanical equivalent of heat = 778 ft.-lb. in this case.

EXAMPLE. Dry steam is supplied to a nozzle at 200 lb. per sq. inch absolute. The exit pressure of 1 lb. per sq. inch absolute. Calculate the final dryness of the steam and the velocity of efflux from the nozzle assuming that the steam is initially at rest and that the flow is adiabatic and frictionless.

To determine the final dryness the entropy quantities will be used. The heat drop will be calculated by the method given on page 157.

$$x_2 = \frac{1{\cdot}5538 - 0{\cdot}1323}{1{\cdot}9724 - 0{\cdot}1323} \quad \text{See page 316.}$$
$$= \frac{1{\cdot}4215}{1{\cdot}8401}$$
$$= 0{\cdot}773$$

Total heat at A, before expansion $= 1{,}199{\cdot}5$ B.Th.U. per pound.
„ „ at B, after „ $= 69{\cdot}7 + 0{\cdot}773 \times 1{,}036{\cdot}1$
$= 69{\cdot}7 + 800{\cdot}7$
$= 870{\cdot}4$ B.Th.U. per pound.

$\therefore$ Heat drop $= 1{,}199{\cdot}5 - 870{\cdot}4$
i.e. $H_R = 329{\cdot}1$ B.Th.U.

$$\therefore \quad V = \sqrt{2g\,J\,H_R}$$
$$= \sqrt{2 \times 32{\cdot}2 \times 778 \times H_R}$$
$$= \sqrt{2 \times 32{\cdot}2 \times 778 \times H_R}$$
$$= 223{\cdot}8 \times \sqrt{H_R}$$
$$= 223{\cdot}8 \times \sqrt{329{\cdot}1}$$
$$= 223{\cdot}8 \times 18{\cdot}11$$
$$= 4{,}052 \text{ feet per second.}$$

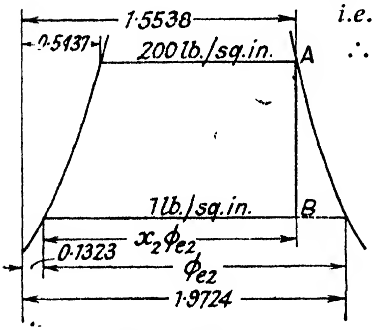


FIG. 124.

It should be noted that such a steam velocity is rarely met in practice. The large range of pressure is divided into portions to give approximately equal heat drops and equal increments of velocity.

Reference to Article 96 and to the worked example on page 155 will show that

$$H_R = \frac{n}{n-1}\,P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right] \text{ foot pounds.}$$

Therefore $V = \sqrt{2g\,\frac{n}{n-1}\,P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$ and this equation can be used to calculate the steam velocity when the value of n is known.

EXAMPLE. Steam expands through a nozzle according to the law $PV^{1\cdot3} = C$. The initial pressure is 150 lb. per sq. inch absolute and the exit pressure is 20 lb. per sq. inch absolute. Calculate the velocity with which the steam issues from the nozzle assuming that the steam is at rest just before the inlet section.

$$V = \sqrt{2g\frac{n}{n-1}P_1V_1\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \sqrt{2 \times 32\cdot2 \times \frac{1\cdot3}{0\cdot3} \times 144 \times 150 \times 3\cdot015\left[1 - \left(\frac{20}{150}\right)^{\frac{0\cdot3}{1\cdot3}}\right]}$$

$$= \sqrt{18{,}173{,}937\cdot6\left[1 - \frac{1}{7\cdot5^{0\cdot2307}}\right]}$$

$$= \sqrt{18{,}173{,}937\cdot5\left[1 - \frac{1}{1\cdot592}\right]}$$

$$= \sqrt{18{,}173{,}937\cdot5\ [0\cdot372]}$$

$$= \sqrt{6{,}760{,}704\cdot8}$$

= 2,600 feet per second.

184. Effect of Friction. The walls of a nozzle are made as smooth as possible, but even when this has been done they are not devoid of some degree of roughness. Thus no actual nozzle is without its frictional effect. There are also frictional effects in the body of the steam itself.

The net result of these effects is that the steam velocity at exit from an actual nozzle is always somewhat less than the ideal velocity which would be generated with the same heat drop.

Let V_a = actual steam velocity at exit from the nozzle in feet per second.

V_t = ideal or theoretical velocity at exit from the nozzle in feet per second.

Then the ratio $\dfrac{\text{actual exit velocity}}{\text{ideal exit velocity}}$ is known as the *velocity coefficient* of the nozzle. The value of this coefficient varies from 0·92 to 0·98 and is usually denoted k.

Thus actual velocity at exit $= k \times$ ideal exit velocity

$$= k \times \sqrt{2g\,J\,H_R}.$$

EXAMPLE. If the velocity coefficient of the nozzle in the last exercise, Article 183, is 0·96, what is the actual velocity of the steam as it leaves the nozzle?

Actual steam velocity $= 0{\cdot}96 \times 2{,}600$
$= 2{,}496$ feet per second.

The principal object of the nozzle is to convert the available heat energy of the steam into kinetic energy. In a frictionless nozzle all the heat drop would be converted. Friction reduces the velocity of the steam and the actual kinetic energy of the steam at exit from the nozzle is less than the ideal kinetic energy.

The ratio, $\dfrac{\text{actual kinetic energy per lb. at exit}}{\text{ideal kinetic energy per lb. at exit}}$ is the *nozzle efficiency* and this quantity will be denoted η_n.

Using the above notation, we have

Actual kinetic energy at exit per pound of steam

$$= \frac{V_a^2}{2g} \text{ foot pounds.}$$

Ideal kinetic energy at exit per pound of steam

$$= \frac{V_i^2}{2g} \text{ foot pounds.}$$

$$\therefore \quad \eta_n = \frac{\dfrac{V_a^2}{2g}}{\dfrac{V_i^2}{2g}} = \frac{V_a^2}{V_i^2} = \left(\frac{V_a}{V_i}\right)^2 = k^2$$

Also $\dfrac{V_i^2}{2g} = JH_R$

$$\therefore \quad \eta_n = \frac{V_a^2}{2g\ JH_R}$$

or $\quad V_a^2 = 2gJ\ \eta_n\ H_R$

i.e. $\quad V_a = 223{\cdot}8\sqrt{\eta_n H_R}$ when H_R is in British Thermal Units.

EXAMPLE. Dry steam is supplied to a nozzle which expands it from 180 lb. per sq. inch absolute to 14·7 lb. per sq. inch absolute. The expansion is adiabatic. Find the velocity of the steam at exit from the nozzle if the efficiency of the nozzle is 0·9.

The process of adiabatic expansion with friction is shown by the line AC in fig. 125. The line AB is the ordinary frictionless adiabatic expansion line.

$$\text{Dryness at B} = \frac{1{\cdot}5554 - 0{\cdot}31186}{1{\cdot}7573 - 0{\cdot}31186}$$
$$= 0{\cdot}860$$

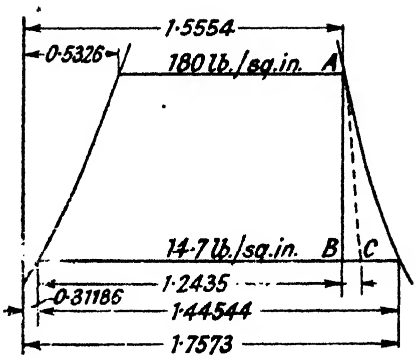


FIG. 125.

Total heat at A before expansion $= 1{,}198$ B.Th.U.

„ „ „ B after „ $= 180 + 0{\cdot}860 \times 970{\cdot}7$
$= 1{,}015$ B.Th.U.

$\therefore$ Heat Drop $= 1{,}198 - 1{,}015$
$= 183$ B.Th.U.

Actual energy converted $= 0{\cdot}9 \times 183$
$= 164{\cdot}7$ B.Th.U.

$\therefore$ Exit velocity of steam $= 223{\cdot}8 \times \sqrt{164{\cdot}7}$
$= 2{,}872$ feet per second.

Assuming that no heat enters or leaves the steam during expansion in the nozzle, the heat not converted into kinetic energy remains in the steam at exit. The result is that the steam is actually drier than it would be at exit from a perfect nozzle when the pressure limits are the same. This affects the specific volume of the steam at exit and this quantity is important when the nozzle dimensions are to be found. In the above example, the dryness at the end of the adiabatic frictionless expansion is 0·860. Actually, however, 0·1 of the heat drop remains in the steam as additional heat.

Therefore the improvement in dryness due to friction

$$= \frac{0{\cdot}1 \times 183}{970{\cdot}7}$$
$$= 0{\cdot}0188$$

$\therefore$ Actual dryness at exit $= 0{\cdot}860 + 0{\cdot}0188$
$= 0{\cdot}8788$

The volume of one pound of the wet steam at exit
$= 0{\cdot}8788 \times 26{\cdot}8$
$= 23{\cdot}55$ cubic feet.

In fig. 125, C is the end point of the actual condition line AC. The length BC represents the evaporation which would be produced by 18·3 B.Th.U. at the atmospheric exhaust pressure of this example. It is necessary to point out that this heat has been degraded to a lower state of temperature and is therefore wasted since 14·7 lb. per square inch is the discharge pressure.

185. The Blade Velocity Diagram. The velocity of steam in the fixed nozzle passages of any turbine is the velocity of the steam relative to the earth. This velocity is known as the *absolute steam velocity.* The velocity of the steam relative to the blade will depend on the speed of the blades, the absolute steam velocity, and the angle which the nozzle axis makes with the plane in which the blade moves. The absolute steam velocity is represented by AB in fig. 126. This velocity can be resolved

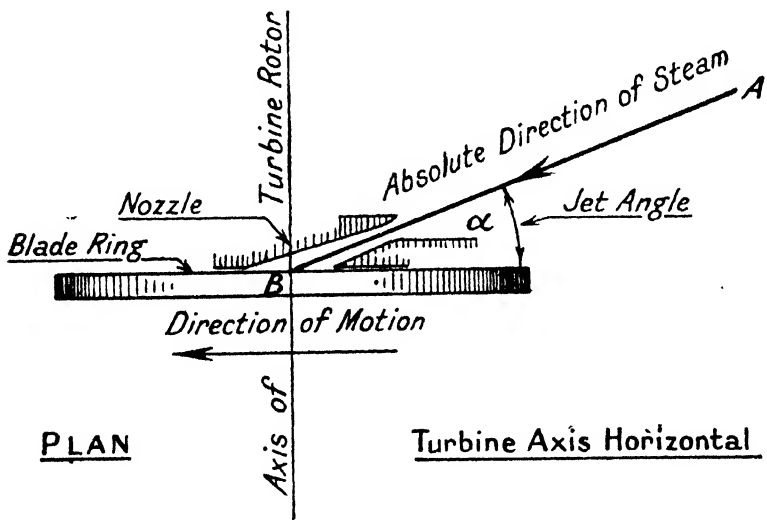


FIG. 126.

into two components, one parallel to the direction of motion of the blade and the other parallel to the axis of the turbine. The first component is known as the *velocity of whirl* and the second is known as the *velocity of flow.* The velocity of whirl is important in calculating the force or work done on the blades and the velocity of flow in calculating the thrust along the turbine axis.

To find the velocity of steam relative to the moving blade, vector velocity diagrams, or equations based upon them, are used.

Let AB (fig. 127) represent the absolute steam velocity at entry to the blades. Set it down to scale, making an angle α with the direction of motion of the blade. This angle is known as the nozzle angle or the inlet jet angle and is the angle at which the nozzles are set. Now set off, to the same velocity scale, the velocity of the blade along the direction of motion of the blade. Let this be CB. Then a straight line AC indicates the direction, magnitude, and sense of the motion of the steam relative to the blade. This diagram is the inlet triangle and is a case of vector subtraction. The blade velocity vector CB is vectorially subtracted from the steam velocity vector AB, giving the vector AC as the difference. Imagine the blade speed to be equal to the whirl velocity then the relative velocity of steam to blade will have a direction at right angles to the direction of the blades, and will be equal in magnitude to the flow velocity. If the blade is stationary, then the relative velocity becomes equal to the absolute steam velocity.

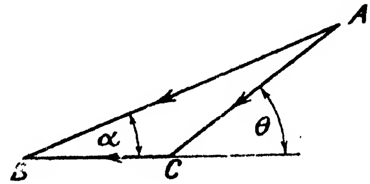


FIG. 127.

The diagram drawn above is the inlet velocity triangle. There is a similar triangle for the exit edge of the blade. This is illustrated in the third example of this article.

The angle θ (fig. 127) is the angle of the blade at inlet and the steam will glide on to the blade without shock when the conditions of turbine rotational speed, turbine diameter, absolute steam velocity and inlet jet angle enable the diagram to produce the angle θ to agree with that of the actual blade. If the blade velocity, steam velocity and inlet jet angle are known, then the blade angle and relative steam velocity can be found.

EXAMPLE. **A steam turbine is 4 ft. mean diameter of blade ring and the nozzle angle is 22°. The turbine speed is 550 r.p.m. and the inlet blade angle is 30°. Draw the velocity diagram and find the absolute steam velocity at entry to the turbine wheel. Also find the inlet whirl and flow velocities.**

Blade speed $= \pi$ DN.

$= 3{\cdot}1416 \times 4 \times 550$ ft. per minute

$= \dfrac{3{\cdot}1416 \times 4 \times 550}{60}$ ft. per second.

$= 115{\cdot}2$ ft. per second.

The velocity triangle, fig. 128, is now set out.

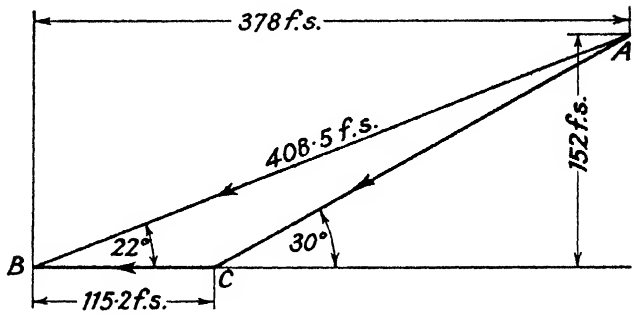


FIG. 128.

Measurement of this triangle gives the following results, absolute steam velocity 408·5 ft. per second; inlet whirl velocity 378 ft. per second; inlet flow velocity 152 ft. per second.

EXAMPLE. The ideal heat available in a turbine nozzle is 288 B.Th.U. The nozzle discharges the steam into a ring of moving blades, the efficiency of the nozzle being 0·92. The nozzle angle is 20° and the blade angle at inlet is 32°. Find the blade velocity assuming no shock at entry.

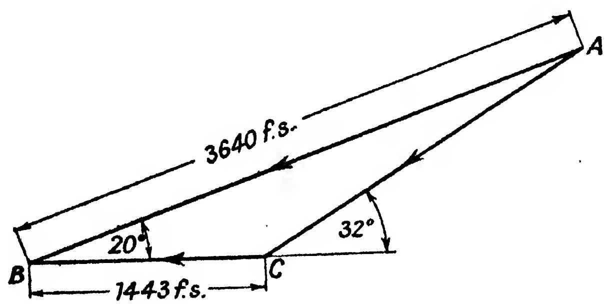


FIG. 129.

Set down any length to represent the unknown blade speed. Draw the line BA, fig. 129, at 20°. Draw CA at 32°. Then AB represents the steam velocity to scale.

This velocity is equal to $\sqrt{2g\,J\,\eta_n\,H_R}$

$$= 223{\cdot}8\sqrt{0{\cdot}92 \times 288}$$

$$= 3{,}640 \text{ ft. per second.}$$

The original diagram, from which fig. 129 is reproduced, was set out so that 3 inches represented the blade speed. On measuring the length of AB it was found to be 7·56 inches. This length represents 3,640 ft. per second, and thus the velocity scale can be found.

$$\therefore \text{ Blade speed} = \frac{3{,}640 \text{ ft. per sec.}}{7{\cdot}56''} \times 3''$$

$$= 1{,}443 \text{ ft. per second.}$$

EXAMPLE. The relative velocity of steam leaving the blades of an impulse turbine is 1,200 feet per second. The blade speed is 460 feet per second, and the angle of the blade at exit is 28°. Draw the exit velocity triangle and find the velocity of whirl, the velocity of flow, and the absolute velocity of steam at exit from the blades. What is the angle which the leaving jet of steam makes with the plane of the turbine wheel?

The diagram in this case is one of vector addition.

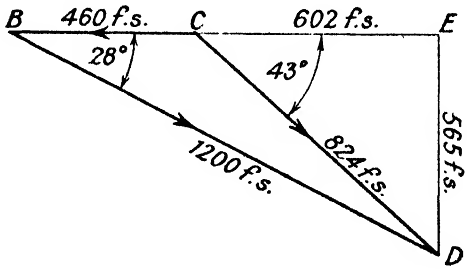


FIG. 130.

Set off the blade speed, 460 feet per second along CB (fig. 130). Also se off 1,200 feet per second along BD making an angle of 28° with the direction of motion. Complete the triangle by drawing CD. Then CE = whirl velocity = 602 feet per second ; ED = flow velocity = 565 feet per second; CD = absolute leaving velocity = 824 feet per second and θ = 43° = exit jet angle.

186.—The Simple Impulse Blade Velocity Diagram. This diagram consists of two velocity triangles, one for the inlet edge and one for the outlet edge of the blades. The arrangement is shown in fig. 131, and this allows the blade shape to be inserted in the combined triangles. The significance of the lines are as follows, all being set down to scale and the angles measured with a protractor.

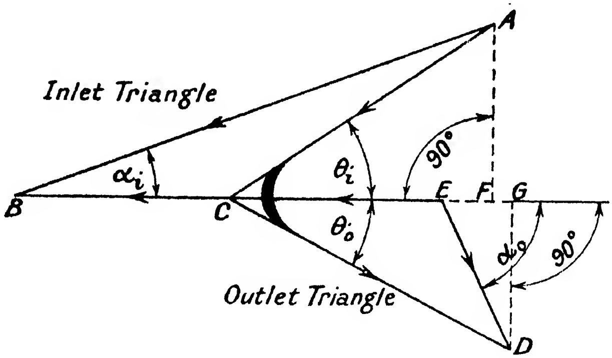


FIG. 131.

AB is the absolute steam velocity at inlet.

CB " " " blade " " "

AC " " relative steam " " " to the blade.

EC " " blade velocity repeated for convenience in the second triangle.

CD " " relative steam velocity at outlet from the blade.

ED " " absolute steam " " "

a_i " " inlet jet angle.

a_o " " outlet " "

θ_i " " inlet blade angle.

θ_o " " outlet " "

If the blade is considered to be frictionless, then the steam leaves it at the same velocity, relative to the blade, as it enters and in this case the length of CD is the same as the length AC. If friction is taken into account then the steam leaves the blade with less velocity than it possessed when entering. In this case, the length CD is less than AC and the ratio $\frac{CD}{AC}$ is the *velocity* *coefficient* for the blade. The inlet whirl velocity is represented by FB and the outlet whirl velocity by EG. The inlet and outlet flow velocities are represented by AF, and GD respectively. The arrows indicate the sense of flow in all cases.

The ratio $\frac{BC}{AB}$ is called the *speed ratio* of the blades. For simple impulse turbines, the value of this ratio is usually about 0·47.

EXAMPLE. An impulse steam turbine nozzle is inclined at 20° to the plane of motion of the blade. The steam leaves the nozzle with a velocity of 2,500 feet per second and the blade speed is 1,200 feet per second. Find the relative steam speed at inlet, the inlet blade angle, the absolute steam speed at outlet, and the outlet jet angle. Neglect friction and assume the blades to be symmetrical, *i.e.* $\theta_o = \theta_i$.

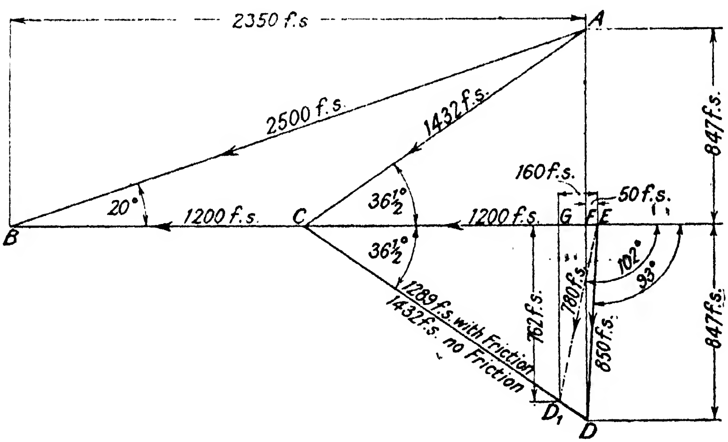


FIG. 132.

Draw the diagram as in fig. 132 with AB = 2,500 feet per second; CB = 1,200 feet per second; and $\alpha_i = 20°$. Then relative steam speed at inlet AC = 1,432 feet per second; inlet blade angle = 36·5°; absolute steam speed at outlet ED = 850 feet per second; and outlet jet angle = 93°.

EXAMPLE. Write down the whirl and flow velocities from the data of the last example (*a*) when friction is neglected, (*b*) when friction reduces the steam speed 10 per cent. whilst the steam is passing through the blade channels.

The whirl velocities will be considered as positive towards the left and the diagram shows that both inlet and outlet whirl velocities are positive.

Measurement of the diagram gives the following results. For no friction, inlet whirl velocity FB = 2,350 feet per second; outlet whirl velocity EF, 50 feet per second; inlet flow velocity AF, 847 feet per second; outlet flow velocity FD, 847 feet per second.

When friction is taken into account, outlet whirl velocity EG = 160 feet per second; outlet flow velocity GD_1 = 762 feet per second; inlet whirl and flow velocities are unchanged. The absolute steam velocities and outlet jet angles change from ED, 850 feet per second to ED_1, 780 feet per second, and from 93° to 102° respectively.

Before leaving the impulse velocity diagram, an important feature which is characteristic of impulse turbine diagrams must be emphasised.

The velocity CD of the steam leaving the blade cannot be higher than the velocity AC of the steam entering the blade. The reason for this is the characteristic of impulse turbines, namely, that there is no fall in pressure from inlet to outlet edge of blades. Hence there is no release of energy of the steam whilst it is in the impulse blade passages.

187. The Reaction Blade Velocity Diagram. In the reaction turbine, the nozzles consist of turbine blades which are fixed to the casing. The moving blades and the fixed nozzle blades are identical, but relatively reversed, as shown in fig. 133. As the steam passes through the length of a reaction turbine, the succeeding sets of blades alter slightly in shape, and considerably in height, but so far as a working pair are concerned, they are identical. It is important to note that there is a pressure fall across both fixed and moving blades with consequent release of

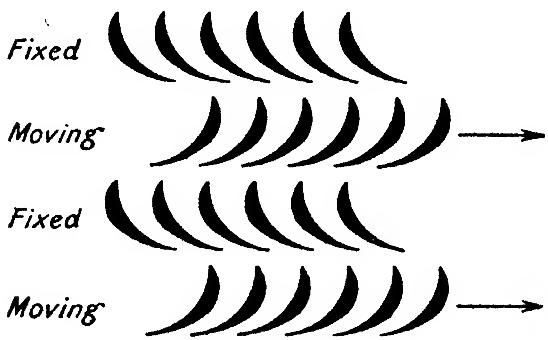


Fig. 133.

energy in each. The release of energy in the moving blade channels causes an increase in steam velocity, whilst the steam is traversing them, and therefore, *the outlet steam velocity relative to the blade is greater than the inlet steam velocity relative to the blade.* The effect of this increase in *relative* velocity is clearly shown in the blade velocity diagram fig. 134, which is lettered to correspond with the impulse diagram of the last article. Note that CD is now greater than AC. Also note that the pressure fall in the moving blade is such as to pull up

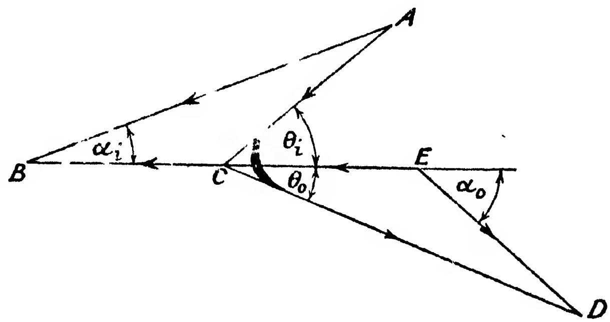


FIG. 134.

the relative velocity from AC to CD so that CD = AB. In other words, the relative velocity of the leaving steam is equal to the absolute velocity of the entering steam. Because the blades are identical, but reversed, the inlet and outlet triangles are identical. Thus BC = CE, AB = CD, AC = ED, $\alpha_i = \theta_o$ and $\theta_i = \alpha_o$.

EXAMPLE. A reaction steam turbine pair has a blade ring 4 ft. 6 in. mean diameter and the turbine speed is 350 r.p.m. The blade speed is 0·7 of the relative leaving velocity of the steam and the leaving edge of the blade is inclined at 20° to the direction of motion. Draw the velocity diagram on the assumption that the steam enters and leaves parallel to the blade face and find (*a*) the absolute steam velocity at entry, (*b*) the relative steam velocity at entry, (*c*) the absolute leaving velocity of the steam, (*d*) the inlet angle of the blades, (*e*) the inlet whirl velocity, (*f*) the exit whirl velocity.

$$\text{Blade speed} = 3{\cdot}1416 \times 4{\cdot}5 \times \frac{350}{60} \text{ feet per second.}$$

$$= 82{\cdot}5 \text{ feet per second.}$$

$$\therefore \text{ Absolute steam velocity at entry} = \frac{82{\cdot}5}{0{\cdot}7}$$

$$= 118 \text{ feet per second.}$$

Only one triangle need be drawn as shown in fig. 135. From this diagram, relative entering velocity = 49 feet per second; absolute leaving velocity = 49 feet per second; inlet blade angle 55·7°; velocity of whirl at inlet 111 feet per second; velocity of whirl at exit = 28·5 feet per second.

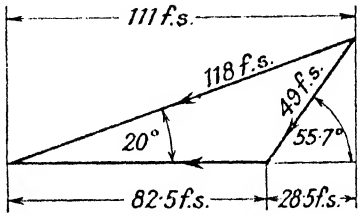


FIG. 135.

188. Calculation of Force exerted and Work done by steam on the blades. To calculate these quantities it is necessary to find the change of momentum of the steam *in the direction of motion of the blades.*

Let V_{wi} = whirl velocity of steam at inlet to blades in ft. per second.

„ V_{wo} = whirl velocity of steam at outlet from blades in ft. per second.

„ W = weight of steam flowing through blade channels in lb. per second.

„ g = engineers unit of mass = 32·2 lb.

Then momentum at entry per second $= \frac{W}{g} V_{wi}$ units

„ „ at outlet „ $= \pm \frac{W}{g} V_{wo}$ units

$\therefore$ Change in momentum per second $= \frac{W}{g} V_{wi} - \pm \frac{W}{g} V_{wo}$

$$= \frac{W}{g} [V_{wi} \pm V_{wo}]$$

This quantity is the force on the blades in pounds. The + sign in the last expression is used when the whirl velocity at outlet is opposite in sign to that at inlet as in figs. 131 and 134. The — sign is used when both have the same sign, as in fig. 132.

Example. Find the force on the blades in the example relating to fig. 135, when 4 pounds of steam flow per second. What effect will doubling the steam weight per second have on the blade force other conditions remaining unchanged?

From fig. 135 inlet whirl velocity = 111 feet per second.

" " outlet " " = 28·5 " " "

The signs are of opposite kind.

$$\therefore \text{ Force on blades} = \frac{4}{32 \cdot 2}[111 + 28 \cdot 5]$$

$$= 17 \cdot 3 \text{ pounds.}$$

The force would be doubled with double steam flow.

The calculation of the work done is now a simple one when the blade speed is known.

Let U = blade speed in feet per second.

Then the work done per second = Force × distance moved per second.

$$= \frac{W}{g}\left[V_{wi} \pm V_{wo}\right] U.$$

This quantity is sometimes called the *diagram work.*

Example. What is the horse power of the ring of blades in the last exercise?

$$\text{Work done per second} = (17 \cdot 3 \times 82 \cdot 5) \text{ ft. lb.}$$

$$\therefore \text{ Horse power developed} = \frac{17 \cdot 3 \times 82 \cdot 5}{550}$$

$$= 2 \cdot 59.$$

189. Efficiency of Turbine Blades. The efficiency of turbine blades is defined as the ratio,

$$\frac{\text{Work done on moving blades per pound of steam.}}{\text{Kinetic energy per pound of steam at exit from nozzle,}}$$

both quantities being in the same units.

Let V_i = absolute velocity of steam at exit from the nozzle in feet per second.

Let V_o = absolute velocity of steam at exit from the blades in feet per second.

Then, when friction is neglected,

Work done on blades per pound of steam

$$= \text{Change in kinetic energy per pound of steam}$$

$$= \frac{V_i^2}{2g} - \frac{V_o^2}{2g}.$$

The latter term is known as the *carry over energy*, whilst the first term is the kinetic energy at exit from the nozzle.

$$\text{Therefore, blade efficiency} = \frac{\dfrac{V_i^2}{2g} - \dfrac{V_o}{2g}}{\dfrac{V_i^2}{2g}}$$

$$= \frac{V_i^2 - V_o^2}{V_i^2}.$$

The work can be calculated by the method of the last article, either taking into account or neglecting friction.

The blade efficiency now becomes

$$\eta_b = \frac{\dfrac{W}{g}\left[V_{wi} \pm V_{wo}\right] U}{\dfrac{W\,V_i^2}{2g}}$$

$$= \frac{2U\left[V_{wi} \pm V_{wo}\right]}{V_i^2}$$

This efficiency is sometimes called the *diagram efficiency* because the work done on the blades is calculated from the blade velocity diagram.

EXAMPLE. The speed of the buckets of a de Laval steam turbine is 1,200 feet per second. The inclination of the nozzle to the plane of the wheel is 20° and the velocity of the steam as it leaves the nozzle, is 3,400 feet per second. If there is no shock, find the angle of the bucket vane at entrance. Assuming no friction and the vane angle at the steam exit to be the same as at entrance, find (*a*) the absolute velocity of the steam as it leaves the buckets, and (*b*) the bucket efficiency. U.L.C.I.

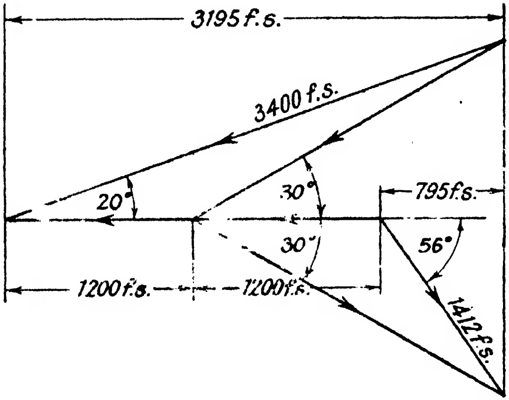


FIG. 136.

From the diagram absolute velocity at exit from buckets
= 1,412 feet per second.

Inlet kinetic energy per pound of steam $= \dfrac{3{,}400^2}{64 \cdot 4}$

$= 179{,}500$ ft. lb.

Outlet kinetic energy per pound of steam $= \dfrac{1{,}412^2}{64 \cdot 4}$

$= 31{,}000$ ft. lb.

$$\therefore \text{Blade efficiency} = \frac{179{,}500 - 31{,}000}{179{,}500}$$

$= 0 \cdot 828$ or $82 \cdot 8$ per cent.

Alternatively : Measurement of the whirl velocity at exit shows it to be 795 feet per second.

$\therefore$ Inlet whirl velocity $= (1{,}200 + 1{,}200 + 795) = 3{,}195$ ft. per second.
Exit ,, ,, $= 795$ feet per sec. and is negative.

$$\therefore \text{Work done on blades} = \frac{1{,}200}{32 \cdot 2} \times (3{,}195 - - 795)$$

$= 148{,}800$ ft. lb.

$$\therefore \eta_b = \frac{148{,}800}{179{,}500}$$

$= 83 \cdot 0$ per cent. as before.

EXAMPLES XXI.

1. **Explain briefly the difference between the steam turbine and the reciprocating steam engine confining your attention to the process of transforming the heat into work.**

2. State the principal advantages of the steam turbine.

3. Explain the terms " impulse " and " reaction " as applied to turbines and also explain why it is necessary to fit a thrust block on a turbine shaft. In which type of machine is the thrust the greater?

4. Sketch a diagrammatic section through a single wheel impulse turbine showing clearly the situation of (*a*) the nozzle chest, (*b*) the nozzles, (*c*) the supply pipe, (*d*) the working blades, (*e*) the blade shrouds, (*f*) the glands, (*g*) the bearings, (*h*) the thrust block, (*i*) the governor driving worm, (*j*) the flexible coupling (*k*) the exhaust passage.

5. Sketch and describe the two principal types of steam nozzle used in turbines and state clearly the circumstances under which each type is used.

6. A steam nozzle is supplied at 150 lb. per square inch absolute and is fully convergent. What is the correct exit pressure if " *n* " in the expansion law is 1·3?

7. Define the terms " nozzle efficiency " and " velocity coefficient." The ideal velocity at outlet from a steam nozzle is 2,800 feet per second. What is the actual velocity of the steam leaving the nozzle if the nozzle efficiency is 0·86?

8. Steam is supplied, dry and saturated, to a turbine nozzle at 120 lb. per square inch absolute the exit pressure being 1 lb. per square inch absolute. Find the ideal velocity with which the steam leaves the nozzle assuming it to be at rest at the inlet section.

9. If the efficiency of the nozzle in the last exercise is 90%, what will be the leaving velocity?

10. The exit area of a nozzle is 2 square inches and the steam velocity is 3,000 feet per second. The volume of 1 lb. of the steam as it leaves the nozzle is 140 cubic feet. Find the weight of steam flowing per second.

11. Steam leaves the nozzles of an impulse turbine with a velocity of 1,185 feet per second, the nozzle angle being 15°. The blade speed is 503 feet per second. Find the relative velocity of the steam entering the blade channels, and the inlet blade angle. If the outlet angle of the blades is 18° and the relative leaving velocity is 575 feet per second, find the absolute leaving velocity and the blade velocity coefficient.

12. The absolute velocity of steam entering a reaction turbine blade ring is 207 feet per second. The jet angle is 20°. Find the velocity of whirl at inlet and also the velocity of flow.

13. Find the work done per pound of steam in example 11, and also the blade efficiency.

14. A reaction steam turbine takes 5·5 lb. of steam per second. The inlet and outlet blade angles are 35° and 20° respectively. The blade speed is 100 feet per second. Draw the velocity diagram and find the horse power developed in one ring of moving blades.

15. What is the increase in steam velocity whilst the steam is passing the moving blade in the last example, and how much heat per pound of steam does this increase represent?

16. Explain the essential difference in principle between the " impulse " and " reaction " turbine. Steam issues from nozzles, inclined at an angle of 20° to the direction of motion with a velocity of 2,400 ft. per sec. on to the blades of a single-stage impulse turbine. The mean diameter of the blade ring circle is 4·5 ft. and the rotor speed is 3,600 r.p.m. The blade tip angles at inlet and outlet are equal. (*a*) Determine the velocity of the blades; (*b*) draw the velocity diagrams for the steam at inlet and outlet from the blade passages assuming no shock at entry and no friction; determine (*c*) the blade angle at inlet and outlet, and (*d*) the work done on the blades per lb. of steam. U.L.C.I.

17. Steam issues from the nozzles of a single-stage impulse turbine at 1,500 ft. per sec. and the nozzles are inclined at 20° to the direction of motion of the blades, the speed of which is 700 ft. per sec. The blade tip angles at inlet and outlet are the same. If the steam is to enter the blades without shock and friction is neglected, determine (*a*) the inlet angle of the blades, (*b*) the force acting on the blades in the direction of their motion, per lb. of steam flowing, and (*c*) the diagram efficiency. U.L.C.I.

18. Steam issues from the nozzles with a velocity of 2,500 ft. per sec., at an angle of 23° on to the blades of a single wheel impulse turbine. The blade tip angles at inlet and outlet are the same and equal to 40°. Draw the velocity diagram (ignoring friction) and determine (*a*) the velocity of the blades, (*b*) the velocity of the steam as it leaves the blades, (*c*) the diagram work per lb. of steam, (*d*) the diagram efficiency. U.L.C.I.

19. In a single wheel impulse turbine, the nozzle angle is 22°, the velocity of the steam as it enters the blades is 2,400 ft. per sec. and the blade speed is 700 ft. per sec. If the blade tip angles at entry and exit are equal, find (*a*) the blade tip angle, (*b*) the change of velocity of the steam whilst passing through the blades, (*c*) the work done per lb. of steam, neglecting friction. What is the velocity of the steam as it leaves the blades? U.L.C.I.

TABLE X

PROPERTIES OF STEAM

Pressure lb./sq. in. absolute	Temperature F.	Specific Volume cu. ft./lb.	Total Heat of Liquid	Latent Heat	Total Heat of Steam	Entropy of Liquid	Total Entropy of dry Steam
p	t	V_s	h	L	H	ϕ_w	ϕ_s
0·5	79·6	643·0	47·6	1,048·5	1,096·1	0·0924	2·0367
0·6	85·3	540·6	53·2	1,045·4	1,098·6	0·1028	2·0214
0·8	94·4	411·7	62·4	1,040·3	1,102·7	0·1196	1·9970
1·0	101·7	334·0	69·7	1,036·1	1,105·8	0·1326	1·9783
2	126·1	173·7	94·0	1,022·2	1,116·2	0·1749	1·9200
4	153·0	90·63	121·0	1,006·7	1,127·7	0·2199	1·8632
6	170·1	61·98	138·1	996·6	1,134·7	0·2473	1·8299
8	182·9	47·35	151·0	988·5	1,139·5	0·2676	1·8065
10	193·2	38·42	161·3	982·5	1,143·8	0·2836	1·7884
14·7	212·0	26·80	180·1	970·6	1,150·7	0·3122	1·7574
16	216·3	24·74	184·5	967·9	1,152·4	0·3187	1·7505
18	222·4	22·17	190·6	964·0	1,154·6	0·3276	1·7411
20	228·0	20·09	196·3	960·4	1,156·7	0·3358	1·7327
25	240·1	16·30	208·6	952·5	1,161·1	0·3534	1·7148
30	250·3	13·73	219·0	945·6	1,164·6	0·3682	1·7004
35	259·3	11·89	228·0	939·6	1,167·6	0·3809	1·6883
40	267·2	10·50	236·1	934·4	1,170·5	0·3923	1·6776
50	281·0	8·516	250·2	924·6	1,174·8	0·4112	1·6597
60	292·7	7·175	262·2	916·2	1,178·4	0·4272	1·6450
70	302·9	6·206	272·7	908·7	1,181·4	0·4412	1·6327
80	312·0	5·472	282·1	901·9	1,184·0	0·4533	1·6219
90	320·3	4·896	290·7	895·5	1,186·2	0·4643	1·6124
100	327·8	4·434	298·5	889·7	1,188·2	0·4742	1·6038
110	334·8	4·046	305·7	884·2	1,189·9	0·4833	1·5963
120	341·3	3·729	312·5	878·9	1,191·4	0·4918	1·5891
130	347·3	3·456	318·8	874·0	1,192·8	0·4997	1·5823
140	353·0	3·222	324·9	869·1	1,194·0	0·5071	1·5762
150	358·4	3·015	330·6	864·5	1,195·1	0·5140	1·5705
160	363·6	2·835	336·0	860·1	1,196·1	0·5205	1·5652

TABLE X. PROPERTIES OF STEAM—*continued.*

Pressure lb./sq. in, absolute	Temperature F.	Specific Volume cu. ft./lb.	Total Heat of Liquid	Latent Heat	Total Heat of Steam	Entropy of Liquid	Total Entropy of dry Steam
p	t	V_s	h	L	H	ϕ_w	ϕ_s
170	368·4	2·677	341·2	855·9	1,197·1	0·5267	1·5602
180	373·1	2·534	346·1	851·9	1,198·0	0·5326	1·5554
190	377·5	2·407	350·[illegible]	847·9	1,198·8	0·5383	1·5509
200	381·8	2·290	355·5	844·0	1,199·5	0·5437	1·5466
220	389·9	2·089	364·2	836·5	1,200·7	0·5537	1·5384
240	397·4	1·918	372·3	829·4	1,201·7	0·5634	1·5311
250	401·0	1·844	376·1	826·0	1,202·1	0·5677	1·5276
260	404·4	1·775	379·9	822·6	1,202·5	0·5721	1·5243
280	411·1	1·651	387·1	816·1	1,203·2	0·5803	1·5179
300	417·3	1·543	394·0	809·8	1,203·8	0·5881	1·5117
350	431·7	1·326	409·9	795·0	1,204·9	0·6058	1·4979
400	444·6	1·161	424·2	781·3	1,205·5	0·6216	1·4857

The properties of steam given in Table X (see page 314) are those calculated by the late Professor H. L. Callendar, and are reproduced by kind permission of Messrs. Edward Arnold & Co. More detailed tables are available and may be obtained on application to Messrs. Arnold.

The liquid heats below 212° F. may be obtained approximately by subtracting 32 from the corresponding temperature. They are, in fact, slightly less, due to the variation in the specific heat of water which is actually less than unity. In practical problems the liquid heat may be taken as equal to the temperature, less 32, in the absence of the table.

The liquid heats are calculated from the equation

$$h = 0{\cdot}99666\,(t-32) + \frac{V_w}{V_s - V_w}\,.\,L - 0{\cdot}0054$$

where t = temperature

V_w = Volume of water /lb.

V_s = Volume of steam /lb.

The evaporation entropy is the difference between the entropy of dry steam and the entropy of the liquid, and is equal to latent heat divided by absolute temperature.

$$\text{i.e. } \phi_e = \phi_s - \phi_w$$

$$\phi_e = \frac{L}{T}$$

$$\therefore \frac{L}{T} = \phi_s - \phi_w.$$

Take the figures at 200 lb. per sq. in. absolute.

$$L = 844{\cdot}0 \text{ B.Th.U.}$$
$$T = 459{\cdot}6 + 381{\cdot}8$$
$$\therefore \phi_e = \frac{844}{841{\cdot}4}$$
$$= 1{\cdot}0030$$
$$\phi_S = 1{\cdot}5466$$
$$\phi_W = 0{\cdot}5437$$
$$\therefore \phi_e = 1{\cdot}0029.$$

Use of the entropy columns to determine dryness after adiabatic expansion.

Case (*a*) dry steam expanding to wet state.

Initial pressure 160 lb. per sq. in. absolute.

Final pressure 20 lb. „ „ „

Total entropy before expansion = 1·5652 Ranks.

„ „ after „ = 1·5652 „

Liquid entropy at 20 lb./sq. in. = 0·3358.

∴ Entropy of evaporation after expansion = 1·5652 — 0·3358.
= 1·2294 Ranks.

Entropy of evaporation when completely dry at 20 lb./sq. in.
= 1·7327 — 0·3358
= 1·3969

$$\therefore \text{Dryness after adiabatic expansion} = \frac{1{\cdot}2294}{1{\cdot}3969}$$
$$= 0{\cdot}887.$$

Case (*b*) steam initially of dryness 0·96.

Evaporation entropy at 160 lb./sq. in. = 0·96 (1·5652 — 0·5205)
= 1·0029 Ranks.

Total entropy before expansion at 160 lb./sq. in. and 0·96 dry
= 0·5205 + 1·0029
= 1·5234 Ranks.

∴ Total entropy after expansion to 20 lb./sq. in. = 1·5234 Ranks.

Liquid entropy at 20 lb./sq. in. = 0·3358

Evaporation entropy after expansion = 1·5234 — 0·3358.
= 1·1846.

Evaporation entropy when completely evaporated at 20 lb./sq. in.
= 1·3969.

$$\therefore \text{Dryness fraction after expansion} = \frac{1{\cdot}1846}{1{\cdot}3969}$$
$$= 0.848.$$

ANSWERS

EXAMPLES I (page 8)

1. 3·9 lb. per sq. ft., 0·0271 lb. per sq. in. **2.** 16·42 lb. per sq. in. **3.** 1,254 ft.-lb. **4.** 28,800 ft.-lb. **5.** 15,552 ft.-lb. **6.** 26,611 ft.-lb. **7.** 3,003 ft.-lb. **8.** 1 sq. in = 3,240 ft.-lb., 34·1 lb. per sq. in.

EXAMPLES II (page 14)

1. 75·6 I.H.P. **2.** 960 H.P. **3.** 606 H.P. and 1,099 lb./hr. **4.** 26,580 lb. **5.** 1 H.P. **6.** 94·6 H.P. **7.** 1,671 ft. per sec. **8.** 180·5 I.H.P., 141 B.H.P. **9.** 350 r.p.m. **10.** 2·3 H.P.

EXAMPLES III (page 23)

1. 1,216° F. **2.** 60·6° F. **3.** 3·843 lb., 126·4 B.Th.U. **4.** 35,900,000 ft.-lb., 1,088 H.P. **5.** 1,395 B.Th.U. per lb. **6.** 2,383 B.Th.U. per lb.

EXAMPLES IV (page 35)

1. 70·2 lb. per sq. in. **2.** 320 cu. ft. **3.** 190·5 cu. ft. **4.** 61·7 cu. ft. **5.** 13·3 cu. ft., 1,925 ft.-lb., 53·47 ft.-lb. **6.** 1 lb. **7.** 131° F. **8.** 5·5 lb. **9.** 29·8 cu. ft., 108·5 lb. per sq. in. **10.** PV = 53·2T, 15·1 B.Th.U. **11.** 96 ft.-lb. per lb. per ° C., 2. **12.** 44·2° C. **13.** 942° C.

EXAMPLES V (page 46)

1. 235,200 ft.-lb. **2.** 212·5 C.H.U., 300 C.H.U. **3.** 25·15 C.H.U. **4.** 119,300 ft.-lb. **5.** 12·64 cu. ft., 0·0791 lb. per cu. ft. **6.** 268 C.H.U., 155,500 ft.-lb. **7.** 29,400 ft.-lb. **8.** 0·1996. **9.** 1,086° C., 860 C.H.U. **10.** 55·76 lb. per sq. in.

EXAMPLES VI (page 62)

1. 37·7 lb. per sq. in. **2.** 4·95 cu. ft. **3.** $PV^{1.087} = C$. **4.** 162,000 ft.-lb. **5.** 15,900 ft.-lb. **6.** 252,000 ft.-lb. **7.** 47,000 ft.-lb. **8.** 63·3 C.H.U. **9.** 123·6 lb. per sq. in., 340° C. **10.** 4·83, 263° C. **11.** 61·3 lb. per sq. in., 232° C. **12.** 104·5 lb. per sq. in., 432° C. **13.** 1·27 cu. ft., 491° C., 10 : 1, 112,000 ft.-lb. **14.** 15·4 C.H.U. wall to gas. **15.** 10,886 ft.-lb., 187·8° C. **16.** 17 cu. ft., 45·5 lb., 410 lb. per sq. in., $\gamma = 1{\cdot}406$. **17.** 2,111° C., 1042° C., 39,320 ft.-lb., 27,800 ft.-lb. **18.** 0·1425 cu. ft., 13·9, 501 lb. per sq. in. **19.** 0·742 lb., 47,500 ft.-lb., 34,100 ft.-lb., 13,400 ft.-lb. rejected.

EXAMPLES VII (page 72)

1. 83 per cent. **2.** 41·2 per cent. **3.** 48·5 per cent. **4.** 184° C., 1,126° C. **5.** 46·5 per cent., 18·4 cu ft., per I.H.P. per hour. **6.** 752 lb. per sq. in., 61·2 lb. per sq. in.

EXAMPLES VIII (page 88)

1. 1,264·4 B.Th.U. **2.** 1,213 B.Th.U. **3.** 540,000 B.Th.U. **4.** 555 lb. **5.** 18·4 lb. **6.** 58,200 ft.-lb., 797,500 ft.-lb. **7.** 333° F. **8.** 852,000 ft.-lb., 576,750 ft.-lb., 59,250 ft.-lb., 216,000 ft.-lb. **9.** 829,000 ft.-lb., 567,000 ft.-lb., 62,400 ft.-lb., 199,600 ft.-lb. **10.** 0·925. **11.** 0·96, 0·92. **12.** 872,000 ft.-lb., 67,200 ft.-lb., 754,400 ft.-lb., 53,600 ft.-lb. **13.** 0·98. **14.** 0·987, 0·269 dry. **15.** 0·981 dry, 37° F. superheat. **16.** 0·953. **17.** 0·53 lb., 0·157 dry. **18.** 0·862.

EXAMPLES IX (page 105)

1. 65·5 per cent., 9·76 lb., 13·65 per cent. **2.** 63 per cent. **3.** 9·79 lb. of steam per lb. of coal, 69·3 per cent. **4.** 78·2 per cent., 15·45 lb., 3·92 lb. **5.** 16·75 per cent., 10,000 lb., 20 per cent. **6.** 3·8 barrels, 10,200 B.Th.U. per lb., 76 per cent. **7.** 11,210 B.Th.U., 1,870 B.T.h.U., 84·1 B.Th.U., 78·5 per cent., 1,129 B.T.h.U. **8.** 78·5 per cent., 13·7 per cent., 74·5 per cent., 11·8 per cent. **9.** 6,184 lb. **10.** 71 per cent., 8,380 lb.

EXAMPLES X (page 122)

1. 0·47 per cent. **2.** 1·03 per cent. **3.** 78·4° F. **4.** 4,250 tubes, 2·37 ft. per sec. **5.** 61,644 gall. per hour. **6.** 11·56° F., 98·7 per cent., 0·507 cu. ft. **7.** 174,000 lb. **8.** 3,190 gall. per min., 794 tubes. **9.** 2,630 gallons, 13·75 lb., 3·1 ft. per sec.

EXAMPLES XI (page 133)

1. 66·8 lb. per sq. in. **2.** 50·9 lb. per sq. in. **3.** 278 I.H.P. **4.** 66·8 lb. per sq. in., 50·9 lb. per sq. in. **5.** 11·41 in. dia., 17·11 in. stroke. **6.** H.P. cyl. 9·48 in. dia., L.P. cyl. 16·42 in. dia and stroke. **7.** 12·76 in. dia., 17 in. stroke. **8.** L.P. cyl. 32·1 in. dia., 36 in. stroke, H.P. cyl. 17·3 in. dia., 36 in. stroke. **9.** 8·7 in. dia., 13·05 in. stroke. **10.** 92·7 in. dia. **11.** 14·26 in dia., 40 B.H.P. approx. **12.** 0·8. **13.** 0·645. **14.** 15·3 in. dia., 3 ft. stroke. **15.** 0·35 of stroke. **16.** 1,635 I.H.P. **17.** 1,380 I.H.P., 9·16 per cent.

EXAMPLES XII (page 149)

1. 1·445 ranks. **2.** 1·068 ranks. **3.** 1·01 ranks. **4.** 0·452 ranks. **5.** 1·580 ranks. **6.** 1·277 ranks. **7.** 1·761 ranks. **8.** 0·89, 23·8 cu. ft. **9.** 130 — 0·86, 110 — 0·73, 90 — 0·62,70 — 0·49, 50 — 0·34, 30 — 0·21.

EXAMPLES XIII (page 162)

1. 127,200 ft.-lb. **2.** 225,000 ft.-lb. **4.** 229,000 ft.-lb., 26·75 per cent. **5.** 264,800 ft.-lb., 28·6 per cent. **6.** 0·815. **7.** 0·84, 53 lb. per sq. in. **8.** 0·00719 ranks, 0·895 dry. **9.** 0·805 dry, 59·2 cu. ft.

Examples XIV (page 172)

1. 33½°. **2.** Lead $\frac{1}{16}$ in., 1 in., ¾ in. **3.** 0·76, 0·96. **6.** 47½°, 1·25 in., 1·95 in. 25°. **7.** 0·3 in., 1¼ in., 0·74 stroke, 0·966 stroke, 0·84 return stroke. **9.** 32°, 1 in., 0·325 in. **10.** 30°, — 4°, 124°, 171°, 310°. **13.** 5·45 in., 1·225 in., 32½°, 0·2 in. **14.** 44°, 0·575 stroke, 0·9 stroke, 0·437 of a rev., 1 in., 1·5 in.

15.

Cut-off	0·7	0·7
Release	0·9	0·9
Outside lap	1·15 in.	0·6 in.
Inside lap	—0·15 in.	0·29 in.
Max. opening steam	0·8 in.	1·25 in.
,, ,, exhaust	1·25 in.	1·25 in.
Lead	0·1 in.	0·65 in.
Angle of advance	39°	39°

Examples XV (page 187)

1. 19,800 B.Th.U. **2.** 13,070 B.Th.U. **3.** 11·9 cu. ft., 13·38 lb., 0·0729 lb. per cu. ft., 1,271 B.Th.U. per cu. ft. **4.** 15,300 B.Th.U., 11·8 lb. **5.** 11·22 lb., 1·475 sq. ft. **6.** 5,680 C.H.U. per lb., 10·83 lb. **7.** 2·86 lb. CO_2, 0·45 lb. H_2O, 14,350 B.Th.U., 13,900 B.Th.U. **8.** 11·6 lb., 34·8 lb., 15·08 lb., 886 B.Th.U. **9.** 11·1 lb. **10.** 35,460 lb. **12.** 16·5 per cent., 1,268 C.H.U., 24 per cent. **13.** 11·64 per cent. CO_2, 76·4 per cent. N_2, 11·96 per cent. O_2. **14.** 43·6 per cent., 12·35 per cent.

Examples XVI (page 201)

6. 18·24 H.P.

Examples XVII (page 214)

11. 2,100° C. abs. **12.** 2,760° C. abs. **13.** (*a*) 1,070° C. abs., (*b*) 812° C. abs. **14.** (*a*) 1928° C. abs., (*b*) 1,460° C. abs. **15.** Expansion ratio 8·9, 758° C. abs. **16.** (*a*) 11·47, (*b*) 6·24. **17.** 0·6255. **19.** 738·2 lb. per sq. in., 22·7 lb. per sq. in., 486·4 lb. per sq. in., 26·9 lb. per sq. in. **20.** 83° C., 727° C., 257·2 C.H.U., 57·5 per cent. **21.** 789° C., 1,839° C., 37·2 per cent.

Examples XVIII (page 235)

1. 41·2 per cent. **2.** £1,212. **3.** 30·4 cu. ft. per I.H.P. per hour. **4.** 32·4 per cent., 41·5 per cent., 0·335 lb. per I.H.P. per hour. **5.** 7·4 lb. per H.P. per hour. **7.** 2,172 lb. per hour. **9.** Ratio 5, 31·4 per cent., 47·6 per cent., 66 per cent. **11.** 11·72 lb. per K.W.H., 8·82 lb. per H.P. hour. **12.** 24·6 per cent. **13.** 22 per cent. **14.** 55·5 I.H.P., 46 I.H.P., 31·8 I.H.P., 22·4 I.H.P. **15.** 13·2 per cent. **16.** 35·2 per cent., 68·9 per cent. **18.** 16·1 per cent., 9·8 per cent. **19.** 5,382 lb. per hour. **20.** 47 per cent., 4·9, 31·8 per cent., 67·7 per cent. **21.** 63·6 per cent., 59 per cent. **22.** 54·4 per cent., 136,800 ft.-lb., 360,600 ft.-lb., 288,700 ft.-lb., 288,700 ft.-lb. **23.** 49 per cent., 148 lb. per sq. in., 147° C.

Examples XIX (page 259)

1. I.H.P. 10·32, B.H.P. 8·06, Mech. Eff. 78 per cent., 42·6 cu. ft. per B.H.P. per hour, 23·25 cu. ft. per I.H.P. per hour, 25·4 per cent., 13·86 per cent., 59 per cent. (Brake heat 132·8 + Friction heat 110·9) = Indicated heat 243·7, Jacket heat 351, Exhaust, etc., 365·3, Heat supplied 960 C.H.U. **2.** I.H.P. hours 3·708, B.H.P. hours 1·79, 26·2 per cent., 12·6 per cent. (Brake heat 84·6 + Friction heat 90·3) = Indicated heat 174·9, 26·2 per cent.; Jacket heat 92·4, 13·8 per cent.; Exhaust, etc., 399·9, 60 per cent.; Heat supplied 667·2, 100 per cent. **3.** 79·8 per cent., 34·25 per cent., 44·8 per cent., 20·95 per cent. **4.** I.H.P. 18·15, 20 per cent., Heat supply = 128,000 C.H.U. (Brake heat 20,350 + Friction heat 5,250) = Indicated heat 25,600 C.H.U., Jacket heat 23,800, Exhaust heat 78,600.

6. I.H.P. 1,723, 0·326 lb., 13,480 lb. **7.** 82·9 per cent., 43·6 per cent., 36·1 per cent. **8.** I.H.P. 180·3, B.H.P. 140·5. **9.** Heat supply 31,400 C.H.U. (Brake heat 6,225 + Friction heat 1,980) = Indicated heat 8,205, Jacket heat 13,850, Exhaust, etc., 9,345. **10.** I.H.P. 11·65, B.H.P. 9·6, 82·5 per cent., 47·3 cu. ft. per hour, 10·2 per cent. **11.** Heat supplied 60,200 C.H.U. (Brake heat 8,630 + Friction heat 2,120) = Indicated heat 10,750, Jacket heat 24,500, Exhaust, etc., 24,950. 17·9 per cent., 14·33 per cent. **12.** 29·6 per cent., Heat supply 201,000 C.H.U., Jacket heat 56,400, Exhaust heat, 59,500 (Brake heat 59,500 + Friction heat 25,600) = Indicated heat 85,100. **13.** 81·8 per cent., 30 per cent., 24·5 per cent., Heat supply 1,980 C.H.U. (Brake heat 485 + Friction heat 108·5) = Indicated heat 593·5, Jacket heat 450, Exhaust heat 855, Radiation, etc., 81·5. **14.** 71·5 per cent., 23 per cent., 13·9 per cent. **15.** Heat supplied per min. = 4,175 B.Th.U. (Brake heat 1,290 + Friction heat 797) = Indicated heat 2,087, Jacket heat 1,365, Exhaust heat, etc., 723. **17.** 19·45, 16·7, 86 per cent., 22·1 per cent.

Examples XX (page 273)

3. 54 to 76·4 r.p.m. **4.** 7·46 in. **5.** 219 r.p.m. **6.** 230 r.p.m., 249 r.p.m. **7.** 130 r.p.m., 170·5 r.p.m. **9.** 20 lb. **10.** 2·5 lb. **11.** W = 5·93*w*, 4·35 per cent.

Examples XXI (page 311)

6. 81·86 lb. per sq. in. abs. **7.** 2,598 ft. per sec. **8.** 4,010 ft. per sec. **9.** 3,800 ft. per sec. **10.** 0·297 lb. **11.** 710 ft. per sec., 26°, 185 ft. per sec., k = 0·81. **12.** 194 ft. per sec., 71 ft. per sec. **13.** 18,500 ft.-lb., 85 per cent. **14.** 9·76 H.P. **15.** 90 ft. per sec. increase, 0·648 B.Th.U. **16.** (*a*) 848·2 ft. per sec., (*c*) 30·2°, (*d*) 74,110 ft.-lb. **17.** (*a*) 36°, (*b*) 44·07 lb., (*c*) 88·2 per cent. **18.** (*a*) 1,137 ft. per sec., (*b*) 979 ft. per sec., (*c*) 82,200 ft.-lb., (*d*) 84·8 per cent. **19.** (*a*) 30½°, (*b*) 3,050 ft. per sec. in. direction of motion of blades, (*c*) 66,300 ft.-lb., 1,221 ft. per sec.

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