THE

OXFORD ENGINEERING SCIENCE SERIES

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E. B. MOULLIN D. R. PYE R. V. SOUTHWELL

THE OXFORD ENGINEERING SCIENCE SERIES

General Editors: E. B. Moullin, D. R. Pye, R. V. Southwell

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THE INTERNAL COMBUSTION ENGINE

VOLUME I

BY

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SECOND EDITION

OXFORD UNIVERSITY PRESS
LONDON: GEOFFREY CUMBERLEGE

Oxford University Press, Amen House, London E.C. 4
EDINBURGH GLASGOW NEW YORK TORONTO MELBOURNE
WELLINGTON BOMBAY CALCUTTA MADRAS CAPE TOWN
Geoffrey Cumberlege, Publisher to the University

FIRST EDITION 1931
SECOND EDITION 1937
Reprinted photographically in Great Britain in 1941, 1943, 1946, and 1948 by LOWE & BRYDONE, Printers, Ltd., London from sheets of the second edition

PREFACE TO THE SECOND EDITION

THE temptation to any author called upon for a second edition must always be to bring his book up to date by expanding upon the scheme of the first edition and producing a larger book. Being without the leisure to undertake extensive rewriting, however, I have had to content myself in this issue with providing for essential modifications and with fairly extensive references to original sources when dealing with recent developments.

There are two important directions in which the edition of 1931 has been brought up to date. The chapter on Detonation has been very largely rewritten in the light of recent research, and the section dealing with specific heats, dissociation constants, and the calculation of combustion temperatures, although retaining its original form, has been entirely revised so far as numerical quantities are concerned. New articles have been added on non-volatile liquid fuels and on their combustion in the compression-ignition engine.

In the section on the calculation of combustion temperatures I have been fortunate in having the generous assistance of Mr. R. W. Fenning. He has carried through the complete set of recalculations based upon the most recent work on heat capacities and dissociation constants. It would have been quite beyond me, in present circumstances, to have done this revision myself, and any value the book may have as a work of reference in this connexion it owes, therefore, to Mr. Fenning, and I am indebted to him accordingly. I am indebted to him, also, for kindly reading the proofs of the rest of the book and for valuable criticisms thereon.

D. R. P.

LONDON, 1937.

PREFACE TO THE FIRST EDITION

One may set out to write a book upon the Internal Combustion Engine from one of two points of view. The aim may be chiefly to set down the principles which underlie the design and operation of the engine, or to supply information upon a host of details which the practising designer must have at call. The two aims do not combine well in the same book, and the author who attempts to achieve completeness in both is likely to succeed in neither.

I have approached the subject from the first point of view and in this first volume have mentioned details of design only when they influence or illustrate principles. The book is therefore primarily for the student, and for the engineer before he has become immersed in all the intricacies of manufacture; in the use and behaviour of special materials; in clearances and tolerances; and in the all-governing question of the cost of production.

Although dealing with principles rather than practice, the book aims, nevertheless, at giving the groundwork of a practical knowledge of the subject. In order to keep the length within reasonable limits I have included only so much basic theory as seemed to me essential to the subsequent handling of the practical problems. For a fuller treatment of the theoretical side of the subject the reader should consult Ewing's *Thermodynamics for Engineers*, and work on from that to the fuller treatises on thermodynamics.

No one could write nowadays on the internal combustion engine without drawing largely upon the work of Mr. H. R. Ricardo, F.R.S. I have been fortunate in being closely in touch with him and his work for many years, and however frequently his name may occur in this book, that can by no means represent my full indebtedness to him. I am indebted also to Mr. H. T. Tizard, C.B., F.R.S., for many suggestions at various stages during the progress of the book, and to Dr. A. F. Burstall for kindly reading the proofs and making many valuable criticisms.

In a later volume I hope to deal with some of the special problems of the aero-engine, in so far as these involve general principles; and also with the general problem of engine cooling by air or liquids, as it arises in its most acute form in the supercharged engine.

D. R. P.

London, 1931.

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UNITS AND SYMBOLS USED IN THIS BOOK

V represents volume in cubic feet.
P ,, pressure in pounds per square foot.
t ,, temperature in degrees centigrade.
T ,, ,, ,, absolute.
$T = t+273$.
B.H.P. represents Brake Horse-Power.
I.H.P. ,, Indicated Horse-Power.
B.M.E.P. ,, Brake Mean Effective Pressure.
I.M.E.P. ,, Indicated Mean Effective Pressure.

The centigrade temperature scale will be used throughout.

The unit of heat will be the 'pound-degree-centigrade' or Centigrade Heat Unit (C.H.U.). This is defined as the quantity of heat necessary to warm one pound of pure water through 1° C., from 14° to 15° C.

The unit of work, or energy, will be the foot-pound.

Joule's equivalent will be represented by J and has the value 1,400 with the other units adopted, so that

$$1 \text{ C.H.U.} = J \text{ foot-pounds} = 1{,}400 \text{ ft.-lb.}$$

The unit quantity of a gas will be either one Standard Cubic Foot (S.C.F.), that is, the quantity which occupies 1 cubic foot at N.T.P., or one Mol.

When dealing with the specific heats of gases, these will be expressed in terms of foot-pounds per S.C.F., or, from time to time when convenient, in C.H.U. per S.C.F. and C.H.U. per mol.

One Pound-Molecule, or Mol, of a gas is that quantity which occupies a volume of 359 cubic feet at N.T.P. and has a weight equal to the molecular weight of the gas. (See art. 4.)

I

INTRODUCTORY

ART. 1. The efficiency of heat engines

All heat engines which are to act as prime movers consist of certain rigid parts which go through regular cycles of movement, and, associated with them, a gaseous 'working substance' which constantly becomes used up and has to be renewed. It is the function of the heat engine to produce mechanical effort by the conversion of heat into work; and it is the part of the working substance to be the vehicle by which heat is taken into the engine at certain points of its cycle of operations, and rejected at others. The heat which the engine has been unable to convert into work is rejected in the expelled working substance, of which the usefulness is over, and the fraction of the whole heat energy supplied with the working substance which has been converted to its equivalent work is the thermal efficiency of the heat engine.

The heat supply is derived as a rule from the combustion of fuel, and, since the fuel has to be bought, the price to be paid for our motive power will depend upon what fraction of the fuel heat, be it from gas, petrol, or coal, is recoverable as power at the crankshaft or turbine rotor, that is to say upon the overall thermal efficiency of the power plant.

It will be assumed in this book that the reader is familiar with the elements of thermodynamics, and in particular with Carnot's demonstration that the efficiency of the heat engine depends upon the range of temperature through which the working substance can be cooled during a cycle of operations. In Carnot's imagined engine all the heat is received by the working substance at a single high temperature T_1 absolute, and all the heat that is rejected is rejected at a single low temperature T_2. Carnot shows that these conditions are necessary to get the greatest possible efficiency, and that when they are fulfilled the highest efficiency which any imagined engine working between these temperatures could possibly achieve is given by the fraction

$$\frac{T_1-T_2}{T_1}.$$

Although the conditions for Carnot's ideal can never be fulfilled in practice, and actual efficiencies are in consequence well below that of the Carnot engine for the same highest and lowest temperatures, yet the importance of a wide temperature range, as foreshadowed by the Carnot analysis, is fully confirmed in practice.

It is of some interest to compare the internal combustion engine, from this point of view, with heat engines employing steam as the working substance. We shall see that combustion of the fuel actually within the engine cylinder allows of a much wider range between the highest and lowest temperatures in our heat engine, and we can anticipate, therefore, a higher efficiency in the conversion of the fuel heat into mechanical energy.

Steam-using engines are handicapped, in the first place, by the fact that all heat has to be got into the steam through metal walls capable of withstanding high pressures. Until recently this limited maximum steam temperatures to about 300° C., and although materials have improved very much of late in regard to maintenance of strength at high temperatures, so that a steam pressure of 1,900 lb. per sq. inch is now contemplated in conjunction with a temperature of 500° C., as an upper limit of temperature this is yet about two thousand degrees below what the internal combustion engine is accustomed to. Here neither temperature nor pressure become high until the working substance is inside the engine cylinder, and heat is then generated with practically no loss, in the very body of the gas to be heated. The only limits to the maximum temperature are the quantity of heat which combustion of all the oxygen present can generate, and the specific heat of the gases to be heated. The actual limit imposed by these factors is about 2,800° C.

So far as T_1 in the Carnot formula is concerned, therefore, the internal combustion engine is at a very great advantage. There may appear at first sight to be no reason why the lower limit of temperature should not be the same for both, namely, the average temperature of the surroundings of the engine. The steam turbine, with its high vacuum condenser at a temperature of 40° C., does in fact succeed in expanding its working substance almost down to the temperature of its surroundings. Any one who has stood near the exhaust pipe of a high duty internal combustion engine must have been struck with the contrast. The working substance roars out at a temperature of the order of 1,000° C., carrying with it a great amount of heat to sheer waste. Many attempts have been made to utilize some of the heat in the exhaust gases, and so to cool them below the minimum temperature which is practically possible by simple expansion. These attempts have been on the whole unsuccessful hitherto, and we may take 500° C. as about the lowest temperature attainable for the exhaust gas under full load conditions, in an internal combustion engine, when expansion is carried out in a single cylinder. Lower temperatures might be reached if the expansion of the working substance were extended by doing it in two stages, first in a high- and then in a low-pressure cylinder. The high average temperatures, however, of the internal combustion cycle impose very strict limitations upon what can be done with the working substance. Transference from one cylinder to another, of gases at temperatures over 1,000° C., is a very different problem from that of doing the same with steam at 150° C. It is a primary necessity, when high gas temperatures are to be associated with a moving, and therefore lubricated, mechanism, to retain the greatest possible simplicity of mechanical arrangement. There have been very many attempts to obtain improved thermal efficiencies which have failed, through an omission to keep the relentless need for mechanical simplicity as a guiding principle. Compound expansion is certainly a possibility for the internal combustion engine, but hitherto it cannot be said that any design has shown much promise of combining the improved efficiency which it seeks, with the reliability which it cannot do without.

If we compare, then, the ranges of temperature available in the internal combustion cycle as it is to-day, with that in the most modern steam turbine plants, using a high vacuum condenser, we have

in the internal combustion engine . . . $\left\{\begin{array}{l} T_1 = 3073 \text{ abs.} \\ T_2 = 773 \text{ ,,} \end{array}\right.$

in the steam turbine . . . . . . $\left\{\begin{array}{l} T_1 = 773 \text{ ,,} \\ T_2 = 313 \text{ ,,} \end{array}\right.$

The corresponding Carnot efficiencies are:

for the internal combustion engine . .	75	per cent.
for the steam turbine	59·5	,, ,,

This last efficiency, moreover, applies only to the turbine itself. From even the most efficient boilers not more than 85 per cent. of the fuel heat reaches the turbine in the steam, and the overall efficiency of the plant will be only the product of those for turbine and boiler separately.

The highest brake thermal efficiencies achieved in practical working up to the present day are, for a turbine, 33·8 per cent., and for an internal combustion engine at its full load output 40 per cent. As pointed out above, however, it is the overall efficiency of the turbine power plant, namely $33{\cdot}8 \times 0{\cdot}85 = 28{\cdot}7$ per cent., which gives a figure directly comparable with the brake thermal efficiency of the internal combustion engine, where the whole process of power production is completed within the boundaries of the engine itself, from the chemically stored energy of the fuel to the turning effort available at the shaft.

A low consumption of fuel per shaft horse-power, of course, is only one of the important factors in the economics of power production. The cost of the fuel and the first cost of the plant, besides repairs and labour, have all to be considered, and in fairness to the steam turbine it must be admitted that the cost per ton of the fuel which it consumes is not more than about one-fifth of that of the fuels suitable for burning in the internal combustion engine, with their necessary purification to avoid leaving any ash, or other undesirable residual products. Furthermore, the internal combustion engine is complicated when compared with the steam turbine, which is about the simplest possible piece of mechanism, with its one, perfectly balanced, rotor. And the turbine can be built, too, in units up to 50,000 horse-power without introducing problems of design much more difficult to deal with than in units of small size; whereas the thermal and lubrication problems of the internal combustion engine increase very seriously in the larger sizes.

For large power plants where ample floor space is available and the load is a fairly regular one, the internal combustion engine will not be able to compete with the steam turbine for many years to come. Its field of usefulness is rather for the smaller stationary, and the mobile, power unit, of which frequent stopping and restarting is required, and from which a large output of power is demanded, combined with a limitation upon space and weight.

Art. 2. The working substance

A study of the working substance of the internal combustion engine is concerned only with the gaseous and liquid phases of matter, and of these chiefly with the former. It is true that attempts have been made to burn pulverized solid fuel in an engine cylinder, but the project has not reached a point when it is of practical importance. We shall deal from time to time with the behaviour of substances as they change from the liquid to the gaseous phase, but for the most part we shall be concerned with substances so far removed from the liquid state that they behave like the permanent gases.

In either turbine or piston engine the working substance goes through a series of changes of pressure and volume, during which it becomes cooled, and, in the process, imparts mechanical effort to the moving parts of the machine. When it is steam, the working substance remains chemically the same throughout all stages as it passes through the machine, although it changes from the physical state of superheat, when it approximates in behaviour to a perfect gas, to the state of a partially condensed vapour. The mixture of gases which compose the working substance of an internal combustion engine, on the other hand, remain throughout far removed from the state of vapour, but, unlike steam, they do not remain chemically the same throughout a cycle of the engine. The air and fuel taken into the cylinder become, after combustion, nitrogen and the products of the chemical reactions involved in the burning of the fuel. Broadly speaking, the combustibles are carbon and hydrogen, and about one-quarter of the cylinder contents by weight, after combustion, consists of carbon dioxide (CO_2) and water vapour (H_2O). The latter is, of course, well above its critical temperature, and behaves like a permanent gas. During combustion there is in general a change in the total number of reacting molecules present. This has to be taken into account in engine calculations and may be as much as 5 per cent. of the total, and either positive or negative, according to the chemical nature of the fuel.

The fact that the internal combustion engine does not, like the steam engine, have to carry a complete supply of its working substance with it, but picks up by far the greater part of it from its surroundings and only has to carry with it $\frac{1}{12}$ to $\frac{1}{15}$ of the total weight in the form of fuel, is of the highest importance in aircraft where a minimum weight of the power plant, and of its supplies, is essential.

ART. 3. The idea of a perfect gas

In making calculations about the behaviour of the real gases, it is convenient to imagine a gas which conforms exactly to certain laws which are only very nearly true of the actual gases we are dealing with. This ideal type of gas, about which it is justifiable to make exact calculations, we call a 'perfect' gas. The common gases, hydrogen, oxygen, nitrogen, while very nearly identical in behaviour with the perfect gas, all show slight deviations at ordinary temperatures; and these deviations become more and more pronounced at very low temperatures and very high pressures, as an approach is made towards the conditions at which liquefaction would take place.

The pressure, volume, and temperature relationships of a perfect gas which conforms to the laws of Boyle and Charles are summarized in the well-known formula

$$PV = RT \tag{1}$$

in which R is a constant. The numerical value of this constant depends upon the molecular weight of the particular gas in question, and upon the quantity of gas of which the volume is represented by V.

According to Avogadro's principle, equal volumes of all gases at N.T.P. contain equal numbers of molecules. It follows that if we have a series of quantities of different gases, each quantity having a weight in pounds equal to the molecular weight of the gas, then each of these quantities will comprise the same number of molecules and will occupy the same volume. For example, 2 lb. of hydrogen, 32 lb. of oxygen, and 28 lb. of nitrogen will each occupy the same volume; and this volume at N.T.P. is found to be 359 cubic feet. The average molecular weight of air, which is almost entirely composed of oxygen and nitrogen in the proportion 1:3·76,[†] is 28·95, and this number of pounds of air at N.T.P. will, therefore, also occupy 359 cubic feet.

If, now, the volume V in the equation $PV = RT$ be taken always as the volume in cubic feet of 1 lb. of dry gas, we can calculate the value of R for 1 lb. of air by putting in the appropriate numerical values, as follows:

$$14{\cdot}7 \times 144 \times \frac{359}{28{\cdot}95} = R.273,$$

whence

$$R = \frac{14{\cdot}7 \times 144 \times 359}{273 \times 28{\cdot}95} = 96{\cdot}0,$$

and the equation

$$PV = 96T \tag{2}$$

can be used to determine the state of 1 lb. of dry air, assuming it to behave like a perfect gas, when any two of the conditions P, V, and T are known. In this equation it must always be remembered that the units are pounds, feet, and degrees centigrade *absolute*.

In order to derive the appropriate equation for any other gas it is convenient to have equation (1) in the more general form

$$PV = \frac{GT}{m}, \tag{3}$$

[†] More exactly, following Kaye and Laby, the percentages by volume are: oxygen 21·0, nitrogen 78·05, argon 0·95, carbon dioxide 0·03 to 0·3, and traces of rare gases.

in which G is a constant having the value 2779, using the same units, and m is the molecular weight of the gas. The constant G, having the same value whatever gas is represented, is known as the 'Universal Gas Constant', and in order to derive the value of R for any particular gas, this universal constant must be divided by its molecular weight. Thus for air:

$$R = \frac{G}{m} = \frac{2779}{28{\cdot}95} = 96{\cdot}0. \tag{4}$$

The constant R represents a certain quantity of 'energy per unit mass' corresponding to a certain change of temperature, and the units in which it is measured are 'foot-pounds per pound per degree centigrade'. This can easily be seen if one imagines 1 lb. of air to be contained in a cylinder and heated, while at constant pressure P_1, from T_1 to T_2, so that it expands from V_1 to V_2.

It will then do a number of foot-pounds of work equal to

$$P_1(V_2-V_1) = R(T_2-T_1)$$

by equation (1), and it follows that if the heating had been such that the rise of temperature (T_2-T_1) was just 1° C., then the work done, $P_1(V_2-V_1)$, would have been equal to R. The constant R in equation (1) therefore expresses the work done in foot-pounds by 1 lb. of air while heated through 1° C. under constant pressure conditions.

The constant G is also expressed in foot-pounds per degree centigrade, but refers to a different quantity of gas. If equation (3) above be written

$$P(mV) = GT, \tag{3 a}$$

it becomes clear that G expresses the work done in foot-pounds by m pounds of gas while being heated through 1° C., just as R in equation (1) expressed the same thing for one pound. The unit of quantity where G is concerned is therefore that same quantity as was mentioned earlier, which has a weight in pounds equal to the molecular weight, and a volume at N.T.P. equal to 359 cubic feet. This unit of quantity is of so much importance that it deserves a short article to itself.

Art. 4. The pound-molecule or mol.

There is a unit of quantity for gases known as the 'gram-molecule' which is familiar to all those who employ the C.G.S. system of units. This unit of quantity has a weight in grammes equal to the molecular weight, and a volume at N.T.P. of 22,412 cubic centimetres. Its equivalent in British units is that quantity, introduced in the last article, which has a weight in pounds equal to the molecular weight and a volume at N.T.P. of 359 cubic feet.

It will be important to get a really clear and concrete idea of this unit of quantity of a gas, and it is helpful, in doing so, to think of it as being roughly a 7-foot cube when at N.T.P. This 7-foot cube (really 359 cubic feet), by Avogadro's principle, will always contain the same number of molecules whatever the gas, and will weigh 2, 32, 28, etc. pounds, according to whether the gas is hydrogen, oxygen, nitrogen, or whatever. This unit of quantity, the pound-molecule, will be called, for brevity, the 'mol.'.

Reverting for a moment to equation (3 a) of the last article, we see that (mV) represents the volume of 1 mol. of the gas at any temperature T and pressure P, and an equation $PV = GT$ is therefore the equation of state of any gas when the unit of quantity is the mol., just as $PV = RT$ was the equation when 1 lb. was involved. But whereas R would be different for 1 lb. of each gas, according to its molecular weight, the constant G is the same for all gases. In $PV = RT$ the quantity of gas involved has always the same weight but various volumes at N.T.P. In $PV = GT$ it has always the same volume at N.T.P. but various weights.

The numerical value of G, as given in the last article, is 2,779 ft.-lb. per mol. per 1° C. In C.G.S. units its value is 84,750 gram-centimetres or 1·985 gram-calories per gram-molecule per 1° C. It should be noted also that the same figure, 1·985, expresses G in C.H.U. per pound molecule. It is useful to remember that all figures for heat quantities in C.H.U. per mol. in British units are numerically the same as in C.G.S. units.

ART. 5. The specific heats of a gas

When unit mass of any substance has its temperature raised by 1° C., a certain amount of heat is required; but the magnitude of the amount will depend upon other conditions apart from the rise of temperature. In particular, when it is a gas that is being heated, the amount of heat required to raise the temperature 1° C. will be greater, when the gas is heated under conditions so that it can expand while the pressure is maintained constant, than when it is contained in a closed vessel at constant volume.

We shall be concerned, in practical calculations, with the heating and cooling of gases under conditions in which neither the pressure nor the volume remain constant, but to enable us to deal with this more complicated case it is necessary to consider first the two special cases of heating at constant volume and constant pressure.

When 1 lb. of gas is heated at constant volume from T_1 to T_2, all the heat communicated to it must go to increase the stock of internal energy in the gas, and if we use the symbol K_v for the specific heat under these conditions, then

$$\begin{aligned}\text{heat communicated} &= \text{increase of internal energy}\\ &= K_v(T_2-T_1)\ \text{C.H.U.}\end{aligned}\tag{5}$$

When heated under conditions of constant pressure P, the gas will expand, say from V_1 to V_2, and in so doing will have to do external work (e.g. in pushing a piston out) equal to $P(V_2-V_1)$ ft.-lb. If an equivalent amount of heat were not being communicated from outside, this external work could only be done by drawing upon some of the internal energy in the pound of gas. Under the conditions of the imagined experiment, therefore, in which the temperature is to be raised from T_1 to T_2, and at the same time external work $P(V_2-V_1)$ has to be done, the amount of heat to be added, if K_p is the specific heat under these conditions, must be

$$\begin{aligned}K_p(T_2-T_1) &= \text{increase of internal energy}\\ &\qquad +\text{heat equivalent of external work done}\\ &= K_v(T_2-T_1)+\frac{P}{J}(V_2-V_1).\end{aligned}\tag{6}$$

In this equation the last term represents the external work done expressed in heat units. The increase of internal energy which, by Joule's Law, to be discussed in the next article, is always the same for the same rise of temperature, is represented in equation (6) by the same expression as it was in equation (5) when the heat was added to a quantity of gas contained in a vessel of constant volume.

Since, by equation (1),

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

it follows that

$$K_p-K_v = \frac{R}{J},\tag{7}$$

which relates the difference between the two specific heats of any gas to the constant R in the equation of state; and this, it will be remembered, is the universal gas constant G divided by the molecular weight m of the particular gas. When the mol. is the unit of quantity, and specific heats are expressed in C.H.U. per mol. per degree centigrade, then

$$K_p - K_v = \frac{G}{J}.$$

The ratio of the two specific heats K_p/K_v is a quantity of great importance in thermodynamics, and is denoted by the symbol γ.

A further simple relationship which follows at once is

$$\gamma - 1 = \frac{R}{J.K_v}. \tag{8}$$

When we are dealing with 1 lb. of air at ordinary temperature, the values of K_p and K_v are found by experiment to be

$K_p = 0{\cdot}2413$ C.H.U. per lb., or 27·24 ft.-lb. per S.C.F.

$K_v = 0{\cdot}1727$,, ,, ,, ,, 19·50 ,, ,, ,,

The ratio γ for air is therefore 1·396, and the difference $K_p - K_v$ which equals 0·0686 C.H.U. per lb., corresponds with the values $R = 96{\cdot}0$ and $J = 1400$.

In engine calculations it is usually more convenient to work with units of volume rather than mass, and to take, therefore, the standard cubic foot or the mol. as the unit of quantity. It is customary then to speak of 'volumetric heat' rather than specific heat, and to express this either as C.H.U. per mol. or ft.-lb. per S.C.F. The values for air given above when expressed as C.H.U. per mol. become

$$K_p = 6{\cdot}995$$

$$K_v = 5{\cdot}01$$

$$\gamma - 1 = \frac{G}{J.K_v} = \frac{2779}{1400 \times 5{\cdot}01} = 0{\cdot}396.$$

ART. 6. Internal energy. Joule's Law

According to the First Law of Thermodynamics 'when mechanical energy is produced from heat a definite quantity of heat goes out of existence for every unit of work done; and, conversely, when heat is produced by the expenditure of mechanical energy the same definite quantity of heat comes into existence for every unit of work spent'.

When the hot working substance expands in an engine cylinder and does work in pushing the piston out, it follows from the above law that a corresponding amount of heat energy must have gone out of existence. If we suppose for the moment that there has been no interchange of heat with the cylinder walls, it follows that the heat which has disappeared must have been drawn from the stock of heat energy which was inherent in the working substance before expansion began; in other words from its internal energy.

Any quantity of gas in a definite state has a definite quantity of internal energy associated with it. There is no way of assessing the total stock of energy contained by a gas, but we shall see later that the amount of it depends only upon the temperature, and we shall therefore be on safe ground in choosing an arbitrary zero and saying that the stock of internal energy of a gas is the amount over and above what it contains at 0° C.

The gas might have been losing or gaining heat through the cylinder walls while the expansion was going on, and in that case the loss of internal energy would have been the algebraic sum of the work done and this external loss or gain.

In writing down the equation which expresses the energy balance in symbols it is customary to make the symbols represent a gain of heat from the walls and a gain of internal energy. We then have

Heat received by the working substance
= Heat equivalent of work done by the piston
+gain of internal energy.

If these three quantities are represented by the symbols Q, W, and E, then for a small movement of the piston during which an amount of external work dW is done, the energy equation will be

$$dQ = \frac{dW}{J} + dE \qquad (9)$$

which equation is the expression in symbols of the First Law of Thermodynamics, enunciated at the beginning of this article.

For adiabatic expansion $dQ = 0$, and therefore

$$\frac{dW}{J} = -dE \qquad (10)$$

which expresses the fact stated above, that when there is no heat interchange with the cylinder walls the amount of external work done by the expanding gas is equal to the loss of internal energy which it suffers.

According to Joule's Law of Internal Energy: 'The internal energy of a given quantity of a gas depends only upon its temperature.'

Since pressure, volume, and temperature completely define the state of a quantity of gas, the only other factors upon which the internal energy *could* depend are the pressure and volume. Joule, in his well-known experiment, allowed a gas to change its pressure and volume in such a way that it did no external work and neither received nor gave out heat; and in consequence could not have changed its stock of internal energy. It was found that in these circumstances its temperature, after things had settled down, had not changed. The inevitable conclusion is that there is a certain temperature which is always associated with a certain stock of internal energy, and that the energy cannot be altered, no matter what changes of pressure or volume take place, unless the temperature is altered. In other words, the internal energy depends upon the temperature, and upon nothing else.

The temperature of a gas on the kinetic theory is defined in terms of the speed of the individual molecules, and the physical interpretation of Joule's law is that all the internal energy which a gas in a definite state possesses, it holds in virtue of the independent movements of its molecules, and that no part is associated with mutual forces between the molecules and therefore with their average distance apart.

As with the other gas laws, there is a slight departure from Joule's Law on the part of all the real gases, and it can only be taken as being strictly true for the imaginary perfect gas which conforms to the laws of Boyle and Charles. The departures of real gases from Joule's Law are not important in internal combustion engine theory, but the amounts of these departures, which are defined by what is called the Joule-Thomson effect, are very valuable as a means of exact investigation of the properties of the real gases.

ART. 7. The representation of pressure-volume changes

It is not proposed to deal fully in this book with all the various conditions under which a gas may change its state, nor with the equations and diagrams representing these changes, but only to summarize the more important formulae and results and to assume that the reader is already familiar with the proofs, which are given in all text-books on heat.

We have first the equation

$$PV = RT,$$

already given in art. 3, which represents with sufficient accuracy the relationship between the pressure, volume, and absolute temperature of a given quantity of any of the gases with which we shall be concerned.

Whenever the pressure and volume of a given quantity of gas are altered by compression or expansion in an engine cylinder, and whether the gas be gaining or losing heat while the change takes place, it is found that the pressure and volume are related by an equation of the form

$$PV^n = C. \tag{11}$$

Such a relationship between P and V is shown plotted as the full line on the diagram, fig. 1. When such a pressure-volume change takes place in a cylinder against a moving piston, the work done by the expanding gas (or upon it, if the change be a compression) is represented by the shaded area under the curve, and in symbols is equal to

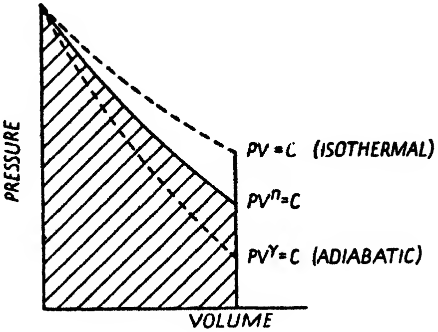

Fig. 1. Various forms of expansion curves.

$$\int_{V_1}^{V_2} P\,dV = \frac{P_2V_2-P_1V_1}{n-1}. \tag{12}$$

The value of the constant n depends upon two things: (1) the specific heats of the gas of which the pressure and volume are changing, and (2) whether the gas is gaining or losing heat by conduction through the cylinder walls during the change. The numerical values to be given to n for practical engine calculations are fully considered in art. 11. There are, however, two special conditions known as 'isothermal' and 'adiabatic' expansion or compression, which are of great theoretical importance, although no complete pressure-volume change in an actual engine conforms, even approximately, to either condition. During an isothermal change $n = 1$, and for an adiabatic change $n = \gamma$, where γ is the ratio of the specific heats of the gas at constant pressure and constant volume. (See art. 5.)

The expressions for the work done during a change from V_1 to V_2 under these two special conditions are:

Isothermal expansion:

$$\text{work done} = \int_{V_1}^{V_2} P\,dV = RT_1 \int_{V_1}^{V_2} \frac{dV}{V} = RT_1 \log_e \frac{V_2}{V_1}. \tag{13}$$

Adiabatic expansion:

$$\text{work done} = \frac{P_2 V_2 - P_1 V_1}{\gamma - 1} = \frac{R}{\gamma - 1}(T_2 - T_1). \tag{14}$$

It will be observed that in the particular case of isothermal (or 'equal temperature') expansion, the specific heats of the expanding gas do not affect the relation between P and V, nor the work done. In adiabatic expansion $n = \gamma$, and γ will be smaller the higher the specific heats of the expanding gas. It will be seen from fig. 1 that the smaller the value of n (or γ, if it is an adiabatic expansion) the flatter is the expansion curve. This indicates that during an adiabatic expansion a gas of high specific heat will, for a given change of volume and when expanding from the same initial pressure, do more work than one of low specific heat and will be at a higher temperature at the end. In other words, a given loss of internal energy involves less drop of temperature and pressure. During an isothermal expansion there is no change of internal energy, and heat has to be supplied from some external source of an amount equivalent to the work done, that is

$$\frac{R}{J} T_1 \log_e \frac{V_2}{V_1} \text{ C.H.U.}$$

The pressure, volume, and temperature relationships during an adiabatic expansion may be given here for convenience in alternative forms, as follows:

$$PV^{\gamma} = \text{constant} \tag{15}$$

$$TV^{\gamma-1} = \quad ,, \tag{16}$$

$$TP^{\frac{1-\gamma}{\gamma}} = \quad ,, \tag{17}$$

It may be useful, also, to summarize the formulae expressing the energy quantities in the Carnot cycle, as follows:

Heat received by the working substance at $T_1 = RT_1 \log_e r$ ft.-lb.;

,, rejected ,, ,, ,, ,, $T_2 = RT_2 \log_e r$,, ;

Work done during the cycle $= R(T_1 - T_2)\log_e r$ ft.-lb.;

where r represents the ratio of isothermal compression and expansion.

It follows that the efficiency of the cycle is given by the ratio

$$\frac{T_1-T_2}{T_1}.$$

Art. 8. The variation of the specific heats of gases

Calculations in connexion with internal combustion engines centre round the changes of temperature and pressure of the working substance while it gains or loses heat and is compressed or expanded. These calculations are much complicated by the fact that the specific heats of the gases employed vary considerably with temperature. They vary also, but to an unimportant extent, with the pressure.

In engine calculations we are directly concerned, not so much with specific heats, as with the closely related quantity of internal energy; and it is of some importance to become thoroughly familiar with the conception of a certain definite quantity of internal energy being always associated with any given quantity of a gas at any one temperature; the quantity being always the same at the same temperature, however the specific heats may vary, except for the slight deviations from Joule's Law. If the specific heat K_v were the same at all temperatures, then the internal energy would be proportional to the temperature measured from the arbitrary zero of 0° C. Actually K_v increases at high temperatures so that the internal energy and temperature are related by a curve, as in fig. 13, but it must be emphasized that although internal energy is by no means proportional to temperature for the real gases, this does not mean that they do not very closely conform to Joule's Law, which only states that at any given temperature the internal energy per pound will always be the same whatever the volume and pressure may be.

When a quantity of gas is heated at constant volume the gain of internal energy may take the form of an increase of kinetic energy of translation, or rotation, of the molecules as a whole; or, where there is more than one atom, of internal vibration of the atoms within the molecule. In monatomic gases, such as argon and helium, only energy of translation is possible, and it can be deduced from the kinetic theory of gases that when molecular energy is confined to energy of translation, the ratio of the specific heats, γ, should be 1·667. The value found experimentally for these monatomic gases is in fact almost exactly this, while for diatomic gases, such as nitrogen and oxygen, the ratio is also found to agree closely with its theoretical value of 1·40. This latter value is deduced from the supposition that such a diatomic molecule may be pictured as equivalent to a dynamic system consisting of two small masses connected by a light bond. If we imagine this bond as being capable of slight elastic extension, so that when in violent motion some intra-molecular vibration is possible, this enables us to account for the slight rise of specific heat of the diatomic gases with temperature, by supposing that the increase of internal energy associated with a 1° C. rise of temperature becomes greater at high temperatures because an appreciable amount of energy is absorbed by this intra-molecular vibration. It is of interest that the monatomic gases, for which intra-molecular vibration is impossible, exhibit no appreciable rise of specific heat with temperature.

From a corresponding but more complicated dynamic equivalent for molecules with more than two atoms, it can be deduced that gases with these larger molecules should exhibit a ratio of specific heats of 1·33. The measured value of γ for the triatomic gases with which we are concerned, CO_2 and H_2O, is rather less than this, on account of the vibratory energy of the component atoms being appreciable even at ordinary temperatures; with the result that the specific heats are higher than if an increase of internal energy contributed only to an increase of movement of the molecule as a whole.

The complex structure of the atom as now conceived, with its positive nucleus and attendant electrons, cannot, of course, be fully represented by any mechanical model; but it is still allowable, nevertheless, to discuss the energy of atoms and molecules as has been done above in terms of dynamic equivalents. An almost complete explanation of the observed relationship between temperature and energy in gas molecules of different types has been provided by Planck's Quantum Theory, and by means of it numerical values of the specific heats of gases can be related to the emission and absorption bands observed when the gases are examined spectroscopically. Of recent years it has become generally accepted that the specific heat values for gases at high temperatures calculated from Planck's theory and based on spectroscopic observations can be relied upon as substantially accurate for engineering purposes.

For a very clear and interesting discussion of gaseous specific heats in the light of the kinetic theory, the reader is referred to Ewing's *Thermodynamics for Engineers*, ch. vii (2nd edition). A discussion of the experimental values found for the specific heats of gases at high temperatures, and the methods used to determine them, will be deferred until the end of Chapter III.

II

ENGINE CYCLES

ART. 9. The ideal air cycle

Before proceeding to calculations about the changes which occur in a real engine, complicated as they are by the changing specific heat of the working substance and by exchanges of heat with the cylinder walls, it is useful to form a clear mental picture of what takes place in a simplified form. In order to do this we shall suppose that the working substance is a perfect gas for which K_p and K_v are always the same as they are for air at ordinary temperatures, and that there is no interchange of heat with the cylinder walls.

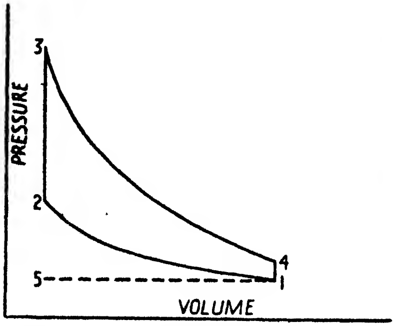

FIG. 2. The 'constant volume' cycle.

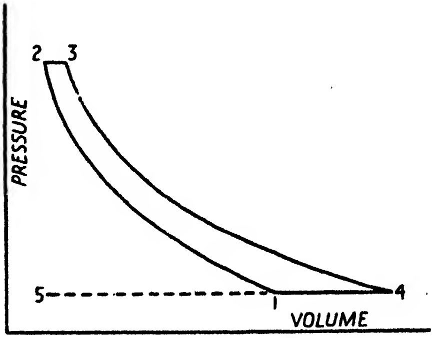

FIG. 3. The 'constant pressure' cycle.

For the efficient conversion of heat into mechanical work it is important that the working substance, after being heated by combustion of the fuel, should be cooled by expansion through as wide a range of temperature as possible. The pressure cannot, by expansion, be usefully lowered below atmospheric pressure (in practice it cannot usefully be lowered below about five atmospheres) and it is therefore important that before combustion takes place the air or combustible mixture should be compressed to as high a degree as possible, as a means to its subsequent expansion.

We will first consider two simplified forms of the cycle through which the working substance might be taken, in the first of which, illustrated in fig. 2, it is supposed to be instantaneously heated at constant volume (2–3), after adiabatic compression (1–2) by the inward movement of the piston, and cooled again at constant volume (4–1) after adiabatic expansion. In the second cycle, illustrated in fig. 3, there is no further rise of pressure after the adiabatic compression (1–2) is complete, the heat being added at such a rate during the first stages of expansion that the pressure remains constant until combustion is complete at the point 3, after which it drops, according to the ordinary law of adiabatic expansion of a perfect gas, down to atmospheric pressure.

Just as no real compression or expansion is adiabatic, so no heating or cooling of the working substance in a real engine takes place either at constant volume or constant pressure, but it is convenient to build up our ideal cycles out of changes which occur under conditions in which the results are easily calculable. These theoretical results, as we shall see, form a valuable guide as to what may be expected from the real engine.

In making calculations about the thermal efficiency of the ideal cycles illustrated in figs. 2 and 3, we are concerned only with the four processes represented by full lines which enclose the area 1234 on the diagram. Any complete diagram of the operations in a real engine cylinder would have to show the processes of getting the gases into and out of the cylinder. Although consideration of these practical aspects of the subject is deferred for the present, it may be pointed out here that in the well-known Otto, or 'four-stroke', cycle, the diagram would include lines such as those shown dotted in figs. 2 and 3, to represent, by the processes 1–5 and 5–1, the expulsion of the used products of combustion during the exhaust stroke, and the in-drawing of a fresh charge for the next cycle, both processes being supposed to take place at atmospheric pressure. It should be noted that in fig. 3, while the full movement of the piston is represented by the line 5–4, it is only during the operation 1–5–1 that any valves are to be imagined as open.

Besides the heating process 2–3 in each cycle, and the adiabatic compression and expansion, we have the fourth stage 4–1, during which we imagine heat to be abstracted while either the volume or the pressure remains constant. Reduction of pressure and reduction of volume take place in the two cycles respectively, each being appropriate to the reduction of temperature which is effected by the abstraction of heat. The process 4–1 is paralleled in a real engine by the drop of pressure which occurs after the exhaust valve opens and before the exhaust stroke begins.

Turning now to the calculation of the efficiency of an ideal engine working upon the Otto cycle of fig. 2, the pressure, volume, and absolute temperature of the gas in the cylinder before compression begins will be represented by P_1, V_1, and T_1, and similarly for the other points of the cycle. The only gains or losses of heat by the working substance take place while it is heated from T_2 to T_3 and cooled from T_4 to T_1, processes 1–2 and 3–4 being adiabatic.

The area of the diagram represents the net work done by the piston, and is equivalent to the difference between the heat supplied to, and given up by, the working substance. In other words, the heat equivalent of the net work done per cycle is

$$K_v(T_3-T_2)-K_v(T_4-T_1), \tag{18}$$

and the efficiency of the cycle $= \dfrac{\text{heat equivalent of net work done}}{\text{heat supplied}}$

$$= 1-\frac{T_4-T_1}{T_3-T_2}. \tag{19}$$

If the common ratio of compression and expansion, $\dfrac{V_1}{V_2}=\dfrac{V_4}{V_3}$ is denoted by r, then it follows from equation (16) that

$$\frac{T_1}{T_2}=\frac{T_4}{T_3}=\left(\frac{1}{r}\right)^{\gamma-1};$$

and, by a simple transformation, also,

$$\frac{T_4-T_1}{T_3-T_2}=\left(\frac{1}{r}\right)^{\gamma-1},$$

and the efficiency of the cycle reduces to

$$\eta = 1-\left(\frac{1}{r}\right)^{\gamma-1}. \tag{20}$$

In the constant pressure cycle of fig. 3, the heat supplied to the working substance after compression is $K_p(T_3-T_2)$, and the heat abstracted from it after expansion is $K_p(T_4-T_1)$. Compression and expansion being again adiabatic, and these expressions identical with those for the constant volume cycle, except for the substitution of K_p for K_v, it follows that an identical expression gives the thermal efficiency of this cycle as was found for the other, r being in each case the ratio of adiabatic compression and expansion.

The remarkable and important fact about the expression for the efficiency of these cycles, given in equation (20), is that if we could build an engine which truly followed either of them, then the theory shows that its thermal efficiency would depend upon nothing but the ratio in which the gas was compressed and expanded. If we had two engines of the same expansion ratio, in which the temperatures were very different, owing either to a different initial temperature T_1 or to different amounts of heat being added during the process 2–3, that would make no difference to the thermal efficiency, which would be the same in each. One engine might do more work than the other, but that work, whatever it was, would always be the same fraction of the total heat supplied during the process 2–3.

No real engine can of course follow the cycles very closely, because no compression or expansion can be made truly adiabatic; because the working substance is not a perfect gas; and because heat cannot be supplied to it at strictly constant volume or pressure; but what is true of the ideal engine is also true of all real engines, namely, that the ratio of expansion is of primary importance and that the higher we can make it, the higher will be the thermal efficiency. The ideal engine affords a standard, or high-water mark, of efficiency with which real engines may be compared, but which they can never hope to reach. The cycle of the ideal engine will be referred to as the 'air cycle', and its efficiency as the 'air standard efficiency'. The name is a little unfortunate, since a perfect gas, and not air, would be needed to carry it out, but the name has become generally accepted.

TABLE 1

Air standard efficiencies.

Ratio of expansion r	*Thermal efficiency* $1-\left(\frac{1}{r}\right)^{\gamma-1}$
2	0·240
3	0·353
4	0·423
5	0·471
6	0·508
7	0·537
8	0·561
10	0·598
12	0·626
14	0·648
16	0·666
18	0·682
20	0·695

In table 1 are given the values of the air standard efficiency calculated for a wide range of expansion ratios, and taking the value of $\gamma = 1{\cdot}396$ which, as we have seen in art. 5, is its value for air at ordinary temperatures.

The calculated efficiency figures have been plotted against expansion ratio in fig. 4, from which it will be seen that there is a rapid gain of efficiency as the expansion ratio is increased up to about 5 to 1 and that afterwards the gain is progressively less for each step. We shall see in Chapter VI that the thermal efficiencies achieved in practice are about two-thirds of those of the air standard, but that the curve of efficiencies actually obtained at different expansion ratios lies very nearly parallel to that of fig. 4, and to an intermediate one based upon calculations which do not assume the use of a perfect gas, but take account of the properties of the real working substance as determined by experiment.

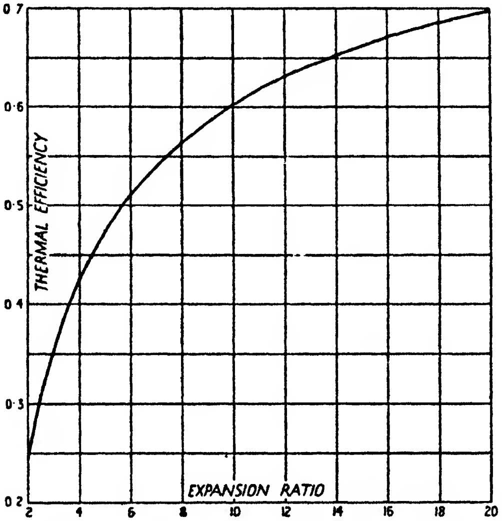

FIG. 4. Air standard efficiencies at different expansion ratios.

For reasons which will appear later, the limit of the compression, and therefore expansion, ratio in petrol engines is at present, except in a few special engines, about 6:1. The higher ratios, up to 16:1, are only used in engines which do not compress a mixture of fuel and air, but nearly pure air, the fuel being injected into the cylinder when the compression is almost complete.

It is convenient very often to speak of compression and not of expansion ratio, and in the great majority of engines, as in the air cycle, the ratios of compression and expansion are very closely the same; but it must always be remembered that it is the *expansion ratio* which really controls thermal efficiency. Attempts have been made to design engines in which the two are not the same, the object being to avoid the difficulties associated with a high compression ratio and yet retain the efficiencies proper to a large expansion. The relentless demands of mechanical simplicity, however, have prevented any important developments along these lines.

Art. 10. The approximation of ideal cycles to real conditions

Of the two ideal cycles illustrated in figs. 2 and 3, the first affords a reasonably close approximation to what takes place in real engines. The only conspicuous difference between the real and the ideal lies in the fact that during the process 4–1 in the real engine, the quantity of working substance in the cylinder, instead of remaining the same, as in the ideal cycle, is reduced to something like $\frac{1}{5}$ by expansion through the open exhaust valve. This difference, however, is immaterial so far as calculations of efficiency are concerned. Heat is removed from the cylinder during the process 4–1 and at the same time the pressure is reduced to atmospheric. These changes are the only ones that affect the calculations, and whether or not they are accompanied by an exchange of working substance can make no difference.

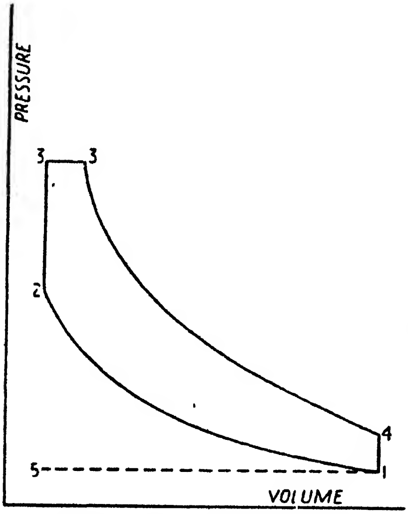

Fig. 5. The composite cycle.

The second, constant pressure, cycle has often been called the Diesel cycle, although there was never very much justification for this, and in view of modern developments of the Diesel engine there is now less than ever. The cycle which Diesel aimed at was in fact a Carnot cycle. This he hoped to achieve by injecting fuel into his cylinder, after a high degree of adiabatic compression, at such a rate as to maintain a constant temperature during the early stages of expansion, followed by adiabatic expansion after the fuel injection was completed. In the early experimental Diesel engines it was never found possible to do this, but it was found possible to maintain something like constant pressure (with of course a rising mean temperature) during the early part of the expansion. For this reason the cycle of fig. 3 came to be spoken of as an ideal for the Diesel cycle. There was of course never any likelihood or intention of extending the expansion process 3–4 down to atmospheric pressure, and to get over this difficulty a third ideal cycle has been suggested, which is illustrated in fig. 5.

In this cycle part of the heat is supposed to be added instantaneously at constant volume between 2–3, and the remainder at constant pressure 3–3′. Adiabatic expansion along 3′–4 then occurs until the working substance has regained its original volume. The cycle is completed by constant volume cooling 4–1 as in that of fig. 2.

In this cycle, let

$$\alpha = \frac{P_3}{P_2} \quad \text{and} \quad \beta = \frac{V_{3'}}{V_3}.$$

Then the magnitudes of α and β will depend upon what fraction of the total heat available we assume to be communicated at constant volume, and what at constant pressure.

The efficiency of the cycle can be shown to be given by the formula†

$$\eta = 1-\left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{\alpha\beta^{\gamma}-1}{(\alpha-1)+\gamma\alpha(\beta-1)}\right].$$

This composite cycle suffers from the drawback that its shape and efficiency depend upon an arbitrary division of the total heat available into that given to the working substance at constant volume and at constant pressure. Apart from this disadvantage it affords a reasonably good guide for calculations about the behaviour of some types of high compression engines with injected fuel. As we shall see in Chapter V, modern developments of this type of engine have tended towards constant volume combustion, so that an indicator diagram would show a sharp peak at the maximum pressure, with no sign of a constant pressure period (see fig. 21 on p. 117).

The limit upon our ability to employ constant volume combustion with a high compression ratio and injected fuel arises, of course, from the very high peak pressures which would be developed. In practice the injection of fuel has to be so controlled that the peak pressure does not rise above some agreed maximum, and it is in representing these conditions that the composite cycle of fig. 5 is useful for the calculation of ideal efficiencies. The necessary assumption about the proportion of heat added at constant volume and at constant pressure must be governed by reference to the maximum pressure which is to be allowed.

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† For proof see Appendix I, p. 287.

In table 2 and fig. 6 are given the calculated ideal efficiencies of the composite cycle for compression ratios between 10 and 20 to 1, and in obtaining these it has been assumed, apart from the usual assumptions as to the working substance being ideal air with no heat loss, that the same quantity of heat has been added at each compression ratio, and that this quantity is that which can just be added at constant volume at the lowest ratio, 10:1, without exceeding the arbitrary maximum pressure of 1,000 lb. per sq. inch; this maximum being about the extreme limit allowable in practice. At each ratio above 10:1 less and less heat can be added at constant volume before the pressure limit is reached, and the remainder is then assumed to be added at constant pressure.

TABLE 2

Air standard efficiencies of the composite cycle of fig. 5 on the assumption of a constant total amount of heat supplied and a constant maximum pressure of 1,000 *lb. per sq. inch.*

Pressure and temperature before compression begins are assumed to be normal atmospheric and 100° C.

Ratio of compression r	*Compression*						*Efficiencies*	
	Temp. T_2	*Press.* P_2	$\alpha = \frac{P_3}{P_2}$	T_3	T_3'	$\beta = \frac{T_3'}{T_3}$	*Composite cycle*	*Constant vol. cycle*
10	933	366	2·735	2550	2550	1	0·60	0·598
12	1000	470	2·13	2130	2479	1·165	0·623	0·626
15	1093	645	1·55	1694	2422	1·434	0·640	0·657
18	1175	830	1·205	1415	2402	1·70	0·648	0·682
20	1223	962	1·04	1272	2396	1·88	0·651	0·695

The available ratio for adiabatic expansion, after this heating at constant pressure, is $V_4/V_{3'}$ instead of V_4/V_3. In other words, the expansion ratio, instead of being equal to the compression ratio, r, is now only equal to r/β. While r increases from 10 to 20, the expansion ratio increases only from 10 to 10·62. It follows from this approximate constancy of the expansion ratio, and the fixed maximum pressure, that the final pressure P_4 in the composite cycle is almost unaffected by the increase of compression ratio from 10:1 to 20:1. The effect of the increase is a small drop in the final pressure from 40·2 to 37·1 lb. per sq. inch absolute.

Since the same total heat is assumed to be supplied at each compression ratio, the working out of the efficiencies on this assumption is equivalent to examining what advantage in power output there is to be gained by raising the compression ratio above 10:1, at which all the heat can be supplied at constant volume, if the maximum pressure is to be limited to 1,000 lb. per sq. inch, with the consequent limitation which this imposes upon the expansion ratio. Comparison of the two curves in fig. 6 shows very clearly the effect of a limitation on the maximum pressure in not allowing the efficiencies to rise with the compression ratio as they would with constant volume combustion, when compression and expansion ratios are equal.

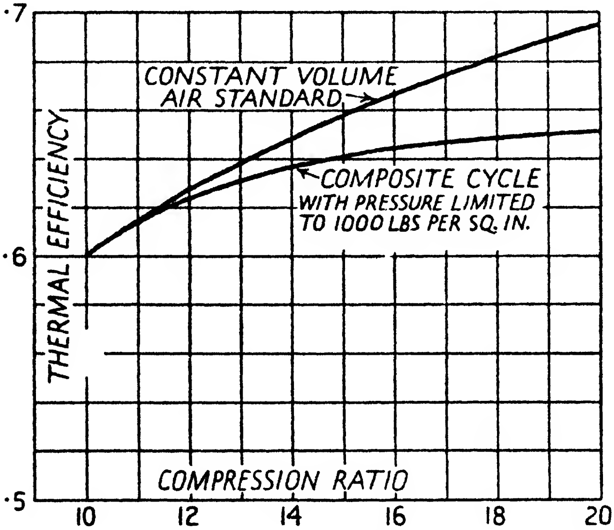

FIG. 6. Comparative theoretical efficiency of the 'constant volume' air standard, and the composite, cycles.

Some of the leading figures in the efficiency calculations are given in the table, from which it will be seen that, while at the lowest compression ratio the pressure is increased at constant volume in the proportion of 1:2·735, at the highest ratio the increase is only 1:1·04, and almost all the heat is added at constant pressure. At first, as the ratio is raised from 10 to 12 the efficiency improves fairly rapidly; but above 15:1 the increasing proportion of the expansion stroke which is taken up by the process of constant pressure combustion means that the advantage of the higher compression ratios is nullified, and the efficiency of the composite cycle is almost stationary. This follows, of course, upon the approximate constancy of the expansion ratio, r/β, which will be found to be almost stationary too.

The comparison is interesting, and of some practical importance as affording some indication of the altered relation between compression ratio and ideal thermal efficiency, when the new factor of a limited maximum pressure has been introduced. The question is further considered in Chapter VI, when the more practical limits of thermal efficiency will be examined, taking into account the properties of the real working substance, and its difference from the ideal air which we have hitherto postulated.

ART. 11. The value to be taken for n in $PV^n = C$

For a true adiabatic compression or expansion the index n would be equal to γ, the ratio of the specific heats, and for air at ordinary temperatures this is 1·396. No real compression or expansion is truly adiabatic, and no working substance with which we are concerned has specific heats which are either constant or equal to those of air at ordinary temperatures, and it becomes a matter of some importance, in engine calculations, to settle upon the most satisfactory value of n to take for practical purposes. Any value taken must be an average for the range of temperature covered, since the specific heats will be varying during the process, and so also will the rate of heat loss. The heat loss, too, will depend upon the relation between engine speed and turbulence of the gases in the cylinder, so that any accepted value for n must be a compromise covering many variables.

The range of variation will be much wider for expansion than for compression curves; so wide indeed that it is of doubtful value to attempt to give an average working value for n during expansion curves. It is for calculations of the pressures and temperatures at the end of compression, however, that a working value for n is chiefly useful, and for the present we shall confine the discussion to that aspect. In the next article an outline is given of the method for dealing with expansion curves.

Even for compression curves no one value of n can be accepted as satisfactory for all, but it is possible to divide up the different conditions for which we need to know the value into three well-defined classes of engines, and to accept an average value for each class of satisfactory accuracy. The three classes are:

(*a*) Engines with petrol-air mixtures. Compression ratios up to 7 : 1.

(*b*) Engines with coal gas-air mixtures. Compression ratios up to 7 : 1.

(c) Engines compressing air only in ratios from 12 to 16:1, such as are required for compression-ignition operation.

It must be understood that for all the classes there are factors about which it is impossible to generalize, and which have an important effect upon the compression pressure reached in any particular engine. These factors, apart from the barometric pressure, are the cylinder wall temperature, the valve timing, and the length of the induction pipes. Depending upon a combination of the last two, the pressure in the cylinder at the beginning of compression may be appreciably below, or appreciably above, the outside atmospheric pressure. The latter possibility arises from the 'ramming pipe' effect of the momentum acquired by the moving column of air in the induction pipe. As an example of how large the effect of valve-timing may be, an experiment may be quoted in which a large air-cooled cylinder of compression ratio 12:1 gave a compression pressure 415 lb. per sq. inch absolute when motored round cold at 1,100 r.p.m., and only 402 when tested just after full-load running. At this speed gas leakage was quite negligible and the drop in compression pressure was doubtless due to cylinder expansion when tested hot, which gave an earlier closing of the inlet valve, and lower volumetric efficiency. The drop in volumetric efficiency was sufficient to lower the compression pressure from 415 to 402 lb. per sq. inch, in spite of the augmentation of the pressure due to hot cylinder walls.

In the face of so large an uncertainty, which may cause wide differences of compression temperature and pressure in engines of the same measured compression ratio, and even in the same engine at different times, the only satisfactory plan, in making analyses for the calculation of theoretical efficiencies, will be to assume truly adiabatic compression from atmospheric pressure. Under these conditions $n = \gamma$, and γ can be accurately obtained from the mean specific heat of the gases in the cylinder up to the compression temperature.

A point may be noted here which will be more fully gone into in art. 37, namely, that the gases *in* the cylinder, before compression, are composed of two parts mingled together: the fresh charge drawn in during the suction stroke, and the residual exhaust gas left in the clearance space of the cylinder after the previous stroke. The residual exhaust gas contains some 20 per cent. by volume of the triatomic gases CO_2 and H_2O, of high specific heat, and for accurate work the presence of these cannot be neglected in calculating the mean specific heat of the contents of an engine cylinder during compression. In a 5 : 1 compression ratio engine the presence of the residual exhaust gas will increase the mean specific heat of the cylinder contents between 100° C. and the compression temperature by about 0·5 per cent., and will therefore reduce the value of $\gamma - 1$ in the same proportion (see equation (8), p. 10).

Apart from the conditions to be assumed in theoretical analyses, however, the best average values of n to be used for finding compression pressure and temperature in the three classes of engines (*a*), (*b*), and (*c*) will now be discussed.

Before dealing at some length with the question of heat loss, which affects all types, it will be convenient to clear up the question of the effect of the fuel vapour in class (*a*), and to get that out of the way. In a petrol and air mixture in the proportion to give just complete combustion of both, there is about 7 per cent. by weight of heavy hydrocarbon vapour, of which the high specific heat will have a marked effect upon the value of n. Class (*a*) differs in this respect from the other two, for in class (*c*) air only is drawn into the cylinder, while in class (*b*) the presence of even 20 per cent. of coal gas, composed as it is to the extent of about 75 per cent. of the diatomic gases hydrogen and carbon monoxide, will affect the mean specific heat of the cylinder contents only to a small extent. The other 25 per cent. of the coal gas would be mostly methane, and if we assume it is all that, the effect of this would be to raise the mean specific heat of a 1 : 4 gas-air mixture above that of pure air by 2 per cent. This is an outside figure, for it is not often that the gas-air ratio would be so high as 1 : 4. It would be commonly more like 1 : 8.

Tizard[1]† has pointed out that the specific heats at constant volume of all the paraffin hydrocarbon vapours from pentane upwards (see art. 16) are very nearly the same and equal to 0·48 C.H.U. per pound between 0° and 300° C., and that the ratio of specific heats, γ, can therefore be calculated for a series of petrol-air mixtures of varying proportions. The values so found are shown plotted in fig. 7, and it will be seen that they vary from 1·35 for a mixture containing only 80 per cent. of the fuel for complete combustion, to about 1·32 for a mixture with 50 per cent. excess of fuel. The value at the 'correct' mixture, as it will in future be called, is about 1·34. These values of γ would be the values of n to be used for the compression of petrol-air

† See p. xiv for all references to original work.

mixtures if all heat loss to the cylinder walls could be suppressed. Heat loss, however, has an important bearing on the calculations of compression pressures at all ratios, and will now be considered in some detail.

In the paper just referred to, Tizard gives the results of a large number of experiments made with the compression machine of which some description is given in art. 34 of Chapter V. In this machine a single rapid compression of air or combustible mixture can be made, so that a knowledge of the initial and final pressure, and of the volume ratio of compression, enables a value of n to be calculated. The values so found for compression ratios below 7:1 have been added to fig. 7 and are shown by the dotted line. The values vary with the mixture strength just as the calculated values of the upper curve, and the vertical distance of the dotted curve below the curve of γ values represents the effect of heat loss. For the correct mixture the observed value of n was 1·31, and it dropped to 1·29 for a mixture containing 50 per cent. excess of fuel. In this machine the gas before compression was in a stagnant condition, whereas in an engine there is always a considerable amount of turbulent motion of the air which will assist heat loss. To set against this the time of compression in Tizard's experiments was very much longer than in any high-speed engine, being about 0·1 sec. as compared with 0·03 sec. in an engine running at 1,000 r.p.m., and

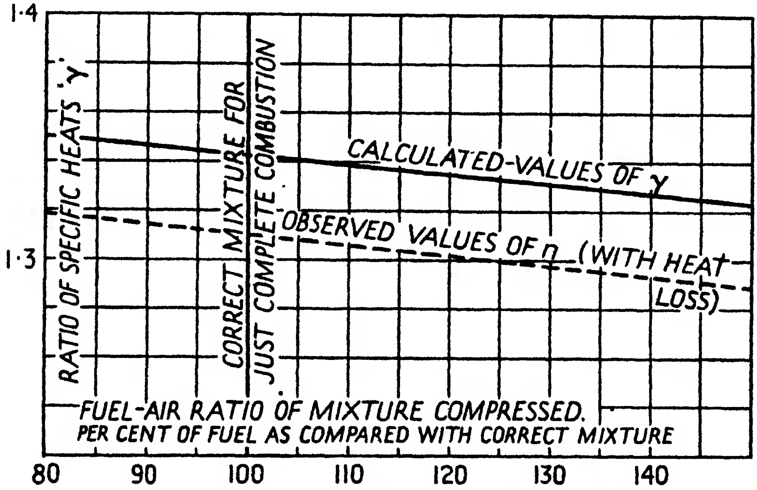

Fig. 7. Values of γ and n for petrol-air mixtures.

we must now examine the combined effect upon heat loss of the opposing factors of turbulence and speed.

Before passing to the high-speed engine with its extreme degrees of gas turbulence it is of interest to record the results of a further series of experiments[16] made with the compression machine to ascertain the effect of time of compression upon the pressure reached, and upon the value of n. The results of a series of compressions of air and of hydrogen in a volume ratio of 6·09 : 1 are shown in fig. 8. The more pronounced effect of heat loss with hydrogen, owing to its greater conductivity, will be noticed. It was impossible, for mechanical reasons, to make the time of compression less than 0·05 sec., but the observations indicate that the less time there had been for heat loss the nearer would the pressure ratio have approached to its true adiabatic value 12·5.

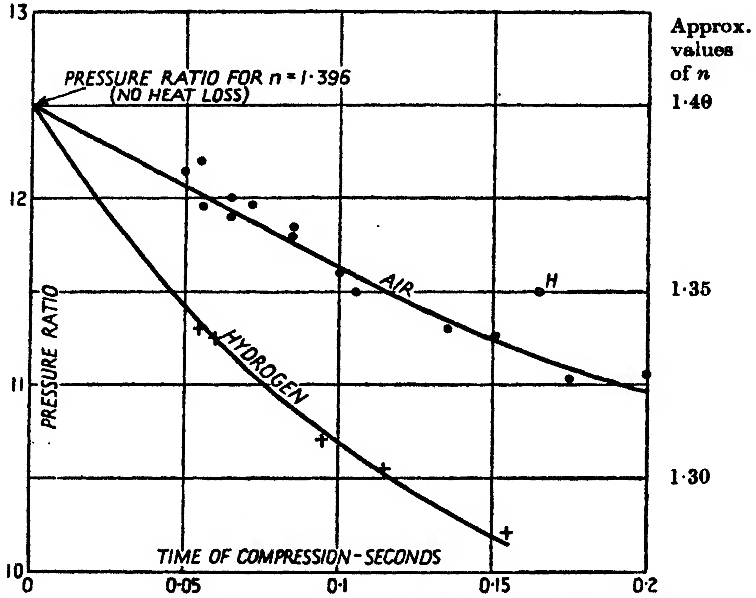

FIG. 8. Pressures reached by compression at varying rates in the volume ratio 6·09 to 1.

Hopkinson[3] examined the compression and expansion curves for a large, slow-running engine of compression ratio 6·37, and found that the average value of n while compressing air only was 1·35. It varied widely during the stroke, averaging 1·41 in the first and 1·31 in the second half. The speed of the engine was 180 r.p.m., giving a time of compression 0·166 sec., and the corresponding pressure ratio for $n = 1{\cdot}35$ has been added to fig. 8 at the point H. The proportionate loss of heat was evidently rather less in Hopkinson's engine, in spite of some turbulence, but this may easily be understood as being due to the very much larger size of cylinder of his engine, this having been 11·5×21 inches, bore and stroke, as against 3×8 inches in the compression machine.

It might be supposed, after an examination of fig. 8, that for engines of speeds over 1,000 r.p.m., that is, for times of compression of 0·03 sec. and under, the pressure ratio would approach very closely to the adiabatic value. This, however, would be to reckon without the effect of turbulence. The effect of this, and speed, will best be understood by an examination of table 3, in which are given a number of carefully observed values of n in the compression machine and in engines of classes (*b*) and (*c*) at various speeds. In some cases the effect of speed may be masked to some extent by a change of pressure before compression, as it was not in all cases possible to get an exact check upon this; but the values for n appear to be consistent and are probably correct to two places of decimals.

TABLE 3

Values of index n under various conditions for compressions of air only.

Compression ratios (volume)	*Speed or time of compression*	*Machine in which observed* *Compression machine*	*Poppet-valve engine*	*Sleeve-valve engine (low swirl)*	*Sleeve-valve engine (high swirl)*
6·09	0·1 sec.	1·36	..	..	..
6·00	1500 r.p.m.	..	1·375	..	..
13·8	0·1 sec.	1·31	..	..	..
12·5	1000 r.p.m.	..	1·32	..	..
12·5	1800 ,,	..	1·325	..	..
13·8	1000 ,,	..	..	1·305	1·29
13·8	1500 ,,	..	..	1·315	1·30

A comparison with the values of the pressure ratio of fig. 8 will show, in the first place, that in the engine of 6:1 ratio the value of n at 1,500 r.p.m. is no higher than that in the compression machine when the time was 0·06 sec. In other words, turbulence must have trebled the rate of heat loss, since the time occupied at 1,500 r.p.m. was 0·02 sec.

In columns 5 and 6 of the table, the effect of turbulence is illustrated in another way. These figures were obtained upon a sleeve-valve engine in which the method of entry of the air into the cylinder produced a vigorous swirling motion about its axis, and the rate of swirl could be measured, and varied at will. Under the conditions designated 'high swirl' the rate of circulation of the air was about 250 revolutions per second, and under low swirl conditions half as rapid. Doubling of the swirl rate reduced the pressure ratio of compression by 4 per cent. at each speed, other conditions remaining steady. Another interesting comparison is between the compression machine and the engine, at an identical compression ratio of 13·8 to 1, in which stagnant compression in 0·1 sec. gave the same pressure ratio, within the accuracy obtainable, as low swirl conditions in the engine occupying 0·025 sec.; one may conclude, therefore, that the organized swirl of the air had increased the rate of heat loss about four times above that of the stagnant compression, as against the trebling of it by the unorganized turbulence of the poppet-valve cylinder. It should be noted, however, that this comparison is subject, so far as the engine is concerned, to uncertainty as to what the effective compression ratio really was. The measured ratio was 13·8, but indicator diagrams did not give with certainty the moment when the pressure in the cylinder was atmospheric.

Enough has now been said to give grounds for settling upon the most satisfactory average figures to adopt for the index n in the three classes of engines (*a*), (*b*), and (*c*), as set out on p. 26.

In settling upon a figure for class (*a*) we have the facts that in the compression machine the observed value of n with a correct petrol-air mixture was 1·31, with a compression time of 0·1 of a second, and this figure rose to 1·36 when the compression was of air only. As compared with 1·375 for the compression of air only at 6:1 and 1,500 r.p.m., we may accept 1·33 as the best average value for high-speed petrol engines in class (*a*), of which the speeds will generally be above 1,500 r.p.m. The proviso must be added that the correct figure may easily be as low as 1·31 for rich petrol-air mixtures, and may be 1·335 or 1·34 for weak ones.

For class (*b*) 1·37 will be taken, on the ground that 1·375 was observed with pure air at 6:1 compression, and that the presence of methane in the coal gas, helped to a slight extent by the residual exhaust, may be expected to produce a lowering of the value of $n-1$ amounting to 1·5 to 2 per cent. Gas-engine speeds are for the most part well below 1,500 r.p.m., at which the figure of table 3 was observed, but to set against this the cylinder sizes are larger. These two factors will influence the heat loss during compression in opposite directions, and it seems reasonable therefore to accept the figure of table 3, modified as above, as being applicable with sufficient accuracy to the whole class of gas engines.

Class (*c*) must be subdivided for poppet-valve and sleeve-valve engines, for among high-compression engines with injection of fuel those with sleeve-valves will be of the high swirl type. An appropriate figure will therefore be 1·30, whereas in the corresponding poppet-valve engine the average appears to be more nearly 1·32.

It must be understood that these figures are purely rough average values of n for the practical purpose of calculating compression pressures and temperatures, assuming that the full compression ratio is effective and that the pressure in the cylinder when the compression stroke begins is atmospheric. Values of n are influenced both by heat loss and by the specific heat of the gases during compression to the extent indicated, but the valve-timing and form of induction system may have an influence as great as either of these factors in deciding the real compression pressure in any one particular engine.

Art. 12. The calculation of cycle temperatures

As a preliminary to the methods to be given in the next chapter for calculating the exact condition of the working substance in an engine, allowing for dissociation, there will be given in the present article the calculated explosion temperatures, and the temperatures after expansion, on the assumption that no dissociation takes place, and the effect of dissociation upon the explosion temperatures will merely be stated. The engine will at first be taken as one of 5:1 compression ratio, employing the typical hydrocarbon benzene (C_6H_6) as fuel, in which the conditions of the combustible mixture before compression are normal atmospheric pressure and 100° C.

When benzene is mixed with air in the correct proportion to give just complete combustion of all the C to CO_2 and all H_2 to H_2O, then the heat generated is 57·5 C.H.U. per standard cubic foot of the mixture. It is more convenient, however, to work with the pound-molecule, or mol., as the unit of quantity, and the heat generated on combustion per mol. of mixture will then be

$$57{\cdot}5 \times 359 = 20{,}640 \text{ C.H.U.}$$

If this quantity of heat is represented by Q, and if q is the heat energy given to the combustible mixture during the previous compression in the ratio 5:1, then the 'energy content', E_T, of the products of combustion per mol. of original mixture, at absolute temperature T, is given by

$$E_T = Q+q \text{ C.H.U. per mol.}$$

E_T is measured from the arbitrary zero of 100° C., and q is of course the work done in compressing the mixture. For the correct benzene-air mixture and 5:1 compression $q = 1{,}590$ C.H.U. per mol. and therefore

$$E_{T_3} = 20{,}640+1{,}590 = 22{,}230 \text{ C.H.U. per mol.}$$

In order to calculate the maximum temperature T_3 of the cycle (fig. 2, p. 17) we need to know the mean specific heat of the products of combustion for the range of temperature from 100° C. up to the maximum.† In engine calculations it is always more convenient to work with units of volume of the working substance rather than weights, and since 'specific heat' is the accepted term for the heat necessary to raise the temperature of unit *mass* of a substance one degree, it is proposed to use the term 'volumetric heat' henceforward when a certain volume of gas is the unit of quantity. Volumetric heats will be expressed in terms of ft.-lb. per S.C.F. or of C.H.U. per mol. The mean volumetric heat of the mixed products of combustion in an engine may be calculated from a knowledge of the mean volumetric heats of N_2, CO_2, and H_2O, as given in table 10 on p. 87, or in the case of the correct benzene-air mixture where dissociation is disregarded it may be taken from a curve such as that of fig. 15 on p. 88, which has been drawn for the products of combustion of such a mixture.

E_T for the combustion products per mol. of original mixture can be very nearly expressed by an equation of the form

$$E_T = 640{\cdot}4\left[\frac{T-373}{100}\right]+4{\cdot}05\left[\frac{T-373}{100}\right]^2 \qquad (21)$$

for the ranges of temperature from 100° C. up to maxima lying between 2,000° and 3,000° C.

The maximum temperature T_3 abs. is therefore given by

$$22{,}230 = 640{\cdot}4\left[\frac{T_3-373}{100}\right]+4{\cdot}05\left[\frac{T_3-373}{100}\right]^2,$$

whence $T_3 = 3{,}301°$ abs. $= 3{,}028°$ C.

During the expansion stroke, the temperature of the working sub-

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† During adiabatic compression the mean specific heat of the combustible mixture is sufficiently nearly the same as that of its products of combustion, up to the compression temperature.

stance will fall through more than 1,000° C., and the volumetric heat of the gases will vary widely during the process. The value of γ, if the expansion curves be represented by $PV^\gamma =$ constant, must also vary very much, but the final temperature T_4 can be calculated with little loss of accuracy by a method which assumes a mean constant value of γ throughout the expansion, corresponding to a mean K_v. The calculation is made by guessing a value for γ, calculating T_4 from equation (16), according to which

$$T_4 = T_3 \div 5^{\gamma-1},$$

and then finding the mean value of K_v which corresponds to the temperature found. For this purpose E_{T_4} is first found from equation (21) and then we have

$$K_v = \frac{E_{T_3} - E_{T_4}}{T_3 - T_4}.$$

Finally we have to find γ corresponding to this value of K_v by equation (8), and if it does not agree with the value previously guessed, the calculation is repeated, starting with the new value of γ.

For example, take $\gamma = 1{\cdot}235$. Then

$$T_4 = 3{,}301 \div 5^{0\cdot235} = 2{,}261 \ (1{,}988^\circ\,\text{C.}),$$

$$E_{T_4} = 640{\cdot}4 \times 18{\cdot}88 + 4{\cdot}05(18{\cdot}88)^2 \quad \text{(from eq. (21))}$$

$$= 12{,}090 + 1{,}445 = 13{,}535.$$

The mean value of K_v between T_3 and T_4

$$= \frac{22{,}230 - 13{,}535}{3{,}301 - 2{,}261} = 8{\cdot}360 \text{ per mol. of original mixture.}$$

But since there are 74·4 mols. of products for 73·4 mols. of mixture, the mean K_v for products $= 8{\cdot}360 \div 1{\cdot}014$, whence

$$\gamma = 1 + \frac{1{\cdot}985 \times 1{\cdot}014}{8{\cdot}360} = 1{\cdot}2407.$$

If the calculation is repeated, using $\gamma = 1{\cdot}241$, the final temperature $T_4 = 2{,}240$ is obtained and the value of γ checks correctly.

The final temperature for the calculated cycle is therefore 1,967° C. The value of $E_{T_4} = 13{,}370$ and hence the work done during the adiabatic expansion must be

$$E_{T_3} - E_{T_4} = 22{,}230 - 13{,}370$$

$$= 8{,}860.$$

The net work done during the cycle

= work done during expansion—work of compression

= 8,860—1,590

= 7,270 C.H.U. per mol. of original mixture.

This figure for the net work done per cycle, when divided by the heat supplied on combustion, 20,640 C.H.U., gives the calculated efficiency of the cycle, which in the example taken is found to be 35·2 per cent.

The calculations so far have been based on the assumption that the full heat of combustion of the benzene-air mixture, 20,640 C.H.U. per mol., will actually be developed when ignition takes place. It is found, however, that when a quantity of CO_2 and H_2O molecules are raised to the very high temperatures produced in an engine cylinder, a certain proportion of them, depending upon the temperature and to a less extent upon the pressure, become split up according to the equations:

$$2CO_2 \rightleftharpoons 2CO + O_2, \tag{22}$$

$$2H_2O \rightleftharpoons 2H_2 + O_2, \tag{23}$$

and a condition of equilibrium is established in which the complete molecule and its component parts coexist together in certain definite proportions, depending chiefly upon the temperature.

It follows that when a combustible mixture of air with any hydrocarbon is burnt, there will come a point, if the temperature becomes high enough, at which the union of oxygen with carbon and hydrogen, which generates the heat, will cease, although only partially complete, and at the maximum temperature there will be not only CO_2 and H_2O but also some CO, O_2, and H_2 in an uncombined state. At these high temperatures the five gases can coexist in equilibrium and the temperature reached will be lower than the 3,028° C. calculated above, because the stoppage of the oxidation of carbon and hydrogen when only partially complete will mean that the full heat of combustion has not been developed at the maximum temperature. When the temperature in the engine drops below the maximum, on account of expansion, the proportion of CO_2 and H_2O which can remain in a dissociated condition rapidly diminishes, and all carbon and hydrogen is soon completely oxidized, provided always that there is no excess of fuel present, which would result in there not being enough oxygen to go round. The method of calculating the amount of heat actually developed at the maximum temperature, when dissociation is allowed for, will be explained in the next chapter. In the correct benzene-air mixture which we are considering the lowering of the maximum temperature amounts to about 320° C., 3,028° becoming 2,705° C. when dissociation is allowed for.

ART. 13. Three cycles compared

The methods of the last article provide means for calculating the temperatures at each point of a cycle, with the exception of the correct explosion temperature when dissociation has been allowed for. In the present article three typical cycles will be examined, and the temperature and pressure calculated for each significant point of the diagram. The effect of dissociation will, as in the last article, be simply stated, the calculations for this being deferred to the end of Chapter III.

The three cycles to be examined are:

(*a*) a petrol engine of compression ratio 5 to 1;
(*b*) a gas engine of compression ratio 7 to 1;
(*c*) a compression-ignition engine of compression ratio 12 to 1.

Although the calculated diagrams will be subject to the condition of no heat loss to the cylinder walls, they will in other respects approximate fairly closely to the real conditions, and afford an interesting comparison of the temperatures and pressures to be expected in the three types of engine. The heat supply from combustion of the fuel is not the same in the three cycles. In the first it is taken as the heat generated by combustion of a correct benzene-air mixture; in the second by combustion of a correct mixture of the typical coal gas of which an analysis is given in table 5 on p. 44; and in cycle (*c*) it has been assumed that sufficient fuel is injected to combine with 80 per cent. of the air in the cylinder. This keeps the calculated diagram in relation to real conditions, for it has never hitherto been found possible in practice to consume more than about 75 per cent. of the air present in the cylinders of a compression-ignition engine. The diagrams are shown side by side in fig. 9, each drawn to a base of equal stroke length and to a common pressure scale. The conditions before the beginning of compression are in each diagram taken as 100° C. and standard atmospheric pressure. It might be objected that the initial temperature of the three cycles would probably not be the same in practice, and this is true; in art. 36 of Chapter V it will be shown that 100° C. is about right for the gas engine of 7:1 compression, but that for the lower compression petrol engine the larger proportion of hot residual gas remaining in the *cylinder will result in a temperature more like 120° C.* In the compression-ignition engine, *mutatis mutandis*, the initial temperature will in like manner be below 100° C. For this preliminary comparison, however, there is much to be said for assuming a common starting temperature. It brings out, for example, that there is a steady *fall* of maximum temperature with rise of compression ratio; and the calculated differences would be increased somewhat by taking starting temperatures more appropriate to each ratio. If dissociation had not been allowed for, the maximum temperatures would have been about 3,028° (as was found in the last article), 2,942°, and 2,481° C. for the three cycles. It should be noted of course that the fall of maximum temperature at the higher ratios is due, not to the rise of ratio, but to the other conditions which happen to go with it. In the gas engine less heat is generated per standard cubic foot of correct combustible mixture than in a petrol engine; and in the compression-ignition engine there is the assumption that only sufficient liquid fuel is injected to combine with 80 per cent. of the air present. In this cycle, also, some heat of combustion is converted into work during the constant pressure expansion and is ineffective in raising the temperature of the cylinder gases. This expansion at constant pressure while combustion goes on reduces the effective expansion ratio from 12:1 to 7·25:1, but even so the final temperature T_4 is about 360° C. below the gas engine and 550° C. below the petrol engine. The lower exhaust temperatures of the compression-ignition cycle are very noticeable in practice, as are also

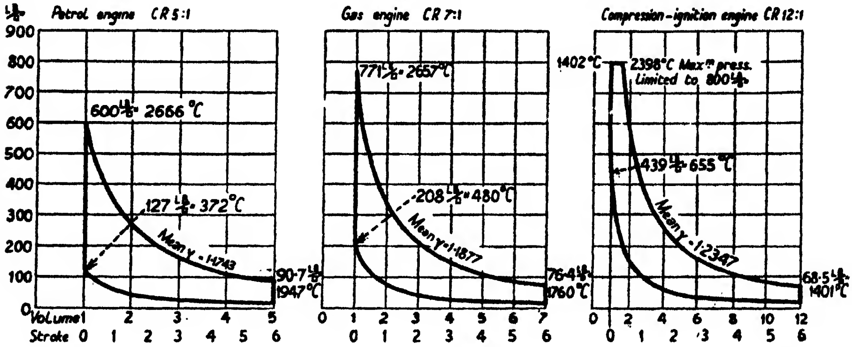

FIG. 9. Calculated cycles for three types of engine. Properties of the real working substance, and also dissociation, have been allowed for, but the cylinder walls and piston have been assumed impervious to heat.

the generally cooler cylinder conditions when measured during full-load operation.

The temperatures shown on the three diagrams of fig. 9 are intended only to bring out, at this early stage, the different characteristics of the three cycles. They are not given as being the temperatures in an actual engine, as nearly as can be calculated after full allowance has been made for practical conditions. For example, besides the assumption of a common initial temperature of 100° C., referred to already, it seemed advisable, for the sake of simplicity, to assume that the cylinder in each engine was filled only with combustible mixture, undiluted with residual products of combustion from the previous stroke. This has the advantage that the calculated temperatures are the same as those which will be obtained, in Chapter III, when the combustion of certain specified fuel-air mixtures, represented by the chemical equations (28) and (29) on pp. 46 and 63, will be considered. It should be realized that the presence of residual products has a very substantial effect in the way of lowering the maximum temperatures. At a compression ratio of 5:1 the weights of residual gas and of combustible fuel-air mixture present in the cylinder during compression will be in the ratio of about 1 to 14 (see table 20); and to a first approximation, therefore, the rise of temperature on combustion will be reduced, by the presence of that proportion of inert gas, in the ratio 14/15, that is by some 150° C. At higher ratios of compression the effect of the inert residual gas will be less marked. The methods of Chapter III can be applied to calculate the true maximum temperature in any particular case, as soon as the conditions in regard to compression ratio, fuel-air ratio, etc., have been defined.

The presence of residual gas in an engine cylinder is of great importance in connexion with the phenomenon of detonation (see Chapter IV). It may be noted here for future reference that every 1 per cent. dilution of a combustible mixture with residual gas will diminish the temperature rise on combustion in a like proportion, i.e. by some 20° C. or more.

The higher the maximum temperature, in the cycles of fig. 9, the higher will be the mean volumetric heat of the expanding gases and the greater the combustion during the expansion stroke—and these features are reflected in the falling values of the 'mean γ' during expansion as marked on the diagrams.

The final temperatures, and the values of the 'mean γ', have been determined by a method similar to that given in the last article. In this case, however, the difference in the energy content of the products of combustion at the beginning and end of expansion has been augmented by the heat of combustion during the expansion, and hence the resulting 'apparent K_v' is considerably increased.†

During compression the value of γ is that corresponding to the mean volumetric heat, throughout the respective temperature ranges, for the benzene-air mixture in cycle (*a*), for the gas-air mixture in cycle (*b*), and for air only in cycle (*c*). The values of K_v and γ are given, for the three cycles, in table 4.

TABLE 4

Values of the mean volumetric heat K_v, and of γ, during the compression strokes of the three cycles shown in fig. 9.

Cycle	*Gas compressed*	*Compression ratio*	*Temp. range. 100° C. up to*	*Mean K_v. C.H.U. per mol.*	γ	*Heat of adiabatic compression. C.H.U. per mol.*
a	air + benzene vapour	5	372° C.	5·847	1·340	1590
b	air + coal gas	7	480° C.	5·50	1·361	2090
c	air only	12	655° C.	5·41	1·367	3000

Apart from the compression ratio and the rise of temperature, the maximum pressures in cycles (*a*) and (*b*) are influenced by a change in the number of molecules present in the cylinder when combustion takes place. In cycle (*a*) this change is an increase of 3·6 per cent., and in (*b*) a decrease of 4·7 per cent. These figures have, of course, been affected by dissociation, which increases the number of molecules, at the maximum temperature, over and above what it would be if no dissociation took place. This point should become quite clear after reading the next chapter.

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† See Ref. 7, pp. 35 and 36.

III

FUELS AND THEIR COMBUSTION

ART. 14. Chemical equations and heats of reaction

Of the two chief components of air, nitrogen can be regarded as the 'sleeping partner', for it plays very little part in the chemical reactions which go on in an engine cylinder. It is merely drawn in, heated up through the activities of the oxygen, expanded, and then expelled through the exhaust valve without undergoing much chemical change. It is well to keep in mind from the beginning that, expressed by weight, very nearly three-quarters of the gaseous working substance in an engine cylinder consists of this inert nitrogen.

During the process we call burning, whether of a coal fire or of fuel in an engine, the active partner oxygen goes through a process of chemical combination with the elements of the fuel. These are thereby oxidized with the evolution of a definite amount of heat which depends upon the nature of the fuel. It is usual to speak of the fuel, but there is no reason why we should not equally speak of the air, being burnt, and it is often convenient to do so. For example, we say of a fire that we can get no more heat when the fuel is all burnt; but this is because the amount of air available is unlimited. In an engine cylinder, on the other hand, the amount of air is limited, and we often have an excess of fuel—in common parlance a 'rich mixture'—and in these circumstances the limit to the amount of heat produced is reached when the air is all burnt, and the excess of unburnt, or partially burnt, fuel passes out through the exhaust valve and is wasted.

The two elements forming the bulk of all fuels are carbon and hydrogen. Of these, hydrogen burns only to water, but carbon may be completely oxidized to carbon dioxide (CO_2) or only partially, if there is insufficient oxygen, to carbon monoxide (CO).

The several combustion processes, and the heat evolved by them, are expressed symbolically in the chemical equations:

$$(a)\quad \underset{(12)}{\text{C(graph.)}}+\underset{(32)}{O_2} = \underset{(44)}{CO_2}+94{,}270^{\dagger}\ \text{C.H.U.} \tag{24}$$

$$(b)\quad \underset{(24)}{2\text{C(graph.)}}+\underset{(32)}{O_2} = \underset{(56)}{2CO}+53{,}320\ \text{C.H.U.} \tag{25}$$

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† If the CO_2 were formed from amorphous carbon instead of graphite, this quantity would be about 3·2 per cent. greater. The heat of reaction in (b) would also be increased.

$$(c)\quad \underset{(56)}{2CO}+\underset{(32)}{O_2} = \underset{(88)}{2CO_2}+135{,}220 \text{ C.H.U.} \tag{26}$$

$$(d)\quad \underset{(4)}{2H_2}+\underset{(32)}{O_2} = \underset{(36)}{2H_2O}(\text{gas})+115{,}640 \text{ C.H.U.} \tag{27}$$

In each case the reaction is supposed to take place at constant pressure and to start and finish at room temperature.

Each letter in the chemical equations represents an atom of carbon, oxygen, or hydrogen, as the case may be, and the figures in brackets, which express the combining proportions by weight of each reacting body, are arrived at by multiplying the atomic weights (see table 38, p. 282) by the number of atoms present. Except at very high temperatures the atom of oxygen never exists independently, but only as the molecule O_2, and therefore in equations (*b*), (*c*), and (*d*) we introduce the minimum amount of fuel necessary to combine with a molecule of oxygen, as being the smallest separately existing unit.

The figures, besides giving the proportions by weight, are also the actual weights of the reactants in pounds which will produce the quantities of heat shown to the right of each equation, expressed in C.H.U. The reactants in equations (*c*) and (*d*) are all gases, and by Avogadro's hypothesis the volume of each gas will be in proportion to the number of molecules. Thus two volumes of CO combine with one volume of O_2 to form two volumes of CO_2. Moreover, if we choose to take as the unit of volume the 'pound-molecule', or mol. (359 cubic feet at N.T.P., see art. 4), then equations like (*a*), (*b*), (*c*), and (*d*) can be interpreted as expressing either combining weights or combining volumes, and the same figure for the heat of the reaction is equally true, whichever way is the most convenient to make use of the equation. This is the peculiar value of the pound-molecule as a unit. For example, equation (*d*) may be read either as telling us that 4 lb. of hydrogen combine with 32 lb. of oxygen to give 36 lb. of water vapour, and develop 115,640 C.H.U.; or as telling us that 2 mols. of hydrogen (718 cubic feet) combine with 1 mol. of oxygen (359 cubic feet) and produce 2 mols. of water vapour (718 cubic feet) with the same evolution of heat, namely 115,640 C.H.U.

Since by definition 1 mol. of any gas has a weight equal to the molecular weight of the gas, it follows that the number of mols. involved in any equation is the same as the number of molecules, and the number can be read off as soon as the equation is correctly written down. The volumes given above in cubic feet are what the volumes involved would be if reduced to N.T.P. They are given to remind the reader that at any moment the mol. can be turned into a concrete and familiar quantity, but as a rule there is no need to complicate the equation by doing so.

In equation (*d*) the water produced is supposed to have remained in the form of vapour, for this is the condition in which it leaves an internal combustion engine. If the vapour has been allowed to cool down and condense, the equation, of course, no longer expresses volumes, although the combining weights may still be used. It must be remembered, however, that an additional amount of heat will be yielded up to the surroundings on account of the latent heat of condensation. This amounts to 583 C.H.U. per pound, and we must therefore add a further $36 \times 583 = 21{,}000$ C.H.U. to the heat of the reaction if the steam after formation is allowed to condense.

If we extract from equations (*a*), (*b*), and (*d*) the heat derived from burning 1 pound of hydrogen to water vapour, and of carbon to CO_2 and CO, we obtain the following figures:

1 lb. of hydrogen burnt to H_2O (vapour) yields 28,910† C.H.U.
1 lb. of carbon (graphite) burnt to CO_2 yields 7,860 C.H.U.
1 lb. of carbon (graphite) burnt to CO yields 2,220 C.H.U.

The point to be noticed here is the very large heat of combustion of hydrogen, when this is compared with that of carbon on a weight basis; and also the comparatively small fraction of the total heat of combustion of carbon to carbon dioxide, which is evolved during the partial combustion to the monoxide. The high calorific value of hydrogen has an important bearing upon the volatile fuels for internal combustion engines, in that the aromatic series of hydrocarbons (see art. 16), which in many ways are ideal as fuel, suffer to the extent of about 10 per cent. in calorific value per pound as compared with the paraffin series, because of the lower proportion of hydrogen in their molecules.

Art. 15. Gaseous fuels

There are about six different types of gaseous fuel which are used for power generation. These are set out in table 5 so that their various compositions may be compared. It must be understood that the

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† Or allowing for the molecular weight of hydrogen being greater than 2, viz. 2·0156, this figure becomes 28,690.

percentages given for the constituents are no more than rough average figures. The composition of each type of gas may vary according to the details of its production.

TABLE 5

	Composition percentages by volume					
Type of gas	*Hydrogen*	*Methane, and other hydro-carbons*	*Carbon monoxide*	*Nitro-gen*	*Carbon dioxide*	*Calorific value (lower) in C.H.U. per st. cubic foot*
'Town gas' or coal gas	48	22·5	19	6	4·5	257
Coke-oven gas . .	53	30	10	5	2	220
Producer gas . .	15	2	28	50	5	85
Blast-furnace gas .	4	1	25	65	5	59
Natural gas . .	0	95	0	4	1	512
'Oil' gas . . .	6	85	0	6	3	600–820

On the gases in the table, ordinary town gas and coke-oven gas are very similar in composition, as indeed would be expected, for they are both produced by the heating of coal in retorts, the only difference being that town gas is produced at a gasworks with coke as a by-product, whereas the chief concern of a coke-oven plant is to produce a hard coke suitable for iron smelting, and here the gas is a by-product. So much so, that for many years all the gas not used in the coke making was burnt away to waste. Since the manufacture of the coke required for smelting one ton of pig iron sets free some 10,000 cubic feet of gas, and the annual output of pig iron is measured in millions of tons, one may get some idea of the waste of valuable fuel that was involved.

Producer gas is generated, usually in close proximity to the engine which is to use it, either by forcing or sucking steam and air through a mass of highly heated anthracite or coke, which are both nearly pure carbon. The water becomes split up into hydrogen and oxygen, and the latter, together with the oxygen in the air which accompanies the steam, combines with the carbon so as to give partial oxidation to carbon monoxide, CO. The heat generated during the formation of the CO maintains the temperature of the producer. The resulting gas is of low calorific value, since it contains a lot of inert nitrogen, but the simplicity and self-contained nature of the engine-cum-producer plant, combined with the high efficiency with which the gaseous fuel can be used, have given to producer gas great commercial importance.

Because of its low heating value per cubic foot it cannot usefully be employed where an engine of high power and small bulk is required. For further information upon gas-producer plants the reader should consult the books named under reference (4) on p. xiv.

Blast-furnace gas, as its name implies, is a by-product from the smelting of pig-iron. Its chief combustible constituent is half-burnt coke in the form of carbon monoxide. The heat evolved in burning one cubic foot of carbon monoxide to CO_2 is 188 C.H.U., as compared with 271 C.H.U. if the same volume of CO_2 had been formed direct from coke. The quantity of nitrogen present, and the fact that its only combustible is already partially burnt, means a very low calorific value, but where power is required in the neighbourhood of blast furnaces a large saving can be effected by using the gas which would otherwise be wasted. Large, slow-running engines are necessary, but it has been established that round about 25 horse-power can be derived per ton of iron produced per day.

Natural gas, which usually consists of nearly pure methane, forms an admirable engine fuel of high calorific value. It is obtained in enormous quantities from wells in the United States, and in the neighbourhood of oil-fields in other parts of the world. Its utility is limited to the neighbourhood where it is found owing to the cost of pipe-lines to transport it to any great distance. It is largely used in some districts in America for town lighting as well as power generation.

Oil gas[5] is a gas which was formerly made in large quantities for the lighting of railway trains. It is produced by the 'cracking' of oils by heating them to a temperature between 700° and 900° C. in a closed retort, the cracking temperature being varied according to the density of the gas required. Its composition is not unlike that of natural gas, but it contains, besides methane, a considerable proportion of ethylene, C_2H_4, and other higher hydrocarbons which give it a greater density and calorific value per cubic foot than natural gas. Its density can be varied according to the cracking temperature between limits of about 0·72 and 1·05, relative to air.

As an example of the chemical equation representing the combustion of a gaseous fuel we will take the typical coal gas of which the composition is given in table 5. We will further assume that of the 22·5 per cent. of hydrocarbons given in column 2, 21 per cent. was methane, CH_4, and that the other 1·5 per cent. was benzene, C_6H_6. If an analysis of the gas showed that it contained other more complicated constituents the equation could be extended accordingly, but as a rule such refinements are not necessary.

Remembering that the number of molecules of each gas in the equation is to be proportional to the volume of each constituent, we write down the equation as follows for one mol. of the gas with just enough air to oxidize the carbon and hydrogen in it completely to CO_2 and H_2O. Since the total volume of all the constituents of the gas is to add up to 1 mol., the numbers of molecules of each constituent are conveniently taken as equal to the fractions of a mol. of each one present, according to the percentage composition given in table 5, thus

$$\begin{aligned}
&\overset{\text{(GAS)}}{[0{\cdot}48H_2+0{\cdot}21CH_4+0{\cdot}015C_6H_6+0{\cdot}19CO+0{\cdot}06N_2+0{\cdot}045CO_2]}\\
&+\overset{\text{(AIR)}}{[0{\cdot}8675O_2+3{\cdot}263N_2]}\\
&=\overset{\text{(From GAS)}}{[0{\cdot}06N_2+0{\cdot}045CO_2]}+0{\cdot}945H_2O+0{\cdot}49CO_2+3{\cdot}263N_2\\
&=0{\cdot}945H_2O+0{\cdot}535CO_2+3{\cdot}32N_2. \qquad (28)\dagger
\end{aligned}$$

We can from this equation immediately write down the change of specific volume of the fuel-air mixture which has occurred on combustion, for if the total number of molecules before and after are added up, the total before combustion will be found to be 5·13 as compared with 4·80 for the final mixture. The ratio of specific volumes is therefore 0·936, and we find there has been a contraction of 6·4 per cent. on combustion.

In order to calculate the calorific value of the coal gas we have to know either the heats of combustion, or the heats of formation, of the two hydrocarbons CH_4 and C_6H_6. The heats of combustion can be applied directly to find the heat generated when the chemical reaction represented by equation (28) takes place or, if we have the heats of formation, then equations (24), (26), and (27) of the last article can be used to ascertain that

the formation of 1 mol. of H_2O vapour yields	57,820 C.H.U.
„ „ 1 „ CO_2 from C (graphite) yields	94,270 „
„ „ 1 „ CO_2 from CO yields	67,610 „

During combustion of the gas, 0·30 mol. of CO_2 was formed from

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† Dissociation is for the present neglected. As will be shown in art. 24 these products of combustion would at the maximum temperature contain a certain proportion of free oxygen and uncombined CO and H_2. Complete combustion to CO_2 and H_2O would not be achieved until after the temperature had fallen somewhat on account of expansion.

carbon and 0·19 from CO. Therefore during the combustion of 1 mol. of the coal gas there were

$$\begin{aligned} 0{\cdot}945\times 57{,}820 &= 54{,}650 \text{ C.H.U. produced by formation of } H_2O \\ 0{\cdot}3\times 94{,}270 &= 28{,}280 \quad ,, \quad ,, \quad ,, \text{ of } CO_2 \text{ from C} \\ 0{\cdot}19\times 67{,}610 &= \underline{12{,}850} \quad ,, \quad ,, \quad ,, \text{ of } CO_2 \text{ from CO} \\ & 95{,}780 \end{aligned}$$

This gives a total of 95,780; but the heat of combustion of 1 mol. of the coal gas is less than this because heat is given out to the extent of 18,150 C.H.U. per mol. when methane is formed from C (graphite) and $2H_2$, and the breaking up of the CH_4 molecules will absorb the same amount of heat. Liquid benzene from C (graphite) and hydrogen has a negative heat of formation of 11,600 C.H.U. per mol. and this will be increased by $104\times 78 \doteq 8{,}100$ when the benzene is in vapour form.

After subtracting $0{\cdot}21\times 18{,}150$ and adding $0{\cdot}015\times 19{,}700$ we arrive at the net total of 92,260 C.H.U. as having been produced by the combustion of 1 mol. of the gas. One mol. is 359 cubic feet at N.T.P., and we therefore obtain 257 C.H.U. per standard cubic foot as the lower calorific value of the gas.

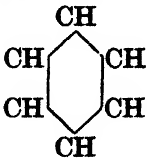
Other useful data which can be readily obtained from the equation are the density of the gas, and the ratio of gas to air in the 'correct' mixture, that is to say the mixture represented in equation (28), where combustion is complete as regards both fuel and air; all carbon and hydrogen having been oxidized to CO_2 and H_2O respectively, and no free oxygen remaining. This ratio by volume of gas to air can be read off at once from the number of mols. of gas and air, namely 1 and 4·13, before combustion, so that the fuel-air ratio in the correct mixture is 1:4·13. To obtain the density of the gas it is only necessary to multiply the number of mols. of each constituent by its molecular weight. This gives 14·47 as the total weight of 1 mol. of the gas. But the weight of a mol. of air is 28·95 (see art. 3), and therefore the density of the gas relative to air is 0·50.

ART. 16. Liquid fuels

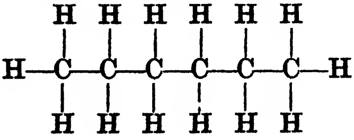
Although there is no clear line of demarcation between them, it is convenient to divide liquid fuels into the volatile and non-volatile classes. The former are those fuels which are normally used with a carburetter, and are carried into the engine cylinder with the air in a partially evaporated condition. The non-volatile class are the 'heavy oils' which are used in the Diesel type of engine by squirting into the cylinder through a spraying nozzle. For the present we shall be concerned with the volatile fuels only, discussion of the non-volatile class being deferred until art. 21, p. 56.

All of these, with the exception of alcohol, which contains some oxygen, are mixtures of what are called Hydrocarbons. These hydrocarbons, all similar in so far as their molecules contain nothing but carbon and hydrogen atoms, differ among themselves as to (*a*) the arrangement, and (*b*) the number, of atoms in their molecules. Difference of arrangement divides the whole field of hydrocarbons with which we are concerned into certain characteristic families, the most important being the Aromatics, the Naphthenes, and the Paraffins. The compounds in the first two of these families are of a type known as 'ring compounds' and those of the second are 'straight chain' compounds, these names being representative of the way in which the carbon atoms are associated in the molecules. The difference of arrangement is illustrated diagrammatically below, where an aromatic, Benzene, and a *normal*-paraffin,† Hexane, each with six carbon atoms in the molecule, have been taken as typical examples of a ring and a chain compound.

Ring compounds, e.g. benzene, C and H atoms associated thus:



Straight chain compounds, e.g. hexane, C and H atoms associated thus:



$$\begin{array}{ccccccccccccccc} & & H & & H & & H & & H & & H & & H & & \\ & & | & & | & & | & & | & & | & & | & & \\ H & - & C & - & C & - & C & - & C & - & C & - & C & - & H \\ & & | & & | & & | & & | & & | & & | & & \\ & & H & & H & & H & & H & & H & & H & & \end{array}$$

Besides the aromatics, naphthenes, and paraffins there are found also in those petrols which are produced by cracking processes considerable quantities of the Olefine family.

Each family of compounds has well-defined characteristics both as to their chemical reactions, and behaviour in an engine. As regards engine performance, the important feature in which they differ is in readiness to detonate, or knock. In this respect it is interesting that

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† References for the present are to n-paraffins only. The different characteristics exhibited by some of the isomeric paraffins, in which a single straight chain is replaced by branching chains, are referred to in arts. 19 and 29.

the ring compounds, aromatics and naphthenes, are superior to the rest. The aromatics are by far the best, the *n*-paraffins the worst, and naphthenes occupy an intermediate position. Olefines when freshly manufactured are good from an anti-knock point of view, but are somewhat unstable, and so difficult to obtain otherwise than as admixtures with the other types that their characteristics have not been quantitatively worked out. The olefines are of a class known as 'unsaturated' bodies, by which is meant that the number of hydrogen atoms in the molecules, in relation to the number of carbons, is not sufficient to satisfy all the valencies of the associated carbons, with the result that as a class the olefine molecules are chemically very active and exhibit a tendency to 'polymerize' or join together and form larger and more complicated molecules. When freshly manufactured the olefines make an excellent fuel, but during storage, more particularly under the influence of light and warmth, their tendency to polymerize may lead to the formation of gumlike substances which, in sufficient concentration, would render a spirit unsuitable as an engine fuel. The polymerizing tendency, however, varies greatly for different olefines, and it has been found possible to 'crack' the higher paraffins to olefines under conditions in which almost no polymerizing bodies are formed.

In each family there is a range of bodies, differing as to the number of atoms in the molecule; and in each family, as the number increases, the boiling-point of the substance is found to be higher. Benzene, with six carbon atoms, is the smallest molecule of the aromatic family, but the paraffin family starts at methane, with one carbon atom, and the first four members of it are gases at ordinary temperatures. Table 6 gives the formulae, specific gravities, and boiling-points of certain members of each series which are of major importance in volatile fuels for internal combustion engines. An average good class of aviation spirit would contain some 50 per cent. of mixed paraffins between pentane and decane, 30 per cent. of aromatics, and 20 per cent. of naphthenes. Cyclohexane is the only one of the latter class given in the table since it is typical, and also it is the only one which is available in bulk and of good purity, so that its characteristics in an engine have been separately determined. The naphthenes as they occur naturally in petrols are extremely difficult to separate from the other constituents, but cyclohexane is now produced synthetically in considerable quantities.

TABLE 6

Hydrocarbons of different types and boiling-points which go to make up a volatile liquid fuel.

(*Data from Landolt and Börnstein's 'Tabellen'*)

PARAFFIN

	Formula	*Density at 20° C.*	*Boiling-point*
			° C.
Methane	CH_4	gas	−164
Ethane	C_2H_6	„	−88·3
Propane	C_3H_8	„	−44·5
Butane	C_4H_{10}	„	0·6
n-Pentane	C_5H_{12}	0·6263	36·0
n-Hexane	C_6H_{14}	0·660	68·7
n-Heptane	C_7H_{16}	0·684	98·4
n-Octane	C_8H_{18}	0·7027	125·6
n-Nonane	C_9H_{20}	0·718	150·7
n-Decane	$C_{10}H_{22}$	0·730	174·0
General formula	C_nH_{2n+2}		

AROMATIC

	Formula	*Density at 20° C.*	*Boiling-point*
Benzene	C_6H_6	0·879	80·0
Toluene	C_7H_8	0·866	110·5
m-Xylene	C_8H_{10}	0·864	139·2
General formula	C_nH_{2n-6}		

NAPHTHENE

	Formula	*Density at 20° C.*	*Boiling-point*
Cyclohexane	C_6H_{12}	0·7784	80·9
General formula	C_nH_{2n}		

A typical distillation curve for a volatile spirit is given in fig. 10. There is a small proportion of very volatile spirit which begins to distil over at 35° C. This is very valuable for promoting easy starting in cold weather. The bulk of the spirit distils over at temperatures between 60° and 120° C., and only a few per cent. of the 'heavy fractions' remain by the time the temperature has reached 160° C. The 'end point' comes at about 180° C.

Each of the series of compounds shown in table 6 continues right on, with rising boiling-points, through kerosenes and Diesel fuel oils, until we come to the constituents of the mineral lubricating oils which have boiling-points in the region of 400°–500° C.† Accurate analysis of any but the volatile fractions is at present beyond the skill of the chemist, but even the heavy lubricating oils exhibit a characteristic behaviour in any engine according as they are derived from 'paraffin base' or 'naphthenic base', or other types of crude oil.

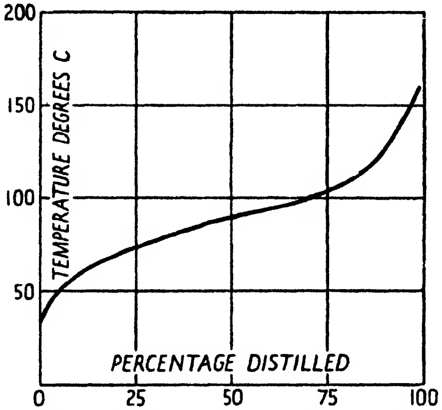

FIG. 10. Typical distillation curve for aviation spirit.

The crude oil as it comes from the wells contains every variety of hydrocarbon, from thick, tarry stuff, down to methane, which is blown off as a gas in enormous quantities. The crude oil is heated in a still, and the various 'fractions' which distil over are condensed in different receivers according to their boiling-points, and the purposes for which they are to be used: the light petrol first, then the kerosene fractions, then gas-making oils and fuel oils, and finally the lubricating oils.

ART. 17. The constitution of 'benzol' and 'petrol'

These are both colloquial terms which are used for substances of which the composition may vary very widely. Broadly speaking, however, benzol is a mixture of the first three members of the aromatic series, chiefly benzene, while petrol contains members of three, and possibly four, different families of hydrocarbons, with a preponderance of the paraffin series. It is only petrols produced by a 'cracking process' which in general contain members of the olefine series.

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† Most of the constituents of a lubricating oil would probably begin to decompose before any true boiling-point was reached.

Aromatics are very much superior to *n*-paraffins in regard to the important matter of their tendency to detonate. Naphthenes and olefines, in this respect, occupy an intermediate position between the other two. To set against the virtues of the aromatics in regard to detonation, their chief member, benzene, has the comparatively high freezing-point of +5·4° C. and a calorific value per pound about 10 per cent. lower than the paraffins. In spite of these drawbacks a certain proportion of aromatics is practically essential in any fuel for a high-duty engine, unless some 'dope' is used in their stead.

The superiority of petrol derived from wells in certain parts of the world such as Borneo lies in the fact that the oil as it comes from the wells happens to contain an unusually large proportion of the aromatic series. Petrol from the Persian wells, on the other hand, contains a large proportion of the paraffin series, little of the naphthenes, and only 10–12 per cent. of the aromatics, and benzol, therefore, or other material to inhibit detonation, has to be added in order to bring this petrol up to the standard required to-day. The addition of benzol to petrol simply increases the proportion of aromatic hydrocarbons of which the natural petrol already contains a certain amount, which depends upon its country of origin. The proportion of aromatics cannot be increased beyond about 50 per cent., if the fuel is to be used in conditions where it may be cooled to temperatures below about −20° C., on account of the danger of their separating out as solids and so blocking fuel pipes and carburetters.

ART. 18. The sources of petrol and benzol

After removal of the more volatile fractions from the crude oil by direct distillation, that which remains behind would become a drug upon the market unless some use could be found for it. In consequence the method of treating the heavier fractions by 'cracking' has become more and more prominent in recent years. It has been found that if the heavy fractions are passed through tubes at a high pressure and temperature, and in the absence of air, the products which finally emerge contain quite a large proportion of the lighter fractions suitable for petrol. Under the high pressure and temperature of the cracking plant the large molecules become split up into two or more smaller ones of low boiling-point hydrocarbons. The cracking can be made to take place while the oil, during its passage through the plant, is either in the liquid or evaporated condition ('liquid phase' and 'vapour phase' processes) by manipulation of the pressures and temperatures of the plant. The products of the cracking process usually contain a large proportion of the unsaturated olefine series, which if diluted with straight run spirits form an excellent engine fuel, but have an evil propensity to form gummy substances during storage. The gums which are deposited are derived from a kind of reverse of the cracking process —an agglomeration of small molecules to form large and complicated ones—which can be minimized by careful purification and by the addition of substances which act as chemical stabilizers; but no really satisfactory, or at least satisfactorily cheap, way of treating cracked spirits has so far been worked out for rendering them suitable for high-duty engines when not diluted with straight-run spirits, for the gum formation to which they are liable is always accompanied by a serious decrease in the anti-knock value. There is always the danger, also, that the gummy substances will be deposited upon the stems of inlet valves and lead to the valves sticking open.

Coal is composed, besides carbon, of a wide range of hydrocarbons from thick, heavy tars up to benzene and other volatile spirits, which can be distilled off by heating, and collected, much as they are after distillation from crude oil. Distillation from coal is the chief source of benzol, for the volatile fractions collected are found to consist almost wholly of members of the aromatic series. The products of the distillation vary according to the temperature at which it is carried out. From high-temperature retorts, at about 1,350° C., the products are a dry coke and coal gas, which consists of a large proportion of hydrogen and carbon monoxide, with a comparatively small proportion of benzene and other hydrocarbons. If the coal is distilled at a lower temperature, not higher than 650° C., the process yields 'coalite', which still retains a fair proportion of the heavy hydrocarbons; a large proportion of distilled hydrocarbons extending over a wide range of boiling-points; and very little hydrogen. The hydrogen in coal gas is derived from the breaking up of hydrocarbons during the high-temperature distillation, apart from a small amount of free hydrogen which is present in the coal.

ART. 19. Synthetic petrol

A process for synthesizing hydrocarbons by the reaction of hydrogen with coal has been developed both here and in Germany, and many thousands of tons of petrol per annum are now produced in this way.

The importance of the process to this country lies in the fact that it is able at the same time to utilize our coal resources and to set us free from the present necessity of importing nearly all our liquid fuel. In this process hydrogen, at a pressure of about 250 atmospheres, is brought into contact with powdered coal in retorts at a temperature of 450°–480° C., in the presence of a catalyst. Direct combination of the carbon and hydrogen takes place, with the formation of a great variety of hydrocarbons. The powdered coal, before being introduced into the retorts, is mixed into a paste with heavy oils which are themselves products of the hydrogenation process. In this way a continuous procedure is maintained: coal and the heavy products of previous hydrogenation are continuously introduced into the retorts, to yield liquids of every degree of volatility. Of these the volatile fractions are drawn off and separately condensed while the non-volatile which remain behind are made to serve their turn as the vehicle for introducing further coal. The process is fraught with practical difficulties, not the least of which is that of obtaining a material for the retorts which will withstand the severe conditions for a reasonable length of time.

TABLE 7

Showing yield of products from the hydrogenation of 1 *ton of coal.*

Quantity of hydrogen used 114 lb., or 5·1 per cent. of the weight of coal.

	lb.	
Fraction 0–170°	83	1,130 lb. of ash-free dry product per ton of coal.
,, 170–230°	208	
,, 230–270°	197	
,, 270–310°	105	
,, 310–360°	208	
Pitch	329	
Gas benzene	42	
Gas	325	
Unconverted coal	363	
Water	179	
Coal ash	161	
Loss	154	
	2,354	

Official figures[6] from an experimental plant are given in table 7, from which it will be seen that although from one ton of coal about ½ ton of liquid hydrocarbons of some sort were obtained, there were only 83 lb., or about 10 gallons, of spirit with a boiling-point below 170° C.

The working of a large-scale commercial plant has been described by

Gordon,[52] who states that to produce 1 ton of petrol 3·65 tons of coal are required; and that the overall thermal efficiency of the plant works out as 43 per cent., taking 8,050 as the calorific value of the coal and 11,000 as that of the petrol produced.

The type and the yield of the light spirits can be varied to some extent by alteration of the catalyst, of the temperature of the retorts, and of the rate of feed of the raw materials. And it may also be increased by further treatment of some of the less volatile products of the primary hydrogenation, either by simple cracking or cracking combined with further hydrogenation. These extra treatments, however, all increase the cost of production. Although the production of motor spirit from coal on a large scale is now technically an accomplished fact, experience, or at any rate published information, is not yet sufficient to make it possible to draw conclusions as to the basic cost of production compared with petrol from the oil wells. The two main factors in the cost of synthetic petrol are (*a*) the expense of producing the necessary hydrogen, in the first place, and (*b*) the large capital expenditure on the huge steel retorts, or converters, and their accessories, capable of withstanding a pressure of 250 atmospheres.

Apart from this synthesis from coal, the demand for fuels of very high 'anti-knock' quality has led to the synthesis of certain special compounds on a large scale for admixture with the ordinary products of distillation, of which the most important is iso-octane. This is an isomer of the normal octane, a straight-chain paraffin, in which the single chain of eight carbon atoms is replaced by a 5-atom chain with 3 methyl (CH_3) groups attached. It is produced by hydrogenation of di-iso-butylene, a compound usually obtained as a by-product from the cracking process. Another special compound which may be produced in sufficiently large quantities to be important is iso-propyl-ether.

ART. 20. The alcohols as fuels

Mention must be made of methyl alcohol, CH_4O, and ethyl alcohol, C_2H_6O. Although at present not extensively used as engine fuels, the alcohols might at any time assume a vast importance if the supply of the plain hydrocarbon fuels should fail, for they can fairly easily be manufactured from vegetable matter, and are even now manufactured on a large scale from water gas, both in this country and abroad, by high-pressure synthesis in the presence of a catalyst. We shall see, too, that the alcohols exhibit certain unique and most valuable properties when used in an engine which even now, in spite of their high cost, *lead to their employment for special purposes.*

Both ethyl and methyl alcohol are highly volatile liquids, having boiling-points of 78° and 65° C. respectively. Their specific gravities are each about 0·79;† and a further point of interest lies in their latent heats of evaporation which are about three and three and a half times that of petrol respectively.

The great weakness of the alcohols as fuels is their low heating value. The burning of one pound of commercial ethyl alcohol only produces 6,360 C.H.U., as against about 10,000 for petrol. Apart from the quite unavoidable result of having to carry about 50 per cent. greater weight of fuel with an engine to achieve the same total effect, this means that to compete with petrol on a basis of cost, alcohol would have to be purchasable at about two-thirds of the price, instead of, as at present, about three times that figure.

ART. 21. Non-volatile liquid fuels

The recent rapid development of the high-speed compression-ignition engine has made it clear that the characteristics of the fuel used, both chemical and physical, may be no less important in their effect upon the performance of this type of engine than they are in the petrol engine.

A discussion of the combustion phenomena in the compression-ignition engine, and their relation to the nature of the fuel used, will be found at the end of Chapter V. The present article will be devoted to a short review of the characteristics of the non-volatile class of fuels, appropriate to this type of engine. Until recently there has been little exact information available, and even now it is very incomplete, largely because the analysis of an oil into its constituents, each in a pure state, becomes progressively more difficult as boiling-points and molecular weights increase.

The sources of the non-volatile or 'heavy-oil' types of fuel are generally the same as those of petrol, except that a considerable quantity of 'shale oil' is derived from the distillation of shale, and this is a source which yields very little of the volatile spirits. Both physically and chemically the shale oils can be matched by very similar oils produced in the normal way by distillation from crude oils, and the shale-oil industry is therefore to be regarded simply as a modest

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† The commercial spirits are always rather higher owing to the presence of water.

additional source of supply of a certain chemical type of oil. It forms a satisfactory engine fuel, and has the added interest of being the only natural oil available in this country.

If one includes the slow-running engines, the range of volatility and viscosity of the fuel oils is very wide indeed, for engines can be built which will burn almost any oil that is fluid enough to be injected through a nozzle. High-speed engines, however, are limited to the less viscous oils. It was stated in art. 16 that the 'end point' for a volatile spirit to be used in a carburetter would be about 180° C., and for aviation spirit it is usually limited to about 160° C. As compared with this, the distillation range for heavy oils begins at about 215°–220° C. and finishes in the neighbourhood of 350° C. for high-speed engines, although in some designs the final boiling-point may be considerably higher.

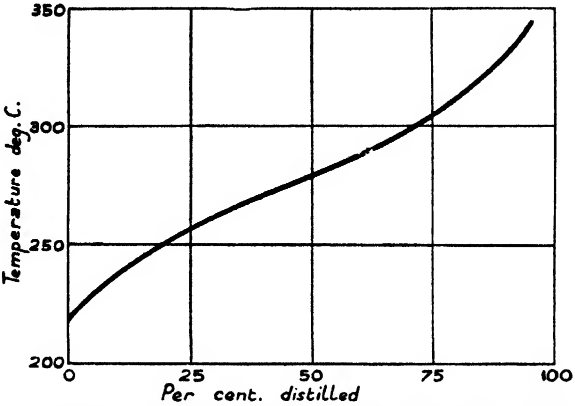

FIG. 11. Typical distillation curve for high-speed Diesel engine fuel.

A typical distillation curve of an oil which has been largely used in high-speed engines is shown in fig. 11. It must be added that such a curve may have a much less definite meaning for a heavy oil than it has for a volatile spirit, because it is very possible that decomposition, by cracking of the larger into smaller molecules, may occur at the higher temperatures. Most heavy oils, for this reason, leave a considerable weight of solid residue behind, when the final boiling-point has been reached. The vapours distilled, once cracking has begun, will of course be chemically quite different from the original oil.

In almost every respect the desirable qualities in a heavy oil are the reverse of those in a volatile spirit. A high volatility and a high temperature of spontaneous ignition, or 'self-ignition temperature' (S.I.T.), which are both virtues in a volatile spirit, are to be avoided in a fuel for compression-ignition engines; and this makes the greater chemical stability of the aromatic series, so welcome in benzol, an almost prohibitive drawback to having any large proportion of aromatics in a heavy-oil fuel.

It has been said above that the boiling-point range among the heavy oils used for direct injection may be wide; but this does not mean that in any one fuel it is not an advantage to restrict the range within certain limits. What the limits should be will depend upon the speed and type of engine it is intended for and upon the question of cost; for a 'narrow cut' is necessarily more expensive, and may be prohibitively so.

For high-speed engines the most satisfactory boiling-range will probably be about 220°–350° C., with a distillation curve as shown in fig. 11. A less volatile oil will tend to be inconveniently viscous at low temperatures for high-speed operation of fuel pumps and injection nozzles; while, on the other hand, it has been suggested that an oil is the more prone to produce 'Diesel knock' (the equivalent of detonation in a petrol engine) the greater its volatility, and this, therefore, makes a high initial boiling-point a desirable feature. Moreover, a low-viscosity oil may be expected to lead to more rapid wear in the fuel pumps: an important point, since the correct metering of the fuel to the cylinders depends upon the maintenance of very fine clearances in the pumps.

Since the oil has to be ignited in the engine cylinder by the heat of compression of the air only, a low self-ignition temperature is of primary importance. If a laboratory apparatus could be devised for examining the S.I.T. of oils which reproduced satisfactorily the conditions of self-ignition in an engine cylinder, it is probable that fuels might be graded for suitability and smooth running in terms of this one property. Actually, however, it has been found advisable to fall back on their behaviour in a test engine for assessing the quality of fuels, comparing them with the behaviour of mixtures of two pure standard substances, cetene and α-methylnaphthalene, the one a good and the other a bad fuel.[53, 54] The 'rating' of the fuel is expressed as its 'cetene number', i.e. the percentage of cetene in the mixture giving similar performance to the fuel to be tested, just as the 'octane number' indicates the quality of a volatile fuel in regard to detonation. It has been found possible to grade the majority of types of Diesel fuel satisfactorily in this way, but Boerlage[53] and others have pointed out that where 'dopes' have been added to lower the S.I.T. of the fuel the only conclusive test of quality will be one made on the engine in which the fuel is to be used.

The nearest approach to a satisfactory S.I.T. apparatus for heavy oils is probably that described by Helmore[51] in which a fine jet of the oil is sprayed into heated air at atmospheric pressure and of known temperature. Although the apparatus cannot be trusted to give a satisfactory method of grading fuels any more than an observation of the 'aniline point', it serves to bring out an important characteristic of the combustion of such fuels. For example, no simple observation of the temperature at which ignition occurs can be made, because it is found that large errors will occur unless the time-interval between the moment of injection and the moment of ignition is taken into account. There is, for every oil, a limiting temperature below which it will never ignite, but if the air into which it is injected is only slightly hotter than this, the spontaneous ignition may occur only after several seconds delay. An important source of uncertainty in the working of the apparatus is the extent to which surface effects at the hot walls of the containing vessel may promote ignition with some oils more than others.

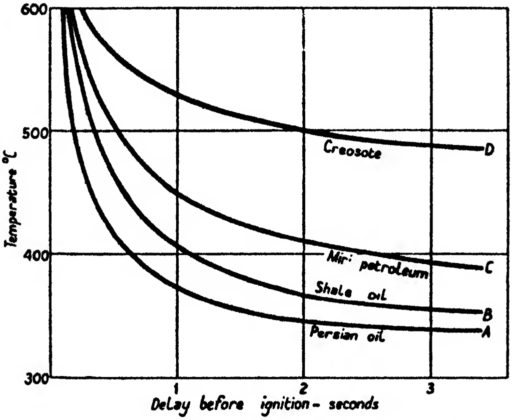

FIG. 12. Variation of ignition delay with temperature.

The delay, or induction period, before ignition is of the greatest importance in the burning of all hydrocarbon fuels, and much will be heard of it in the later chapters of this book. The curves of fig. 12 show the observed values of the delay before ignition for several oils when tested in Helmore's apparatus. It will be seen that the minimum temperature at which spontaneous ignition would occur varied from

340° C. for oil *A* to 485° C. for oil *D*. Ignition only occurred at these temperatures after 3 or 4 seconds delay, and even when the air into which oil *A*, for example, was injected was as much as 60° C. above the minimum ignition temperature, there was a delay of 0·6 sec. between injection and ignition.

At a temperature of 600° C. there is little to distinguish the different oils from one another, but as the temperature is lowered the delay of each one before ignition rapidly increases when its minimum S.I.T. is approached.

It may be objected that delays of a second or more can bear little relation to behaviour in an engine where combustion is complete in a few thousandths of a second. There is a good deal of evidence, however, that the characteristics at any rate of the mineral oils, as expressed by their S.I.T.s and delay periods observed at atmospheric pressure in the way described, can be correlated with engine performance, in spite of the speeding-up process in the cylinder. Of the four oils shown in fig. 12, that marked *A*, with S.I.T. 340° C., was a high-grade Persian gas oil with a high percentage of paraffins, excellent for smooth running in an engine; *B* was a shale oil, also with a high percentage of paraffins and nearly, but not quite, as satisfactory in most engines as oil *A*; *C* was an oil from a crude with a high proportion of aromatics, and was found to be very unsatisfactory as an engine fuel; while *D* was a creosote, or coal-tar, oil, distilled from coal, which could not be made to ignite in an engine at all when tried as a fuel by itself.

From comparative data given by Dicksee[48] on the behaviour of a number of oils in the same type of engine it is clear that the satisfactory oils are those which combine an S.I.T. in the neighbourhood 340°–360° C. with a comparatively low specific gravity, between 0·82 and 0·86. The higher density oils, with specific gravities from 0·86 to 0·94, are those which contain a larger proportion of aromatics and naphthenes. These all have S.I.T.s ranging from 380°–475° C. and their classification as regards their behaviour in the engine ranges from 'poor' to 'very bad'.

ART. 22. Calorific values and latent heats

By calorific value† is meant the heat measured in a calorimeter when unit weight of a fuel is burnt with air or oxygen, and the products of

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† If the volume of the products of combustion differs from that of the original unburnt constituents, then a distinction should be drawn between determinations of calorific value made at constant pressure and those made at constant volume, for in the former case the result is affected by work done on or by the atmosphere.

combustion are cooled down to room temperature. In these circumstances the measured heat value includes the latent heat of condensation of the water vapour formed by combustion of the hydrogen in the fuel. Take away this latent heat and we get the heat available from the reaction when the final product is *water vapour*. But it is in this form that the H_2O is expelled from the engine, for the exhaust gases are far above the condensation temperature. It is therefore customary, when calculating the heat supplied to an engine, to use this 'lower calorific value', which may be roughly defined as the heat measured in a calorimeter at room temperature, less the latent heat of evaporation of the water formed during combustion. This heat amounts to 15·4 per cent. of the gross heat of combustion—at constant pressure—of the hydrogen in the fuel.

The heat-producing part of nearly every fuel we are concerned with, liquid or gaseous, is either hydrogen by itself or carbon and hydrogen combined in the molecule of a hydrocarbon. The alcohols and carbon monoxide are the only exceptions. It has been mentioned that, as compared with other liquid fuels, alcohol has a very low heat value, and we may regard this as being due to the fact that some of its combustible material is already combined with oxygen in the fuel molecule itself, so that it is not available for producing heat by oxidation with air.

All liquid fuels are evaporated before they are burnt in an engine cylinder, and we need no longer think of them as different from gaseous fuels except in one particular: the possession of a latent heat of evaporation. This affects matters in two ways. In the first place it introduces a modification into the figure we must take as the effective calorific value of the fuel. We have seen above that when calculating the heat supplied to an engine we must subtract from the calorific value, as measured in a calorimeter, the latent heat of the water formed by condensation after combustion. In addition to this correction the latent heat of the fuel itself comes in in the following way. Part of the heat developed during combustion in a calorimeter is used up, when the fuel is a liquid, in turning the liquid into a vapour. In other words, if the same fuel had been supplied to the calorimeter in the form of a vapour, a higher heating value would have been recorded, by an amount equal to the latent heat of the liquid fuel. Now when a liquid fuel is supplied to an engine by means of a carburetter it is, in effect, supplied in the form of a vapour because evaporation is practically always complete by the time the inlet valve has closed. In estimating the heat supply to the engine *cylinder*, therefore, we must, when dealing with a volatile liquid fuel, add to the lower calorific value the latent heat of the fuel itself. In the case of pure benzene, for example, the lower calorific value is 9,612 C.H.U. per lb., and the latent heat 97 C.H.U. The effective heating value to be taken in engine calculations is therefore 9,710† C.H.U. per lb. The foregoing argument applies only to carburetter engines employing volatile liquid fuel. In engines of the Diesel type, where the fuel is injected into the cylinder in the liquid state, the available heat per pound of fuel supplied is simply the lower calorific value.

The second effect of the latent heat of a liquid fuel is in cooling the fuel-air mixture as it passes from the carburetter into the cylinder. As already stated, evaporation is substantially complete, with all fuels except alcohol, by the time the inlet valve is closed, and meanwhile the latent heat of evaporation has been supplied partly from the hot induction pipes, valves, and cylinder walls, and partly by the incoming air itself. That part of the necessary heat which has been drawn from the air is sufficient to produce a considerable lowering of its temperature. As regards the drop in air temperature, when the *whole* of the heat is drawn from the air and evaporation is complete before the inlet valve closes, if temperature of evaporation is taken as 10° C. we get

drop for correct air-benzene mixture = 33° C.,

drop for correct air-heptane mixture = 25° C.

The drop for a petrol-air mixture would be a little more than for heptane, depending on the percentage of aromatics which it contains. Ethyl alcohol has at 10° C. a latent heat of 223 C.H.U. per lb., or about 2½ times that of an average petrol. In consequence it forms an exception in that, although of high volatility, it is as a rule impossible to get sufficient heat supplied to the mixture to evaporate all the fuel before the valve closes.

This cooling by evaporation of the fuel has a very appreciable effect on the maximum power output with all liquid fuels, and in the case of alcohol the cooling is so large that it also raises the thermal efficiency of an engine above the best obtainable on petrol at the same compression ratio, through a lowering of the temperatures of the working substance all round the cycle.

Data in regard to the heats of combustion of both liquid and vapour-

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† Or slightly less at 100° C.

ized fuels are given in tables 39 and 40 on pages 284 and 285, and these should be studied in connexion with the further note on calorific values in art. 25.

Art. 23. Combustible mixtures

One example of a chemical equation, representative of the combustion of an average coal gas with air, has already been given in art. 15. In the reaction there represented, just enough air was associated with the fuel for all the hydrogen of the mixed hydrocarbons to be burned to water and all the carbon to carbon dioxide. When this is so, the fuel-air mixture is said to be 'correct'. If there be not enough air with the fuel in a combustible mixture to allow of complete oxidation the mixture is 'rich' (in fuel), and if there is more than enough air, so that free oxygen remains after combustion, it is 'weak'. The ratios by weight and volume of the fuel to air in the correct mixture will depend upon the nature of the fuel, but for any fuel and any mixture strength, once the chemical equation is written down, the combining proportions can at once be ascertained.

As examples of the combustion of liquid fuels, let us take two which for simplicity will be assumed to be pure benzene, C_6H_6, and pure heptane C_7H_{16}. The one is the chief constituent of benzol and the other a typical member of the paraffin series in petrol. Complete combination of the fuel and air would take place according to the equations:

$$\underset{\substack{\text{wts. } 156\\ \text{vols. } 2}}{2C_6H_6\,(\text{vapour})} + \underbrace{15O_2 + 56{\cdot}4N_2}_{\substack{2060\\ 71{\cdot}4}}$$

$$= \underbrace{12CO_2 + 6H_2O\,(\text{vapour}) + 56{\cdot}4N_2}_{\substack{2216\\ 74{\cdot}4}} + 1{\cdot}515^{\dagger}\times 10^6\ \text{C.H.U.} \tag{29}$$

$$\underset{\substack{\text{wts. } 200\\ \text{vols. } 2}}{2C_7H_{16}\,(\text{vapour})} + \underbrace{22O_2 + 82{\cdot}8N_2}_{\substack{3020\\ 104{\cdot}8}}$$

$$= \underbrace{14CO_2 + 16H_2O\,(\text{vapour}) + 82{\cdot}8N_2}_{\substack{3220\\ 112{\cdot}8}} + 2{\cdot}154\times 10^6\ \text{C.H.U.} \tag{30}$$

In these equations, as before, the left-hand side shows the associated weights and volumes of fuel and air before combustion, and the right-

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† This value, which has been used throughout this book, is about 0·34 per cent. less than the probable value.

hand side the products after combustion is complete. The particular values given for the heats of combustion in these equations are those which apply when the initial temperature of the 'mixture' on the left-hand side is 100° C.—the reaction proceeds in such a manner that no energy is communicated to, or received from, the surrounding atmosphere, e.g. in a closed vessel—and the final temperature of the products of the combustion is 100° C. Such calorific values are required in connexion with the calculation of maximum explosion temperatures in the next article—calculations in which we determine, in effect, what the condition of these products would be in regard to temperature, pressure, dissociation, etc., if these specific quantities of heat were re-imparted to them.

The combustion of a rich or a weak mixture may be represented, say for benzene, by writing

$$2(1+\alpha)C_6H_6+15O_2+56{\cdot}4N_2 = A\,CO_2+B\,CO+C\,H_2O+D\,H_2+E\,O_2+56{\cdot}4N_2 \qquad (31)$$

in which α is the 'degree of richness'. If the mixture is 5 per cent. rich, $\alpha = +0{\cdot}05$; if it is 5 per cent. weak, $\alpha = -0{\cdot}05$, and so on for other percentages.

What the coefficients A, B, C, D, E will be, that is to say what will be the proportions of the various gases in the products of combustion, will depend upon the temperature, and to a less extent the pressure, produced by the combustion. Were it not for the phenomenon of dissociation there would never be in the same mixture pure oxygen together with hydrogen and CO. But, as stated in Chapter II, a proportion of the CO_2 and H_2O molecules will become split up into CO, H_2, and O_2 when temperatures are high; and the proportion dissociated increases rapidly above 2,700° C. At these temperatures the three gases exist together in equilibrium. As soon as the temperature in the cylinder falls, on account of expansion, recombination will take place so to leave some residual oxygen if the mixture is weak and CO and H_2 if it is rich. The amount of dissociated water vapour even at the maximum temperature is not great under ordinary engine conditions, but the dissociation of CO_2 is an essential factor in explaining certain well-known features of engine behaviour.

Dissociation will be dealt with more fully in the next article, and it will suffice here to state that by the use of certain experimentally determined figures concerning the conditions under which CO, H_2,

CO_2, and O_2 can coexist in equilibrium—the 'dissociation constants'—we are able to calculate the five coefficients A, B, C, D, E of equation (31), when the temperature and pressure in the cylinder are known, and so to find the proportions of the gases in the products of combustion. It may be mentioned again that the maximum temperature reached will itself be affected by the dissociation of the CO_2, since all the available heat of combustion of the fuel will not have been generated so long as there exist together uncombined CO, hydrogen, and oxygen.

In the products of the combustion of a correct mixture of benzene and air (eq. (29)) there will be about 4 per cent. by volume of CO at the maximum temperature, rising to 15 per cent. when there is a 50 per cent. excess of fuel ($\alpha = +0{\cdot}5$). If the mixture be 20 per cent. weak ($\alpha = -0{\cdot}2$), there will be about 4·7 per cent. of free oxygen, falling to 2·2 per cent. at the correct mixture. In the 50 per cent. rich mixture the free oxygen at the maximum temperature is practically zero, while in the 20 per cent. weak mixture the CO at the maximum temperature is about 1·3 per cent.

ART. 24. Calculation of the products of combustion, allowing for dissociation

It was shown by Tizard and the present writer[7] that the well-known characteristic of a petrol engine, that it yields its maximum power when running on a fuel-air mixture about 10–20 per cent. richer in fuel than that giving complete combustion (see art. 36), is just what would be expected from theory when dissociation is taken into account. A full account of the methods of calculation will be found in the original paper to which reference is given above, together with results worked out for typical fuels. We had to rely at that time upon the experimental results which were then available on the values of volumetric heats and of the dissociation constants drawn from the work of Nernst, Bjerrum, and others.

Recently a large amount of work has been done in the U.S.A. and elsewhere on the determination of these quantities from the band spectra of the various gases. As these values have, in general, received considerable support from explosion experiments, they have been used in the calculations which follow, and hence the results given here will differ from those to be found in reference (7). The particular values of the heat capacities and dissociation constants that have been drawn upon are those that have been collected together and published by Lewis and von Elbe.[56]

The maximum temperature in a petrol engine cylinder may be of the order of 2,500°–2,700° C., and the gases in the cylinder under these conditions will, in the main, be a mixture of N_2, CO_2, H_2O, CO, H_2, and O_2. The last five are able to coexist in equilibrium at these high temperatures, as stated above, in certain definite proportions which depend mainly upon the temperature. The proportions in which the different gases will be present are defined by two 'dissociation constants', K_1 and K, keeping the same symbols as in the original paper.

The constant K_1 connects the partial pressures (in atmospheres) of CO_2, CO, and O_2 existing together in a mixture of gases at absolute temperature T in the following way:

$$\frac{(P_{\mathrm{CO}})^2 \times P_{\mathrm{O_2}}}{(P_{\mathrm{CO_2}})^2} = K_1 \tag{32}$$

and

$$\log_{10} K_1 = 8{\cdot}593 - \frac{28{,}600}{T}, \tag{33}$$

where P_{CO}, etc., represent the partial pressures of CO, CO_2, and O_2, which will, of course, be proportional to the numbers of molecules of each gas present in the mixture.

A further relationship between the partial pressures of the gases present is provided by the second constant K in the following equations:

$$\frac{P_{\mathrm{CO}} \times P_{\mathrm{H_2O}}}{P_{\mathrm{CO_2}} \times P_{\mathrm{H_2}}} = K \tag{34}$$

and

$$\log_{10} K = 1{\cdot}2065 - \frac{1{,}100}{T}. \tag{35}$$

Values of K_1 and K evaluated for the range of temperature 2,000°–3,000° C. abs. are given in Table 8.

TABLE 8

Values of the dissociation constants K_1 and K at various temperatures.

Temperature	K_1	K	*Temperature*	K_1	K
°C. abs.	from equation (33)	from equation (35)	°C. abs.	from equation (33)	from equation (35)
2,000	$1{\cdot}963 \times 10^{-6}$	4·534	2,600	$3{\cdot}917 \times 10^{-3}$	6·074
2,100	$9{\cdot}419 \times 10^{-6}$	4·816	2,700	$1{\cdot}002 \times 10^{-2}$	6·298
2,200	$3{\cdot}917 \times 10^{-5}$	5·088	2,800	$2{\cdot}393 \times 10^{-2}$	6·512
2,300	$1{\cdot}439 \times 10^{-4}$	5·350	2,900	$5{\cdot}395 \times 10^{-2}$	6·717
2,400	$4{\cdot}742 \times 10^{-4}$	5·601	3,000	$1{\cdot}148 \times 10^{-1}$	6·915
2,500	$1{\cdot}422 \times 10^{-3}$	5·841			

N.B. Values of K_1 at intermediate temperatures must be determined from equation (33).

Returning now to equation (31) of the last article, the products of combustion were represented by

$$A\,CO_2+B\,CO+C\,H_2O+D\,H_2+E\,O_2+56{\cdot}4N_2.$$

In order to find out the quantities of each gas present at the maximum temperature we have to evaluate the five coefficients A, B, C, D, and E. Three equations can be written down at once by equating the number of atoms of the various elements before and after combustion. Taking $\alpha = 0$ in equation (31), we have

$$\begin{aligned} A+B &= 12 \\ 2A+B+C+2E &= 30 \\ 2C+2D &= 12. \end{aligned}$$

Remembering that the partial pressure of a gas is proportional to the number of molecules of it present, we obtain from equations (32) and (34) two further equations to complete the five necessary for the five unknowns, as follows:

$$\frac{B^2}{A^2}\times(P_{O_2}) = K_1 \tag{32 a}$$

and

$$\frac{B\times C}{A\times D} = K. \tag{34 a}$$

Now P_{O_2}, in equation (32 a), we can only evaluate as being a certain fraction $\frac{E}{X}$ of the maximum explosion pressure, where X is the total number of molecules present at the maximum temperature, namely,

$$X = A+B+C+D+E+56{\cdot}4.$$

The maximum pressure P_e is at present unknown, as is also the maximum temperature T_e abs.; but we have the following relationship between them:

$$P_e = \frac{X}{73{\cdot}4}\times 5\times\frac{T_e}{373}\text{ atmospheres,} \tag{36}$$

remembering that the number of molecules before combustion in equation (29) was 73·4, that the compression ratio of the engine was taken as 5:1, and the initial temperature as 100° C.

From henceforward the true solution for the five coefficients is made indirectly. A maximum temperature is guessed, and values for the five coefficients are found. This particular solution for the amount of CO_2, H_2O, and CO formed by combustion will correspond to a definite amount of heat generated, and to a definite volumetric heat of the products of combustion. The question is: Will this amount of heat generated, and this mean volumetric heat, result in the temperature we originally guessed? If not, we have to guess a new temperature nearer the calculated one and go through the same series of operations, until the final temperature calculated coincides with the original temperature guessed. Thus in the benzene-air combustion suppose the temperature is first guessed as 2,500° C.

Then we have

$$T_e = 2{,}773$$

$$K_1 = 0{\cdot}019$$

$$K = 6{\cdot}45$$

$$P_{O_2} = \frac{E}{X} P_e = \frac{E \times 5}{73{\cdot}4} \times \frac{2{,}773}{373} = 0{\cdot}506E$$

$$\frac{B^2}{A^2} E = \frac{K_1}{0{\cdot}506} = 0{\cdot}0375$$

$$\frac{B \times C}{A \times D} = 6{\cdot}45.$$

We now have the five equations for the five coefficients, and can obtain our first solution,

$$A = 10{\cdot}09$$

$$B = 1{\cdot}91$$

$$C = 5{\cdot}83$$

$$D = 0{\cdot}17$$

$$E = 1{\cdot}04$$

The heat generated during the formation of these products of combustion from air and benzene will be the heat of combustion of 2 mols of benzene vapour (2 × 757,400 C.H.U.) less the heat of combustion of 1·91 mol of CO (1·91 × 67,350) and 0·17 mol of H_2 (0·17 × 57,660). The net heat generated at the maximum temperature, therefore, amounts to 18,750 C.H.U. per mol of original fuel-air mixture. The mean volumetric heat of the products of combustion between 100° and 2,500° C. is 7·346 C.H.U. per mol of original mixture. During compression in the ratio 5:1 from the starting temperature 100° C., 1,590 C.H.U. per mol are communicated to the mixture. The total temperature rise will therefore be

$$\frac{18{,}750 + 1{,}590}{7{\cdot}346} = 2{,}768 \text{ degrees}$$

and the maximum explosion temperature 2,868° C., which is far too high, the guessed temperature having been 2,500° C.

A guess of 2,705° C. for the final temperature leads to

$$\frac{B\times C}{A\times D} = K = 6{\cdot}876$$

$$\frac{B^2}{A^2}E = \frac{K_1}{0{\cdot}544} = 0{\cdot}180$$

and the solution of the five equations for A, B, C, D, and E leads to a composition of the products of combustion at the maximum temperature,

$$9{\cdot}01CO_2+2{\cdot}99CO+5{\cdot}72H_2O+0{\cdot}28H_2+1{\cdot}635O_2+56{\cdot}4N_2.$$

It will be seen that this solution shows considerably more dissociation, owing to the higher guessed temperature, and in consequence the heat generated is less, being 17,670 instead of 18,750 C.H.U. per mol. The mean volumetric heat of the mixture over the higher temperature range will be greater, 7·394 against 7·346, and on both accounts, therefore, the finally calculated temperature is lower than before. It is

$$\frac{17{,}670+1{,}590}{7{\cdot}394}+100 = 2{,}705^\circ \text{ C.},$$

which is the temperature guessed and may be accepted as the true final temperature, after allowing for dissociation, when a correct mixture of benzene and air is compressed in the ratio 5:1 and then burned at constant volume. Examination of the products of combustion shows that of the carbon in the benzene 25 per cent. has not been oxidized beyond the CO stage at the maximum temperature and that 4·6 per cent. of the hydrogen will be still uncombined.

The fuel-air mixture chosen in this case was the correct one of equation (29). If the mixture had been rich or weak the appropriate value of α would be used in equation (31), and the only difference in working out the maximum temperature would be that two out of the five equations for finding the five coefficients would be slightly different numerically. Under the new conditions,

$$A+B = 12(1+\alpha)$$

$$2C+2D = 12(1+\alpha).$$

From this point onward the process of calculation would be identical, as it is, also, when the calculation is made for the gas-engine mixture of which the results were given in art. 13. The solution for the products of the gas-air combustion represented by equation (28), at the maximum temperature reached, 2,691° C., is

$$0{\cdot}408CO_2+0{\cdot}127CO+0{\cdot}904H_2O+0{\cdot}041H_2+0{\cdot}084O_2+3{\cdot}323N_2$$

instead of the simpler products yielded by complete combustion as given by the right-hand side of equation (28) where dissociation was neglected. The volume ratio σ at the maximum temperature is $\frac{4 \cdot 89}{5 \cdot 13} = 0 \cdot 953$ instead of 0·936 as was given in art. 15 (p. 46) for the condition when combustion was complete. The former is the figure to be used in calculating the maximum pressure.

The calculation of the maximum temperature in the compression-ignition cycle, illustrated in fig. 9, is also made in a somewhat similar way, except that here the maximum pressure instead of being dependent on the temperature to be found, and therefore unknown, has been arbitrarily fixed as 800 lb. per sq. inch.

The stages in this cycle are firstly the compression of 71·4 mols of air, secondly the injection and combustion of sufficient fuel to raise the pressure to 800 lb. per sq. inch at constant volume, thirdly the injection of the remainder of the fuel and its combustion—so far as equilibrium conditions permit—at a constant pressure of 800 lb. per sq. inch, and fourthly the expansion of the products to the original volume whilst further combustion of partially burnt fuel proceeds in accordance with the temperatures and pressures that pertain. The temperature at the end of each stage can be determined from the energy of the cylinder contents and the number of mols present.

It has already been specified that the fuel† to be injected is to be limited to 80 per cent. of that required for the complete combustion of the air, and since this fuel is in liquid form at, say, room temperature, the heat available from its combustion is given by:

$$\underset{\text{at }18^\circ\text{ C.}}{C_6H_6\text{(liquid)}} + \underset{\text{at }100^\circ\text{ C.}}{7\tfrac{1}{2}O_2} = \underbrace{6CO_2 + 3H_2O\text{(vapour)}}_{\text{at }100^\circ\text{ C.}} + 748{,}300\ \text{C.H.U.}$$

Considering in turn each of the first three stages of the compression-ignition cycle, we already know from table 4 (p. 40) that the energy supplied to the cylinder contents (71·4 mols of air) during their adiabatic compression is 3,000 C.H.U. per mol and that their temperature and pressure at the end of compression are 655° C. (928° abs.) and 439 lb. per sq. inch respectively.

During the second stage some of the fuel is injected and its combustion takes place at constant volume, and we have to determine what the quantity of fuel must be to raise the pressure of the cylinder contents

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† Benzene has been selected merely for comparative purposes.

from 439 to 800 lb. per sq. inch. The temperature T_3 at the end of this constant-volume combustion will be

$$T_3 = T_2 \times \frac{P_3}{P_2} \times \frac{71{\cdot}4}{X_3} \quad \text{(see fig. 5 for meaning of suffixes),}$$

and since X_3, the number of mols at T_3, will be somewhat greater than 71·4, T_3 will be somewhat less than

$$928 \times \frac{800}{439} = 1{,}691^\circ \text{ C. (abs.).}$$

Hence the fuel to be injected is that required to give a temperature rise of somewhat less than $1{,}691-928 = 763^\circ$ C.

But from table 41 on p. 286 it is seen that 1 mol of benzene vapour ($\alpha = -0{\cdot}5$) gives a temperature rise of 1,527° C. and therefore it may be assumed, as a first approximation, that the fuel required is less than $\frac{763}{1{,}527} = 0{\cdot}5$ mol.

Assume the quantity is 0·48 mol and determine the composition of the products of the reaction

$$0{\cdot}48C_6H_6\,(\text{liquid}) + \underbrace{15O_2 + 56{\cdot}4N_2}_{71{\cdot}4 \text{ mols}}$$

when their temperature is about 1,690° C. (abs.) and their pressure 800 lb. per sq. inch.

It will be found that at this comparatively low temperature and high pressure the dissociation of CO_2 and H_2O is so small as to be negligible, and hence the products of combustion will be

$$\underbrace{2{\cdot}88CO_2 + 1{\cdot}44H_2O + 11{\cdot}4O_2 + 56{\cdot}4N_2}_{72{\cdot}12 \text{ mols}}.$$

The change in the number of mols is from 71·4, before the fuel was injected, to 72·12, and therefore

$$T_3 = 928 \times \frac{800}{439} \times \frac{71{\cdot}4}{72{\cdot}12} = 1{,}675^\circ \text{ C. abs.}$$
$$= 1{,}402^\circ \text{ C.}$$

The estimated quantity of fuel, 0·48 mol, must be verified by seeing whether the heat content of the products between 100° C. and 1,402° C. is equal to the heat supplied by combustion and compression.

The heat of combustion is $0{\cdot}48 \times 748{,}300$ and of compression $71{\cdot}4 \times 3{,}000$, giving a total of 573,400 C.H.U., whereas the heat content of the products between 100° and 1,402° C. is almost the same, namely 573,600 C.H.U., or 440·55 × 1,302.

The quantity of fuel has, therefore, been correctly estimated, and we have, at the end of stage 2,

$$T_3 = 1{,}402^\circ\,\text{C.},\quad X_3 = 72{\cdot}12,\quad P_3 = 54{\cdot}4\ \text{atm.}$$

During the third stage, that of combustion at constant pressure, the remainder of the fuel is injected and the final products will be those resulting from the combustion of 1·6 mols of fuel in 71·4 mols of air when the temperature is $T_{3'}$ and the pressure 54·4 atm. (800 lb. per sq. inch).

The composition of the products is obtained from the equation

$$1{\cdot}6C_6H_6+15O_2+56{\cdot}4N_2 = ACO_2+BCO+CH_2O+DH_2+EO_2+56{\cdot}4N_2$$

by the method already described, but in this case it must be borne in mind that external work is done during the constant-pressure combustion and this will absorb some of the heat generated.

Before proceeding with the calculation of the coefficients A, B, C, etc., it may be noted that

$$P_{O_2} = \frac{E.P_{3'}}{X_{3'}} = \frac{54{\cdot}4E}{X_{3'}}$$

and hence guesses must be made of both $T_{3'}$ and $X_{3'}$.

Assume that the final temperature $T_{3'}$ is 2,426° C. (2,699° abs.) and the corresponding number of mols $X_{3'}$ is 74·11. Then

$$\frac{B\times C}{A\times D} = 6{\cdot}295$$

$$\frac{B^2}{A^2}E = \frac{K_1}{0{\cdot}734} = 0{\cdot}0135$$

and the solution of the five equations for A, B, C, D, and E gives the following composition of the products of combustion at the maximum temperature,

$$9{\cdot}03CO_2+0{\cdot}57CO+4{\cdot}75H_2O+0{\cdot}05H_2+3{\cdot}31O_2+56{\cdot}4N_2,$$

the total mols being 74·11 as assumed. A check is now made to see whether the heat supplied by combustion and compression is balanced by the energy content of the products between 100° and 2,426° C. plus the external work done. The heat of combustion

$$(1{\cdot}6\times 748{,}300-0{\cdot}57\times 67{,}350-0{\cdot}05\times 57{,}660)$$

plus the heat of compression (71·4 × 3,000) gives a total of 1,370,200 C.H.U., whereas the energy content of the products between 100° and

2,426° (521·4×2,326) plus the heat equivalent of the external work (1·985[2,699×74·11−1,675×72·12]) gives a total of 1,370,300 C.H.U.

Since these totals are in agreement, the final temperature $T_{3'}$ has been correctly guessed as being 2,426° C.† (2,699° abs.), and the rise of temperature at constant pressure is therefore 1,024° C.

As regards the external work done it may be noted that during the constant pressure combustion stage the number of mols changes from X_3 to $X_{3'}$, due partly to the injection of the remainder of the fuel and partly to the combustion processes. The external work done in going from X_3 mols at T_3 to $X_{3'}$ mols at $T_{3'}$ can be found from the volume changes that take place at the constant pressure P_3.

The volume changes from V_3 to $V_3 \times \frac{T_{3'}}{T_3} \times \frac{X_{3'}}{X_3}$.

The external work done per mol of gas at V_3

$$= P_3 V_3 \left(\frac{T_{3'} X_{3'}}{T_3 X_3} - 1\right)$$

$$= RT_3 \left(\frac{T_{3'} X_{3'}}{T_3 X_3} - 1\right)$$

and the external work done by X_3 mols

$$= R(T_{3'} X_{3'} - T_3 X_3).$$

In the fourth—or adiabatic-expansion—stage, the volume changes from

$$V_{3'} = V_3 \times \frac{T_{3'}}{T_3} \times \frac{X_{3'}}{X_3} = \frac{2{,}699}{1{,}675} \times \frac{74{\cdot}11}{72{\cdot}12} V_3 = 1{\cdot}656 V_3$$

to $V_4 = 12V_3$, and the ratio of expansion is therefore

$$\frac{12}{1{\cdot}656} = 7{\cdot}246 : 1.$$

The calculation of the final temperature T_4 is made by a modification of the method explained in art. 12. The modification is required to allow for the recombination that takes place during the expansion stroke when we are dealing with gases containing dissociation products.

It has been found (ref. 7, p. 36) that a close approximation to a more rigid solution is obtained by assuming not only that γ has a mean constant value throughout the expansion stroke but that the value of γ is based on the *apparent* volumetric heat of the expanding gas mixture. This *apparent* volumetric heat includes both the normal change in

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† This figure differs slightly from that given in fig. 9 because in that calculation the formation of NO was taken into account (see p. 75).

heat content with change in temperature and also the heat energy arising from the chemical action which the change in temperature permits.

In calculating the final temperature T_4 we shall therefore assume that the expansion curve can be represented by $PV^{\gamma} =$ constant and that γ is computed from the *apparent* volumetric heat of the expanding gases.

A value for T_4 is guessed and the usual equations solved for A, B, C, D, and E, bearing in mind that

$$P_{O_2} = \frac{E}{X_4} P_4$$

and $$P_4 = \frac{1 \times T_4 \times X_4}{T_1 \times 71{\cdot}4} \text{ (approximately)}\dagger$$

and hence $$P_{O_2} = \frac{E.T_4}{T_1 \times 71{\cdot}4} \text{ (approximately).}$$

If T_4 is assumed to be 1,397° C. (1,670° C. abs.), the composition of the products is

$$\underbrace{9{\cdot}60CO_2 + 4{\cdot}8H_2O + 3{\cdot}00O_2 + 56{\cdot}4N_2.}_{73\cdot8 \text{ mols}}$$

The total energy content (heat+chemical) above 100° C. of these products is $486{\cdot}1 \times 1{,}297 = 630{,}500$ C.H.U., whereas that for the products at the beginning of the expansion is

$$521{\cdot}4 \times 2{,}326 + 0{\cdot}57 \times 67{,}350 + 0{\cdot}05 \times 57{,}660 = 1{,}254{,}270 \text{ C.H.U.}$$

The numbers of mols at the beginning and end of the expansion are 74·11 and 73·80 respectively, the mean being $73{\cdot}95_5$.

The mean *apparent* K_v between $T_{3'}$ and T_4

$$= \frac{\text{total energy at } T_{3'} - \text{total energy at } T_4}{\text{mean number of mols} \times (T_{3'} - T_4)}$$

$$= \frac{1{,}254{,}270 - 630{,}500}{73{\cdot}95_5 \times 1{,}029} = 8{\cdot}20$$

Hence $$\gamma = 1 + \frac{R}{K_v} = 1 + \frac{1{\cdot}985}{8{\cdot}20} = 1{\cdot}242.$$

The correctness of the guessed value (1,397° C.) of T_4 is now checked

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† Strictly $$P_4 = 1 \times \frac{T_4}{T_1} \times \frac{X_{3'}}{71{\cdot}4},$$

but the approximate value is sufficiently precise at these comparatively low temperatures.

by seeing whether it is in agreement with the value obtained from the expansion equation,

$$T_4 = T_{3'}\left(\frac{V_{3'}}{V_4}\right)^{\gamma-1}$$

$$= \frac{2,699}{(7{\cdot}246)^{0{\cdot}242}} = 1,671^\circ\ \text{C. (abs.)}$$

$$= 1,398^\circ\ \text{C.}$$

This differs by only 1° from the temperature assumed and hence the temperature T_4 at the end of the expansion can be taken as 1,398° C.

Returning now to the 'constant-volume' cycle, the maximum temperatures for air-benzene mixtures—varying in fuel content from 50 per cent. weak to 50 per cent. rich—have been worked out and the results are given in table 41 and in fig. 13.

This table and graph are similar in form to those given in the original paper,[7] but they have been extended to include the results of calculations in which the formation of nitric oxide is taken into account. Nitrogen is not the inert gas that has been assumed up to the present, for under high-temperature conditions appreciable oxidation—to nitric oxide (NO)—occurs, as will be seen from column 7 of table 41. Since heat is absorbed when nitric oxide is formed, the maximum explosion temperature is depressed.

The calculation of the composition of the products of combustion, when nitric oxide is taken into account, requires two more equations to correspond with the increase of two in the number of unknowns. The general equation for the correct air-benzene mixture is

$$2C_6H_6+15O_2+56{\cdot}4N_2$$
$$= A\,CO_2+B\,CO+C\,H_2O+D\,H_2+E\,O_2+F\,NO+G\,N_2,$$

and the additional equations are

$$G+\tfrac{1}{2}F = 56{\cdot}4$$

and
$$\frac{(P_{NO})^2}{P_{N_2}\times P_{O_2}} = \frac{F^2}{G\times E} = K'.$$

K' can be obtained from the data presented by Lewis and von Elbe,[56] or from the following expression, which represents those data fairly closely over the temperature range 2000° to 3000° C. absolute:

$$\log K' = 1{\cdot}348 - \frac{9,475}{T}.$$

As to the heat of formation of NO—at 100° C. and at constant volume—the following equation indicates that it is negative,

$$NO = \tfrac{1}{2}N_2 + \tfrac{1}{2}O_2 + 21{,}530 \text{ C.H.U.},$$

and hence the heat absorbed when 1 mol of nitric oxide is formed from ½ mol of nitrogen and ½ mol of oxygen is 21,530 C.H.U.

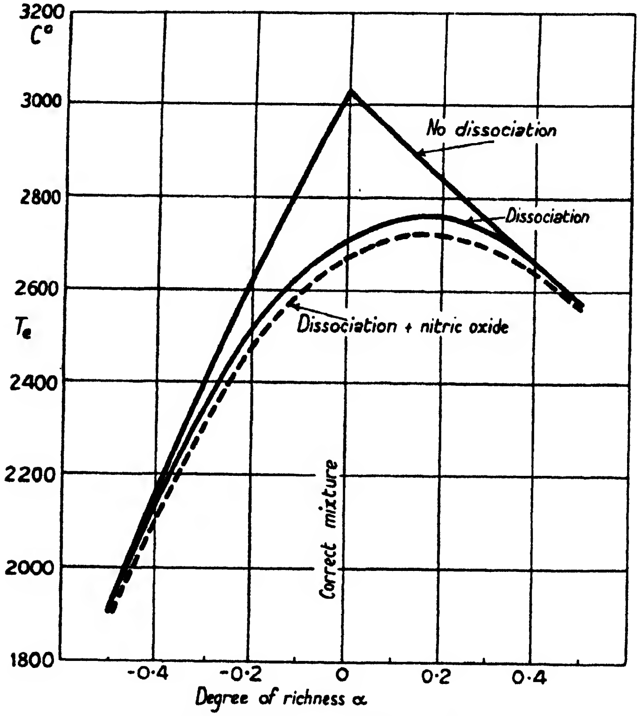

FIG. 13. Calculated maximum temperatures for benzene-air mixtures.

The calculation of the values of the seven coefficients A, B, C, D, E, F, and G is made in much the same manner as that already outlined for the determination of the first five when F is neglected and G is equal to the nitrogen content of the original mixture. As to the results, the majority are given in table 41, but the effect of nitric oxide is also shown on the three cycles of fig. 9.

Returning from this digression on nitric-oxide formation to the general results of the air-benzene calculations, the way in which the maximum temperature varies with the fuel-air ratio can be seen in fig. 13, in which the curve of calculated temperatures when dissociation is allowed for is seen beneath the approximately straight line variation which the temperatures would exhibit if no dissociation took place. In these circumstances the greatest amount of heat would be generated by the combustion of the correct mixture, and for either weak or rich mixtures, on either side of this, the maximum temperature would fall away in the linear manner shown in fig. 13.

It will be seen that owing to dissociation the highest calculated temperature on the curve is that for a mixture some 20 per cent. richer in fuel than that giving complete combustion, and that over a wide range of mixture strength on the rich side the temperatures show only a slow variation. Under comparable conditions the power of an engine will vary with the maximum temperature and the change in the number of mols due to combustion. The present writer has, in fact, found[7] that for conditions similar to those represented by table 41, the product of the ratio of the number of mols at maximum temperature to the number in the original mixture and the rise in temperature on explosion (i.e. $\sigma(T_3-T_2)$) gives a figure which is closely proportional to the net work done per cycle. If this product is plotted it will be found that the resulting curve reproduces the well-known power characteristic of a petrol engine when the fuel ratio is varied, and hence affords substantial confirmation of the calculations and theory on which they are based (see especially fig. 33, p. 135).

Goodenough,[35] using a different method of calculation, has arrived at figures for the maximum temperatures of benzene-air mixtures which closely confirm those obtained by the method given here. He has carried out a very complete theoretical analysis of the combustion of a variety of fuels, and the reader who is interested in the mathematical treatment of the conditions of chemical equilibrium should consult the reference given. This paper also includes a critical examination of the experimental data upon heats of combustion and volumetric heats which is of much value, but subsequent work has already shown the need for revision of some of the conclusions. The recent revision of the dissociation constants has already been mentioned, and the question of volumetric heat values will be fully dealt with in art. 27. For the present it will be sufficient to conclude by saying that it would be premature to consider that the values used in this book for the volumetric heats and dissociation constants have been unquestionably established or to claim *absolute* accuracy for any of the calculated temperatures;

but this does not, of course, affect the usefulness of the relative values as a guide to understanding the behaviour of an engine.

When really certain figures for the volumetric heats become available, the time will be ripe for a complete revision of all the calculations, using the most accurate figures for all the physical and chemical data. Until that day arrives one may say that the figures given have all the accuracy which the imperfect data can provide.

Art. 25. A further note on calorific values

In art. 22 a definition of the calorific value of a fuel is given which is sufficiently precise for most engineering calculations and this is followed by a discussion of the corrections that must be applied when the fuel is initially in a state of vapour and when the steam formed during its combustion is not condensed.

Although this preliminary discussion may have been sufficient to indicate how the heat of combustion associated with one set of conditions can be used to determine that relating to another, it will probably help to clear up any outstanding difficulties if the matter is considered in more detail and put on a fundamental basis. The principle involved is that of the conservation of energy, and this principle can be readily applied to equations representing chemical reactions if particulars are available from which heat energy contents or changes of heat energy content can be determined.

In the case of liquid benzene, for example, the heat of combustion *at 18° C.* and *at constant volume* may be taken, in round figures, to be 780,000 C.H.U. per mol. The temperature is specified to indicate that the fuel and the air or oxygen for its combustion are initially at 18° C., and that the products of the combustion are finally at the same temperature; whereas the term 'constant volume' implies that the combustion and subsequent cooling of the gases take place in a closed vessel of constant volume and hence are free from the losses or gains of energy associated with volume changes. We can, therefore, write that, at 18° C. and constant volume,

$$C_6H_6\,(\text{liquid}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{liquid}) + 780{,}000 \text{ C.H.U.},$$

or, in words, that the heat of the chemical reaction plus the heat-energy content of the reactants at 18° C. is equal to the heat-energy content of the products at 18° C. plus the heat of combustion.

Since the heat of the chemical reaction is constant, all we need to know for determining the variation in the heat of combustion with change in conditions is the effect of that change of conditions on the difference of the heat-energy contents of the reactants and products.

A knowledge of the following energy quantities will enable us to write down the heats of combustion of benzene under various conditions. The latent heat of benzene at 18° C. can be taken as 8,138 C.H.U. per mol, the work done during its evaporation as 578 C.H.U. per mol, and hence the heat required to evaporate this fuel under closed-vessel conditions as 7,560 C.H.U. per mol. The corresponding quantities for water are 10,504, 578, and 9,926 at 18° C. and 9,710, 740, and 8,970 at 100° C. The mean volumetric heats of O_2 and CO_2 over the temperature range 18° to 100° C. are about 5·2 and 7·3 respectively at constant volume, whereas under constant-pressure conditions these will be increased by about 1·985. The volumetric heat of benzene vapour is less well established, but its mean value at constant volume over the temperature range 18° to 100° C. can be taken as about 19 C.H.U. per mol.

Using the above data in connexion with our basic equation

$$C_6H_6\,(\text{liquid}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{liquid}) + 780{,}000 \text{ C.H.U.}$$

at 18° C. and constant volume, we obtain:

(i) $C_6H_6\,(\text{vapour}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{liquid}) + 787{,}560$ C.H.U. at 18° C. and constant volume;

(ii) $C_6H_6\,(\text{vapour}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{vapour}) + 757{,}780$ C.H.U. at 18° C. and constant volume;

(iii) $C_6H_6\,(\text{vapour}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{vapour}) + 757{,}400$ C.H.U. at 100° C. and constant volume, the heat-energy content of the reactants having been increased by $7{,}560 + 82(19 + 7{\cdot}5 \times 5{\cdot}2)$ and of the products by $82(54 + 6 \times 7{\cdot}30) + 3 \times 8{,}970$;

(iv) $\underset{\text{at } 18^\circ \text{ C.}}{C_6H_6\,(\text{liquid})} + \underset{\text{at } 100^\circ \text{ C.}}{7{\cdot}5O_2} = \underbrace{6CO_2 + 3H_2O\,(\text{vapour})}_{\text{at } 100^\circ \text{ C.}} + 748{,}300$ C.H.U. at constant volume;

(v) $C_6H_6\,(\text{liquid}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{liquid}) + 780{,}870$ C.H.U. at 18° C. and *constant pressure*, the heat-energy content of the O_2 having been increased by $7{\cdot}5 \times 1{\cdot}985 \times 291$ and the heat-energy content of the CO_2 by $6 \times 1{\cdot}985 \times 291$;

(vi) $C_6H_6\,(\text{vapour}) + 7{\cdot}5O_2 = 6CO_2 + 3H_2O\,(\text{vapour}) + 757{,}490$ C.H.U. at 18° C. and constant pressure.

Enough examples have been given to illustrate the method, and it merely remains to point out that the heat of combustion represents the

TABLE 9

Calorific values and heats of combustion in C.H.U. per mol when the steam formed is uncondensed.

1	2	3	4	5	6	7
Fuel	*Conventional lower calorific value of liquid fuel* $H_P - 539G$	*Conventional lower calorific value of vapour or gas* $H_P + L_P - 539G$	*Heat of combustion of vapour at 25° C. and constant pressure* $H_P + L_P - 583G$	*Heat of combustion of vapour at 25° C. and constant volume* $H_V + L_V - 550G$	*Heat of combustion of vapour at 100°C. and constant volume*	*Heat of combustion of vapour at 100°C. and constant pressure*
n-Pentane	780,780	787,000	782,245	783,425	783,280	781,800
n-Heptane	1,072,720	1,081,700	1,075,360	1,077,135	1,077,090	1,074,870
Toluene	901,150	910,695	907,310	907,890	907,790	907,050
Ethyl alcohol	297,530	307,650	305,270	305,860	305,835	305,095
Hydrogen	..	58,610	57,815	57,520	57,670	58,040
Methane	..	193,370	191,785	191,785	191,715	191,715
Butane	..	693,380	635,415	636,305	636,180	635,070

heat available for any particular purpose when the conditions before, during, and after combustion are those specified.

The conventional 'lower calorific value' of a fuel is obtained by deducting the latent heat of steam *at 100° C.* from a calorific value determined at *room temperature*, and further there seems to be a failure to discriminate between calorific values obtained under constant-pressure and constant-volume conditions. If this discrimination were made and the latent heat of steam at atmospheric temperature—and at constant pressure or constant volume—were deducted, a more reasonable value would be obtained. This is illustrated by the figures given in table 9, in which H_P and H_V represent the heats of combustion of the liquid fuel at constant pressure and constant volume respectively, and at room temperature, the steam formed being condensed; L_P and L_V represent the heats of vaporization of the fuel at room temperature and under constant-pressure and constant-volume conditions respectively; G represents the weight of steam formed by combustion; 539 and 583 are the latent heats of steam at constant pressure and at 100° C. and 25° C. respectively; and 550 is the latent heat of steam at 25° C. when no external work is done during vaporization. Suitable variations must be made to the figures 583 and 550 if the calorific value was determined at a temperature other than 25° C.

The practical agreement between the figures in columns 5 and 6 on the one hand and those in columns 4 and 7 on the other, shows that a variation in temperature between 25° C. and 100° C. has very little effect on the heat of combustion. If, therefore, 'lower calorific values' were obtained, as in columns 4 and 5, by subtracting the latent heat of steam at the temperature at which the calorific value was determined, they would represent accurately the heats of combustion at that temperature and also apply fairly closely to considerably higher temperatures. These values would, in fact, be 'true' lower calorific values.

In the data relating to liquid and gaseous fuels given in tables 39 and 40 the 'true' lower calorific values have been given in place of the conventional values.

Art. 26. Closed-vessel experiments. Volumetric heats

As was pointed out in art. 12 it is convenient, when dealing with gases, to speak of their 'volumetric heats' and to express these in foot-pounds per standard cubic foot or C.H.U. per mol. In the next article the values of the volumetric heats of gases will be given in these units, and in the present discussion of the method of determining them by explosion experiments in a closed vessel of constant volume the same units will be adopted.

If a combustible mixture, like one of those discussed in previous articles, were enclosed at some predetermined pressure and temperature in a vessel of constant volume, and ignited, a pressure-time record would be obtained, of which S 79 and S 84 in fig. 14 on p. 84 are two typical examples, in which the rate of combustion varied considerably. The maximum pressure recorded would depend upon a number of factors. If it were a benzene-air mixture, the heat liberated would be equivalent to about 80,500 ft.-lb. per S.C.F. If this heat were all developed with no loss to the walls of the vessel, and if the volumetric heat of the contained gas were the same at all temperatures as it is for air at 15° C., namely 19·5 ft.-lb. per S.C.F., then the temperature rise in the vessel would have been about 4,000° C., and the final pressure would have been nearly 15 times the starting pressure. The recorded pressure would in fact have been about half of this, and the contributory causes of the 'lost pressure', in approximate order of importance, are as follows:

(1) The greater volumetric heat of the products of combustion as

a whole and the increase of that volumetric heat with temperature. Both these effects are largely due to the water-vapour and carbon-dioxide portion, which amounts to about a quarter of the whole.

(2) The cessation of the chemical reaction when equilibrium conditions are reached. This means that, for a given mixture, the extent to which the reaction can proceed is dependent on the temperature and the pressure—the higher the temperature the greater the suppression of the formation of CO_2 and H_2O. Pressure is a less important factor and acts in the reverse direction.

(3) Loss of heat to the walls of the vessel by radiation and conduction before the instant of maximum pressure is reached. The amount of this heat loss will depend on the temperature and composition of the products, the amount of gas turbulence, the rate of spread of flame, the point in the vessel at which ignition is started, and the nature (polished or black) of the surface of the vessel walls.

(4) Loss of heat due, possibly, to the cooling action of the walls of the vessel delaying the combustion of the thin layer of mixture in contact with them.

(5) Loss of pressure due to a probable temperature gradient in stagnant mixtures, the gradient extending from the region of the sparking-plug—where it tends to be highest—to the last inflamed portions of the charge.

It has also been suggested, under the title of 'after-burning', that the molecular readjustment involved in the formation of water vapour and carbon dioxide takes an appreciable time to complete itself, and that there may not have been time before maximum pressure for all the heat available to be developed. This is more than a suggestion that the flame may not have spread completely throughout the vessel, so that unburnt fuel may still exist at the maximum pressure. This theory presupposes that a chemical action, once started, needs a length of time to complete itself which is of an altogether different order from those times associated with atomic and molecular movements in any other connexion. There may be some truth in this theory, but it must be admitted that no very conclusive evidence has ever been put forward that chemical action is incomplete at the point of maximum pressure. There is, moreover, a good deal of evidence against the view that a chemical action, once begun, takes an appreciable time to complete itself. For example, the fact that the shortest time between ignition and maximum pressure in a closed-vessel experiment, except with hydrogen-air mixture, is about 0·03 sec.; and that it is possible, when the spread of flame is accelerated by turbulence as it is in an engine cylinder, to get sufficiently complete combustion in about $\frac{1}{10}$ of this time to give a peak pressure like that exhibited on the indicator diagram illustrated in fig. 20; these two points taken together afford very strong evidence that shortening the time of combustion in an engine is almost entirely a question of speeding up the spread of the flame throughout the volume of the combustible gases. And this it would not be, if the chemical action itself, once well started, took an appreciable time before it could be completed.

The effect of increased turbulence at higher speeds in accelerating the spread of flame in an engine cylinder has been directly observed by Glyde,[29] and in the experiments of Withrow, Lovell, and Boyd,[30] referred to more fully in art. 31. In these experiments, moreover, direct evidence was obtained that combustion was confined within a comparatively narrow zone, spreading out rapidly from the ignition point, and that in the rear of this combustion zone the chemical action was complete, as indicated by the disappearance of all the oxygen.

Of the suggested causes of the lost pressure in a closed-vessel explosion, by far the most important is the increase of volumetric heats; and a measurement of the maximum pressures reached in experiments of this type affords the only practicable way of measuring the mean volumetric heats of gases up to really high temperatures. In the simplest possible form of such an experiment, which would not, however, be practicable, we might imagine a mixture of two volumes of carbon monoxide and one of oxygen exploded together. From the recorded rise of pressure, the highest mean temperature, and hence the rise of mean temperature, say from T_1 to T_2, could be calculated. If H ft.-lb. per S.C.F. be the heat liberated by the combustion, and h be the amount of heat lost to the walls of the vessel before maximum pressure, the mean volumetric heat of carbon dioxide between the temperatures T_1 and T_2 will be

$$\frac{H-h}{T_2-T_1}.$$

This particular experiment would not be practicable because the combustion would be so slow, and the time to maximum pressure so long, that the lost heat, h,* which is always a difficult quantity to estimate, would be too large a proportion of the whole to allow of an accurate result. The typical pressure-time records of fig. 14 exhibit the remarkable effect of adding so little as 0·1 per cent. of hydrogen-oxygen mixture, in speeding up the rate of combustion of the naturally slow-burning carbon monoxide-oxygen mixture. In the case of S 85, an attempt was made to localize the hydrogen-oxygen mixture in the region of the spark, and the peculiar shape of the diagram must be attributed to the resulting lack of homogeneity of the gaseous contents of the vessel.

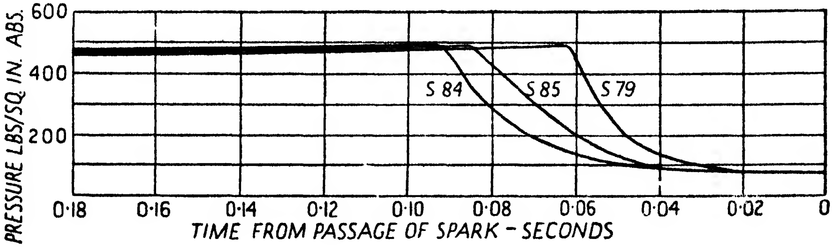

FIG. 14. Superimposed pressure-time records, showing the effect of small additions of a mixture (*b*) represented by $7H_2+O_2$ to another (*a*) represented by $7(CO+0{\cdot}0008H_2)+O_2$.

S 84 mixture (*a*) only.
S 79 mixture (*a*) + about 0·12 per cent. of (*b*).
S 85 mixture (*a*) + about 0·09 per cent. of (*b*) localized.

A correct estimate of the heat lost before maximum pressure is the great difficulty in volumetric heat determinations, but this is not the only source of uncertainty. If dissociation were present, for example, the full value of *H* would not be developed at the moment of maximum pressure and temperature, and the result would be lower maxima. Whether these lower maxima, moreover, were produced by a failure to generate all the heat available, on account of dissociation, or by a genuine rise of volumetric heat of the gases to be heated, the result would be the same, and it would be impossible to distinguish between the two causes of lost pressure in any one experiment. It is possible, however, to minimize the amount of dissociation by having a large excess of one of the reacting gases present, and this has been done in the mixtures of which the pressure-time records are given in fig. 14. The gas used in this case—namely carbon monoxide—is fairly stable at high temperatures, and hence its addition in large quantities does not give rise to any secondary effects. Hydrogen, on the other hand, splits up much more readily into its atomic form, and hence the addition of a large excess of H_2 for the purpose of reducing the dissociation of

H_2O may lead to appreciable quantities of atomic hydrogen being formed. Further, it is now recognized that water vapour may split up into 2OH and H_2 in addition to $2H_2$ and O_2, although no account has been taken of this reaction in the calculations of art. 24.

Therefore, in determining both the composition of the products of combustion at the maximum temperature and the heat generated, all possible reactions must be considered, and even then the calculations may be in error owing to the assumption of a mean temperature of the gases in place of a very considerable temperature gradient.

The experimental determination of volumetric heats is discussed further in the next article, and references are there given to original sources which must be consulted for full descriptions of the methods employed.

In engine calculations we are concerned not so much with volumetric heats, as such, but with the amount of internal energy to be associated with the working substance at every temperature. We need the relation connecting internal energy and temperature, and the exact relation will differ according to the composition of the working substance. It is mainly the water-vapour and carbon-dioxide parts of it which are responsible for the increase of the volumetric heat with temperature, and the internal energy-temperature relation will therefore be different for rich and weak mixtures, for with the latter the proportion of the triatomic gases in the products of combustion will be less. For the correct mixture of coal gas and air of equation (28) the proportion was 30 per cent. by weight, but for a mixture 50 per cent. weak it will be only 16 per cent.

The first step towards the determination of the *internal energy-temperature* relation for any particular working substance is to obtain a series of values for the mean volumetric heats of its constituents over different temperature ranges from some agreed starting-point. This point is conveniently taken at 100° C., so as to eliminate any complication due to condensation of the water vapour in the products of combustion. The mean volumetric heat of any particular mixture of gases can be quickly calculated from the volumetric heats of the separate constituents.

Art. 27. Experimental values for the volumetric heats of gases

Apart from closed-vessel explosions, measurements of the volumetric heats have been made by two other methods: firstly by heating the gas while flowing uniformly at constant pressure, generally atmospheric, and secondly by experiments in an engine in which gas is alternately compressed and expanded, and the changes of mean temperature, as deduced from the pressure record of an indicator diagram, are correlated with the measured amounts of work done by or upon the gas by the piston. Experiments of the latter type were made by Clerk[9] up to temperatures of about 1,400° C., but their accuracy depends very largely upon a correct allocation of heat loss to the cylinder walls between the compression and expansion strokes. In view of the varying and quite unknown degree of air turbulence in the cylinder, and the great dependence of heat loss upon this factor, any certain determination of the rates of heat loss from moment to moment appears too difficult of achievement for the method to yield results of the accuracy we have to aim at.

Constant-pressure experiments have also been limited to a maximum temperature of about 1,400° C., and it is doubtful whether anything higher could usefully be attempted owing to the very great technical difficulty of carrying out accurate temperature measurements under these conditions. Very accurate figures have, however, been obtained by Swann at 20° and 100° C. in constant-pressure experiments upon air and carbon dioxide. These results afford valuable datum line figures for the volumetric heats at low temperatures, as follows:

	At 20° C.	At 100° C.
Air . . .	19·5	19·7 ft.-lb. per S.C.F.
Carbon dioxide .	26·9	30·2 ,, ,, ,,

A great number of experiments of the closed-vessel explosion type have been carried out by many different workers in attempts to measure mean volumetric heats of the contents of the vessel between the initial and the maximum temperature,[10] and these have recently been supplemented by indirect determinations of volumetric heats from the band spectra of the gases concerned. Some of the workers[55] in this latter field claim a high degree of accuracy in the case of diatomic gases, and the more difficult problem of the triatomic and polyatomic gases is receiving and has received a large amount of attention. In 1935 the position was summarized by Lewis and von Elbe in a paper published by the American Chemical Society.[56] In this paper the authors give tables of heat capacities and equilibrium constants derived from spectroscopic data and also full references to the sources of their figures. The figures appear, in fact, to represent the present state of knowledge and they have been used in the deduction of the volumetric heats given in table 10 and in the formulae for equilibrium constants given in art. 24. The adoption of these values must not be interpreted as an endorsement of their absolute accuracy, for time alone can show whether all the energy levels have been discovered.

TABLE 10

Mean volumetric heats between 100° C. and t° C. in foot-pounds per standard cubic foot.

To convert to C.H.U. per mol, divide by 3·90.

100° *C. up to*	500°	1,000°	1,500°	2,000°	2,500°	3,000°
Carbon dioxide . .	35·38	39·58	42·17	43·95	45·25	46·37[†]
Water vapour . .	25·74	28·55	31·08	33·27	35·10	36·66[†]
Nitrogen . . .	20·26	21·76	22·80	23·60	24·18	24·67
Carbon monoxide .	20·38	21·98	23·13	23·87	24·45	24·96
Oxygen[†] . . .	21·76	23·50	24·67	25·62	26·44	27·09
Hydrogen . .	19·60	20·14	20·93	21·80	22·60	23·23

A simple assumption which is often made in regard to the volumetric heats of gases is that they vary in a linear manner with the temperature—in other words that the volumetric heat K_v at any temperature t is given by an equation of the form

$$K_v = (K_v)_0 + \alpha t.$$

For practical purposes, however, it is not the volumetric heat *at* any particular temperature which we require, but the mean value over a range of temperature, from which to calculate the change of internal energy associated with that change of temperature. The working substance for which we have to know the changes of energy, moreover, is a mixture of diatomic gases, carbon dioxide, and water vapour. And although it is possible to represent the variation of the mean volumetric heat of the diatomic gases with fair accuracy by a simple linear equation of the type given above, it is not possible to do so at all for the triatomic gases, nor really satisfactorily for any practical working substance over the full range of temperature from 100°–3,000° C. For approximate calculations it is convenient and sufficiently satisfactory to work with a curve such as that shown in fig. 15, in which the full line curve gives the internal energy at every temperature of one standard cubic foot of the products of combustion of the correct benzene-air mixture. The

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[†] At 1,500° C. Johnston and Walker's value for oxygen (*J. Am. Chem. Soc.*, **57**, 1935), is about 0·5 per cent. below the value given here, and at 2,500° C. this discrepancy amounts to nearly 2 per cent. in the same direction.

curve has been constructed from the composition of the mixture as given by the right-hand side of equation (29), and the mean volumetric heats as given in table 10.

The products of combustion, and therefore the energy-temperature curve, will necessarily be different for every fuel and every mixture strength, but fortunately, since at least three-quarters of the working substance is always nitrogen, the variation is not great. In order to show the possible range of variation there has been added to fig. 15 the dotted curve, showing the energy relationship for a coal gas-air mixture 50 per cent. weak. The products of such a combustion have about the maximum proportion of diatomic gases, and therefore the minimum volumetric heat values, likely to be met with in practical working.

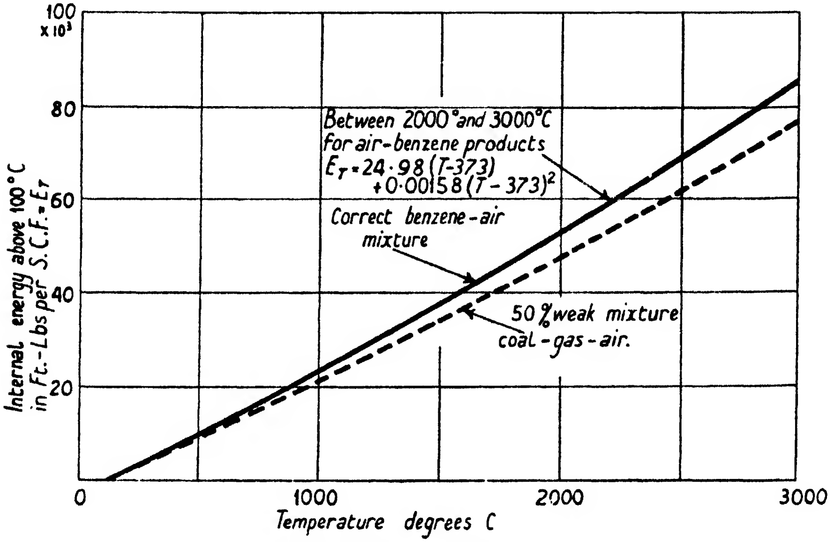

FIG. 15. Internal energy-temperature curves.

The corresponding values of temperature and internal energy for the two curves shown in fig. 15 are given in table 11. By the use of this table the curves can be quickly plotted to a large scale for practical use.

For accurate work, the mean volumetric heat of the actual combustion products must be determined from a large-scale plotting of the individual values of the volumetric heats given in table 10.

Various empirical formulae have been proposed from time to time in attempts to represent the changes of K_v for CO_2 and H_2O throughout the whole range up to high temperatures.[36] In the present state of

TABLE 11

Corresponding values of temperature and internal energy for the products of combustion of a correct benzene-air mixture, and of a coal gas-air mixture 50 *per cent. weak.*

To convert to C.H.U. per mol, divide by 3·90.

Temperature	*Correct benzene-air mixture*	*Coal gas-air mixture 50 per cent. weak*
°C.	ft.-lb. per S.C.F.	ft.-lb. per S.C.F.
100	0	0
500	9,750	8,300
1,000	23,200	20,800
1,500	37,800	33,900
2,000	53,100	47,600
2,500	69,050	62,350
3,000	85,700	76,650

experimental knowledge, however, such attempts are in danger of being premature and misleading. Formulae of this kind are not necessary for the purposes of the internal-combustion engineer, and it seems wiser to await more certain experimental data before defining the probable changes of K_v in any characteristic equation.

IV

DETONATION

ART. 28. The nature of detonation. Its difference from pre-ignition

Detonation, also known as 'knocking' or 'pinking', affords an interesting example of something which was familiar to the practical engineer a long while before its far-reaching importance was realized. There was even a good deal of rough and ready knowledge about its occurrence, and every rider of an air-cooled motor cycle knew that if his engine began to knock he must close his extra air throttle and run with a richer mixture.

It was Ricardo who first showed that of all the factors which limit the power output and efficiency of engines using volatile liquid fuel and electric ignition, detonation is the most important. In later discussions upon engine performance we shall return to it repeatedly, and it will be well to get beforehand a mental picture of what it means, and some knowledge of the conditions which produce it. The occurrence of detonation is very much bound up with questions of engine design and speed, but recent research has shown that essentially it is a chemical and physical problem, and this aspect of it can usefully be explained at the present stage, leaving until later the influence of engine details.

The noise which has given the name of 'knocking' to detonation is like that produced by a sharp ringing blow upon the metal of the engine cylinder. Formerly it was supposed to be actually due to some such mechanical cause, which might be derived, it was supposed, from a looseness between moving parts. Experiment soon proved this supposition to be untenable, and in place of a mechanical blow between two solid parts of the engine, we now imagine the noise to be caused by a blow delivered against the cylinder wall by a wave of high pressure travelling at great speed through the gas which forms the working substance. If it is difficult to imagine a noise being set up in this way, it should be remembered that the noise of a hammer blow is due to the vibrations set up by the sudden and *local* high pressure produced under the hammer; and the fact that sharp local high pressures are in fact produced in the gaseous contents of a cylinder when detonation occurs has often been demonstrated.

In an engine cylinder combustion is initiated during normal working by an electric spark timed to occur at a definite instant towards the end of the compression stroke. The correct timing varies according to engine speed and other factors, but ignition should never start before the spark has passed. If it does do so, on account of overheated sparking-plug points, or scraps of stray incandescent carbon, then *pre-ignition* is occurring: a state of affairs which seldom persists for long without violent thumping and excessive pressures being developed before the end of the compression stroke, which cause a loss of power and may bring the engine to a standstill. Detonation, on the other hand, is something which follows after ignition has been started by the spark in the normal manner, and if it is not violent it may persist for long periods without adversely affecting the running of the engine. If the detonation becomes violent, however, it is very apt to lead to conditions in which the sparking-plug points become overheated, and so to prepare the way for pre-ignition to occur. The two phenomena, however, are totally distinct from one another. Detonation *follows*, while pre-ignition, as its name implies, *precedes* the spark.

So long as no detonation takes place, the flame started by the spark spreads steadily throughout the combustion space at a speed which depends mainly upon gas turbulence, as well as upon the pressure and temperature before ignition. As the flame spreads from the point of ignition the heat generated causes that part of the charge already inflamed to expand, compressing the remainder. With a detonating fuel self-ignition may occur, and the rate of spread of the flame throughout the last portions of the charge to burn becomes almost instantaneous.[39] Under these conditions a 'detonation wave' is set up, in the front of which the gas pressure may rise locally to a figure far in excess of the average pressure in the cylinder. Pistons have actually been broken from time to time by the hammer blow which this detonation wave is capable of delivering, and from its less violent manifestations is derived the characteristic 'knock' when the wave front reaches the cylinder wall.

As already stated, the onset of detonation is much influenced by engine speed and cylinder design. Features like the shape of the combustion chamber, and the relative positions of the sparking-plug and exhaust valve, have a marked influence; but these points may be left for later discussion, and for the present we shall consider more particularly the fundamental physical and chemical circumstances in which a detonation wave is set up, and the views of different authorities upon their relative importance. Before doing so, however, it will be advisable to review the facts about the behaviour of different fuels in regard to detonation.

ART. 29. Preliminary facts about fuels and 'dopes'

It has been found that in the same engine, under the same conditions of compression ratio and temperature, the tendency to detonate varies widely according to the fuel used. With coal gas, for example, detonation has never been observed; and the three types of hydrocarbons found in the volatile liquid fuels, namely aromatics, naphthenes, and paraffins, take this order of merit in regard to detonation.

A most potent factor in promoting detonation is a rise in the compression ratio, and Ricardo has coined the phrase Highest Useful Compression Ratio, or H.U.C.R., as a way of expressing the merit of different fuels in a quantitative manner. It means the highest compression ratio at which the fuel can be used in an engine, with fully open throttle, without detonation occurring. Any such figure of merit depends upon the engine used for the test, as well as upon other things like speed, ignition advance, and inlet air temperature; but, being easily apprehended, it has a value for expressing the relative merit of fuels, provided the conditions of the test are properly defined, and will be used in this preliminary survey. The method, now widely accepted, of rating fuels for detonation behaviour against mixtures of pure iso-octane and *n*-heptane will be dealt with in Chapter VII.

The figures of H.U.C.R. given in table 12 have been found for certain pure substances by Ricardo and other workers using the former's 'E. 35' variable compression research engine, running at 1,500 r.p.m., under comparable conditions as regards temperature, ignition advance, etc.

A good average commercial petrol would show a figure of about 5·0, and an aviation spirit about 6·0, under the same conditions. Commercial 'paraffin' is composed of hydrocarbons boiling between about 100° and 300° C., and would probably have a H.U.C.R. in the neighbourhood of 3·5.

It will be seen that apart from octane and nonane the members of the paraffin series given in table 12 show a steady fall of H.U.C.R. as their molecular weights and boiling-points increase. The figures in this table were obtained before the vital importance of the form of the molecule, as distinct from the mere number of carbon atoms, was fully

TABLE 12

H.U.C.R. values in Ricardo E. 35 engine under standard conditions. That is to say at 1,500 r.p.m. full throttle, 30° ignition advance, two sparking-plugs diametrically opposite, cooling water outlet temperature 60° C., heat input to the carburetter 1,350 watts, and with the fuel-air ratio which gives maximum detonation.

Fuel	*H.U.C.R.*	*Molecular weight*	*Boiling-point or distillation range*
Aromatic Series:			°C.
Benzene (pure)	6·9†	78	80
Toluene (99% pure)	7·0	92	110
Xylene (91% pure)	>7·0	106	84–140
Naphthene series:			
Cyclohexane (93% pure)	5·9†	84	80·8–81
Paraffin series:			
Propane	>8	44	−38
i-Butane	7·5	58	+ 1
n-Pentane	5·85	72	36
n-Hexane	5·2	86	69·5–71·5
n-Heptane	3·75	100	98
Octane	4·6	114	126·5–135
Nonane	3·9	128	151·8–154·6
Ethyl alcohol	>7·5	46	78

† With these fuels pre-ignition occurred at these ratios but no detonation.

realized. Since Ricardo's early work a great number of pure chemical substances have been examined for their detonation behaviour[40] in an engine, and the result has been to confirm the fall of H.U.C.R. exhibited in table 12, but also to show that the isomers of the normal paraffins, in which the single straight chain of carbon atoms is replaced by one or more short branching chains, have totally different properties and in general detonate much less readily. Iso-octane, for example (2-2-4-trimethylpentane) is a powerful anti-detonant. When added to *n*-heptane it is nearly as effective as benzene in reducing the tendency to detonate.[41]

Later experiments with pure *n*-octane and *n*-nonane have shown that their H.U.C.R. values are much below that of *n*-heptane. The higher values shown in table 12 were doubtless due to the octane and nonane having been mixtures of the normal with isomeric molecules. The isomers, besides having the same molecular weight, have nearly the same boiling-points as the *n*-paraffins. There are seven possible isomers of heptane and eighteen of octane, so that the difficulty of exact identification leaves plenty of room for anomalous behaviour. Some indication of the purity of the substances tested can be found in the distillation range, where a definite boiling-point is not given. But a narrow range, or even a definite boiling-point, is no evidence of the absence of isomers.

Any spirits composed wholly of *n*-paraffins would be intolerable on account of detonation, and all commercial spirits, therefore, contain a certain proportion of aromatics and naphthenes, the former being added, if necessary, in the form of benzol, which is to all intents and purposes a non-detonating fuel. By additions of it, the H.U.C.R. of any other fuel can be raised in proportion to its total aromatic content. To improve the H.U.C.R. of an inferior fuel, however, up to the level required for a high duty aero engine, it is necessary to add as much as 40 per cent. of benzol, and this is undesirable from several points of view. Not only is benzol relatively scarce and therefore expensive, but also its heating value per pound is 10 per cent. lower than that of the paraffin hydrocarbons, and it freezes in the neighbourhood of −10° C.† instead of away down below −50° C.

Several substances have been discovered which, when added to petrol in even a very small proportion, have a pronounced anti-knock effect. The most important of these is the well-known lead-tetra-ethyl $Pb[C_2H_5]_4$, or lead ethide as we may call it for short, discovered by Midgley and Boyd,[11] a colourless liquid which decomposes when heated to 210° C. and will readily dissolve in petrol. In the pure state it is extremely poisonous. Lead ethide mixed with a proportion of ethylene dibromide, $C_2H_4Br_2$, is the 'ethyl fluid' of commerce, the second constituent being added in order to prevent the deposition of lead inside the engine after the decomposition of the lead ethide. The lead and bromine combine and are expelled with the exhaust gas as lead bromide, the greater part of which is not deposited until the gas comes into contact with the cooler parts of the exhaust system.

This so-called 'dope', ethyl fluid, when added to petrol in the relatively minute proportion of 1/900 by volume (5 c.c. per gallon) produces a rise of H.U.C.R. of about 15 per cent., and is equivalent to the addition of 30 per cent. by volume of benzol.

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† Pure benzene freezes at +5° C., but commercial benzol contains toluene and xylene, and these lower the freezing-point. According to the British standard specification the freezing-point of benzol shall not be higher than −14° C.

There are several other dopes, such as the iron and nickel carbonyls, $Fe(CO)_5$ and $Ni(CO)_4$, which are in the same class, but not so effective, and they are no more free from the liability to leave deposits in the cylinder. Apart from these organic compounds of metals, a great variety of non-metallic substances have been experimented with, in the hope of discovering a dope which is not subject to the two great drawbacks of ethyl fluid, its poisonous nature and its propensity for leaving harmful deposits in the engine. For although the association of ethylene dibromide with the lead ethide in ethyl fluid greatly reduces the amount of the deposits, it is not completely effective in preventing them altogether. When used in a proportion not greater than 1/900 by volume, which is the limit allowed commercially, ethyl fluid appears to be fairly innocuous, at any rate in engines for moderate duty; but when the proportion is raised much above this, or the power output and cylinder temperature of the engine are increased by supercharging, troubles from fouled plugs and valves are not slow to appear. Even if the proportion of ethyl fluid is only 1/900 a simple calculation will show that a 50 horse-power engine during a 10 hours run will have something like $\frac{1}{4}$ lb. of metallic lead supplied to it in the fuel, and even a very small proportion of this left behind in the cylinders cannot fail to lead to trouble in time. This addition of ethyl fluid in the proportion 1/900 raises the H.U.C.R. of an average petrol about 0·8 of a ratio, and as there are large quantities of straight-run petrol with an H.U.C.R. of 4·8 which one would like to raise to 7 or higher, it is clear that there is still a great opening for the ideal dope when it appears.

Several of the non-metallic substances, out of the wide field which has been explored in the search for an ideal dope, are much more effective than benzol in suppressing detonation, but not so much so as to place them in the same class as ethyl fluid, or to justify the distinction of being called a dope. A few of the most effective are given in table 13, which shows the percentage increase of H.U.C.R. when 5 per cent. by volume of each compound is added to an average good class petrol. The corresponding figure for benzene is conveniently 1·0, so that the figures in column 4 also give the relative effectiveness to that. Ethyl fluid, on the same scale, would be represented by the figure 700 and the pure lead ethide by about 1,000.

A full account of researches on a great variety of materials will be found in the references[12, 40] given, and it will be sufficient here to say that lead ethide has so far, among substances commercially producible,

TABLE 13

Percentage increase of H.U.C.R. obtained by adding 5 per cent. by volume of the anti-knock substance to an average good class petrol.

Name of substance	*Chemical formula*	*Boiling-point*	*Increase of H.U.C.R.*
		°C.	per cent.
Benzene	C_6H_6	80	1·0
Aniline	C_6H_7N	184	21·7
Methyl aniline . .	C_7H_9N	194	22·2
Ethyl aniline . .	$C_8H_{11}N$	204	10·4
Benzyl aniline . .	$C_{13}H_{13}N$	310	9·5
Toluidine . . .	C_7H_9N	200	18·3
Xylidine . . .	$C_8H_{11}N$	215	22·3
Cresol	C_7H_8O	190	5·8
Phenol	C_6H_6O	181	4·4

no serious rival, and that none of the non-metallic dopes are sufficiently effective, in comparison, to make it worth while to develop their use on a large scale, in spite of the great advantage of an absence of any non-volatile products arising from their use which could form deposits in the cylinders.

ART. 30. Engine experiments. Physical and chemical observations

Exhaustive researches have been made on almost every aspect of detonation as it can be observed directly in an engine. It has been found, for example, that its occurrence is always associated with an increase in the heat loss to the surfaces of the combustion chamber, and that there is therefore nearly always some loss of efficiency and power. Withrow and Rassweiler[39] observed maximum temperatures in a detonating engine about 300° C. higher than with no detonation owing to the more rapid completion of the combustion. During expansion, however, the fall of temperature was more rapid, so that the exhaust-gas temperature was lower under detonating conditions. Free carbon generally appears in the exhaust gases, and the occurrence of quite a slight degree of detonation may largely increase the rate of formation of gummy deposit in the piston-ring grooves. This last effect is shown even when the cooling of the cylinder has been increased so as to maintain a constant cylinder-wall temperature with and without detonation. Failure to burn all the carbon, even when there was no excess of fuel, was observed also in the compression-ignition machine to be described in art. 34. When the conditions were such as to produce the extremely rapid combustion that occurs in a detonating engine, fine powdery carbon was invariably thrown down.

If detonation is allowed to become very violent, any surface of aluminium alloy exposed to the combustion becomes eroded, in much the same way that the metal of a gun barrel is eroded. The increased rate of heat loss, already mentioned, is probably to be associated with the scouring effect of the hot gases, rather than with any large increase of their temperature; and this may also be responsible for the curious erosion observed, although chemical effects such as the presence of oxides of nitrogen are probably potent in this connexion, and possibly, also, in the promotion of piston-ring gumming.

The conclusions which it is possible to draw from engine observations are very limited, however, as regards the real nature of detonation. One can scarcely get farther than the conclusion that whether or not detonation will occur depends on the chemical nature of the fuel and on the initial temperature and pressure, and rate of combustion, of that part of the working fluid first ignited. This portion, as it heats up, compresses before it the still unburnt portion, and the latter may or may not be brought to the state of nearly instantaneous combustion which we associate with the name detonation, according to whether the fuel has a low or a high 'anti-knock' value. A simple conclusion, but one which cannot stand by itself on close examination, might be that a low and a high anti-knock value could be identified with a low and a high spontaneous or self-ignition temperature of the fuel when mixed with air and compressed. Such a conclusion is untenable because there are fuels which do not detonate readily in an engine and yet which show a very low self-ignition temperature when tested under conditions of sudden adiabatic compression.

It is necessary to interpose here that the figure obtained for the self-ignition temperature of a fuel depends very much on the nature of the experiment made to determine it. This may be of the type devised by Moore,[14] in which a drop of the fuel falls into a heated platinum crucible; or a fuel-air mixture may be suddenly compressed, and the lowest temperature may be calculated at which the heat of compression is sufficient to ignite the mixture;[2] or again, the fuel-air mixture may be made to flow through a tube heated in an electric furnace, and the temperature noted at which combustion begins to occur, as indicated by the appearance of products of the reaction at the end of the tube. Each method will give a different result. The temperatures found by *the crucible method will in general be a good deal higher than those* obtained in the other two methods. This would be expected, for although it is difficult to define and interpret exactly what is happening in the crucible, one may be fairly sure that there will be intense local cooling as a drop evaporates before combustion, and the effect of this will be to necessitate a crucible temperature higher than the minimum needed to produce ignition.

The other two methods do not really aim at observing the same thing. The temperature reached in the compression apparatus is a genuine self-ignition temperature, at which inflammation ensues, with a rapid completion of the chemical action. In the electric furnace, on the other hand, the first signs of combustion, as evidenced by the appearance of oxidized products, occur at a temperature well below that at which any general inflammation sets in. It might be expected that these temperatures of initial combustion would be lower than the self-ignition temperatures observed in the compression machine, and this is the case.

TABLE 14

Temperatures of initial combustion (T.I.C.) observed in passing fuel-air mixtures through an electric furnace, and spontaneous ignition temperatures (S.I.T.) observed by adiabatic compression.

Fuel	*T.I.C.*	*S.I.T.*
	° C.	° C.
Pentane	295	316–336
Hexane	266	307
Heptane	230	284–298
Octane	215	275–297
Nonane	210	..
Decane	210	..
Benzene	670	373–380
Toluene	550	..
Xylene	540	..
Aniline	502	..
Ethyl alcohol	520	..
Ether	145	212
Carbon disulphide	195	253

In table 14 are given some figures for the temperatures of initial combustion as observed by Mardles[15] with the electric-furnace method, and for self-ignition temperatures in the compression machine, as found by Tizard[1] and the present writer.[16] It will be seen that corresponding figures for the former are from 30° to 60° lower than the latter, and that both sets of figures agree in showing reactivity on the part of the paraffins at much lower temperatures than with the non-detonating aromatics, aniline, and alcohol; and also that within the paraffin series the greater propensity to knock of the larger molecules is associated with lower ignition temperatures. The behaviour of benzene in the electric furnace was most extraordinary, for it showed no preliminary stages of combustion at all. There was no sign of oxidation until a temperature was reached at which complete inflammation took place. This is the more significant when considered in conjunction with the fact, noted in table 12, that benzene was found to give pre-ignition at very high compression ratios without any previous detonation. There is here a clear suggestion that detonation is promoted by what happens during the early stages of chemical action in an engine cylinder, before any general inflammation has set in. This suggestion forms the basis of the theories discussed in the next article.

To return for a moment to the physical phenomena of detonation, we have said in art. 28 that it is accompanied by a wave of high pressure travelling through the burning gases and being reflected from the cylinder walls, and that this is set up when the rate of burning of a part of the charge becomes enormously accelerated. Egerton[13] has pointed out that it is incorrect to think of the detonation wave in an engine as being the same phenomenon as that which is characteristic of detonating combustion in a long tube; but that waves of high pressure are set up in an engine cylinder is now well established. The advent of high-speed photography and of electric pressure indicators devoid of inertia has enabled flame photographs to be made simultaneously with records of the rapid pressure variations which occur in the cylinder when detonation occurs.[43] The perfect synchronism observed between the peaks of the pressure records and the ripples in the flame photographs leaves no room for doubt that these are caused by pressure waves in the burning gas. Taylor[44] and others have measured the frequency of the pressure waves in a detonating engine and have shown that these agree closely with the calculated frequency based on the velocity of sound in the burning gases.

The nearly instantaneous combustion which sets up the pressure waves is confined, as already noted, to the last part of the charge to be burnt. This unburnt residue, or 'end-gas', will be at a pressure of the order of 600 lb. per sq. in. and will have been heated by compression during the early stages of combustion to, or beyond, its self-ignition temperature. If it then burns so quickly that the volume has not time to change appreciably, the local pressure will be of the order of 1 ton per sq. in., and may well suffice to explain a broken piston.

The occurrence of these local high pressures and the characteristic 'knock' which they produce cannot, however, be related simply to the self-ignition temperature of the fuel. To carry our description one stage farther reference must be made to the observations on 'ignition-lag' to be described in art. 34. It is there shown that combustion of a fuel-air mixture raised by rapid compression to its self-ignition temperature only takes place after a delay which may amount even to ¾ sec. The delay becomes shorter as the temperature of compression is raised, but delays of 0·05 sec. were observed even when the compression temperature was 40° C. above the minimum necessary to produce ignition. Since the time to complete the combustion in an engine even at 1,000 r.p.m. is only about 0·01 sec., and much of this is always occupied by the normal advance of the flame front, it is clear that the ignition-delay which is characteristic of a hydrocarbon/air mixture when heated by compression must be fundamentally important in the process of a knocking combustion. The existence of the delay period is sufficient in itself to explain the time element so often observed in a detonating engine, the knock becoming more severe as speed is reduced.

It must not be forgotten that detonation is ultimately a chemical problem. To understand what lies behind the phenomenon of the ignition-lag and give an explanation of the suppression of detonation by the addition to a fuel of a relatively minute proportion of lead ethide we have to study the complex chemical changes which take place in advance of the flame front in an engine cylinder, and may even have begun during the earlier history of the charge from the time when it was first exposed to the hot surfaces of the piston and exhaust valves.

This chemical aspect will be dealt with fully in the next article. If we are content for the present with an engineer's explanation, it can be said that there are two factors on which detonation depends, (*a*) the self-ignition temperature of the fuel, and (*b*) its ignition-lag, when rapidly compressed to a certain temperature. The fuel/air mixture which forms the 'end-gas', compressed first by the piston and later also by the expansion of the first part of the charge to burn, is heated to a temperature well above its self-ignition temperature. If given time it, or some part of it, will ignite instantaneously and produce a knock. On the other hand, the ignition lag may be long enough, or the engine speed and rate of flame travel high enough, to enable the flame front to spread right through the end-gas and consume it by normal combustion before detonation can occur.

From observations made on the compression machine to be described in art. 34 it has been suggested that the differences of ignition-lag observed with different fuels under comparable conditions may be related to differences in the rate of acceleration of the combustion of the fuel with rise of temperature, and Tizard has developed a theory of detonation based on this temperature coefficient of combustion. He pointed out that it was known from observations of the rates of slow chemical reactions at lower temperatures that the rates vary with the absolute temperature T in a way which can be empirically expressed by an equation of the form

$$k = Ae^{-B/T},$$

where k is the velocity constant of the reaction and A and B are constants, B being the 'temperature coefficient'. The values of A and B may be such that the rate of the reaction increases very rapidly with rise of temperature. Many reaction rates are more than doubled by a rise of 10° C. If this temperature coefficient is of the same order for a rapid reaction like the combustion of a hydrocarbon, then the extremely rapid combustion associated with detonation in an engine may be only the normal rate at the flame temperature for certain reactions, of which the temperature coefficient is sufficiently high. According to this view, it will be observed, there is no definite temperature above which ignition will occur and below which it will not. Chemical action will occur at any temperature above the absolute zero, but at ordinary temperatures it may be so slow as to be quite beyond detection.

A difficulty with Tizard's theory as a complete description of detonation is that the controlling factor should be the maximum flame temperature and that a knocking combustion would simply be one in which the spread of the flame front had become enormously accelerated by the combination of a high temperature coefficient with a high flame temperature. The very beautiful flame photographs of Withrow and Rassweiler,[40] however, of which some examples are reproduced in fig. 16, have shown quite definitely that the nearly instantaneous combustion associated with knocking is something quite distinct from normal combustion greatly accelerated. It starts at one or more points well in advance of the flame front (see exposure a 2 in fig. 16) and where the gas, therefore, is far below the flame temperature. The photographs of fig. 16 show most convincingly the rapid combustion under detonating conditions, complete about 4 deg. after the dead centre, as compared with the steady advance of the flame front when a non-detonating fuel (iso-octane) was used. In certain experiments two distinct centres of knocking combustion could be identified, starting independently. They were each shown both on the pressure record and in the flame photograph, and were due to two independent centres of self-ignition, each developing ahead of the flame front in different parts of the combustion chamber.

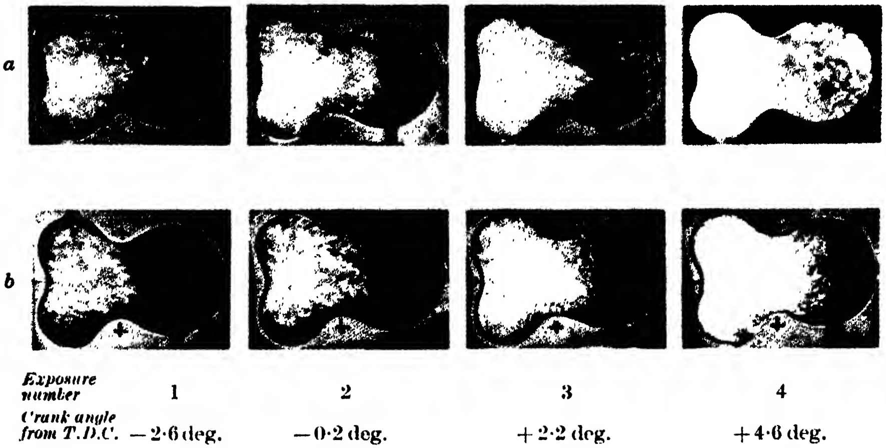

FIG. 16. Comparative photographs of (*a*) a detonating and (*b*) a non-detonating combustion, taken at the rate of 2,250 exposures per sec. Interval between exposures 2·4 deg. of crank revolution. Engine speed 900 r.p.m. Ignition advance 25 deg. Air: fuel ratio (*a*) 14:1 and (*b*) 15:1 by weight.

By courtesy of Messrs. Withrow and Rassweiler.

Although the description of a knocking combustion in terms of the self-ignition temperature of the fuel and its ignition delay under the conditions in the cylinder is satisfactory so far as it goes, it makes no attempt to explain why one fuel knocks and another does not, or why the addition of a minute quantity of lead ethide can profoundly affect the combustion characteristics of a whole gallon of fuel. If one pictures to oneself the mixture of air and vaporized fuel, how can it be that a few sparsely scattered molecules of the dope can affect the combustion of more than a thousand times as many molecules of the fuel? The answer to this problem calls for a chemical approach, and for experimental methods which go farther than is possible in an engine where the observations are limited to certain physical conditions of the combustible mixture treated as a whole.

ART. 31. Detonation as a chemical problem

Conclusions about the exact nature of the chemical reactions which go on in an engine cylinder are still to some extent tentative, chiefly because of the extreme difficulty of their close investigation under working conditions. It is virtually impossible to observe them directly, for the final products expelled through the exhaust valve give little clue to them, and if samples of gas are extracted from the cylinder during a cycle it is very difficult to be certain what further chemical changes the samples may not have undergone by the time they have been cooled and collected for examination.

Withrow, Lovell, and Boyd[30] have succeeded in following the chemical process of combustion to some extent, in its later stages, by examining the rate of disappearance of oxygen from the contents of an engine cylinder during the working stroke. For this purpose samples were extracted for analysis at different points in the cylinder head, and at different times. The great difficulty of this method is to keep the time of extraction of the sample sufficiently short, and to time the sampling valve with sufficient accuracy. In the work referred to, samples were taken over intervals as short as 2 degrees of crank revolution, at a speed of 900 r.p.m. The results are of much interest, as showing that there is a well-defined and comparatively narrow zone of combustion which travels out from the sparking-plug at a velocity depending upon engine speed, in other words upon gas turbulence; and, moreover, that the forward velocity of the combustion zone is less in the neighbourhood of the cylinder walls. These experiments indicate, further, that immediately in the rear of this narrow zone combustion is apparently complete, at least so far as this may be deduced from a complete disappearance of oxygen from the analyses.

As regards detonation, there was evidence of an accelerated speed of travel of the combustion zone when this occurred, but only during the last quarter of the maximum distance to be traversed from the sparking-plug. Detonation appeared to be without influence upon the rate of disappearance of oxygen until about three-quarters of the charge had been burned, an observation which has been confirmed by the flame photographs of Withrow and Rassweiler[40] already referred to, in which the nearly instantaneous combustion was shown to be confined to the last part of the charge to burn; and also by Schnauffer[45] in experiments in which the spread of the flame was traced by the ionization it produced between the points of 24 small spark gaps distributed about the cylinder head. Withrow and Rassweiler's photographs were taken through a quartz window forming the whole upper surface of the engine combustion space. In them the spread of a well-defined flame front can be clearly observed, followed, in the detonating engine, by self-ignition which started well in advance of the flame.

The technique of gas sampling for the study of chemical reactions in the cylinder has been developed by Egerton, Smith, and Ubbelohde.[46] They also succeeded in sampling the gases during a period of only 2 degrees of crank revolution. Besides very consistent analyses of the gases in the cylinder after top dead centre, which confirmed the conclusions of Withrow and his co-workers, Egerton was able to show that important reactions occur before the arrival of the flame front and especially in the neighbourhood of a hot exhaust valve. Earlier observations by Withrow and Rassweiler,[43] using a spectrographic method, had shown that the radiation from the flame of a knocking combustion differed from a non-knocking one, and that intermediate combustion products are formed in advance of the flame fronts in the last part of the charge to burn, and that these are much more in evidence under knocking conditions. Ricardo and Thornycroft[47] detected the presence of aldehydes in the cylinder contents immediately prior to combustion, and found that the addition of lead ethide to the fuel appeared to inhibit their formation. Dumanois and others[57] had also obtained evidence that when an engine was knocking there were aldehydes and peroxides† present, but not otherwise.

These earlier observations have been confirmed and placed on a quantitative basis by the work of Egerton, Smith, and Ubbelohde[46] referred to above. Fig. 17 shows the disappearance of oxygen, with production of CO_2 and CO, and fig. 18 the observed rise and fall in the concentrations of aldehydes and peroxides in an engine when knocking, under the same conditions of speed and throttle as in fig. 17.

The experiments were made on a 'Delco' engine of bore and stroke $2\frac{1}{2} \times 5$ in., which had a flat circular combustion space, with the sparking-plug in a recess at one side and the sampling valve near the opposite

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† The term 'peroxides' is here used to denote oxidizing substances which react with potassium iodide.

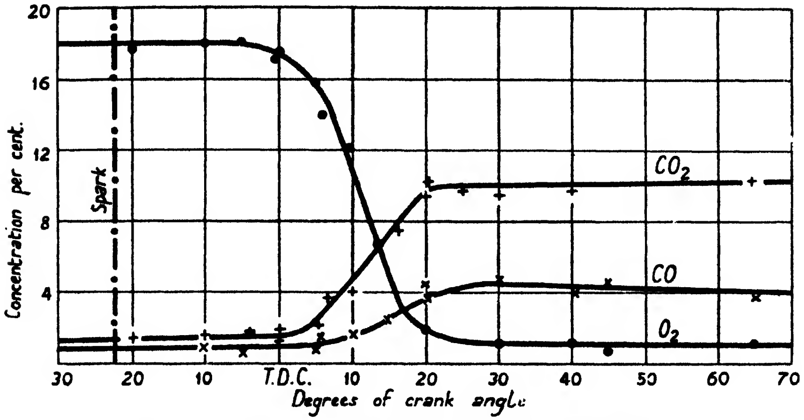


FIG. 17. Composition of cylinder gases, no detonation.

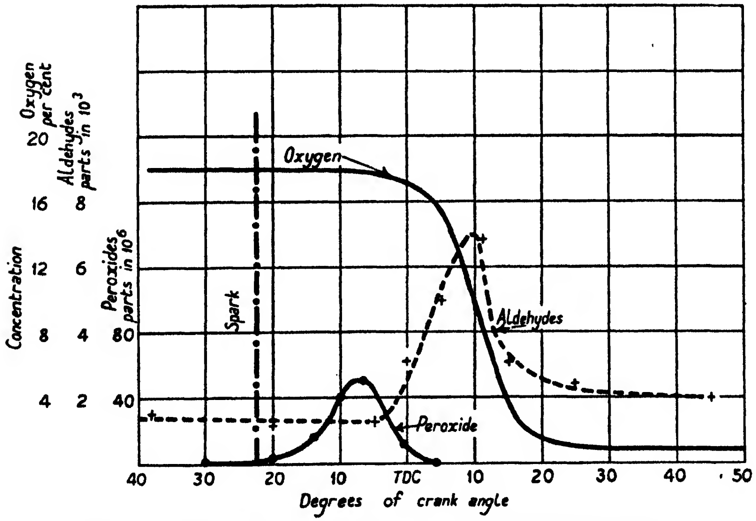


FIG. 18. Composition of cylinder gases, moderate detonation.

end of the same diameter. The distance from the centre of the valve to the sparking-plug was 3½ in. The speed was 600 r.p.m., compression ratio 6·75, and ignition timing 22½° before top dead centre.

It will be observed in fig. 17 that the disappearance of the oxygen occupies about 15° of crank angle, but the steepness of fall of the curve was a good deal affected by throttle opening and whether or not there was detonation occurring. Fig. 17 shows a non-knocking condition near full throttle. When there was violent detonation the slope of the oxygen curve was about three times as great. The great interest of Egerton's work lies in the consistent and reproducible figures obtained for aldehydes and peroxides, of which those in fig. 18 are typical for a condition giving moderate detonation. It will be seen that the peroxide concentration rises to 50 parts per million at 5° before T.D.C., by which time there was almost no oxygen burned, and had dropped to zero 10° later. In a non-knocking combustion there was never more than the merest trace of peroxides, while under violently knocking conditions the concentration rose to 145 parts per million 5° before T.D.C. and fell to zero within the space of 5°.

The variation of the maximum concentration of aldehydes between knocking and non-knocking conditions was far less marked, being only between 5 and 7 parts per 1,000, but the rate of disappearance was much greater under knocking conditions. The maximum concentration was always attained at the moment when the flame front reached the sampling valve. Under all conditions the concentration settled down to about 2 parts per 1,000, reached about 20° after T.D.C. in a knocking combustion, but not till 70° after in a non-knocking one. During most of the last part of the compression stroke the aldehyde concentration was constant at rather more than 1 part per 1,000, and since this cannot be accounted for by aldehydes in the residual exhaust gas it must be due to chemical action as the fuel-air mixture comes into contact with the hot exhaust valves.

When the fuel was benzol no peroxides at all were detected, nor was there any sudden rise or peak in the aldehyde curve. The latter remained constant at 1½ parts per 1,000 right up to T.D.C. and then gradually rose during the next 35° to another steady value of 3½ parts per 1,000. This peculiar behaviour, however, was later shown, as explained below, to be due to impurities in the benzol.

By testing samples of the cylinder gases when various aldehydes had been added to the fuel, it was shown that these produced no increase, but rather a decrease of peroxide formation, nor had the presence of aldehydes in the fuel any marked influence on the tendency to detonate. The peroxides, therefore, were not oxidation products formed from the aldehydes, as might, indeed, be concluded from the relative position of the peaks of the curves in fig. 18.

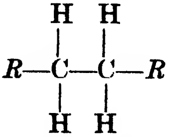
In further experiments Egerton showed that the chief peroxide of which evidence had been obtained was nitrogen peroxide, NO_2, and that its apparent rapid disappearance at T.D.C. was due to two causes. In part it was a genuine effect due to the rapid disappearance, after this point, of the oxygen necessary for peroxide formation; but in part it was merely that the detection of the NO_2 by the liberation of iodine from KI became inhibited by the presence of sulphur dioxide, derived from small quantities of sulphur compounds in the fuel. It was proved, also, that the non-appearance of NO_2 with benzol was due to the same cause. When pure, sulphur-free, benzene was used NO_2 was found to be formed in increasing quantities throughout the stroke.

There are strong reasons for thinking that the NO_2 owes its origin to surface reactions at the hot exhaust valves. In spite of the fact that the amount of it detected always increased with the intensity of detonation there was very little evidence that it had more than a secondary effect as a promoter of detonation. The organic nitrites and nitrates are powerful pro-knocks, but NO_2 itself was not found to be so when deliberately introduced, and furthermore there are fuels like cyclohexane which gave rise to a relatively high proportion of NO_2 and yet are very good from the point of view of detonation. All one can say, therefore, is that with some fuels the same causes which produce NO_2 give rise to the conditions necessary for detonation to occur. There seemed good reason to suspect, however, that certain organic peroxides, known to be potent knock producers, might accompany the formation of the NO_2. Special methods were therefore worked out for detecting organic peroxides in the presence of NO_2, from which it was concluded that although the quantities are minute and their existence transient, certain organic peroxides behaving like ethyl hydrogen peroxide are found in the region where knock occurs just prior to the passage of the flame. Detonation, or knocking combustion of the last part of the charge to burn, is due to the violent break-up of these peroxides through their power of promoting chain reactions (see below) in a part of the combustible mixture which has been 'pre-sensitized' by the formation of these products of partial combustion. The type of reaction to which knocking is due is nearly always found to be one which is inhibited by the presence of lead ethide, and no theory of detonation can be satisfactory which does not provide an explanation of this inhibitory action by a few sparsely scattered molecules of the dope distributed throughout the fuel-air mixture. The first attempt at an explanation was Callendar's theory[12] of 'nuclear drops'. He pointed out that whereas steam tends to be condensed during a sudden expansion and re-evaporated by a compression, the reverse is true of certain hydrocarbon vapours. They can be condensed by a sudden compression. Callendar suggested that during the compression stroke of an engine the small quantity of non-volatile fractions of a petrol, which would be the last to evaporate, would remain as a comparatively small number of nuclear drops, and that even if the vapour pressure of the fuel were too low to allow any recondensation upon them on account of the compression, this tendency of hydrocarbons to recondense with rise of pressure would at any rate prevent complete evaporation before the end of compression. It was known that low volatility and high molecular weight, in the paraffin series, were associated with low ignition temperature and a readiness to detonate, and Callendar supposed that the tendency of any fuel to detonate was bound up with the persistence of these nuclear drops of low self-ignition temperature.

When a dope such as lead ethide was present it would naturally become concentrated upon the nuclear drops after evaporation of the lighter fractions, since its own boiling-point is of the same order, about 210° C. The number of nuclear drops would of course be small compared with the number of fuel molecules, and if we imagine that the dope associated with each drop can act as some kind of inhibitor upon the chemical activity of the drops, we have at least removed the difficulty of the apparently wide field of effectiveness of each dope molecule.

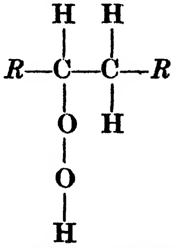
There are obvious difficulties about accepting the theory. There are the awkward facts, for example, that certain gaseous fuels which can produce no nuclear drops can yet be made to detonate, and that when the experiment was tried of evaporating liquid hexane completely before admission to an engine cylinder, its detonating qualities were unaltered. The nuclear-drop theory, therefore, can be shown to be inconsistent with some of the major experimental facts, and we have to seek some other explanation of the widespread effectiveness of some of the metallic dopes.

Egerton[12, 13] was led by his own and other experimental results to frame a theory which fits in well with the conclusions to be drawn from the experiments on low-temperature oxidation, and at the same time explains the action of dopes. He suggested that chemical action starts at a comparatively small number of 'centres of high energy' wherever two specially active fuel and oxygen molecules happen to have encountered one another and combined together. The first result of such an encounter is the temporary formation of one of the unstable organic peroxides by the incorporation of the oxygen molecule in the fuel molecule. One may illustrate what is supposed to take place by the following symbolic picture of the molecular changes. Suppose the fuel molecule before it encounters an oxygen molecule is represented by a formula of the type



$$\begin{array}{ccccccc} & & H & & H & & \\ & & | & & | & & \\ R & - & C & - & C & - & R \\ & & | & & | & & \\ & & H & & H & & \end{array}$$

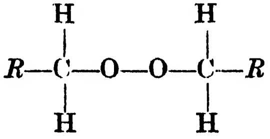
in which *R* is a short way of representing some combination of carbon and hydrogen atoms depending on the molecular weight of the particular hydrocarbon. If this were hexane, for example, each *R* would represent CH_2CH_3 (see art. 16). Now after collision with an oxygen molecule it is supposed that this may become incorporated so as to form a compound molecule, either of the type



$$\begin{array}{ccccccc} & & H & & H & & \\ & & | & & | & & \\ R & - & C & - & C & - & R \\ & & | & & | & & \\ & & O & & H & & \\ & & | & & & & \\ & & O & & & & \\ & & | & & & & \\ & & H & & & & \end{array}$$

alkyl hydrogen peroxide

or



$$\begin{array}{ccccccccccc} & & H & & & & & & H & & \\ & & | & & & & & & | & & \\ R & - & C & - & O & - & O & - & C & - & R \\ & & | & & & & & & | & & \\ & & H & & & & & & H & & \end{array}$$

dialkyl peroxide.

Whichever peroxide is formed the compound molecule will be an unstable body in a high energy state, and a variety of things may happen to it. The new molecule may collide with another fuel, or another oxygen, molecule, with which it will react and produce further highly active products. These products in turn may collide with, and activate, other fuel molecules, and so on, a 'reaction chain' being set up. Egerton has suggested that low-temperature oxidation, such as may occur in advance of the flame front, can only make progress through the setting up of reaction chains of this kind, since the average molecular energy is below that corresponding to the self-ignition temperature. Anything which can break, or stop, the chain of reactions, will inhibit early oxidation and the formation of the highly active peroxides in a quantity sufficient to lead to detonation.

There will be only a small number of these reaction chains at first, for the starting of each will depend either upon the presence of a hot surface or upon the collision of fuel and oxygen molecules both of an energy well above the average molecular energy, which defines the temperature of the mixture. *We have seen how the product of one of these collisions may set going a chain of reactions which release more and more energy and so propagate the chain. But if one of the early collision products had met with a comparatively inactive metallic* peroxide such as lead peroxide PbO_2, instead of meeting another fuel or an oxygen molecule, much less energy would have been liberated and the chain would have got no farther. Now during the compression stroke of an engine the lead-ethide molecules $Pb(C_2H_5)_4$ are certainly decomposed, and the first result will be the formation of PbO_2. We have here, then, a possible way in which the reaction chains may be broken, and Egerton has made a further series of observations of great importance in the way they help towards an explanation of the great effectiveness of certain of the metallic dopes. It had already been found by several observers that in the metallic dopes it was the metal radical that was the essential part, and Egerton showed that all the metals which possess an anti-knock effect have the common property that they form *two* oxides, a higher and a lower, for example PbO_2 and PbO; and that a state of equilibrium can exist between these at the temperatures in an engine cylinder towards the end of compression. This means that when the PbO_2 molecule has reacted with a compound fuel-peroxide molecule, with mutual destruction and rearrangement and the reduction of the PbO_2 to PbO, the latter can be regenerated to PbO_2 by the next suitable impact with an oxygen molecule. Any such regeneration will mean that a single atom of lead from a lead-ethide molecule may be effective again and again in helping to break a reaction chain.

It was mentioned earlier, and illustrated in table 13, that some organic substances, for example aniline and the aromatic amines, are powerful anti-knocks, although too feeble, when compared with the very active metallic compounds, to justify their being classified as dopes. Their behaviour when oxidized is very similar to that of the metallic dopes, but of course we cannot expect any of the effectiveness derived from the regenerative behaviour described above, because the whole of the dope would undergo combustion itself at the temperatures at which it would have to operate. The relative ineffectiveness of the organic anti-knocks, therefore, fits in well with Egerton's theory. If the theory is sound, one may add, there is little hope of discovering the ideal dope;

for this would be one with all the effectiveness of lead ethide which yet would add nothing new to the products of combustion. In other words, it would be a dope which contained only carbon and hydrogen and could be trusted to burn away completely; whereas it is an essential of Egerton's theory that an effective dope should contain an indestructible radical of which the level of oxidation can be raised and lowered by successive impacts with oxygen and fuel-peroxide molecules.

ART. 32. Some further facts about fuels and dopes

Much detailed information is now available about the interaction between fuels, dopes, and lubricating oils in an engine, which is of practical importance and of some theoretical interest, although it does not add much to our knowledge of the fundamentals of detonation. It has been found,[17] for example, that successive additions of ethyl fluid to a petrol are not equally effective. The curve *AB* in fig. 19 shows the increase of H.U.C.R. through additions of ethyl fluid† up to 1/200 by volume (23 c.c. per gallon) to a petrol of which the H.U.C.R. when undoped was 4·75. In the same figure the curve *AC* shows the effect of benzol added to the same fuel.

It has also been found[18] that small additions of lubricating oil to a fuel, of the order of 1 per cent., and especially of the vegetable oils castor and rape, are able to destroy almost completely the anti-knock effects of the iron- and nickel-carbonyl dopes. In a fuel doped with ethyl fluid, on the contrary, not only do these vegetable oils not have any bad effect upon detonation, but they actually appear to improve the anti-knock value of the doped fuel at normal intake temperatures. The mineral oils have a less marked effect on all the dopes in the way of reducing their anti-knock value, but on ethyl fluid rather less than the others. It will be seen from fig. 19 that the addition of 10 c.c. per gallon raises the H.U.C.R. from 4·75 to 6. The presence of 8 per cent. of mineral oil in the fuel will lower this figure to about 5·6, and other amounts in proportion. There are certain differences between mineral oils dependent upon the crude oils from which they are derived, and for information as to these details reference should be made to the original papers cited.

It might at first sight be supposed that the effect, at any rate of the mineral oils, in reducing the H.U.C.R. of a doped fuel was due not so much to an effect of the oil on the dope, as to a natural propensity

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† The ethyl fluid contained 65 per cent. by volume of tetra-ethyl lead.

of the large hydrocarbon molecules of the oil to set up detonation on their own account. This, however, does not seem to be so, in the first place because even large additions of oil are found never to reduce the H.U.C.R. of a doped fuel below that of the same fuel without the dope, and secondly because, when added to an undoped fuel of naturally high H.U.C.R., they do not produce the same deleterious effect.

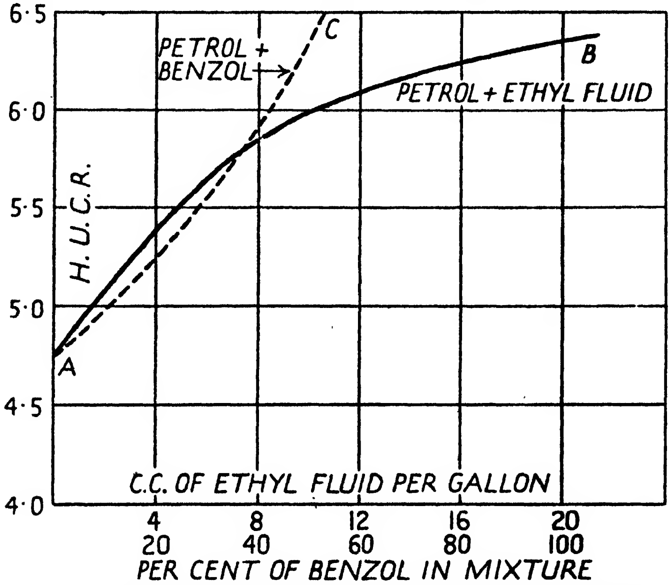

FIG. 19. H.U.C.R. of petrol with added ethyl fluid † and benzol.

A high temperature of the indrawn fuel-air mixture might be expected to promote detonation through the consequent increase of compression temperature. This it does, but experiments with the mixture raised to temperatures up to 90° C. have brought out certain interesting differential effects of temperature. A high induction temperature, for example, is more detrimental to the high anti-knock qualities of a fuel with much added benzol than if the H.U.C.R. is obtained by added ethyl fluid. It will be seen, in the figures of tables 15 and 16, that a rise of induction temperature from 10° C. to 90° C. produces, apart from any added oil, a fall of H.U.C.R. from 6 to 5 in the benzol mixture, but only from 6 to 5·25 in the ethyl-fluid mixture. With 8 per cent. of added mineral oil the fall of H.U.C.R. is from 5·75 to 4·7 with the benzol mixture, and only 5·55 to 4·9 with ethyl fluid. With 8 per cent. of

† See footnote, p. 111.

TABLE 15

Petrol and ethyl fluid. H.U.C.R. at 10° C. and 90° C. induction temperatures with various proportions of added castor or mineral oil in solution.

Percentage of oil in fuel-oil mixture	*Castor oil H.U.C.R.*		*Mineral oil H.U.C.R.*	
	at 10° C.	at 90° C.	at 10° C.	at 90° C.
0	6·0	5·25	6·0	5·25
4	6·05	5·05	5·75	5·05
8	6·1	5·0	5·55	4·90
12	6·2	4·95	5·35	4·80

TABLE 16

Benzol in petrol. H.U.C.R. at 10° C. and 90° C. induction temperatures with various proportions of added castor and mineral oil in solution.

Percentage of oil in fuel-oil mixture	*Castor oil H.U.C.R.*		*Mineral oil H.U.C.R.*	
	at 10° C.	at 90° C.	at 10° C.	at 90° C.
0	6·0	5·0	6·0	5·0
4	5·95	4·9	5·85	4·85
8	5·9	4·8	5·75	4·70
12	5·85	4·75	5·65	4·55

castor oil the effect of temperature on the benzol mixture is just about the same but on the ethyl-fluid mixture it is greater, producing a drop of 6·1 to 5·0 as against 5·55 to 4·9. Column 2 of table 15 exhibits the remarkable fact that 12 per cent. of added castor oil improves the anti-knock value of the ethyl-fluid mixture at 10° C. induction temperature from 6·0 to 6·2.

The bearing of these experiments, in which lubricating oils were added to the fuel and introduced into the cylinder with it, upon practical engine conditions, is rather difficult to assess. It will depend upon how much lubricating oil can get dispersed, during normal running, throughout the fuel-air mixture. It has been estimated that, if the oil consumed by an engine be expressed as a percentage of the fuel consumed, then an amount of oil equal to 1·5 per cent. of the fuel by weight is required for lubrication and is without effect on detonation; and that about half the remainder may be taken as affecting detonation in the degree shown by tables 15 and 16. Since the oil consumed by an engine may be 5 or 6 per cent. of the fuel, one may conclude that the effect of lubricating oil on detonation may become appreciable in practical working. Detonation may be just as serious even if confined to one or two cylinders of a multi-cylinder engine, and with an overall consumption of lubricating oil amounting to 6 per cent. of the fuel, it is always possible that certain cylinders are getting more than a fair share and that in them the consumption may be 10 or 12 per cent., with serious results on detonation. In supercharged engines it is difficult to prevent oil from the supercharger from being carried into the cylinders with the air. It is found that the amount so carried in may be as much as 4 per cent. of the fuel weight, and being swept in by the heated air it will be in the best (or worst!) condition to promote detonation.

V

COMBUSTION IN THE ENGINE CYLINDER

Art. 33. Carburetter and heavy oil engines compared

In every internal-combustion engine, whether it works upon the Otto or Diesel cycle, whether 'two-stroke' or 'four-stroke', and whether the fuel be gas, petrol, or a heavy fuel oil, the essential series of operations is always the same. A cylinder-full of gas, mostly nitrogen, is compressed, heated, expanded, and thrown away. In a Diesel engine, air only is compressed in a volume-ratio of between 12 and 16 to 1, and the temperature of the air is then sufficiently high (550°–650° C.) to ignite the fuel oil when it is squirted in against the compression pressure.

For obvious reasons the carburetter engine, which draws in a combustible mixture of air and gas, or volatile liquid fuel, cannot compress this mixture to anything like the same degree. It is an essential of the carburetter engine that ignition should *not* be brought about by the compression temperature alone, but should await the spark. The compression which can be used is limited further by the onset of detonation, and this differs, as we have seen, according to the fuel used.

It has been emphasized in Chapter II that there is little or no justification for the habit, common to many books upon the subject, of drawing the distinction between the petrol engine and the Diesel cycle as though it were a distinction between engines in which combustion occurs under conditions of approximately constant volume and constant pressure, having ideal indicator diagrams like those illustrated in figs. 2 and 3. With the modern developments of the Diesel engine, conditions have so altered that in these days the real line of demarcation should be between carburetter engines with spark ignition on the one hand, and high-compression engines with auto-ignition and burning a heavy fuel oil, on the other. It is true that constant pressure combustion is approximately realized in some slow-speed engines, but in the high-speed heavy oil engine it becomes necessary to inject the fuel well before the top dead centre to get complete and efficient combustion, exactly as it is necessary to have an early spark in a high-speed carburetter engine, and under these conditions the indicator diagrams may show a rise of pressure after ignition which, seen to a base of piston displacement, looks equally vertical in both types of engine. Early fuel injection leads to better fuel economy, but also to higher maximum pressures, and the limits of early injection are settled by what maximum pressure in the cylinder we are prepared to put up with.

Figs. 20 and 21 are two indicator diagrams, each drawn to the same base of piston displacement and with the same pressure scale. The first is from a petrol engine of compression ratio 5 : 1, running at 1,900 r.p.m., the second from a compression-ignition engine of ratio 14 : 1 at 2,200 r.p.m. The point *A* marks the instant of ignition in the first and the start of injection in the second, the first 35° and the second 15° before the dead centre position, which is marked by *B* on each diagram. Thereafter the dots in each diagram indicate time intervals corresponding to 2° of crank angle. The point of maximum pressure occurred in each engine at just about 12° after the dead centre.

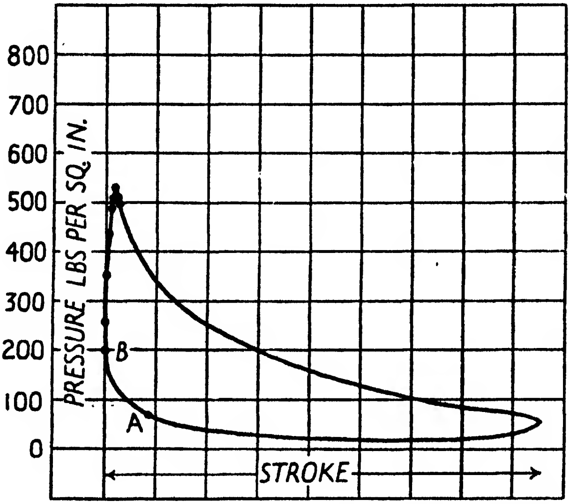

FIG. 20. Indicator diagram from a petrol engine.
Speed 1,900 r.p.m. Compression ratio 5 : 1.

The Diesel diagram, owing to its high compression, is a good deal more sharply pointed than the petrol diagram, and it is clear that, as regards combustion at constant volume, there is little to choose between the two. Apart from pressure differences due to the difference of compression ratio, the two engines behave in a similar manner; and it must be understood that fig. 21 represents a perfectly normal diagram from a well-adjusted high-speed Diesel engine.

We shall see later on in this chapter that for many purposes indicator diagrams drawn to a base of piston displacement like figs. 20 and 21 do not show sufficiently clearly what is going on during the interval from 45° before to 45° after the dead centre. Events during this most important quarter-revolution of the crank can be much more clearly seen if the pressure curve is drawn to a base of crank angle as in fig. 22, where the two sets of conditions represented in figs. 20 and 21 are re-plotted on the same diagram. As before, the points of ignition and of injection are marked *A* on the respective diagrams, and it will be seen that while the petrol engine reaches its maximum pressure 12° after the dead centre, the Diesel does so slightly sooner, at about 10° after.

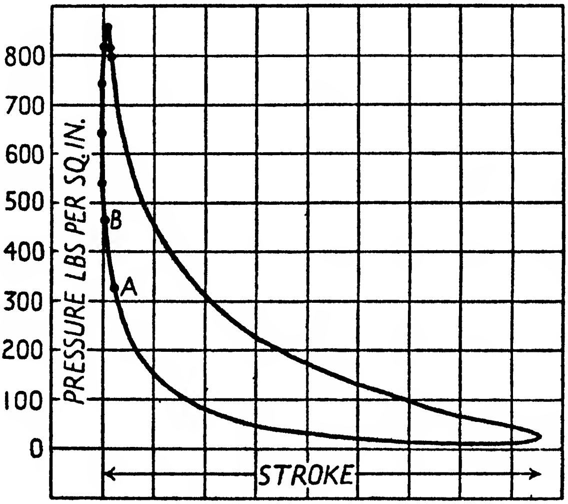

FIG. 21. Indicator diagram from compression-ignition research engine. Speed 2,200 r.p.m. Compression ratio 14 : 1.

It will certainly be surprising, to any one not familiar with this type of diagram, to learn that the indicated mean effective pressure was a good deal greater in the petrol than in the Diesel engine: 136 lb. per sq. inch as against 122 lb. per sq. inch. In explanation of this it must be remembered, in the first place, that areas on the pressure-time diagram of fig. 22 do not represent work, as they do on a *PV* diagram, and that the high pressures round the dead centre only last while the piston is almost stationary; in the second place, during all the period to the left of the dead centre line, the piston is doing negative work against the compression pressure. It follows that although the pressure curve of the petrol-engine diagram lies throughout far below the Diesel diagram, the average *difference* between the pressures to right and left of the dead centre line, when plotted on a base of piston movement, can be greater in the smaller of the two diagrams.

The real value of an indicator, however, in giving us pressure-time records like those of fig. 22, does not lie in the power it gives of calculating the indicated mean effective pressure. We shall see in Chapter VII that with high-speed engines the indicator is of little use for this

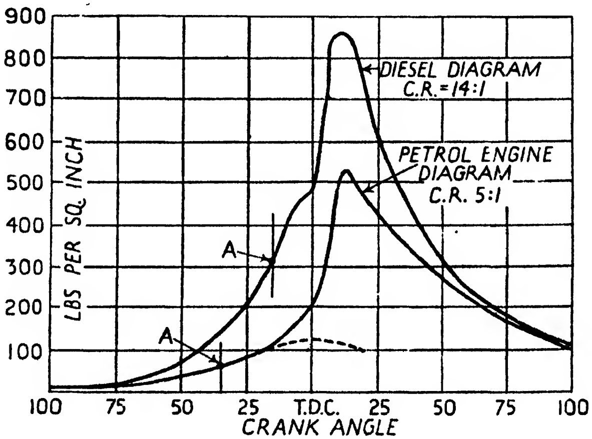

FIG. 22. The indicator diagrams of figs. 20 and 21 plotted upon a base of crank angle before and after the dead centre.

purpose when a high degree of accuracy is required. The purpose for which an indicator record is of supreme importance when plotted to a base of time, or crank angle, is for giving information about the process of combustion in the cylinder from the moment when the spark passes, or injection commences, until after expansion has well started. It is upon the sequence of events during this period that the power output and efficiency of an engine mainly depend. How much valuable information can be gleaned from the pressure-time record will become clear in the next article.

ART. 34. The combustion process in a homogeneous fuel-air mixture

The promotion of rapid combustion in a Diesel engine is very different from that in a homogeneous mixture of air with gas, or the vapour of a volatile liquid, and will be treated separately in arts. 39 and 40. We will, however, so far anticipate what is said there as to point out that in fig. 22. the compression pressure on the Diesel diagram, if no ignition had taken place, would have been about 470 lb. per sq. inch, and that the curve would have followed the indicator diagram so closely as to be indistinguishable except for the last few degrees before the dead centre. With the petrol-engine diagram the compression pressure would have been about 125 lb. per sq. inch, and the compression-expansion curve has been added to the diagram as a dotted line. For the first 15° after passage of the spark there is no appreciable difference, whether combustion is taking place or not. There are two reasons which may contribute to this. In the first place, as the flame front travels from the spark the *mass* of charge consumed is small in the early stages. This will be readily appreciated if we picture what happens when a charge, enclosed in a spherical vessel at say 100° C., is ignited at the centre. When the flame has travelled *one-fifth* of the radius there will be a central sphere of gas at a temperature of, say, 2,150° C. absolute. The mass of this gas will be

$$\frac{1}{125} \times \frac{373}{2,150} = \frac{1}{720}$$

of the whole mass of gas. Hence the rise of pressure will be only about one seven-hundredth of the ultimate rise. Even when the flame has travelled nearly half (0·47) the radius, the pressure rise will be only about two per cent. of its final value.

This is not the whole explanation of the apparent delay period in spark-ignition engines, however, and the subject is sufficiently interesting and important to justify a short digression to describe some compression-ignition experiments in which it appeared in a much more obvious and even dramatic form.[2, 16] The experiments were made by Tizard and the present writer upon a machine of Ricardo's design illustrated in fig. 23. In this machine a combustible mixture is rapidly compressed in the cylinder A, in any desired ratio, which can be varied between wide limits, and the plunger B is then held stationary at the top of compression. The initial temperature before compression can be varied by alteration of the cylinder-jacket temperature, and the compression ratio by screwing the whole cylinder and cylinder head up or down. The movement of the piston is effected through the operation of an internally expanding friction clutch fitted on to a large and heavy flywheel which is rotated at any desired speed by an electric motor. When the clutch is engaged the piston is driven upward through the medium of the toggle motion shown, and when it reaches the top of its stroke it is automatically locked into position. At the same time the outside sheath of the telescopic connecting-rod, which drives the piston up, disengages from the central rod, on which it slides up and down freely so long as the clutch is kept engaged.

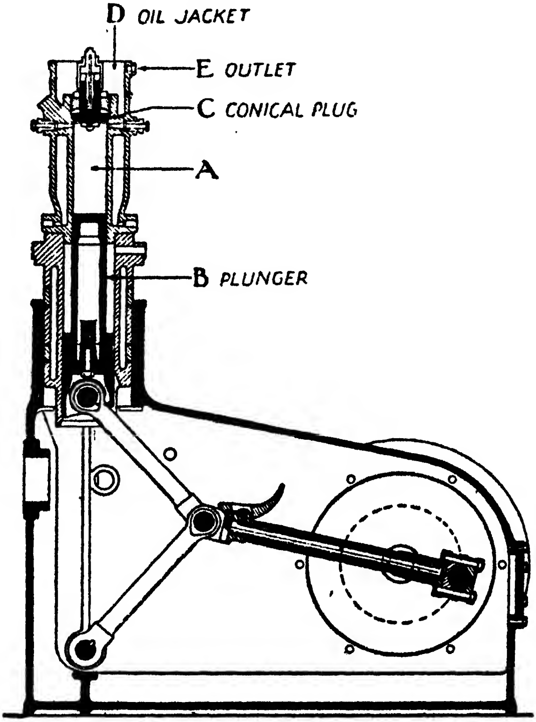

Fig. 23. Machine for producing a single rapid compression.

When the initial temperature and ratio of compression were such that the temperature reached by the adiabatic compression was a little above the self-ignition temperature of the particular fuel used, it was found that a record such as that illustrated in fig. 24 was obtained. The surprising nature of this diagram will be clear when it is understood that the portion *AB* represents the pressure rise due to the motion of the piston, and that this is followed by a period of constant or slightly *falling* pressure *BC*, which in some experiments lasted no less than $\frac{3}{4}$ second. In the record illustrated it was $\frac{1}{4}$ second, the wavy line, which gives the time scale, being the trace of a tuning-fork making 100 vibrations per second. At the point *C*, which is the end of the delay, explosion takes place without any further outside influence whatever. The rapid rise of pressure shown at *CD* is followed by subsequent cooling, everything from the point *B* onwards having happened at constant volume, and with no appreciable gas leakage meanwhile.

Fig. 24. Pressure-time record taken with the compression-ignition machine.

During the delay period *BC*, cooling to the cylinder walls must have been going on, and one can argue therefore that ignition in some form started at *B*, for after that the average temperature must at once have started to fall. During all the delay period minute nuclei of combustion must have been smouldering, and then, after an appropriate period of incubation, burst out into a full-fledged explosion.

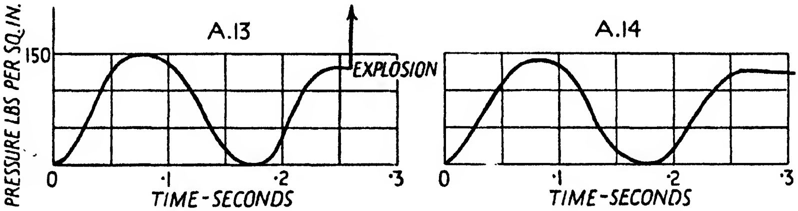

Fig. 25. Two exceptional records obtained from the compression-ignition machine.

It must be understood that this period of incubation was not a matter of chance. With all liquid hydrocarbon fuels it was found to be very consistent and to depend upon the amount by which the compression temperature exceeded a certain minimum below which ignition would not occur at all. As the compression temperature was raised above this minimum, so the delay period was shortened.

How remarkably persistent these nuclei of combustion could be was proved by the accidental and very surprising records reproduced in fig. 25. In each of these two experiments, *A*. 13 and *A*. 14, the clip which should have held the piston failed to act the first time, and in consequence the first compression was followed by an expansion to atmospheric pressure, and then by a second compression, at the end of which the clip held fast. The second compression in *A*. 13 is to a lower pressure by reason of heat loss to the walls, and possibly some slight gas leakage, during the first. Since the temperature during the second compression must have been considerably lower than during the first, and was also lower than that in the second experiment *A*. 14 in which no ignition occurred at all, it appears certain that nuclei of combustion started at the moment of the first compression in *A*. 13 and were able to survive a drop in the average gas temperature down to rather below 15° C., from which initial temperature the first compression was made.

TABLE 17

Illustrating the constancy of the delay period with widely varying amounts of ignition advance.

Speed . . . . . . 1,800 r.p.m.
Compression ratio 7:1
Maximum power mixture.

Ignition advance	*Approximate temperature at the moment of spark*	*Period before visible pressure rise*
Degrees before dead centre	° C.	Degrees of crank revolution
39	314	20
34	335	18
29	356	15
24	377	15
19	397	15½
14	415	15
9	427	13

The records shown in figs. 24 and 25 were obtained with volatile liquid fuels, but the same phenomena were also exhibited by gases such as hydrogen and acetylene, although the results were less regular and the delays for the most part shorter.

This phenomenon of delayed combustion which was demonstrated so clearly in the compression machine no doubt contributes to the period of 15°–20° of crank angle after the spark has passed in a petrol engine during which combustion has no appreciable effect upon cylinder pressure. Indeed, the likelihood is that it is mainly responsible for it. In an engine the mixture during this period is violently turbulent, whereas in a closed-vessel experiment or in the compression machine it is stagnant. The reason for the slow rise of pressure in a closed-vessel experiment given on p. 119, therefore, cannot be applied closely in an engine, where the rate of spread of the flame is extremely rapid as soon as combustion has fairly started.

Although turbulence has a profound effect as soon as this stage is reached, it has little or none during the delay period (see fig. 27). The approximate constancy of the delay period under widely different conditions of temperature and pressure is illustrated in a beautiful set of diagrams reproduced in fig. 26, which were obtained by Ricardo. These were all taken at one time, during which the engine was running with exceptional steadiness and consistency, and from the very perfect diagrams obtained it was found that with the various ignition timings marked on the curves, the periods which elapsed before there was an appreciable pressure rise due to combustion were as given in table 17.

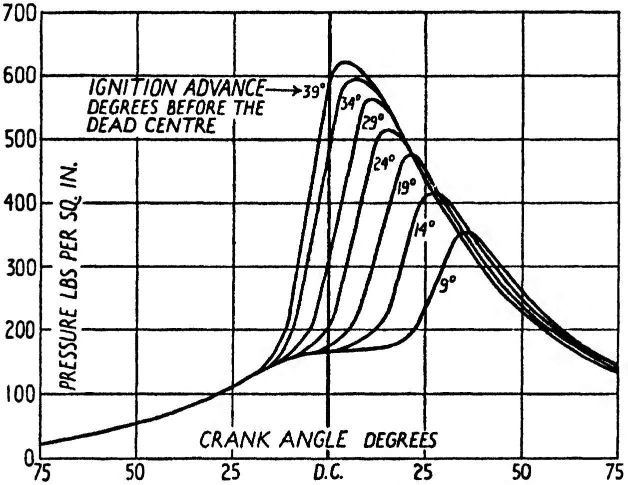

FIG. 26. Series of indicator diagrams showing the effect of changing ignition advance. Fuel, petrol. Maximum power mixture strength. Speed 1,800 r.p.m. Compression ratio 7:1.

Between the diagram with the earliest and that with the latest spark the pressure was approximately doubled at the moment the spark passed, having risen from 84 to 180 lb. per sq. inch, while the temperature as shown in the table had risen by about 113° C. In spite of these wide variations of temperature and pressure at the moment the spark was passed the delay was roughly constant throughout. Its average value for the seven diagrams was 16° or 1/675 second. It follows therefore that the 'ignition advance' to compensate for the delay period, expressed as it usually is in terms of crank angle, must be increased in proportion to the crankshaft speed.

This ignition advance to compensate for the delay period is only a part of the full advance necessary, for the time occupied by the pressure rise has also to be allowed for, so that the peak of the pressure curve shall be reached before expansion has seriously begun. Experience has fixed 12° after the dead centre as a good standard to adopt for efficient working. If the peak pressure is reached later than this, it means that some heat of combustion is not developed until a point when the remaining ratio of expansion is seriously reduced, and therefore also the efficiency with which this late-developed heat can be utilized in the engine.

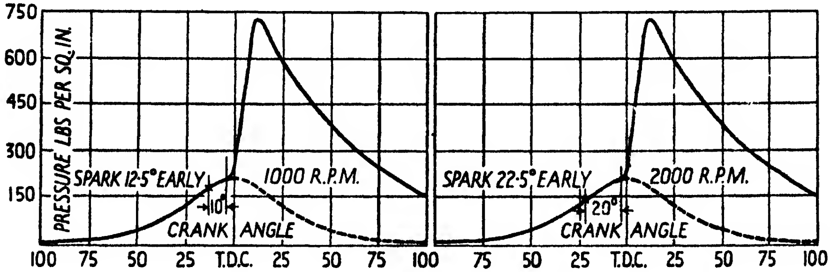

FIG. 27. Diagrams showing increase of delay period with increase of speed when expressed as degrees of crank revolution.

As soon as the delay period is over and general combustion has set in, the rate at which it proceeds is directly dependent upon turbulence. Without turbulence the rise of pressure would be so slow that it could not be completed in the time available. Fortunately, as engine speeds increase, so also do the turbulence and the rate of pressure rise, so that the number of degrees of crank revolution occupied by the rise remains substantially constant, and there is no difficulty about maintaining satisfactory early peak pressures at high speeds,. provided increased allowance (expressed as crank angle) is made for the delay period which remains, throughout, a constant length of time.

The features described above are very well illustrated by two diagrams given by Ricardo[19] and reproduced in fig. 27. It will be seen that the pressure rise begins in each diagram 2½° before the dead centre and is completed about 10° after it, although the speed of revolution in one is twice that in the other. The time occupied by the pressure rise was therefore just halved when the speed was doubled. On the other hand, 20° of ignition advance had to be allowed for the delay period at the higher speed against 10° at 1,000 r.p.m.

The allowance for the delay in these experiments was 1/600 second as against 1/675 second in those previously quoted and illustrated in fig. 26. The constancy of the delay period is closer than one would expect, considering that the two sets of experiments were carried out at quite different times and on a different engine. It is to be expected that the delay would vary to some extent with the nature of the fuel used, as well as with the mixture strength.

Reference has already been made in art. 26 to experiments by Glyde,[29] in which the spread of the flame front in an engine cylinder was directly observed. Both in these, and in the experiments of Withrow, Lovell, and Boyd,[30] referred to in art. 31, figures for the rate of spread of the flame under various conditions were obtained. Absolute values for the rate of spread will necessarily depend upon the design of the cylinder head, but both sets of observations confirm the conclusion reached above, that when inflammation has once started, its rate of spread is dependent mainly upon turbulence and is nearly proportional to engine speed. Glyde, using a stroboscopic method of direct observation of the advancing flame front, through a series of small glass windows in the cylinder head, states that for combustion chambers of the Ricardo turbulent type (see art. 61 and fig. 82), with a rate of pressure rise of from 30–40 lb. per sq. inch per degree of crank revolution, the flame velocity may be given as of the order of 30 ft. per second initially, rising to a maximum of 300 ft. per second, at an engine speed of 900 r.p.m. In the ordinary type of combustion chamber, with side-by-side valves, in which the rate of pressure rise was 23 lb. per sq. inch per degree in these experiments, the initial flame velocity was found to be about 25 ft. per second, at 900 r.p.m., rising to 150 ft. per second at its maximum value. The process of combustion occupied about 30 degrees of crankshaft movement in this type of head, and 25 degrees in the turbulent type.

Both in this engine, and in that used by Withrow, Lovell, and Boyd, it was found that the spread of inflammation, after a slow start, reached a maximum speed and then slowed down again as the flame front approached the cylinder wall, provided always that detonation was not occurring. In both series of experiments, also, it was found that the rate of spread near the walls was always less than across the centre of the head. As compared with the maximum flame speeds given above from Glyde's paper, the American experiments show maxima which vary from 50 ft. per second at 500 r.p.m. up to 200 feet per second at 1,200 r.p.m. Since the engine used in these experiments had a plain L-type head, with no special arrangements for promoting turbulence, one may say that the agreement between the two sets of results, arrived at by totally different methods, is remarkably good.

ART. 35. Ignition and ignition timing

As in the last article, so in this one, we shall be concerned only with the carburetter engine. In this type, ignition is now almost universally initiated by a high-tension spark, which is generated at the plug points either by a magneto or by a battery and high-tension coil. For the present all we need to know about these accessories is that they are able to produce, at one or more plugs in the cylinder head, a spark which can be accurately timed to occur at a particular moment, the timing being adjustable to suit the conditions of running. When the spark is produced by a magneto the energy liberated, or in common parlance the 'fatness' of the spark, depends to some extent upon when it occurs. When generated by a battery and high-tension coil, on the other hand, the spark energy is the same under all conditions.

In an engine running at full throttle and on a full power mixture the intensity of the spark matters little. The natural rate of burning of the mixture is so great that a spark of some sort is all that is needed to establish the combustion nuclei, which in due course develop into rapidly growing centres of combustion, and these are swept at high speed throughout the mixture by its own turbulence. If the engine is throttled, however, or the mixture strength is weak, the energy and duration of the spark may be very important. Under these conditions the natural rate of burning of the mixture, that is, the rate of spread of the flame under stagnant conditions, is much reduced, and combustion of the whole cylinder contents in a reasonably short time is much helped by a fat and lasting spark, which then fulfils the function of a torch, past which the sluggishly burning gases are swept, and ignited as they go.

The rapid completion of combustion is helped to some extent even under full power conditions by simultaneous ignition at more than one point; either because of the greater number of nuclei established, or because the spark normally lasts rather beyond the delay period and gives a certain amount of 'torch effect'. The fact that all aero engines carry two plugs per cylinder is not solely a precaution against failure of the ignition system. It is invariably found that a few per cent. more power can be obtained with two plugs than with one, as well as a slightly improved efficiency.

Some interesting experiments were made by Burstall[20] upon the effect of varying the number and placing of the sparking-plugs in one of the well-known 'E. 35' variable compression engines designed by Ricardo.[21] This engine has a flat, circular combustion space with four plug sockets almost equally spaced round the periphery. There are three exhaust and two inlet valves arranged in the flat cylinder head as shown in fig. 28, which also shows the position of the plug sockets in relation to the valves. Burstall found that the position of a plug or plugs in relation to the valves was important. In his experiments, in which coal gas was the fuel, there was a noticeable improvement both in power and efficiency when plugs were placed near the inlet, as against the exhaust, valves. Two plugs showed an advantage over one, except that one plug in position *A* or *B* was superior to two together when these were both near the exhaust valves at *C* and *D*. The most instructive of the quantitative results are summarized in table 18. The deduction which one may perhaps draw from the beneficial influence of having a plug or plugs near to an inlet valve is that the later stages of combustion,

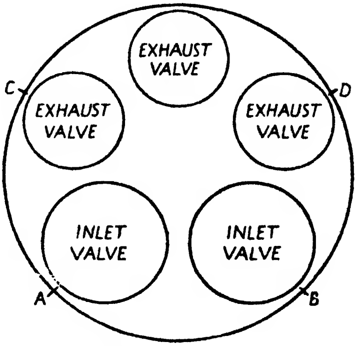

FIG. 28. Plan view of cylinder head of Ricardo 'E. 35' variable compression engine, showing position of sparking-plugs at *A*, *B*, *C*, and *D*.

TABLE 18

Influence on power and efficiency of the number and arrangement of sparking-plugs.

E. 35 engine. Fuel, coal gas.
Speed, 1,000 r.p.m. Compression ratio, 5:1.

No. of plugs and position	*Indicated mean effective pressure*	*Thermal efficiency*
	lb. per sq. inch	per cent.
2 at *A* and *B*	101·6	33·0
2 opposite	100·7	33·0
1 at *A* or *B*	100·7	32·5
2 at *C* and *D*	99·6	33·0
1 at *C* or *D*	97·4	31·7

N.B. The fact that I.M.E.P. and efficiency do not vary together indicates slight changes of mixture strength between the experiments. The figures must therefore only be read as a rough guide to the effect of plug positions.

in the neighbourhood of the hot exhaust valve, were speeded up; and that the effect produced was equivalent to using a greater ignition advance. The experiments were done with a constant ignition advance, and it seems likely that if the amount of this had been adjusted for maximum power in each test, the effect of sparking-plug position might have disappeared in these coal-gas experiments. It must be remembered that, with a gaseous fuel, a possible richening of the mixture near the inlet valves, such as one might expect in a petrol engine, cannot have existed. In a petrol engine such a richening of the mixture might be responsible for improved performance when ignition was at a point near the inlet valves, but not so with gas.

Coal gas being a non-detonating fuel, this factor was not present to affect Burstall's results. With a liquid fuel liable to detonation, Ricardo has shown that this is promoted by firing the charge at a point remote from the exhaust valve, so that the later stages of the combustion take place in its neighbourhood. When the cylinder head is of wide and shallow design, with the exhaust valve and sparking-plug on opposite sides, then the conditions are the worst possible for promoting detonation.

The figures for I.M.E.P. in table 18 were obtained with the same ignition timing and, of course, at the same compression ratio. With a fuel liable to detonation it would be possible to run at an appreciably higher compression ratio with a single plug at *C* or *D* than with one at *A* or *B*, and the optimum arrangement, with such fuels, would depend chiefly upon this factor of detonation.

In a cylinder head with two sparking-plugs and four valves, two inlet and two exhaust, each pair side by side, the optimum arrangement is to have the plugs at opposite ends of a diameter, and each one equidistant from one inlet and one exhaust valve.

It has been mentioned in the last article that to obtain the highest power and efficiency from an engine, it is important that combustion should be as nearly as possible complete before expansion begins. Complete combustion, whatever the arrangement of sparking-plugs, will occupy a certain length of time, which may vary between wide limits according to the fuel-air ratio. With very weak mixtures the burning may be so slow that it must be initiated 40° or 50° before the dead centre, if it is to be reasonably complete before expansion has seriously begun: if the peak pressure, let us say, is to be reached within about 10° of the dead centre. When we adjust the spark timing so that an engine develops its maximum power for any particular speed and fuel-air ratio, we are striking the best compromise between two conditions, namely, we are making the spark sufficiently early to allow combustion to be nearly complete before expansion begins, and not making it so early that the pressure during the last part of compression is seriously raised by early combustion, with a consequent increase in the negative work to be done by the piston.

A too early ignition, besides reducing the power through an increase in the negative work, will cause an excessive rate of pressure rise and maximum pressure, which between them will make a rough-running engine, even if violent detonation does not immediately set in. Over the range of ignition advance from 9° to 39° before the dead centre, covered by the series of indicator diagrams shown in fig. 26, the rate of pressure rise, after ignition was well started, rose from 17 to 36 lb. per sq. inch per degree of crank revolution; while at the same time the maximum pressure rose from 360 to 625 lb. per sq. inch. The brake mean effective pressure in the same series reached a maximum at about 25° advance, but showed a variation of only 1 lb. per sq. inch between 19° and 34° advance.

A simple adjustment for maximum power, therefore, may not be sufficiently sensitive as a method of fixing ignition advance if we wish to study combustion conditions in the cylinder. The merits of the pressure-time form of indicator diagram for studying the effect of ignition timing have already been emphasized in the last article, and a further demonstration of its value is afforded by figs. 29 and 30. The two diagrams in fig. 29 show, by the difference between them, the effect, in a gas engine burning a weak mixture, of increasing the ignition advance by 11° of crank angle, from 52° to 63° before the dead centre. The two diagrams have been re-plotted to a base of piston displacement in fig. 30 and yield an identical mean effective pressure. It follows that the alteration of 11° in the ignition timing produced no difference in the power as measured by the brake, although the point of maximum pressure was reached, in one case 4°, and in the other 11°, after the dead centre.

For studying the effect of a change of ignition timing or of fuel-air ratio, then, upon the rate of burning, it is essential to have some standard more exact than an adjustment for maximum power. It has been mentioned already that, on an average, maximum power is combined with maximum efficiency when the peak pressure is attained at 12° after the dead centre. This was the standard condition adopted by Burstall in his experiments with coal gas which are further illustrated in fig. 31.

In this diagram the curve *AB* shows the ignition advance which it was necessary to give, so as always to develop maximum pressure

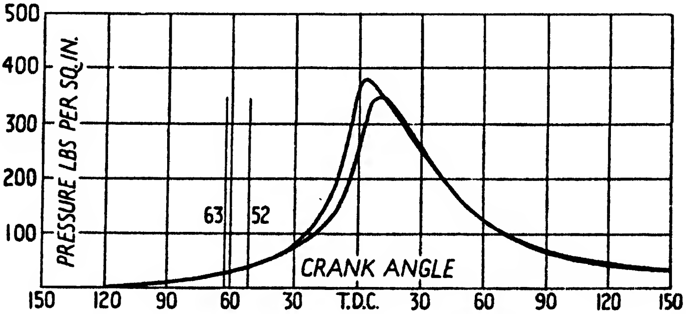

FIG. 29. Two indicator diagrams, with ignition advance 52° and 63° respectively, all other conditions constant. Fuel, coal gas. Speed 700 r.p.m. Compression ratio 5 : 1. Mixture 65 per cent. of correct.

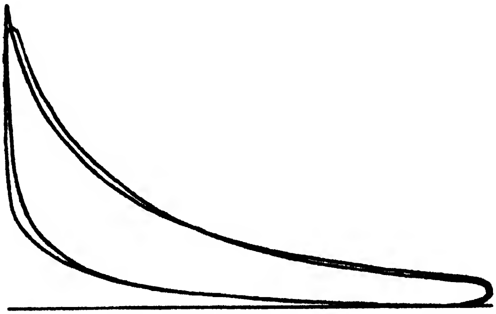

FIG. 30. Indicator diagrams to a base of piston displacement, plotted from the pressure-time records of fig. 29.

12° late, when the fuel-air ratio was varied between 50 per cent. weak and 40 per cent. rich, at 1,000 r.p.m. and compression ratio 5 : 1. The results show how widely the rate of burning differs for different mixture strengths.† It will be observed that, in order to obtain the peak

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† The curves for coal gas in fig. 31, as well as those in fig. 33, have been plotted from Burstall's figures, which assume a volume ratio in the correct mixture of coal gas and air of 1 : 4·5. Accurate determination of the ratio of coal gas to air for complete combustion is a difficult matter on account of the presence in the gas of a small proportion of heavy hydrocarbon vapours, such as benzene. These small quantities are difficult to determine, and yet they have a large influence upon the air required per cubic foot of gas. Subsequent recalculation has indicated that in Burstall's experiments the ratio of gas to air

pressure in the standard position, a minimum spark advance of 30° was required at just about the maximum power mixture of 10–20 per cent. rich (see art. 36), and that as the mixture was weakened the necessary advance steadily increased until for a mixture 46 per cent. weak, 74° was required. It follows that the total time occupied by the burning of this weakest mixture, 86° of crank angle, was just double that of the most rapidly burning mixture, which gives maximum power.

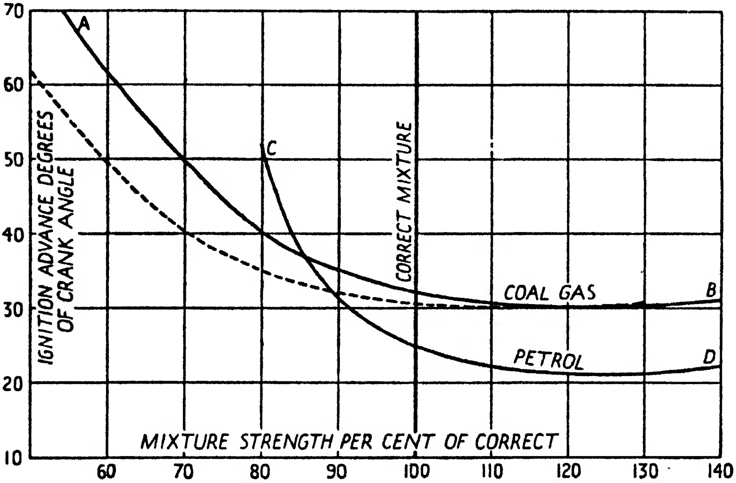

FIG. 31. Ignition advance necessary to develop maximum pressure 12° after the dead centre. Speed 1,000 r.p.m. Compression ratio 5:1.

With the very weak mixtures and early ignition a fair amount of combustion was going on during the last 30° of the compression stroke. This can be seen in fig. 32, which is an indicator diagram taken with the weakest fuel-air ratio of fig. 31. The compression diagram for no ignition is shown dotted, and it can be seen that for the last 30° of the compression stroke the pressure of the full-line diagram is appreciably above the other, and that at the dead centre the extra pressure produced by combustion amounts to 50 lb. per sq. inch. The delay period of nuclear combustion has lengthened out with this very weak mixture to no less than 44°, as compared with 10° at the same speed for a full power petrol-air mixture, as illustrated in fig. 27. The two cases, however, are not really comparable, for in addition to the

in the correct mixture was more nearly 1:4 than 1:4·5, as assumed by him. This means that the curves for coal gas in figs. 31 and 33 should be shifted to the left by about 10 per cent. of mixture strength, to the positions shown dotted.

difference of fuel-air ratio the compression ratio in the petrol engine was higher.

The curve *CD* in fig. 31 gives the same data in regard to ignition advance and fuel-air ratio for a petrol engine, as *AB* gives when coal gas is the fuel. The two curves when placed side by side in this way bring out very clearly the difference in behaviour between liquid and gaseous fuels on the weak side. On the rich side their behaviour is very similar: the minimum ignition advance, 21° for the petrol and 30° for the gas engine at this compression ratio, is required in each case for a fuel-air ratio about 20 per cent. rich. For both fuels this is the most rapidly burning mixture and the one giving maximum power.

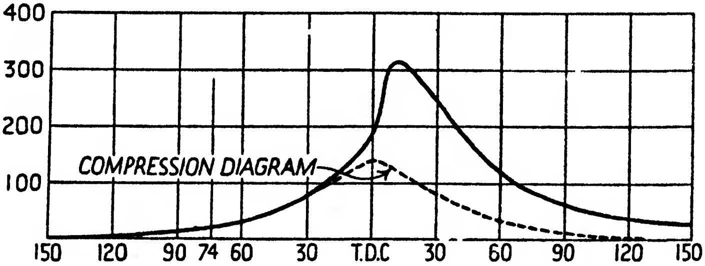

FIG. 32. Diagram with very weak coal gas-air mixture and 74 degrees ignition advance. Speed 950 r.p.m. Compression ratio 5:1. Mixture 54 per cent. of correct.

On the weak side, on the other hand, the behaviour of the two is very different. In the gas engine, provided the appropriate ignition advance is given, the fuel-air ratio can be weakened steadily down to 54 per cent. of the correct, with regular combustion and a rising efficiency. In the petrol engine, on the other hand, when the mixture is weakened from 90 per cent. to 80 per cent. of the correct, a rapid increase of ignition advance from 30° to 50° is called for, and beyond that point no amount of ignition advance can, in general, promote sufficiently rapid and complete combustion to give a peak pressure at the standard point of 12° after the dead centre. It has been found possible in an air-cooled aero engine to maintain steady running down to mixtures only 70 per cent. of the correct, but the operation on these very weak mixtures was found to be very sensitive to the form and temperature of the combustion space. So much so, in fact, as to preclude their use in practical service. At less than 80 per cent. of the correct mixture the combustion, as a rule, becomes hopelessly irregular, and so slow as to be still incomplete at the end of the expansion stroke; a condition which causes the familiar 'pop-back' through the carburetter. Some further discussion of this characteristic difference between the combustion of weak mixtures in gas and petrol engines is given in art. 43, p. 174, and may be referred to at this point.

In discussing fig. 31 we have passed from the discussion of ignition timing to the related question of fuel-air ratio, which will be dealt with more fully in the next article. The relation between the compression ratio and the appropriate ignition timing will be dealt with towards the end of art. 37.

ART. 36. The influence of the fuel-air ratio

Under this heading, again, the heavy oil engine stands completely apart from the gas and the petrol engine. Since the fuel is injected at one or more points in a fine spray which penetrates the air, and burns as it goes, it is clear that ignition of the fuel in a Diesel engine will not be affected by the ratio of the total air in the cylinder to the amount of fuel injected, so long as there is an excess of air. In the carburetter engine, on the contrary, with its approximately homogeneous mixture of fuel and air which has to be ignited locally at one or two points, combustion will only propagate itself and spread from the plug points if the fuel-air ratio is within certain definite limits. In a Diesel engine, if there is more air than is needed to combine with the fuel, the latter will take up as much oxygen as it requires to convert the carbon and hydrogen to CO_2 and H_2O, and the excess air will simply be heated and expanded. In the neighbourhood of the jet the fuel-air ratio must vary from point to point between infinity and zero, and this factor can impose no more limitation upon combustion than it does in a candle-flame. If there is an excess of fuel, the portion injected late will fail to get burnt completely and will pass out through the exhaust valve as carbon or partially burnt products, and produce a black smoke.

It is, perhaps, the most important feature of the Diesel engine that it is able to work efficiently, indeed more and more efficiently, as the fuel-air ratio is reduced. When running at less than full load the air charge remains unaltered, but the fuel injected is reduced. The result is that the temperatures of the working substance throughout the cycle are reduced and the detrimental effects of the rise of volumetric heat at high temperatures are less pronounced. The behaviour of the working substance at the lower range of temperatures is less far removed from that of a perfect gas, and a closer approximation to the ideal air-cycle efficiency is possible.

In the carburetter engine the ratio of fuel to air giving just complete combustion, the 'correct' mixture, is about 1:14·5 by weight for petrol and 1:4 by volume for coal gas; with approximate workable limits of from 1:18 to 1:9 for petrol and from 1:9 to 1:3 for coal gas. The limits will vary with the chemical composition of the fuel, more especially in the case of coal gas.

On the rich side of the correct mixture, the proportion of fuel can be increased, in either a petrol or gas engine, by 40 or 50 per cent. without much affecting combustion, except that of course there will be in the exhaust gases a high proportion of CO and other intermediate combustion products. As we shall see presently, a small increase of power is obtained in either case as the fuel is increased up to a point when the mixture is about 10 to 20 per cent. rich, and falls off slowly beyond that point.

The ability of a petrol engine to accommodate itself to a wide range of mixture strengths on the rich side, and to give nearly constant power all the way between a correct mixture and one 50 per cent. rich, is of great practical importance, for it enables the engine to survive the shortcomings of its carburetter and induction system. There are few carburetters which succeed in maintaining anything like a constant fuel-air mixture at all speeds and throttle openings, and no induction systems which distribute the fuel equally to all cylinders. Happily, owing to the power of accommodation exhibited by the individual cylinders, it matters little to the smooth running of an engine, provided no cylinder gets too weak a mixture, whether some of the cylinders get much too rich a one. But it matters, of course, from the point of view of fuel economy, for in order to ensure that every cylinder gets enough fuel we have to put up with several getting more than they need or can fully make use of. We shall see in art. 42 that this loss due to unequal fuel distribution between the cylinders amounts to about 10 per cent. of the whole fuel consumption, and may be even more.

Turning now to the behaviour of fuel-air mixtures on the weak side, which is really more important, for it shows the pathway to economy in fuel consumption, we saw in the last article that there is an important difference between liquid and gaseous fuels. Although a gas engine can be run steadily and efficiently upon a mixture containing scarcely more than half the fuel for the correct mixture, when a petrol-air mixture is weakened more than 18–20 per cent. the rate of burning becomes so low and irregular that it may continue throughout the expansion and exhaust strokes and ignite the incoming charge for the next stroke, causing a back-fire through the carburetter. Even before the point of weakness has been reached at which irregular firing and popping-back occurs, the working of a petrol engine becomes inefficient through slow burning and failure to generate the full gas pressure early enough in the expansion stroke.

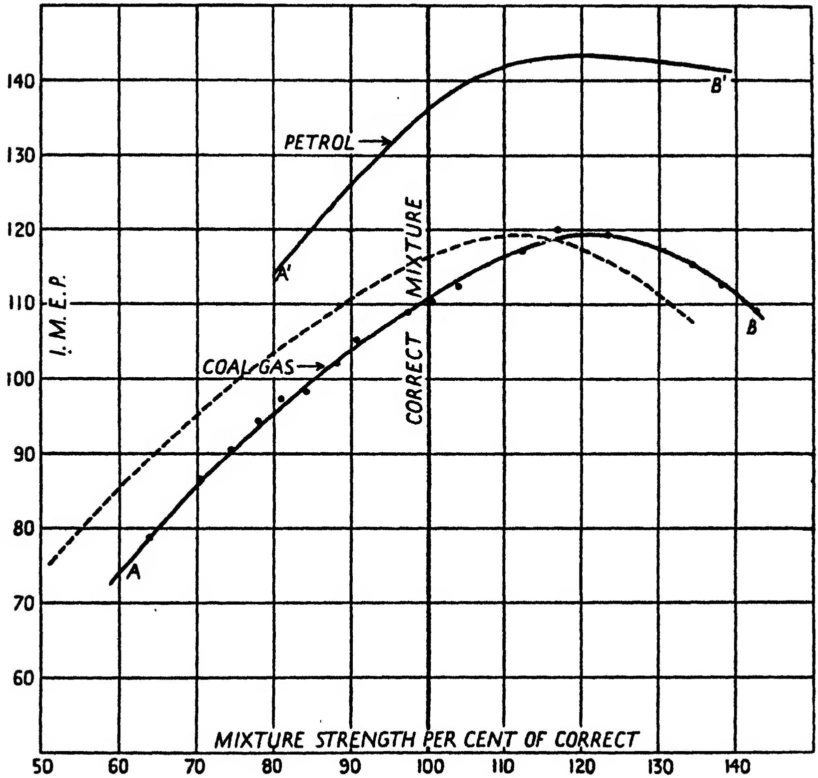

Fig. 33. Indicated mean effective pressures at different fuel-air ratios. Compression ratio 5 : 1. No heat to air intake. Speeds: petrol 1,500 r.p.m., coal gas 1,400 r.p.m.

Turning first to the gas engine experiments of Burstall,[20] the curve *AB* in fig. 33 shows the observed variation of indicated mean effective pressure as the mixture of coal gas and air was varied on each side of the correct mixture,† in an engine running at 1,400 r.p.m. and com-

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† See note p. 130.

pression ratio 5:1, the ignition advance being always adjusted to the standard condition with the peak pressure at 12° after the dead centre. It will be seen that the maximum power was obtained with a mixture about 10 per cent. rich, according to the dotted curve, and that the power was then about 3½ per cent. above that obtained with the correct mixture. On the weak side, that is to the left of the correct mixture line, the M.E.P. of course falls away as the fuel in the mixture is reduced. It does not do so proportionally, however, because the indicated thermal efficiency rose from about 32 per cent. to 36 per cent. for the range of mixtures covered, as is shown by the curve *CD* of fig. 50 on p. 175, and in consequence the power obtained at the weakest mixture was to this extent a larger proportion of the heat supply. There may also be small changes of volumetric efficiency to be allowed for. Power, for constant speed conditions, is of course proportional to the I.M.E.P., and the power of the gas engine, as shown by the left half of the curve *AB*, is therefore well maintained down to mixtures about 50 per cent. weak, and down to this point it has fallen away a good deal less rapidly than the fuel supply. A reduction of the fuel to half that in the maximum power mixture is seen in fig. 33 to have resulted in a drop of power in the ratio of 120 to 80, or to 67 per cent. of the maximum.

The curve *A'B'* in the same figure illustrates the corresponding behaviour, when using petrol, of the same engine under comparable conditions. On the weak side the curve does not extend beyond 20 per cent. weak, at which point running became irregular, but it lies throughout substantially above the gas-fuel curve, and it is instructive to examine the factors which contribute towards the higher mean pressures obtainable with petrol. It will be observed that a higher thermal efficiency is not the cause, for the efficiency curve *C'D'*, in fig. 50, would nowhere lie above the curve *CD* obtained with gas, even if the latter were shifted to the left to allow for the error in the correct fuel-air ratio already referred to.

In the first place, when running on gas, the fuel-air ratio at the correct mixture is about 1:4·1 by volume, which means that of the gases entering the cylinder per stroke only about 4/5 is air, whereas with petrol the fuel enters the cylinder in the form of a heavy vapour displacing only about 2 per cent. of the air which might be drawn in (see art. 23). On this account, therefore, we might expect the power at the correct mixture to be increased when using petrol, over that obtainable with gas, for the quantity of oxygen available, and therefore also the heat per cubic foot of cylinder contents, will be greater. The quantity of available oxygen with petrol and with gas will be, very roughly, in the ratio 0·98/0·8 = 1·22. By reference to tabulated values (pp. 284–5) and to arts. 15 and 23 it will be found that the heats of combustion at constant volume and at 100° C. of the correct petrol-air and coal gas-air mixtures, taking heptane as typical of petrol, are respectively 78,680 and 70,020 ft.-lb. per S.C.F. The ratio of these energies, namely 1·12 to 1, is the chief factor in accounting for the difference in power output with petrol and with coal gas as fuels. There are, however, many other factors which come in to influence the power of the engine. Besides a possible difference of thermal efficiency, the latent heat of the liquid fuel will cool the ingoing mixture. The lower average temperature, and therefore greater weight, of the charge in the cylinder when the inlet valve closes will tend towards an increased power with liquid fuel, but to set against this there is the fact that in most petrol engines there is a certain amount of deliberate heating of the mixture on its way in through the induction system. This facilitates evaporation, and at the same time nullifies the cooling effect of the latent heat. Another difference between the gas and the petrol engine is that the 'choke tube' necessary with a petrol carburetter can be eliminated on a gas engine, and this allows a freer passage for the ingoing air and a higher volumetric efficiency on the score of the higher pressure in the induction system during suction. The mean effective pressure obtainable at any given speed, again, is very dependent upon whether the valve setting is the most appropriate for the speed at which the test is made. It is, therefore, unsafe to draw conclusions from any figure for mean effective pressure, or power, until allowances have been made for a host of possible influences.

In the comparison given in fig. 33 the two tests were made upon the same design of engine, at the same compression ratio, and with no special heating of the ingoing air in either case. There was a slight difference of speed and of valve setting, but these differences can be allowed for, since the volumetric efficiency was known during each set of tests. On gas at the correct mixture it was 76 per cent., and on petrol 77 per cent., so that the effect of the cooling due to fuel evaporation was not all counteracted, when running on gaseous fuel, by the freer air passages referred to above. The differences of thermal and volumetric efficiency, combined with the greater heat energy per cubic foot, should between them account for the higher power with petrol. If the mean pressure observed with coal gas at the correct mixture is increased in the appropriate ratio on account of the last two of these factors, we obtain

$$116 \times 1{\cdot}12 \times \frac{77}{76} = 132,$$

which is about 3 per cent. lower than the observed figure for petrol. The petrol figures were obtained by Ricardo about 1921, and those with coal gas some three years later by Burstall at Cambridge.

To hark back for a moment to the curve *AB* in fig. 33, the increase of power of at least 3½ per cent. with a rich mixture of coal gas over that obtainable at the correct one is especially interesting, for although the same thing had previously been observed in a petrol engine by Ricardo, prior to Burstall's experiments it might always have been argued that the increase of power on rich mixtures was due to some extent to an increased volumetric efficiency brought about by the extra cooling through evaporation of surplus fuel. In a gas engine no such effect is possible, and there is at the same time the increased displacement of air by fuel in the rich mixtures which, if it were not for the occurrence of dissociation, must necessarily reduce the power output appreciably as soon as the fuel-air ratio becomes richer than the correct one.

It has been shown in arts. 12 and 24 that the calculated explosion temperature for the correct benzene-air mixture is reduced, when dissociation is allowed for, from 3,028° C. to 2,705° C.; and furthermore that the highest explosion temperature is not reached when the mixture is correct, but when it is some 20 per cent. richer than this. Over a considerable range of mixture strength, too, there is only a small variation in the explosion temperature, and therefore in the case of explosion pressures—where volume ratios have to be taken into account—the fall away of pressure with increase of mixture strength at the rich mixture end is still further reduced.

The calculated temperatures for the benzene-air mixtures were shown plotted in fig. 13 on p. 76, with and without allowance made for dissociation, and explosion temperatures calculated for coal gas-air mixtures would show the same characteristics. The shape of the curve of temperatures, when dissociation has been allowed for, corresponds closely with that of the curve of power variation against mixture strength which is found in practice, and which is illustrated by the curves *AB* and *A′B′* of fig. 33. Power with the rich mixtures will be influenced, apart from the maximum temperature reached, by the volume ratio on combustion and by changes of volumetric efficiency; but neither of these factors can at all nearly account for the observed variation of power with mixture strength. In a gas engine, as already observed, there should be no change of volumetric efficiency, and there was none in the series of experiments illustrated in fig. 33. For an explanation of the increase of power obtained on rich mixtures in a gas engine, therefore, we are driven to changes in the maximum temperature brought about by the conditions of chemical equilibrium. The similarity of form of the curves of fig. 33 when compared with that of the curve of calculated explosion temperatures in fig. 13, allowing for dissociation—or still better with a curve showing temperature-rise multiplied by volume ratio—affords evidence that the assumptions underlying the calculations are substantially correct. As was explained in arts. 24 and 27, the absolute numerical values of the calculated temperatures must for the present remain somewhat uncertain, through lack of certainty as to the true volumetric heat figures at high temperatures, and some doubt about the accuracy of the equilibrium constants; but this uncertainty does not at all affect the foregoing argument, which relies only upon the *relative* accuracy as between temperatures at different mixture strengths. As soon as the data upon volumetric heats and equilibrium constants have been consolidated we shall be in a position to apply the type of calculation given in art. 24 with confidence in the absolute correctness of the temperatures found.

ART. 37. The influence of compression ratio on combustion

Compression ratio is an important factor in determining the thermal efficiency of an engine, and in general it is advantageous to work with the highest ratio consistent with smooth running and absence of detonation. A full consideration of thermal efficiency and its limitations will be deferred to the next chapter, and for the present we shall deal only with the effect of a change of compression ratio upon combustion in the cylinder.

It was stated in art. 11 that the average value of n in the equation PV^n = constant, during the compression of a fuel-air mixture, might be taken as 1·37 for a gas engine and 1·33 for a petrol engine. In treating of the effects of a change of compression ratio, we shall assume a petrol-air mixture, for it is primarily with this type of engine that alterations of compression ratio are of such vital importance.

Taking $n = 1{\cdot}33$, therefore, we arrive at the figures in the second column of table 19 for the compression pressures at various ratios, on the assumption that at the beginning of compression the pressure in the cylinder is atmospheric. It must, of course, be remembered, as was explained in art. 11, that the pressure in any particular engine will depend upon the speed, valve-setting, type of carburetter and of induction system. The value of n taken will give a good average figure for the actual compression pressure in high-speed petrol engines, but not necessarily an accurate one for any particular engine. In discussing principles, however, what is required is consistency in the assumptions made, and *relative* numerical accuracy between the different sets of conditions, and this the acceptance of an average value of n will provide satisfactorily.

TABLE 19

Compression pressures and temperatures, assuming $n = 1{\cdot}33$ *and the temperature before compression* 100° *C. at all ratios.*

Compression ratio	*Maximum compression pressure*	*Maximum compression temperature*
	lb. per sq. in abs.	° C.
4	93	316
5	125	362
6	159	401
7	195	436

If the contents of the cylinder at the beginning of compression were entirely the fuel-air mixture, the temperatures at the end of compression could be similarly calculated, and we should arrive at the figures in column 3 of the table, on the assumption previously made of a temperature before compression of 100° C. There is, however, the exhaust gas left in the compression space to be allowed for, and the fact that the amount of this diminishes as the compression ratio is raised has a most important influence upon combustion in the cylinder; a greater influence, in fact, than the changes of pressure and temperature which occur through the change of ratio by itself. The effect of the reduction in the proportion of exhaust gas at the higher ratios is two-fold, and the two effects are in opposite directions. The reduction of exhaust gas present has a direct effect in the way of increasing the rate of combustion of a chemically correct mixture, for it means less dilution of the mixture after its entry into the cylinder. On the other hand, the reduced proportion of residual gas means less heating of the incoming mixture and a lower temperature before compression. On account of the changing proportions of residual gas, the range of the compression temperatures between the ratios of 4:1 and 7:1 becomes considerably less than is suggested by column 3 of table 19.

The dilution of the fuel-air mixture at any ratio, upon entry into the cylinder, may be arrived at as follows:

Let P_0 and T_0 refer to the fuel-air mixture before entry, and P and T refer to the final, and diluted, mixture in the cylinder at the end of the suction stroke.

Let S = the volume swept by the piston,
C = the clearance volume,
η_v = the volumetric efficiency referred to fuel-air mixture at P_0 and T_0,
R = the compression ratio $= \dfrac{S+C}{C}$.

The volume of fuel-air mixture which enters the cylinder per stroke, measured at P_0 and T_0 is

$$\eta_v S.$$

The volume occupied by this mixture when the inlet valve has just closed, at the end of the suction stroke, is

$$\eta_v S \frac{T}{T_0} \frac{P_0}{P}.$$

The total volume of gas at P and T is

$$S+C.$$

Therefore

$$\frac{\text{total gas in the cylinder}}{\text{mixture entering cylinder}} = \frac{C+S}{\eta_v S} \frac{PT_0}{P_0 T} = \frac{R}{R-1} \frac{1}{\eta_v} \frac{PT_0}{P_0 T}.$$

If the air in the entering mixture is a fraction m of the total volume, then the 'dilution factor' due to the residual exhaust gas will be

$$\frac{1}{m} \frac{R}{R-1} \frac{1}{\eta_v} \frac{PT_0}{P_0 T}$$

in terms of volume, or of mass, if we neglect the difference between the specific gravity of the residual gas and of air.

For a petrol engine working on the correct fuel-air mixture the value of m would be about 0·98. P/P_0 will as a rule be nearly unity, even at high speeds, and for simplicity will be assumed to be so. T_0/T and η_v will depend (*a*) upon the heat picked up by the ingoing mixture from valves, piston, and cylinder walls, and (*b*) upon the heating by admixture with the residual exhaust gas.

It is legitimate, in estimating the relative importance of (*a*) and (*b*), to imagine that the second stage, of heating by admixture with residual gas, does not occur until the end of the stroke, after the inlet valve has closed. We may do this because it can be shown that, apart from any difference in the volumetric heats of the two portions of the cylinder contents, admixture at constant volume after the valve had closed would cause no change of pressure in the cylinder.

If we picture the fresh charge as remaining separate from the residual gas until after the valve has closed, and meanwhile acquiring an absolute temperature T', then the volumetric efficiency, η_v, will be given by

$$\eta_v = \frac{T_0}{T'}\frac{P}{P_0}.$$

The temperature of the ingoing air would be initially about 15° C., but this will be subject to cooling by evaporation of the fuel on the one hand, and heating from the hot valves, piston, and cylinder walls, as well as in the induction system, on the other. Working backwards from measurements of volumetric efficiency we arrive at the figure of 60° C. as a probable temperature of a correct petrol-air mixture after entering the cylinder, and before mixing with residual exhaust gas. In other words, when $T_0 = 288$, an average figure for T' would be 333. P/P_0 will in practice vary between unity and 0·96, and will be taken as unity.

Turning now to the heating by admixture with residual gas, the temperature of the latter may be taken as 850° C. with sufficient accuracy. If the swept volume of the piston be now taken as 1 cub. ft., then the volume of the clearance space for ratios between 4:1 and 7:1 will be as in column 2 of table 20.

The fresh charge consists of a mixture of fuel and air of volume 1 cub. ft. and absolute temperature T', assumed to be 60° C. The specific gravities of the residual gas and fresh charge will be substantially the same, and therefore the volumes of each at N.T.P. given in columns 3 and 4 of table 20 may also represent their masses. The average value of K_v for the exhaust gas between 100° and 850° C. is about 25·0 ft.-lb. per S.C.F., as compared with about 21·2 for a benzene-air mixture between 60° and 100° C. After mixing, therefore, we obtain, as the temperatures of the mixture before compression, the figures in column 5 of table 20. After multiplying the corresponding absolute temperatures by the adiabatic compression factor, with $n = 1{\cdot}33$, we finally obtain the figures in column 6 as the temperatures at the end of compression.

TABLE 20

Temperatures at the end of compression at different ratios, after allowing for the changing proportion of residual exhaust gas.

Compression ratio	*Volume residual exhaust at* 850°	*Volume ditto at N.T.P.*	*Volume fresh charge N.T.P.*	*Temperature of mixture before compression*	*Maximum temperatures of compression*
	cubic feet			°C.	°C.
4	0·33	0·08	0·82	141	381
5	0·25	0·061	0·82	124	402
6	0·20	0·049	0·82	112	422
7	0·167	0·04	0·82	103	442

By comparing these figures with those previously given in column 3 of table 19, it will be seen that allowance for the diminishing proportion of hot exhaust gas at the higher ratios has narrowed the difference in compression temperatures between 4:1 and 7:1 from 120° C. to 61° C.[†] In this calculation our use of a single temperature of 850° for the residual gas at all compressions will tend to underestimate its effect, for, if anything, the residual gas at the high ratios, besides being less in quantity, will also be cooler. The correction involved can only be very small, for if the temperature to be taken is lower than 850° C., then the quantity of residual gas measured at N.T.P. in column 3 of the table will be greater, and vice versa, but the point serves to show that the figure of 61° C. arrived at for the range of the compression temperatures between 4:1 and 7:1 is probably an outside one. The pressure before, and therefore at the end of, compression is unaffected by the admixture with residual exhaust gas, and apart from secondary effects due to differences of heat loss, and gas leakage, we may accept the figures of table 19 as sufficiently correct for purposes of comparison. The actual values observable in any

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[†] Comparison of the compression temperature at 7:1 ratio in table 20 with that at the same ratio in table 19 must not be taken as indicating a negligible effect of exhaust gas upon the temperature at this ratio. The figures show that the common temperature before compression of 100° C., assumed for all ratios in table 19, is consistent only at 7:1 with a temperature of 60° C. for the ingoing mixture before mixing with the residual exhaust gas.

particular engine will, as with the pressures, depend upon the valve setting, the speed, and the induction system.

With the rise of pressure and temperature at the end of compression we should expect some increase in the rate of burning at the higher ratios, and there is ample evidence that this occurs. The reason, however, is rather a direct effect of the reduced proportion of exhaust gas left behind to dilute the incoming charge than the rise of pressure and temperature. Apart from simple dilution of the fuel-air mixture, the high specific heats of the carbon dioxide and water vapour in the exhaust gas have a retarding effect on combustion, through a lowering of the maximum flame temperature, as has clearly been shown by experiments to be described in the next chapter. In these experiments the effect upon detonation of artificially increasing the normal amount of diluent, by adding cooled exhaust gas to the charge was found to be very marked.

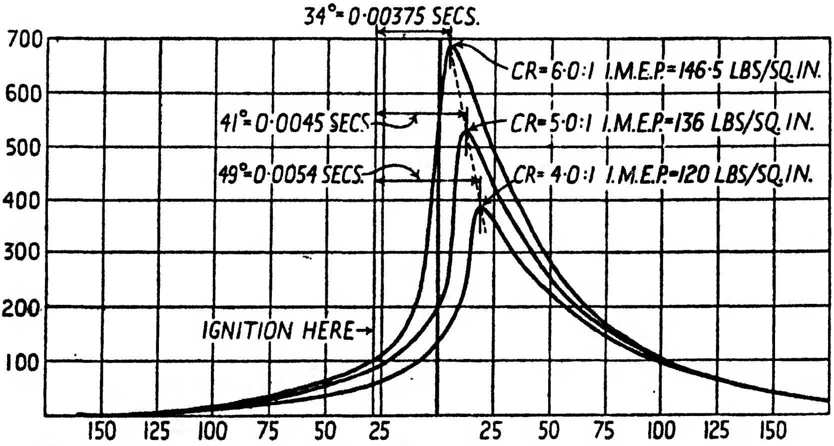

FIG. 34. Indicator diagrams at compression ratios 4, 5, and 6 to 1, with constant ignition advance 30° before dead centre. Speed 1,500 r.p.m. Fuel, petrol. Constant mixture strength.

By way of illustrating the effect on combustion of a rise in the compression ratio, three indicator diagrams drawn to a base of crank angle have been plotted in fig. 34 so as to show how the time taken to reach maximum pressure differs for ratios of 4:1, 5:1, and 6:1. In each test the ignition was timed 30° before the dead centre, and the time occupied to reach maximum pressure is reduced from 0·0054 second at 4:1 to 0·00375 second at 6·1. As an interesting comparison, a similar set of four diagrams taken with coal gas at ratios from 4:1 to 7:1 are reproduced in fig. 35. The engine used was in each set of trials the Ricardo E. 35, but the speed in the coal-gas experiments was 1,000 r.p.m. as against 1,500 r.p.m. in the petrol diagrams. Ignition was again 30° before the dead centre. It will be observed that the time to reach the maximum pressure at 6:1 compression ratio was, in the petrol engine, just half the time in the gas engine. Part of this higher rate of combustion in the petrol-air mixture was due to the higher engine speed and greater gas turbulence in the cylinder; but it will be seen also that the speeding up of the rate of combustion in each series by an increase of compression ratio is about twice as great in the petrol as in the gas engine.

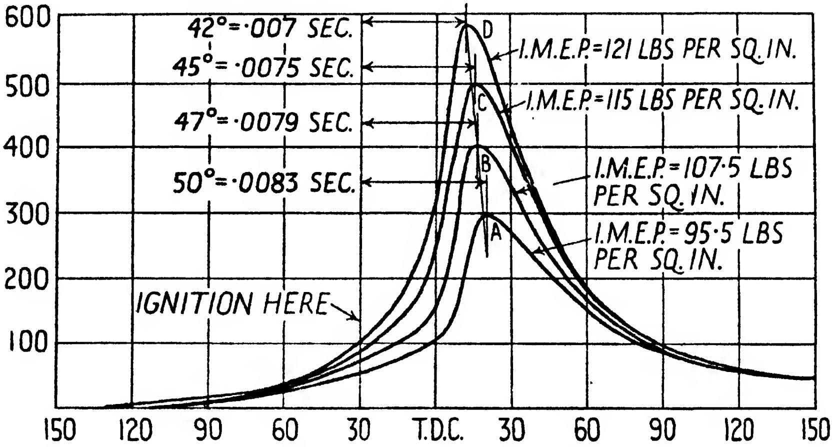

Fig. 35. Indicator diagrams at compression ratios 4, 5, 6, and 7 to 1. Ignition advance 30°. Fuel, coal gas. Mixture strength constant, 5 per cent. weak. Speed 1,000 r.p.m.

A. Compression ratio 4·0:1 *B*. Compression ratio 5·0:1
C. " " 6·0:1 *D*. " " 7·0:1

The maximum and mean effective pressures are considerably lower in the coal-gas diagrams of fig. 35 than in the petrol ones, for reasons given in art. 36 and because the gas-air mixture was 5 per cent. weak. The diagrams for compression ratio 5:1 in each figure correspond to the tests on gas and petrol in fig. 33, the petrol engine at the correct mixture.

Reference to art. 35 will show that in all these diagrams a certain amount of active combustion must have gone on before the dead centre. In the petrol experiments the 'delay' period must have occupied about

15° (see fig. 27) and in the gas experiments about 10°, so that some rise of pressure due to combustion had occurred for about the last 15° of the compression stroke in the petrol diagrams, and 20° in the gas diagrams. In table 21 some measurements from the diagrams are given for the purpose of calculating the ratio of pressure rise in each test, and if the figures for pressures at the dead centre in columns 2 and 3 are compared with those given in table 19 for compression pressures, we obtain some measure of the amount of combustion which had occurred.

TABLE 21

Showing the different rates of rise of pressure after the dead centre, with coal gas and with petrol, at various compression ratios. Figures taken from the diagrams of figs. 34 *and* 35.

Fuel-air mixture with coal gas	5 per cent. weak.
Fuel-air mixture with petrol	Correct.
Speed with coal gas	1,000 r.p.m.
Speed with petrol	1,500 r.p.m.

Compression ratio	*Pressure at the dead centre*		*Maximum pressure*		*Time to maximum pressure*		*Rate of pressure rise*	
	Gas	*Petrol*	*Gas*	*Petrol*	*Gas*	*Petrol*	*Gas*	*Petrol*
					Degrees		lb. per sq. in. per sec.	
4	119	155	315	395	21	19	57×10^3	114×10^3
5	175	215	415	540	19	11	76	260
6	270	515	515	705	16	4	92	430
7	341	..	595	..	13	..	117	..

A comparison of the average rates of pressure-rise between the dead centre and the peak pressure for each diagram has been worked out in columns 8 and 9 of table 21. A comparison of the rates, as between gas and petrol, is of no special interest, for the rates will in any case depend so much upon the mixture strength and upon turbulence. From both aspects the conditions favoured the petrol engine, for its speed was 1,500 as against 1,000 r.p.m. and the mixture was correct, whereas it was 5 per cent. weak in the gas-engine trials.

We are on safer ground in comparing the effect of a rise in compression ratio upon combustion in each series of diagrams than in comparing one series with the other, for in each set both turbulence and mixture strength were maintained constant. There is an increase of nearly fourfold in the rate of pressure-rise with petrol, between 4 : 1 and 6 : 1, as compared with only twofold with coal gas.

So much for the rates of pressure-rise immediately *after* the dead centre. Returning now for a moment to the amounts of burning which had taken place before the dead centre at the various ratios, there are given in table 22 the amounts of excess pressure due to combustion over and above the compression pressures with no ignition, and in the last two columns each of these amounts has been expressed as a percentage of the total rise to the peak pressure. These last figures express in a rough way the percentage of the total available fuel heat which had been generated by the end of the compression stroke.

TABLE 22

Showing the amount of pressure-rise at the dead centre due to combustion at various compression ratios.

Ignition advance 30° throughout.

Compression ratio	*Compression pressure*		*Pressure at dead centre excess over compression pressure*		*Rise of pressure before the dead centre as a fraction of total rise*	
	Coal gas $n = 1{\cdot}37$	*Petrol* $n = 1{\cdot}33$	*Coal gas*	*Petrol*	*Coal gas*	*Petrol*
					per cent.	per cent.
4	98·5	93	20	62	9·2	20·5
5	133	125	42	90	15	22
6	171	159	99	356	29	65
7	212	195	129	..	34	..

It must be remembered that all the diagrams were obtained with a fixed ignition advance of 30° in order to demonstrate the effect of changing only the compression ratio, and one deduction to be drawn from table 22, and the diagrams themselves, is that this 30° was not the most suitable ignition advance for the different conditions. It was certainly too little for the lowest compression ratio, both in gas and petrol, giving, as it does, a peak pressure 20° and 19° after the dead centre respectively, and no doubt a power and efficiency below the best obtainable.

The petrol diagram for 5:1 and the gas for 7:1 each have their peak pressures just in the standard position, so that the ignition advance for these tests was probably correct for maximum power and efficiency. On the gas diagram for 6:1 the peak is only 3° later than the standard position and the difference would probably be imperceptible as affecting the power of the engine. For petrol at 6:1 the advance was definitely too great. The peak pressure occurred 4° after the dead centre, and even if the amount of negative work and the augmented heat loss due to early combustion had not already begun to affect adversely the power and thermal efficiency, it is certain that the running of the engine must have been unnecessarily rough owing to the extremely rapid pressure-rise; and that a rather later ignition would have given a smoother engine with at least as good power and efficiency.

The curve of fig. 36 shows the results of a direct test made with the E. 35 engine to find the optimum ignition timing at a series of compression ratios when using a non-detonating liquid fuel, at 1,500 r.p.m., and with the fuel-air ratio giving maximum power. The optimum advance varies from about 45° at 4:1 down to 20° at 7·5:1.

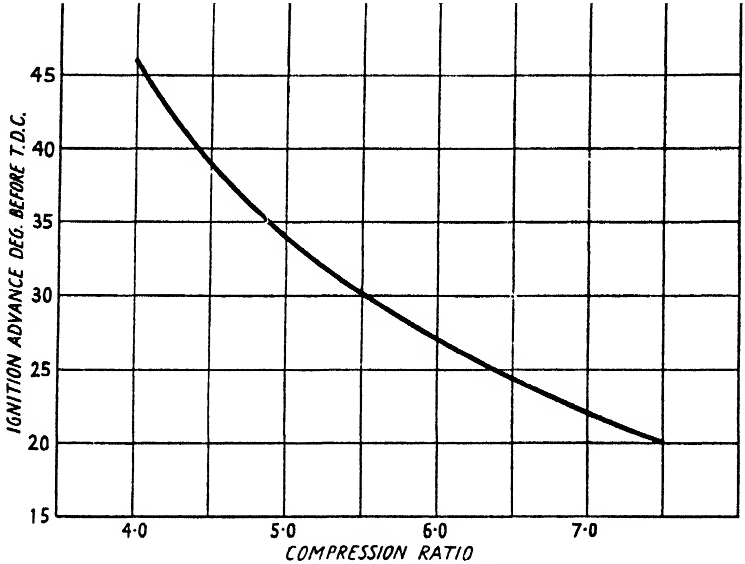

FIG. 36. Variation of optimum ignition timing with compression ratio.

It may be remarked, in conclusion, that a rise of pressure before the dead centre due to combustion does not necessarily mean serious loss of heat to the cylinder walls during compression. Although some small loss by radiation begins at once, as soon as combustion has started, it will not become serious until the highest temperatures have been reached, and as for the conduction loss, this will not begin until the flame front has reached the cool cylinder walls, which may not be until after a considerable rise of pressure has taken place. As soon as the flame front does reach the walls, the loss will be more serious at the higher ratios, for owing to the reduced size of the combustion chamber its ratio of surface to volume is quite appreciably increased. From 10° before to 10° after the dead centre, during which period most of the combustion takes place, the volume in the cylinder is sensibly constant, and for the change in compression ratio from 4:1 to 7:1 the ratio of surface to volume of this combustion space increases from 1·8 to 2·5.

ART. 38. The influence of temperature on combustion

Under the conditions in which the indicator diagrams of figs. 34 and 35 were obtained, the acceleration of combustion at the higher compression ratios was brought about by a combination of different factors: a lower proportion of residual exhaust products, and a higher temperature and pressure when ignition takes place. The importance of the amount of residual exhaust gas has already been emphasized, and with a view to giving some idea of the effect of temperature by itself, an experiment of Ricardo may be quoted, which is illustrated in fig. 37. This shows two indicator diagrams taken at the same speed and compression ratio, and with the same ignition advance of 30°, but with a large difference of temperature between the two, both of cylinder jacket water and of ingoing charge.

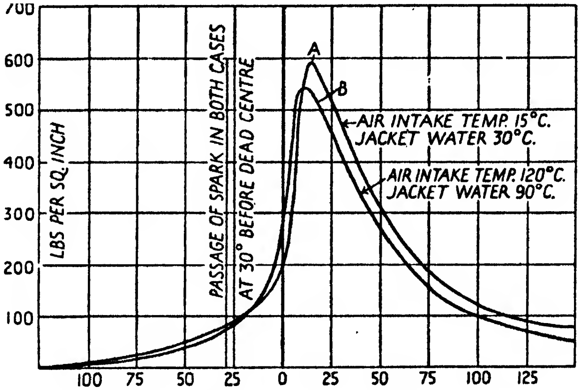

FIG. 37. Two indicator diagrams taken at the same speed, 1,500 r.p.m.; the same compression ratio, 5:1; the same ignition advance, 30°, which show the effect on combustion of a rise of temperature. Fuel, petrol.

The raising of the temperature of the ingoing charge from 15° C. to 120° C., with all other conditions identical, reduced the time required to reach the maximum pressure from 44° to 41° of crank angle, or from 0·0048 to 0·0045 sec. It is impossible to say just what rise of temperature at the top of compression was produced by a rise of 105° C. in the temperature of the ingoing gas, for in arriving at 122° C. as the approximate temperature at 5:1 compression for normal conditions (table 20) a considerable allowance was made for the heat derived from hot cylinder walls, piston, and valves, before the inlet valve closed, and all these allowances would have to be very different when the ingoing gas was heated before entry to 120° C. The temperature before mixing with the exhaust products was, under normal conditions, taken as 60° C., and a good approximation to the new conditions would probably be to assume an increase of temperature of 60° C. in the mixture before compression, and 100° C. at the end of it. This involves the rough assumption that any heat which the ingoing gas at 120° C. picks up from hot valves and piston is counterbalanced by a loss to the cylinder walls, which were about 30° cooler than the gas.

The temperature at the end of compression is in any case not of much importance, but rather the temperature 30° of crank angle before the dead centre, when the spark passes, for in each diagram by the time the dead centre was reached a good deal of heating through combustion had taken place, and the pressure was well above the probable compression pressure of 125 lb. per sq. inch (table 19). Of greater interest than the reduction of the time for the complete pressure-rise from 0·0048 to 0·0045 sec. is the much more marked acceleration of combustion during the early stages, before the dead centre. During the first 30° of crank angle after the passage of the spark the rise of pressure was increased, at the high temperature, from 115 to 204 lb. per sq. inch. In other words, the rate of combustion was nearly doubled by an increase in the gas temperature, at the moment when the spark passed, amounting to about 90° C. if the increase before compression was 60° C.

ART. 39. The process of compression-ignition

When fuel is injected into the combustion space of a cylinder, full of air at a high pressure and temperature, the problem of bringing about its complete and rapid combustion is wholly different from that in a petrol engine, in which a flame spreads from the spark throughout a more or less homogeneous fuel-air mixture.

If the whole of the injected fuel is to be burnt, every particle of it must be brought, in a finely divided condition, into intimate association with its proper quantity of oxygen. This pulverizing, mixing, evaporation, and burning of the fuel has all to be achieved in a time which, even in the comparatively slow-running type of heavy stationary engine, is very short. It is about 1/60 sec. for an engine running at 300 r.p.m. and 1/400 sec. at 2,000 r.p.m. In the heavy stationary type of engine the pulverization and distribution of the fuel is accomplished by squirting it in with a high-pressure air blast. For the light high-speed type of engine, which of late years has been developed for road and air transport, the weight and complication of the necessary air compressors put them quite out of court, and a method of fuel injection has been developed which is independent of high-pressure air, the liquid fuel being squirted in as a fine jet or spray under direct pressure from a pump plunger. In these circumstances we are clearly dependent upon either turbulence of the air in the cylinder, or upon the penetration of the jet itself, for the proper mixing of the fuel and air. In a third method, which is really a combination of these two, the cylinder head incorporates a small 'pre-combustion' chamber into which the fuel is injected in a partially atomized condition, and from which a mixture of burning and unburnt fuel is projected violently into the remainder of the combustion space through a number of small holes.

If we do not rely upon penetration of the fuel jets, but upon air turbulence, then this must take the form of an organized 'air swirl' at an extremely high velocity which either persists throughout the compression stroke or is created towards the end of it, in such a way as to bring all the air into association with the fuel within about 30° of crank revolution, following the beginning of the fuel injection.

Four alternative forms of cylinder head are illustrated diagrammatically in fig. 38, in which (*a*) and (*b*) indicate the essentials of the 'directed spray' and 'pre-combustion chamber' types of cylinder, while at (*c*) are shown a section and a plan view of the type of cylinder employed by Ricardo for producing air swirl in his sleeve-valve engines. At (*d*) is shown another Ricardo design, for use in poppet-valve engines, in which the air swirl is created in a side chamber of the cylinder head by the forcing of the air into it through a narrow orifice. In this form of combustion head the air swirl in the side chamber is chiefly created just at the end of the compression stroke while the fuel is being injected across the chamber in the direction indicated in the figure.

In the sleeve-valve type of engine, on the other hand, the inlet air ports in the walls of cylinder and sleeve are so directed tangentially that the air as it enters is given a rapid circulatory motion round the cylinder axis before compression. At the end of compression this swirling motion is further accelerated when the air is all forced into a combustion space of about half the cylinder diameter. As the fuel is projected axially from a point near the circumference of this combustion space, the highly heated air sweeps past it at such a rate that during about 35° of crank revolution a complete circuit of the combustion chamber is made and all the air, by that time, has been carried past the burning jet. It has been found experimentally that a rate of swirl equal to about ten times the r.p.m. of the crank gives the most satisfactory engine performance, and that it is possible to burn about 75 per cent. of the oxygen of the air present, in an efficient manner.

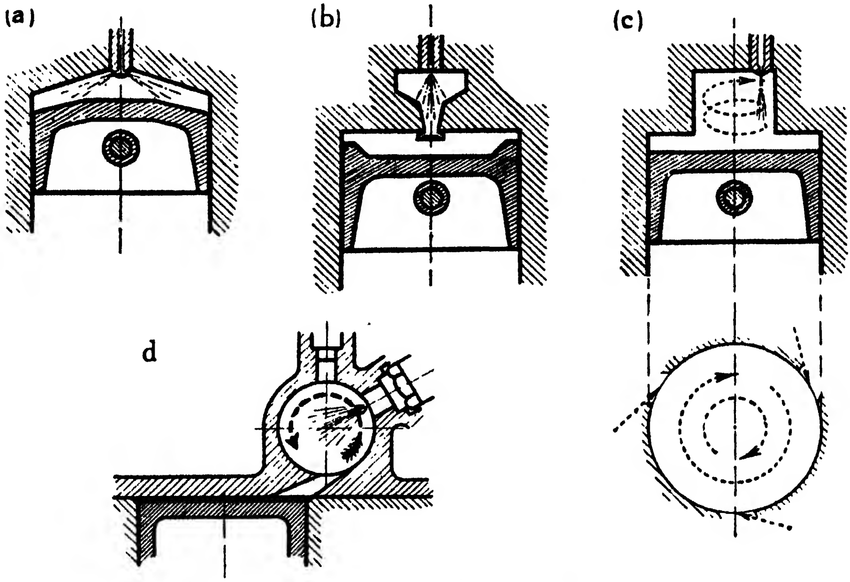

Fig. 38. Forms of combustion chamber, compression-ignition engine.

Fig. 39 shows an actual pressure-time record taken upon a single-cylinder engine employing the air-swirl principle. Fuel injection began 15° before the dead centre, and since the compression pressure, if there had been no combustion at all, would have been about 470 lb. per sq. inch for the compression ratio of 14:1, it is clear that for the first 10° or 12° after fuel had first entered the cylinder it produced no appreciable effect upon the pressure.

We find, therefore, in the compression-ignition process, a 'delay' period exactly analogous to the initial delay in a petrol engine, which was discussed in art. 34. There is, however, a difference during the later stages, for whereas in the petrol engine the combustion of the homogeneous mixture was substantially completed during a second stage, from the end of the delay period to the moment of maximum pressure, in the Diesel engine Ricardo[22] has argued that the combustion process should be regarded as taking place in *three* stages while the injection of the fuel proceeds, whether mixing of the fuel and air is brought about by an organized air swirl or by the directed-spray method.

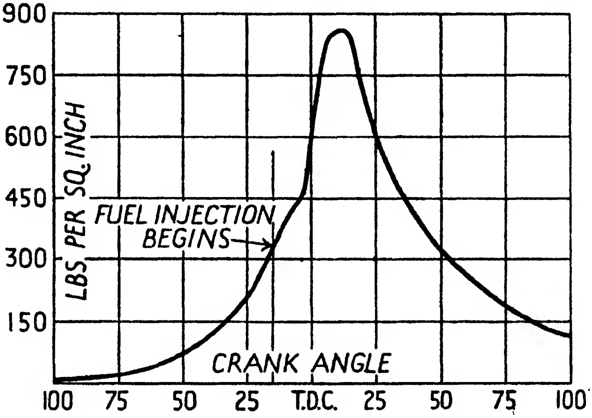

FIG. 39. Indicator diagram from compression-ignition engine employing the air-swirl principle. Speed 1,500 r.p.m. Compression ratio 14:1. Fuel injection commences 15° before dead centre. B.M.E.P. = 112·5 lb. per sq. in.; I.M.E.P. = 136 lb. per sq. in.

The three stages envisaged by Ricardo are as follows:

(1) The delay period, during which combustion is confined to small nuclei and the amount of heat produced is too small to affect the gas pressure.

(2) A period of very rapid combustion, during which the fuel injected during the delay period is burned in an uncontrolled manner.

(3) A period during which the remaining fuel burns as it enters the combustion chamber, of which the contents have by this time been raised to furnace temperatures. The rate of burning and of pressure-rise can be controlled to some extent during this period by the rate of fuel injection.

The periods are numbered 1, 2, and 3 in fig. 40, which shows an artificial indicator diagram in which the pressure-time curve has been given an angular form in order to separate out the three stages. By comparison with fig. 39 the three stages can be quite well distinguished upon the experimental diagram.

Whatever the exact sequence of events may be, it is quite certain that after injection of the fuel has begun there is a delay period of about 12° of crank angle at 1,500 r.p.m., that is 0·0013 second, during which there is no appreciable increase of pressure due to combustion, and that a period of very rapid pressure-rise follows during the next 7° or 8°. The close parallelism with the behaviour of a petrol-air mixture ignited by a spark is quantitative as well as qualitative, for reference to art. 34 will show that the time of delay in those experiments was 0·0015 second. If we cannot do away with this delay, we must continue to allow for it by beginning the injection sufficiently early to allow maximum pressure to be reached before expansion has seriously begun, just as we allow for the same thing in a petrol engine by an early ignition timing. This early injection, however, followed by a very rapid and uncontrolled pressure-rise, is highly unsatisfactory and is to some extent a confession of failure.

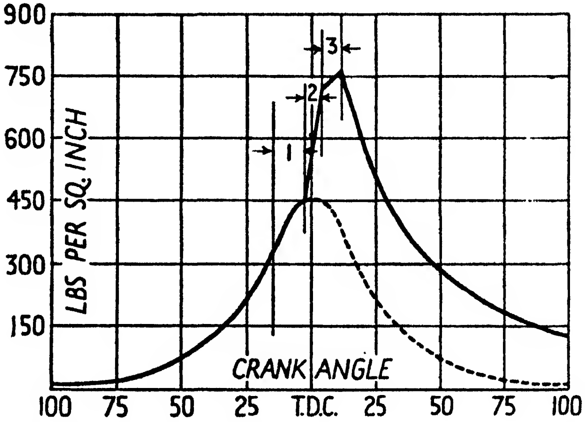

FIG. 40. Pressure-time diagram to illustrate the three phases of the combustion process in a compression-ignition engine.

A factor of great importance in Diesel engine operation is that of 'roughness' of running. It is more readily to be apprehended from experience of an engine on test than defined in words, and even then its external effects may be masked by heavy and rigid mountings for the engine. This will not save the engine, however, from the effects of rough running, which, apart from easily perceived vibration, may take the form of rapid wear of any gears through which the load is transmitted or of damage to connecting-rod and main bearings.

It has been proved experimentally that roughness is not dependent, as it might be expected to be, upon the maximum pressure in the cylinder, but that it is closely associated with the *rapidity* of the pressure-rise during period (2) of the combustion, and the same thing is true, also, in the petrol engine, as we shall see in art. 61. The rate of burning of the fuel already in the cylinder during this period has been spoken of above as being uncontrolled, and for an engine employing the directed-spray method of fuel injection, it is so. With an organized air swirl, however, in which air is brought to the fuel rather than fuel to the air, the rate can be almost perfectly controlled by an alteration of the swirl conditions.

With an engine in which the rate of swirl could be varied between wide limits while running, it has been found that the rate of pressure-rise during period (2) could be accurately controlled by the rate of air swirl, and, with it, the roughness of the engine. Starting from a slow swirl rate of about four times the crank revolution speed, which gives a smooth-running, inefficient, engine with a rate of pressure-rise of 10 lb. per sq. inch per degree of crank angle, maximum and mean effective pressures of 600 and 60 lb. per sq. inch, and a fuel consumption 0·6 lb. per B.H.P. hour, the engine can be transformed by steady degrees, while maintaining the same quantity of injected fuel, into one giving an M.E.P. of over 100 lb. per sq. inch with a fuel consumption of 0·35 lb. per B.H.P. hour. This improvement is associated, step by step, with acceleration of the rate of pressure-rise up to 70 lb. per sq. inch per degree, at which point the engine is intolerably rough. In fig. 41 are shown four diagrams, all taken with the same quantity of injected fuel, and the same injection timing, but with four different rates of swirl in relation to crank revolution speed. In diagram (4) the swirl was excessive, for not only was the engine too rough, but there was a small drop of M.E.P. below that of No. (3), which corresponds to maximum power and efficiency.

The maximum pressures, during the speeding up of the swirl, increased from 600 to 900 lb. per sq. inch, but independent experiments have shown that an increase of maximum pressure by itself does not necessarily produce roughness, and that if the rate of pressure-rise is very great an engine will be rough whatever the maximum pressure. Fig. 42 is an example of a diagram from an engine employing the directed-spray principle. The extremely rapid pressure-rise during period (2) will be noticed. It amounts on this diagram to 85 lb. per sq. inch per degree, and is, of course, quite uncontrollable so far as air movement is concerned. Although the average speeds of air movement in this poppet-valve cylinder were probably much less than in the sleeve-valve cylinder, the very fine atomization of the jet produced by high-pressure injection of the directed spray has more than made up for the lower air speeds so far as rate of burning is concerned.

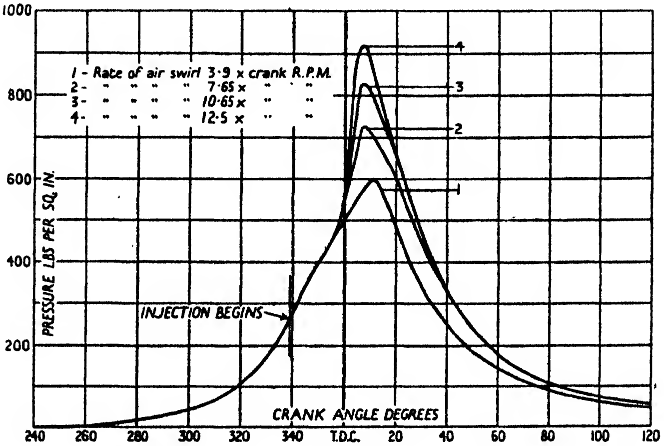

FIG. 41. Diagrams with varying swirl rates, showing the acceleration of combustion by an increase of swirl rate. Engine speed 1,300 r.p.m.

The ideal of the Diesel engine designer should be to eliminate the delay, and merge periods (1) and (2) into one period of steady pressure-rise, during which the fuel burns as it enters the cylinder. This would mean the initiation of vigorous combustion immediately upon entry. Now it must be remembered that although the peak compression temperature of an engine of compression ratio 15:1 is about 600° C., assuming an initial temperature of 100° C. and $n = 1{\cdot}32$ (see art. 11), on the other hand the temperature when the fuel is first injected, 15° early, is only about 530° C. For a 12:1 compression ratio the figures would be 555° C. and 500° C. Actual temperatures, moreover, are likely to be lower rather than higher than those given here, since the initial temperature at these high compression ratios would probably be rather below 100° C., and the figures given in art. 11 indicate that the effective value of the index n is likely to be below 1·32 rather than above it. Tests with the compression machine described earlier showed that on several occasions no ignition occurred when the temperature reached by compression was 450° C., even when the fuel was already evaporated before compression, so that when we recollect that some local cooling must occur by evaporation when the fuel is first injected, it is clear that there is very little temperature margin to provide certain and rapid ignition, and that any lowering of the self-ignition temperature of the fuel will be an advantage provided we do not thereby nullify its value as compared with petrol from the point of view of safety from fire danger. With the intention of lowering their self-ignition temperatures, certain chemicals have been added as 'dopes' to heavy oils with some success,[50] just as dopes are introduced into petrol to prevent detonation.

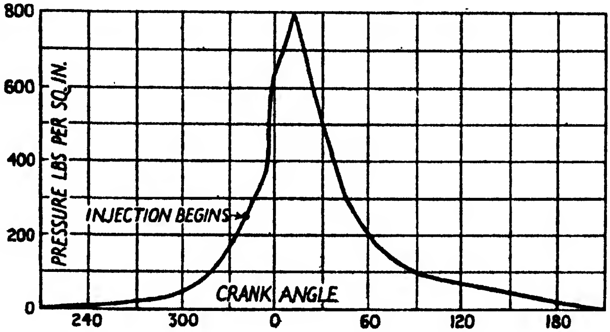

FIG. 42. Diagram from an engine employing the 'directed-spray' principle. Speed 1,000 r.p.m. B.M.E.P. 121 lb. per sq. in. Fuel consumption 0·40 lb. per B.H.P. hour.

The alternative to lowering the ignition temperature of the fuel is to raise the compression temperature of the air. This must be done by some special cylinder head design which enables the air to pick up heat from the metal walls after, but not before, the inlet valve has closed at the end of the suction stroke; for then the gain of compression temperature will be achieved without any loss of volumetric efficiency. Ricardo's 'comet' type of cylinder head illustrated in fig. 38 (d) is well arranged to achieve this raising of the compression temperature of the air, for it is possible to design the entry passage to the auxiliary chamber so as to be thermally insulated from the rest of the cylinder head. The neck through which the air is forced is therefore always at a high temperature and the air is heated on its way into the 'comet' chamber. Some figures showing the beneficial effect of this design on combustion are given at the end of the next article.

If we cannot abolish the delay period, and its remarkable constancy under very varied conditions seems to point to its being something fundamental in the ignition process, then it is desirable to design the fuel-injector mechanism so that a minimum of fuel is injected during period (1), leaving a large proportion for the later stages when combustion is well started and the rate of burning can be controlled to some extent by the rate of injection. If time is a necessary factor in the development of the flame nuclei into full combustion, then it will pay to supply as little fuel as possible while the development is going on, for a microscopic amount of fuel, properly atomized, should suffice to form the centres of the nuclei. The difficulty lies in the fact that the conditions for good atomization at the start of injection are apt to be inconsistent with those of slow initial delivery.

Until we can succeed in reducing the delay period or slowing up the early rate of injection, it must follow that, with an increase of engine speed, more and more fuel will be injected while the delay lasts, for the pump movements will in general be proportional to crankshaft revolutions. Period (3) is a time of roughly constant, or slowly rising, pressure, and the maximum pressure, therefore, is very nearly reached by the end of period (2). It follows that the maximum pressure will be mainly influenced by the quantity of fuel in the cylinder at the end of period (2), and will increase the higher the crankshaft speed. That this is so under any given set of conditions, as regards rate of injection and temperature, has been proved by experiment.

A more detailed study of the delay period follows in the next article.

Art. 40. Diesel knock and the delay period

Recent research has thrown much light upon the problem of promoting satisfactory combustion in the compression-ignition engine at high speeds, in regard both to features of engine design and to fuel characteristics.

It was explained in the last article that with controlled combustion during stage (2) of fig. 40 it is possible to increase progressively the rate of burning and, step by step with this, the roughness of an engine. The point was illustrated in fig. 41. If the uncontrolled type of combustion with its extreme rate of pressure-rise illustrated in fig. 42 occurs in an engine after a long delay period, the result is an intolerable roughness of running accompanied by a loud 'knock' somewhat, but not quite, like the pinking or detonation of a petrol engine.

This 'Diesel knock' as it has been called is undoubtedly due to extremely rapid burning, often of a large part of the fuel charge, if the delay period before it has been long. There is general agreement that Diesel knock is nearly always associated with a long delay and that, failing its elimination, the shortening of this period is of primary importance for the development of a smooth running and reliable engine.

Whether or not detonation and Diesel knock are identical in their physical nature, they are very similar in their effects, and equally bad for the engine. Diesel knock is accompanied by an extremely rapid pressure-rise, and by heavy vibration, with overheating of the piston and cylinder head; and this cannot fail to be followed by a falling off in power, damage to bearings, and possible piston seizure.

As was explained in the last article, the ideal to be aimed at is the instantaneous combustion of the fuel from the moment when it first enters the cylinder. And the possibility of an approach to this ideal must now be discussed from the two points of view, of the fuel and of the engine.

The importance of the chemical composition of an oil, and the bearing of this upon its self-ignition temperature and chemical stability, have already been mentioned in art. 21. There is general agreement that a low S.I.T. must always be the ideal of a good Diesel fuel, and it will hardly be questioned that a fuel of low viscosity and high volatility is likely to be unsatisfactory on account of fuel-pump leakage and wear, even if these physical characteristics have no direct effect in the promotion of knock.

It has been suggested by Dicksee[48] that a high volatility is directly responsible for Diesel knock by reason of more rapid evaporation during the delay period, and simultaneous ignition thereafter of a larger mass of fuel-air mixture than with a less volatile oil. Whether or not volatility may affect matters in this way as between two oils which both give a long delay, there seems to be ample evidence that volatility, as such, has no effect upon the length of the delay, upon which the knock primarily depends. The essential condition for smooth running is that the delay should be short, and the comparative lengths of delay, when different oils are tested in the same engine, depends almost entirely upon the chemical nature of the molecule.

It is true that the seriousness of a long delay may be very different in different engines, although this does not affect the general conclusion that the shorter it is, the better. The development of knock is the combined result of a long delay, followed by a period of rapid burning which is much influenced by gas turbulence in the cylinder. In a high-speed engine a high degree of turbulence is inevitable, and also essential, for upon it we rely for the completion of the main pressure-rise during period (2) (fig. 40) in the short time available. This leads inevitably to roughness and knock, after a long delay; but the same delay in a slow-speed engine may be far less objectionable, for there will be time enough to complete the combustion even with a low degree of turbulence, and in these circumstances the rate of pressure-rise during period (2) may be quite reasonable, though uncontrolled.

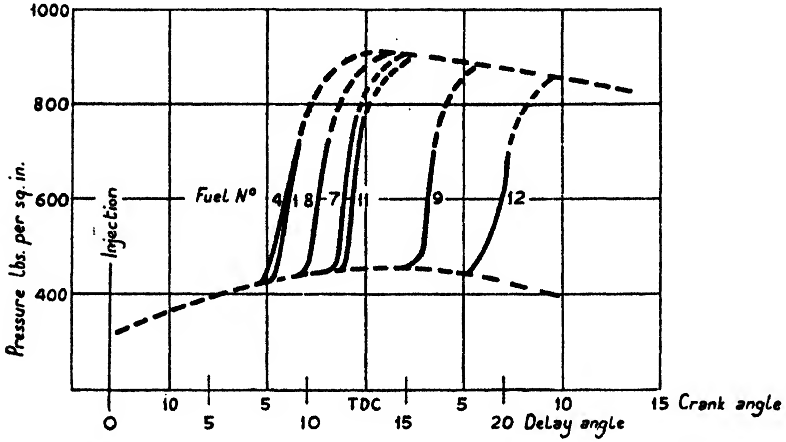

FIG. 43. Variation of delay angle with fuels of different chemical composition.

To illustrate the wide difference of behaviour exhibited in the same engine by oils of different chemical composition, when tested under comparable conditions, the results in fig. 43 have been selected from among those given by Le Mesurier and Stansfield.[49] The figure shows the top part only of a series of indicator diagrams taken upon the same engine, all at 1,000 r.p.m. and with the fuel injection timed to commence at 13° before the dead centre. Water-jacket and air temperatures were also maintained constant, and so was the B.M.E.P.

It will be seen that the delay angle varied from 8° to 19°, and that for fuels (4) to (9) the increasing delay was accompanied by an increasing rate of pressure-rise. With fuel (12) combustion was so long deferred that cooling of the gases by expansion after the dead-centre had set in before the start of rapid combustion, and the result was a less rapid rise of pressure; but also, of course, an inefficient engine, for the available expansion ratio would be reduced by the lateness of the point of maximum pressure.

Le Mesurier and Stansfield have given a further, very interesting, curve showing how the noise and vibration of an engine, as measured by a special instrument, may be correlated with the length of the delay period. This correlation is shown by the curve in fig. 44. Readings of noise and vibration go hand in hand, and are seen to increase rapidly with the delay until this reaches about 13°. These experiments were made upon the same engine and at the same speed as those of fig. 43, where it can be seen that a delay of 13° corresponded very closely to the maximum rate of pressure-rise. With oil No. 12, and others giving even longer delays, there was a reduction of noise and vibration which was in keeping with the lower rate of pressure rise for these very long delays, noted in connexion with fig. 43. It has been observed that in the great majority of engines maximum vibration and rate of pressure-rise coincided with the condition when the delay angle was the same as the total time of fuel injection; in other words, when the whole of the fuel charge was in the cylinder at the instant the delay ended. Those fuels which gave short delay times and smooth running were the ones composed largely of members of the paraffin series, while the long delays were always associated with a large percentage of naphthenes and aromatics—the ring compounds of art. 16.

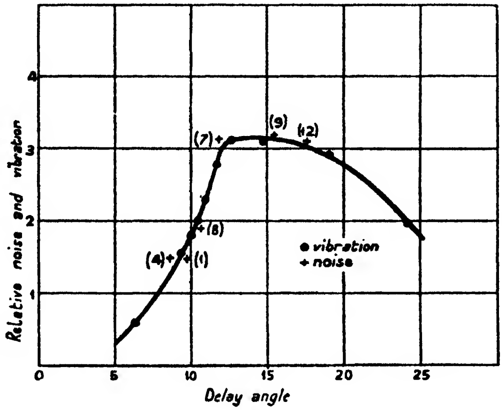

FIG. 44. Variation of noise and vibration with delay angle.

It has been argued that a high volatility should tend to reduce the delay period, because the more readily an oil vaporizes upon first entering the cylinder, the more readily will rapid combustion begin. That this is not so has been shown by direct experiment. Oils No. (1) and (4) of fig. 43, for example, the two best of the group, were of low and medium volatility respectively; while a much more volatile oil of different chemical composition and higher S.I.T. exhibited a delay of more than twice the length. High volatility is therefore no cure for a long delay, which is essentially an induction period dependent upon the chemical and not upon the physical properties of the oil. But if the oil is of a type which gives a long delay, then, according to the theory of Dicksee already mentioned, the more volatile it is, the worse will be the knock produced.

It has further been argued that a more viscous oil, giving a larger size of drop when sprayed, will tend to give a long delay because the large drops will take longer to become heated to the combustion point. This, again, appears to rest upon a fallacy. Experiments made upon the same oil with different spraying nozzles, one to give a fine and the other a coarse spray, showed no difference in the length of the delay period, although the fuel from the fine-spray nozzle burned much more rapidly during period (2). As a further proof of the unimportance of the drop size, an experiment may be quoted in which a heavy lubricating oil was compared with the oil (1) of fig. 43. The lubricating oil, which did not begin to distil below 360° C., formed drops in air of more than four times the size of those of fuel (1), but there was no difference at all in the delay observed with the two oils. The exhaust gas from the engine, however, was tinged yellow when the lubricating oil was used as the fuel, indicating that the large drops were taking so long to burn that combustion was incomplete even at the end of the expansion stroke. One may conclude, therefore, from this, combined with the short delay, that combustion proceeds from the surface of a large drop and that it does not wait for the bulk of the drop to be heated throughout to the combustion temperature.

In discussing the influence of engine design upon combustion, and upon the delay period, one is faced at the outset by the difficulty of dealing with the great variety of designs which exist. The difficulty, however, is not so great as it appears to be at first sight because, by comparison with the influence of compression temperature and pressure, the other possible effects arising from design features are unimportant.

Gas turbulence, for example, will vary widely in different forms of combustion chamber; but although it has a profound effect upon the later stages of combustion it is doubtful whether turbulence, as such, has any direct effect upon the delay. It may have a large indirect effect, however, through its influence on the compression temperature.

The effect of engine design may, therefore, be discussed in terms of the temperature and pressure, or rather density, towards the end of the compression stroke. These are the two factors which largely control the length of the delay with any given fuel, and features of design are valuable or the reverse according to their influence upon them.

The degree to which density and temperature can be increased by increasing the compression ratio is limited by the difficulty of designing a satisfactory shape of combustion space when the compression ratio is very high. There is also the danger of excessive maximum pressures. It is not found practicable to increase the compression ratio beyond about 14–15 to 1 in small, and 12–13 to 1 in large, cylinders.

Any device, on the other hand, which can increase the air temperature at the end of compression, without at the same time causing any reduction of the air charge per cycle, will afford a pure gain in a shortening of the delay period and consequent smoother running. The aim of such a method must be to add heat to the air during compression, after the inlet valve has closed; for any heating earlier than this must mean a loss of volumetric efficiency and of the density of the cylinder contents.

Before giving the results of experiments along these lines, it will be well to give some quantitative results on the shortening of the delay through an increase of air temperature and pressure.

In fig. 45 are shown the results of some experiments by Stansfield upon an engine with a simple open combustion chamber of the type shown at fig. 38 (*a*). The jacket and inlet air temperatures were varied from 10° C. to 100° C., so that in this engine the rise of air temperature was accompanied by a smaller weight of air per cycle. The fuel quantity, however, was adjusted so as to maintain a constant fuel-air ratio. It will be seen that the delay with the inferior fuel No. (12) was reduced to a much greater degree by the rise of air temperature than it was with the good paraffin fuel No. (1). This is in satisfactory agreement with the results of ignition-temperature experiments given in fig. 12, for there it was shown that at high temperatures there was much less difference between the behaviour of the good and the bad fuels.

In fig. 46 are given the results of a series of tests upon a different type of engine in which the air during compression is forced through a neck

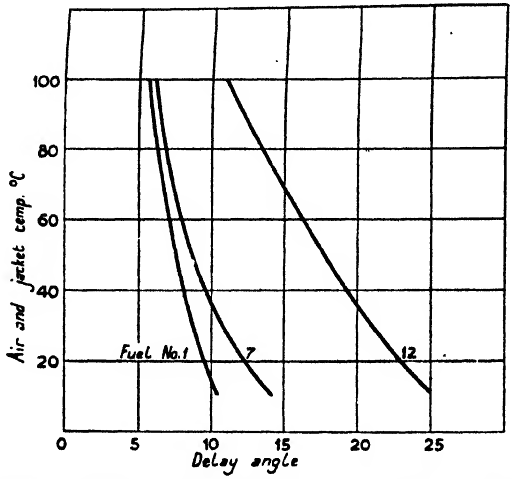

FIG. 45. Variation of delay angle with air temperature.

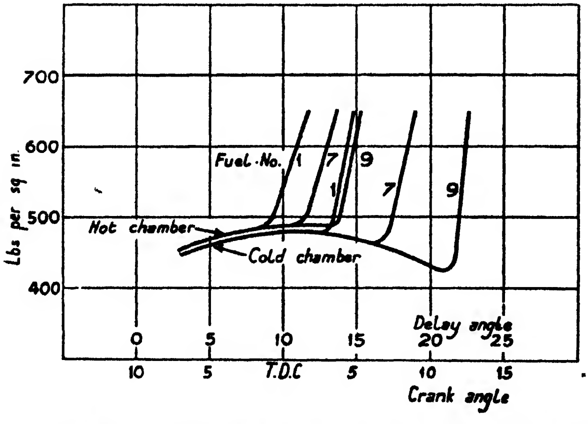

FIG. 46. Variation of delay angle between a hot and a cold auxiliary combustion chamber.

into an auxiliary combustion chamber, as illustrated at fig. 38 (*d*). If the neck and auxiliary chamber are maintained in an uncooled condition, it is possible for a large amount of heat to be derived by the air from the metal during compression, and to increase the compression temperature as much as 200° C. by this means. Fig. 46 shows the reduction of the delay with three different fuels when the compression temperature was raised in this way. It again appears that the inferior fuel No. (9) was improved to a much greater degree than No. (1). There is, however, no reason to think that by air heating the disparity of performance due to the chemical nature of the fuels could be eliminated.

To investigate the effect of pressure upon the delay Le Mesurier and Stansfield arranged an engine with the simple type of combustion chamber, fig. 38 (*a*), so that the inlet air could be throttled, and the compression pressure reduced, without alteration of the compression temperature. Fig. 47 gives the results for the same three fuels as before, Nos. (1), (7), and (12). The superiority of No. (1) will be noted in respect of the smaller increase of delay as the compression pressure was reduced: an increase from 9° to 13° between 450 and 300 lb. per sq. inch, as compared with an increase from 10·5° to 21° with fuel (7). Fuel (12) would not fire regularly at all when the compression pressure was below 370 lb. per sq. inch, but did not show so rapid an increase of delay with falling pressure as fuel (7).

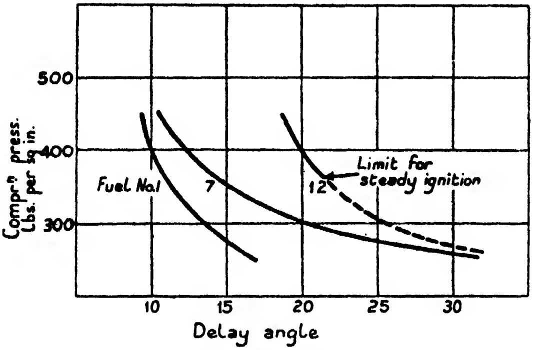

FIG. 47. Variation of delay angle with compression pressure.

An important conclusion to be drawn from these experiments is how widely fuels of different chemical composition may vary in the way they are affected by a fall of pressure; and that their behaviour in this respect cannot be foretold with certainty from measurements of ignition temperature made at one pressure only. It is a point which must be carefully watched in any selection of the most suitable oil for use as a fuel in aircraft engines.

How large an increase of compression temperature can be achieved in the auxiliary-chamber type of engine, by the heating of the air during compression, has been well demonstrated in several series of diagrams given by Dicksee.[48] During tests at speeds between 500 and 2,000 r.p.m. it was found that the compression pressure increased as shown in fig. 48. The estimated compression temperatures are given also, and it will be seen that there was a difference of no less than 200° C. between 500 and 2,000 r.p.m. This rise of temperature was due to the combined effect of the hotter cylinder head and higher gas turbulence which both follow from an increase of engine speed. To achieve so large an amount of air heating the neck of the auxiliary chamber must be uncooled, and the shape designed to give an organized air turbulence and vigorous scouring action.

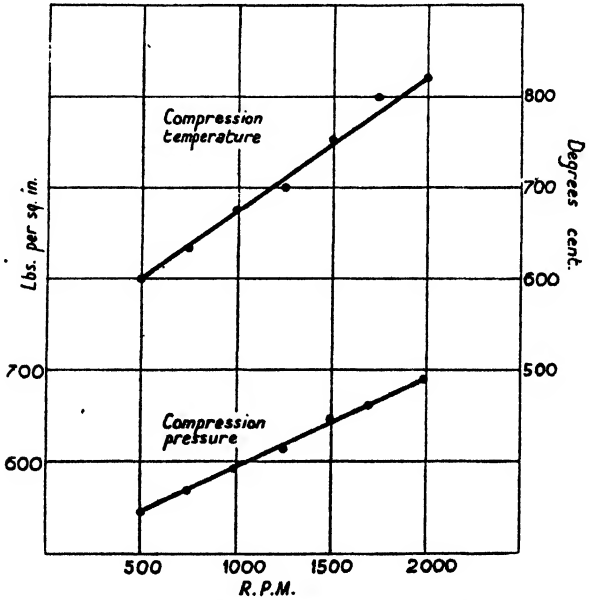

FIG. 48. Variation of compression pressure and temperature with r.p.m.

The exact degree to which a rise of compression temperature shortens the delay time will vary from engine to engine. Uniformity of temperature in the combustion space will depend upon the form and vigour of the turbulence, and the amount of the delay, for a given mean gas temperature, will depend upon whether the fuel jet is placed so as to take full advantage of the temperature-rise by being projected through the hottest parts of the air.

In fig. 49 are shown the delay times, expressed as seconds and as degrees of crank angle, for the range of speeds given in fig. 48, when the combustion chamber of the engine had been modified so as to produce the greatest possible increase of compression temperature. It will be seen that there was a steady shortening of the delay time, measured in seconds, sufficient to keep its value in degrees of crank angle very nearly constant over the speed range from 500 to 2,000 r.p.m. Before modification the same engine gave a similar delay at 2,000 r.p.m., but at lower speeds the heating of the air was less effective, so that at 1,000 r.p.m., instead of the delay occupying the same crank angle it extended over an arc of twice the length, and was therefore of four times the length measured in seconds.

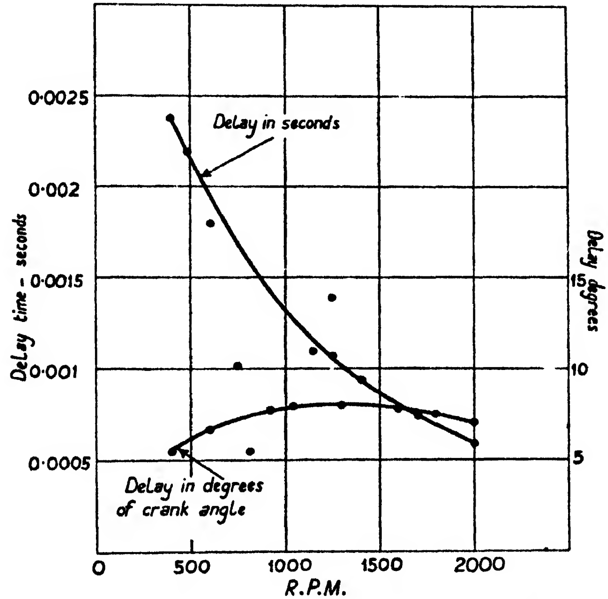

Fig. 49. Variation of delay, in seconds, and also in degrees of crank angle, with r.p.m.

The great possibilities and importance of this method of raising the air temperature during compression will now be realized; for it provides a means of reducing the delay in proportion to an increase of speed, and so of maintaining smooth and efficient running up to much higher speeds than would otherwise be possible. Unfortunately the necessary design of cylinder head, with its separated combustion chamber, presents serious difficulties when the very highest power output per cylinder is demanded, owing to the limitations which it imposes in regard to the placing of the valves. Such limitations, however, are only likely to be serious in aero-engine designs. For all forms of road transport the auxiliary combustion chamber, although usually rather inferior to the open cylinder head with directed spray in regard to fuel economy, has grown steadily in favour by reason of its reliability and satisfactory running under widely varying loads.

VI

THERMAL EFFICIENCY

ART. 41. Fuel consumption per horse-power and thermal efficiency

Any full discussion of thermal efficiency is complicated, so long as we have to deal with volatile liquid fuels, by the all-important factor of detonation. There are, however, a number of questions which can be dealt with on the simplifying assumption that detonation does not arise. Later we shall have to examine the limitations which it imposes.

There are two senses in which the term 'thermal efficiency' is constantly used, a loose and a strict one, and it is very important to be clear about the difference. Thermal efficiency is frequently spoken of in the loose sense as being equivalent to the inverse of the 'fuel consumption per horse-power', whereas it may be no such thing. The fuel consumption per horse-power of an engine might be increased by 50 per cent. without its thermal efficiency, in the strict sense, being diminished at all. The following will make the point clear. Thermal efficiency, strictly speaking, means the efficiency with which the heat actually generated, or capable of being generated, by combustion is converted into mechanical work. If an engine is working upon a correct or a weak fuel-air mixture, the thermal efficiency, 'indicated' or 'brake' as the case may be, is the fraction

$$\frac{\text{work done per minute}}{J\,[\text{effective heat value of fuel supplied per minute}]},$$

but if the fuel-air mixture is a rich one, then the heat supplied per minute is still only that amount which can be generated by combination with all the oxygen present in the cylinder. The surplus fuel is not burnt, and the full amount of its potential heat is never developed. The thermal efficiency of the engine in the strict sense may be the same as when using the correct mixture, but the fuel consumption per horse-power will be greater, and thermal efficiency in the loose sense correspondingly less, for this will have for its denominator the heat value of all the fuel supplied, unburnt as well as burnt.

It is not proposed, in what follows, to keep to the use of thermal efficiency only in the strict sense, but it must not be thought that the point is an academic one, or unimportant. When dealing with the problem of equal fuel distribution between the cylinders of a multi-cylinder petrol engine, for example, it is of the highest importance to have the difference clearly in mind. In such an engine, if distribution were perfectly equal and the mixture correct or slightly weak in every cylinder, the thermal efficiency as ordinarily calculated would be the same as that obtainable from a single cylinder of the engine tested separately. In practice it is seldom possible to obtain so high a thermal efficiency on a multi- as on a single-cylinder engine, because of inequality of distribution. While one or two cylinders may be working on the best fuel-air mixture for high thermal efficiency, the others will be getting an excess of fuel some of which they cannot possibly use. During steady running there can be no cylinders receiving less fuel than corresponds to the most economical mixture strength, for, if so, irregular firing will very soon set in.

If we put up the compression ratio of all the cylinders of an engine we shall improve the true thermal efficiency of each one by a certain amount, but there may still be a large waste of fuel in certain cylinders which is more than the gain we can achieve by putting up the compression ratio. The difference between the apparent thermal efficiency, which corresponds to the inverse of the fuel consumption per horse-power, and the true thermal efficiency, represents the gain which might be effected by simply improving the fuel distribution without altering the cylinders in any way.

ART. 42. Thermal efficiency reckoned from air consumption

Ricardo has pointed out that it is possible to derive a figure for the thermal efficiency of an engine, when working on a rich mixture, from the air consumed, for this will correspond roughly to the portion of the fuel which gets burnt. Instead of thinking of carbon and hydrogen in the fuel as being burnt to CO_2 and H_2O, let us think of the oxygen of the air present as being 'burnt', by the agency of the fuel, to the same CO_2 and H_2O. Provided the chemical nature of the fuel and the proportion of carbon to hydrogen in it does not greatly alter, we might even speak of the 'calorific value' of a pound or a cubic foot of air, meaning the heat generated when a pound of air is used up in turning a certain quantity of the fuel into CO_2 and H_2O. If we assume, for example, that the fuel is pure heptane, then we see from equation (30) in art. 23 that the heat energy generated during the burning of 3,020 lb. of air is

$2{\cdot}154 \times 10^6$ C.H.U. and the 'calorific value' of air is therefore 713 C.H.U. per lb., or 57·3 C.H.U. per standard cubic foot.

It must be noted that this calorific value for air only holds good on the assumption that all carbon and hydrogen burnt is burnt to CO_2 and H_2O. As regards the carbon, some of this in a rich mixture will be burnt to CO only, and the heat derivable when one cubic foot of oxygen is used to produce two cubic feet of CO is only 167 C.H.U., instead of 271 C.H.U. if it had produced one cubic foot of CO_2. The figure 57·3 C.H.U. per S.C.F. for air is therefore only an approximate one for fuel-rich mixtures, the true value will be variable and rather lower; but it is sufficiently accurate to be of practical service, as the figures now to be given will indicate.

Ricardo made a series of experiments in which his single-cylinder engine was run at constant speed and full throttle, with a gradually increasing supply of petrol to the carburetter. These experiments showed that over a wide range of fuel-air ratios on the rich side, up to 35 per cent. rich, the air consumption of the engine was very nearly constant, as is shown by the figures in column 2 of table 23.

TABLE 23

Thermal efficiencies calculated from air consumption when operating on a rich mixture.

Compression ratio 5 : 1.

Fuel-air ratio	*lb. of air per hour*	*Indicated horse-power*	*lb. of air per I.H.P. hour*	*Indicated thermal efficiency from air consumption*
				per cent.
Correct	196	32·0	6·13	32·4
20% rich	198	33·4	5·94	33·4
35% rich	200	33·1	6·04	32·8

Maximum thermal efficiency reckoned upon fuel consumption was 32·1 per cent., with a fuel-air ratio 16 per cent. weak.

The horse-power figures in column 3 exhibit the well-known fact that the power of an engine at full throttle increases slightly as the mixture is made richer, up to a fuel-air ratio 20 per cent. rich, and then begins to fall off again. It must not be taken for granted that the constancy of air consumption shown in column 2 must necessarily be associated with the nearly constant power of column 3; in fact the relationship is quite a chance one. As we shall see later, in a gas engine it does not hold. The rough constancy of air consumption in a petrol engine (in the table actually a slow increase) is due to a fortunate balance between an increase of the whole weight of mixture drawn in, on the one hand, because of the increased cooling by evaporation of a greater weight of fuel, and on the other hand a reduction of the air part of the mixture because a greater volume is displaced by fuel vapour.

The rough constancy of power in column 3 is actually an increase followed by a decrease, instead of a steady increase like the air, and it is due to a balance between a slightly falling calorific value per cubic foot of air, as explained above, and a true thermal efficiency which at first rises and then falls. This is the really important factor, and it will be dealt with more fully in a later article. For the present it will be sufficient to say that the reason for the quite substantial rise of true thermal efficiency for mixtures 10–15 per cent. rich is because of the reduction in the proportion of CO_2 in the products of combustion, and the substitution for it of the diatomic CO, which has a much lower volumetric heat, and one rising less with temperature. At the still richer mixtures the average volumetric heat of the cylinder contents rises again on account of the presence of unburnt fuel vapour of which the volumetric heat is even higher than that of CO_2. With this final rise of volumetric heat both efficiency and power output decline steadily as the proportion of fuel becomes excessive.

To calculate the thermal efficiencies reckoned upon air consumption we have only to remember that the approximate calorific value of air with heptane (and with similar hydrocarbon fuels) is 713 C.H.U. per lb., so that

$$\text{thermal efficiency} = \frac{33{,}000 \times 60}{713 \times 1{,}400} \div (\text{air per I.H.P. hour}).$$

In this way column 5 of table 23 is obtained direct from column 4.

Ricardo has suggested that this figure for thermal efficiency calculated from the air consumption affords a valuable guide to what we may call the 'combustion-chamber efficiency', or the 'intrinsic efficiency', of a multi-cylinder engine. As will be clear from the example to be given presently, this efficiency calculated from the air consumption gives a close approximation to what the best obtainable thermal efficiency of a multi-cylinder engine *would* be, if all the fuel were equally distributed between the cylinders. In other words it gives an indication of the intrinsic efficiency of the cylinders themselves, unconfused by differences of mixture strength from one cylinder to another.

The method is particularly convenient to apply in practice, provided means are available for measuring air consumption. We have seen that with both gas and liquid fuels maximum power is obtained with a fuel-air ratio about 20 per cent. rich. Adjustment for maximum power is an easy one to make, since the fuel-air ratio need not be accurately regulated and the power is observed directly on the brake. Having got the engine running steadily at its maximum power, we then have to make only one observation of the horse-power and the rate of air consumption. To get a reliable figure for thermal efficiency determined from the fuel consumption, on the other hand, we have to take a series of readings of power output at gradually weaker fuel-air ratios and plot a 'consumption loop' (see Chap. VII).

As an example of the utility of this method of calculating thermal efficiency from air consumption for bringing out the wastage of fuel in a multi-cylinder engine through unequal distribution of the fuel between the cylinders, we may quote a further test made by Ricardo upon a six-cylinder aero engine with a compression ratio of 4·7 : 1. This engine showed a constant air consumption of 6·4 lb. of air per I.H.P. hour over a range of fuel-air ratios from 10 per cent. to 25 per cent. rich. The efficiency reckoned upon air consumption therefore works out as 31 per cent. The highest figure for the efficiency which could be obtained by measurement of the fuel consumption was only 28·2 per cent. This was at the most economical *average* fuel-air ratio, which means that if the fuel supply was cut down any lower, some cylinders were then receiving so weak a mixture that the power from them began to fall off rapidly and so increased the average fuel consumption per horse-power for the whole engine.

The difference between the two observed efficiencies on the multi-cylinder engine, 31 per cent. from the air and 28·2 per cent. from the fuel, was due to wastage of fuel through inequality of distribution. We may say that, while the intrinsic efficiency of the cylinders was 31 per cent., the best obtainable efficiency for the whole engine, owing to defects of the induction system, was only 28·2 per cent. This difference in efficiency corresponds to a difference in fuel consumption per I.H.P. hour of from 0·42 lb. to 0·46 lb.

In a single-cylinder engine, where there is no distribution loss, the thermal efficiency from air consumption must be identical with that from fuel consumption at the correct mixture, for the calorific value of air was derived from that of the fuel by taking the heat produced when air and fuel are burnt together in that proportion. It follows that

31 per cent. and 0·42 lb. per I.H.P. hour would have been the figures obtained in an accurate experiment with the multi-cylinder engine at the correct mixture, if distribution had been perfect, and still better figures would have been possible at the most economical mixture strength, about 15 per cent. weak. Actually the best which could be obtained was 0·46 lb. per I.H.P. hour, so that imperfect fuel distribution was responsible for at least 0·04 lb., or 10 per cent. of the total fuel consumption. This 10 per cent. was wasted because it formed part of an excessively rich mixture in certain cylinders, while other cylinders were working upon a fuel-air ratio near the limit of weakness. The excess of fuel in the rich cylinders could not all be used efficiently, and merely served to increase the average fuel consumption per horse-power for the whole engine.

ART. 43. Thermal efficiency and mixture strength

We have seen in art. 36 how combustion in the cylinder is influenced by fuel-air ratio, and we shall now extend our study to the effect of this variable upon thermal efficiency.

In fig. 33 was shown the variation of I.M.E.P., and therefore I.H.P., of a gas and a petrol engine, running at 1,400 and 1,500 r.p.m. respectively, when the fuel-air ratio was varied from the weakest possible to one 40 per cent. rich. In fig. 50 there has been added to the same diagram a curve *CD* which shows how the thermal efficiency on coal gas varied during the same tests.† The placing of this curve of thermal efficiency, *CD*, as well as the curve *AB*, in relation to the correct mixture point on the diagram, has been made on the assumption of a correct mixture ratio of gas to air of 1:4·5 as in the original paper.[20] It seemed preferable, in order to avoid confusion, to give Burstall's results without modification, but, as already mentioned in the footnote on p. 130, the correct mixture in these experiments was probably 1:4·1 rather than 1:4·5. This corrected value accords with the calorific value of the gas, which was accurately known. The ratio 1:4·5 was derived from the analysis of the gas, in which accuracy is difficult as regards the very small but very important proportion of hydrocarbon vapours. A gas with a calorific value of 258 C.H.U. per cubic foot *must* require just about 4·1 cubic feet of air for complete combustion, because within fairly wide limits of fuel composition the combustion of one cubic foot of air always produces the same number of C.H.U.

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† It will be noticed that thermal efficiency is here used in the loose sense so far as the right-hand end of the curve is concerned, where the mixture is rich.

Diagrams in which the heat generated per cubic foot of combustible mixture is taken as the abscissa, such as fig. 51, are unaffected except as to where the correct mixture line is drawn. In fig. 51 it has been drawn at a heat content of 70,500 ft.-lb. per S.C.F., which corresponds to a mixture of 1 cubic foot of coal gas and 4·13 cubic feet of air.

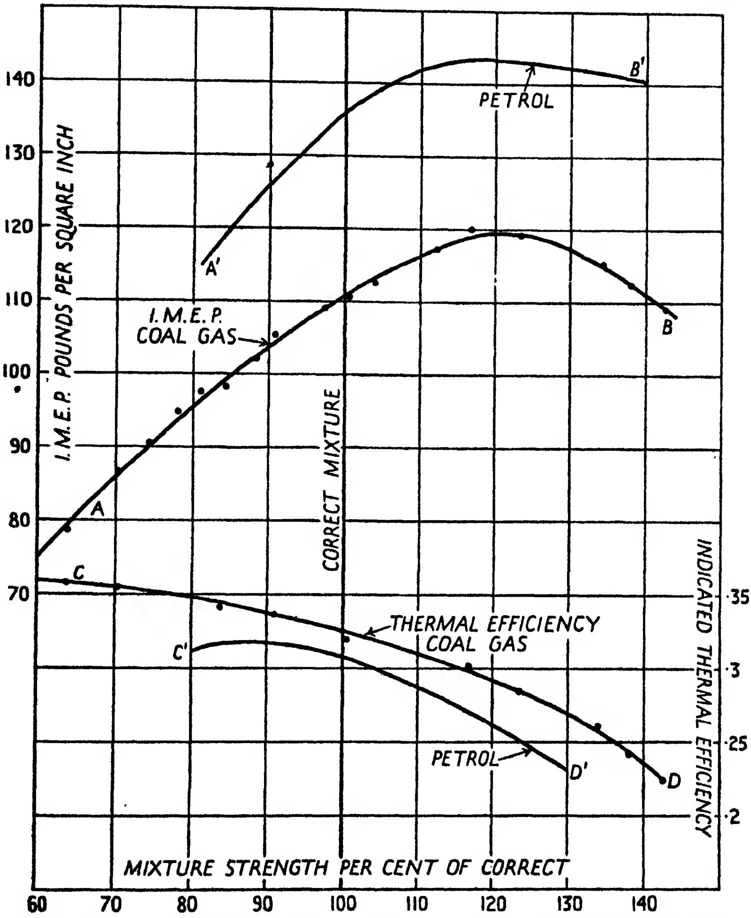

Fig. 50. Indicated mean effective pressures and thermal efficiencies at different fuel-air ratios. Compression ratio 5:1. Speed: coal gas, 1,400 r.p.m.; petrol, 1,500 r.p.m.

From fig. 51 it is seen that the thermal efficiency on coal gas at the correct mixture is about 31·5 per cent. Hence in fig. 50 the coal-gas thermal efficiency curve should be moved to the left, and it will then be found that at the correct mixture the thermal efficiency on coal gas is only slightly higher than that on petrol. This is as it should be, for although the heats developed per S.C.F. of combustible mixture are 56·2 and 50·0 C.H.U. respectively for heptane and coal gas, the heats developed per S.C.F. of the products of combustion are 53·2 and 53·4 C.H.U. The maximum temperatures will therefore be almost the same.

As to the shape of the coal-gas thermal efficiency curve in fig. 50, it will be observed that there is a steady fall of efficiency from the left hand of the diagram as the mixture becomes richer, the change being due at first to higher temperatures and consequently higher volumetric heat of the working substance. Heat loss to the cylinder walls becomes higher also as the average cycle temperature rises. To the right of the 100 per cent. line the mixture is rich, there is not any longer enough oxygen to burn all the fuel, and although the efficiency with which all the heat generated by combustion is turned into work may remain approximately constant, the thermal efficiency as ordinarily reckoned, on the basis of the fuel supplied, continues to fall as the surplus fuel increases.

It may be observed here that if the fuel had been petrol the efficiency curve would have shown a maximum point at about 15 per cent. weak, its left-hand half having taken the shape shown by the curve $C'D'$. If the mixture in a petrol engine is weakened further than this, the loss of power through slow, and very soon through irregular and incomplete, combustion, more than balances any further gain through a lowering of the cycle temperatures. We shall see in art. 45 that it is possible, by working with a stratified charge, to maintain a rising efficiency curve with lower fuel-air ratios on petrol, but so long as the mixture in the cylinder is more or less homogeneous, nothing like the steady rise of efficiency on the weak side which is shown for coal gas by the curve CD can be maintained with volatile liquid fuels. This difference of behaviour between the two types of fuel is probably due in part to the presence in the coal gas of a large proportion of free hydrogen. This has an extremely high rate of combustion with air and is able, even in very weak mixtures, to bring about complete and fairly rapid combustion. The hydrogen portion, in fact, may be regarded as acting like an igniter throughout the rest of the fuel-air mixture.

This, however, is not the whole of the story, for Burstall[32] has shown that it is possible to get a steadily rising efficiency curve when running an engine upon weak mixtures of pure carbon monoxide and air down to 40 per cent. weak; and, moreover, to keep the maximum pressure in the standard position, 12° after the dead centre, provided sufficient ignition advance is given. As much as 90° advance was necessary with the 40 per cent. weak CO mixture at 5:1 compression ratio, as compared with 50° for coal gas under comparable conditions. Free hydrogen is therefore not an essential in the burning of weak mixtures efficiently, but the difference of ignition timing required for pure carbon monoxide and for coal gas reflects the speeding up effect which the hydrogen produces.

Although the best thermal efficiency which he obtained with CO was only about 30 per cent. at 5:1 compression, as compared with 36 per cent. with coal gas, nevertheless Burstall's experiments proved the possibility of getting complete and regular combustion with CO down to weak mixtures, which would be quite impossible with a volatile liquid fuel. This difference one must attribute to the greater simplicity and number, and hence the better distribution, of the CO molecules in the cylinder as compared with those of an evaporated hydrocarbon fuel; and perhaps also to the fact that the early stages of oxidation of the more complex petrol molecule is in the nature of a chain reaction (see art. 31). If this be so, one can the more easily understand the failure of combustion to proceed when the fuel molecules become sparsely distributed, as in a weak petrol-air mixture.

In fig. 51 the efficiency has been shown in relation to the heat available per cubic foot of combustible mixture. In the correct mixture of coal gas and air (taking as before the average coal-gas composition given in art. 15) the heat developed by complete combustion at 100° C. and constant volume was 50·0 C.H.U., or 70,000 ft. lb. per S.C.F. of combustible mixture. In other degrees of richness or the reverse the full heating value of the fuel present will be in proportion to the richness, and we may express the mixture strength, therefore, in terms of the heat energy available in the fuel per cubic foot of mixture, although it does not, of course, follow that all this heat will be developed by combustion if the amount is greater than 50·0 C.H.U. per cubic foot. In fig. 51 the abscissae extend from 0 to 95,000 ft.-lb. per S.C.F. and represent mixtures of coal gas and air varying from that containing no fuel at all at the left-hand side, to a mixture 35 per cent. rich on the right. The vertical line at 70×10^3 corresponds also to the correct mixture. A diagram of this kind was first used by Hopkinson,[3] who pointed out that in the limit, if it were possible to burn such weak mixtures, then, when the heat available had been reduced to zero an engine should approach the ideal air-cycle efficiency, for then there would be no rise of temperature on combustion and no appreciable rise of volumetric heat.

The lowest line on fig. 51 shows the experimental results obtained by Burstall at a compression ratio 5:1 and 1,400 r.p.m. The horizontal straight line *BB* is drawn at a thermal efficiency of 47·1 per cent. which is the air-cycle efficiency for this ratio (table 1). The line *BC* represents what may be termed the 'theoretical limit of efficiency'. Its derivation will be more fully explained in the next article. It is the highest efficiency which any engine could possibly attain, having regard to the real properties of the working substance, if all heat losses were suppressed and combustion were complete so far as temperature and pressure conditions permit. For an expansion ratio of 5:1 this theoretical limit of efficiency has been calculated for four different mixture strengths of a benzene-air mixture,† namely, for the correct mixture, and for mixtures 20 per cent., 50 per cent., and 75 per cent. weak. For

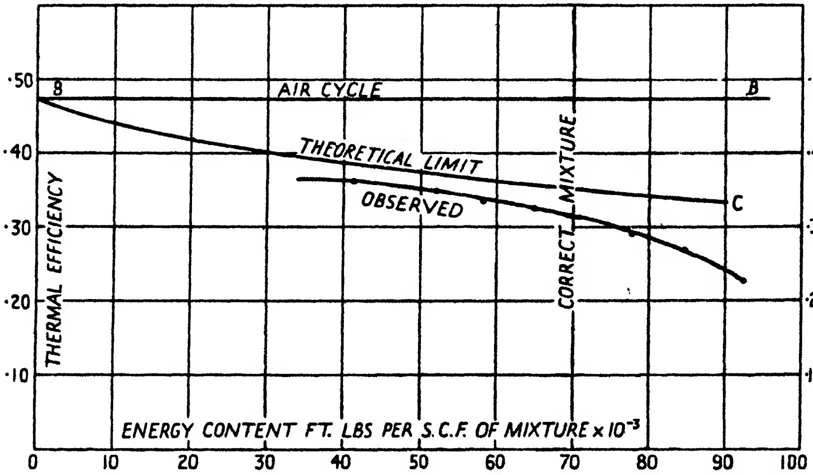

FIG. 51. Efficiencies with different heat contents per cubic foot of mixture entering cylinder. Fuel, coal gas. Speed 1,400 r.p.m. Compression ratio 5:1.

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† The limit curve in fig. 51 has been drawn the same as in fig. 54 and it has therefore been assumed that the theoretical limit of efficiency will be the same at the same heating value per cubic foot of combustible mixture, whatever the fuel. Although only an approximation, this assumption can be justified on the grounds of simplicity where, as here, the purpose of the diagram is merely to illustrate a principle. To obtain the true theoretical limit curve for coal gas corresponding to Burstall's observations, allowance would have to be made for dilution of the charge with exhaust gas in calculating the heat content per S.C.F. of mixture *in* the cylinder.

these mixtures the values found for the calculated efficiency were 34·0 per cent., 35·7 per cent., 38·5 per cent., and 41·6 per cent. The four values are shown on the diagram and the theoretical limit line for all mixture strengths has been drawn through them. At zero fuel it intersects the air-cycle efficiency line *BB*.

Everywhere to the right of the correct-mixture line the mixture is rich. There is a waste of fuel, and the experimental curve falls away from the theoretical limit. To the left, or weak, side of the line the experimental curve shows an efficiency rising at first more rapidly than the theoretical limit curve, but below 50,000 ft.-lb. per S.C.F. there is a gradual falling off due to slow combustion at the very weak mixtures. In all these experiments the ignition was adjusted to give the standard condition of maximum pressure at 12° after the dead centre (see art. 35). To do this with the weakest mixtures required a spark timing actually 74° before the dead centre, and under these conditions the loss of heat during compression begins seriously to affect the thermal efficiency.

The experimental results here given were all obtained with coal gas as fuel. In art. 45 results will be given for petrol, on the same type of diagram, when discussing the possibilities of working with very weak petrol-air mixtures by means of a stratified charge.

ART. 44. The limits of possible efficiency

We saw in art. 9 how it was possible to derive an ideal air-cycle efficiency, and fig. 4 showed how this ideal air standard of efficiency varied with increasing expansion ratio. Now there are different ways in which a standard engine of comparison may be looked upon, when regarded as the ideal with which real and imperfect engines may be compared. The ideal air cycle of art. 9, which postulated a perfect gas as working substance, must, in the nature of things, ever remain far removed from practical possibilities. As an alternative to this ideal air cycle we might adopt as a standard, with which to compare the performance of real engines, one which takes account of the fundamental properties of the real working substance, and only defines and simplifies the working conditions by assuming, for example, that heating of the working substance is instantaneous, and that there is no loss or gain of heat through the cylinder walls. This latter standard then becomes an approachable though not, so far as can be seen, an attainable ideal.

In addition to the ideal air standard, it is convenient to have a standard cycle which can act as a guide to designers in the sense that it shows what is to be expected as the result of a modification in design, such as raising the compression ratio, after taking into account the fundamental physico-chemical conditions in the cylinder but at the same time legitimately simplifying the processes in the engine, by the neglect of practical imperfections, so as to allow results to be calculated with reasonable ease and to make the conditions of calculation independent of any particular engine.

In accordance with these principles Tizard and the present writer,[7] adopting the most accurate data upon calorific values and volumetric heats available at the time (1920), and working upon the known behaviour of CO_2 and H_2O in regard to dissociation at high temperatures, evaluated the explosion temperatures, and temperatures at the end of expansion, for certain typical fuel-air mixtures at expansion ratios of 5:1 and 10:1.

As already mentioned, a re-calculation of much of this work has recently been made using values of volumetric heats and equilibrium constants obtained from spectroscopic data by various investigators, and also taking into account the formation of nitric oxide. On this basis it has been found that for an expansion ratio of 5:1, as compared with the ideal air-cycle efficiency of 47·1 per cent., the theoretical limit of efficiency for the benzene-air mixture giving just complete combustion (see art. 23) is 34·05 per cent. when the real properties of the working substance are taken into account but no loss of heat permitted to the cylinder walls and piston.

At 10:1 expansion ratio, for the same mixture, the calculated efficiency is 44·38 per cent.

The derivation of these figures involves first the calculation of the explosion temperatures by the methods of art. 24; next the final temperatures after expansion by the method of art. 12, but allowing for the attainment of chemical equilibrium by combustion as the expansion proceeds (see art. 24, p. 73); and finally the net work done per cycle, as the difference between the total energy—heat plus chemical—of the working substance before and after expansion, less the work of compression. The denominator of the thermal efficiency fraction is obtained by deducting, from the heat of combustion of the fuel content of the original mixture, the chemical energy of the combustible products in the gases at the end of expansion.

The efficiency for the correct benzene-air mixture was worked out in art. 12, neglecting dissociation, and was there shown to be 35·2 per cent., as compared with the more correct figure of 34·05 per cent. The latter figure would have been reduced to 33·0 per cent. if the chemical energy of the gases at the end of expansion had been disregarded.

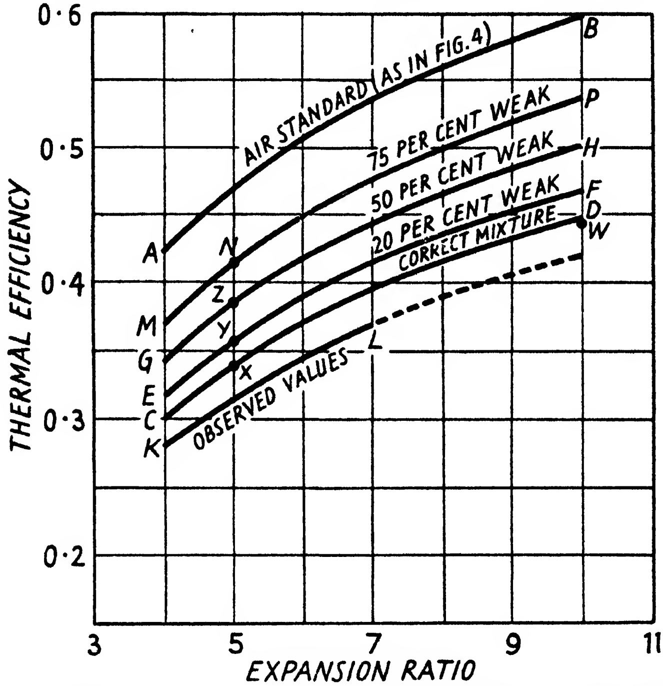

FIG. 52. Variation of thermal efficiencies, theoretical and observed, with expansion ratio.

In fig. 52 the ideal air-cycle efficiencies are shown plotted against expansion ratio in the curve *AB*. The calculated efficiencies for the benzene-air† mixture at 5 : 1 and 10 : 1 are shown at the points *X* and *W*. It will be found that the curve *CD* which is represented by the equation

$$\eta = \left(1 - \frac{1}{r}\right)^{0\cdot2586}$$

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† Benzene (C_6H_6) was chosen as the typical hydrocarbon fuel for making the calculations for several reasons: it is comparatively easy to obtain in a state of high purity and could therefore be used in accurate engine trials for checking the calculations. And moreover the physical data upon calorific value etc. are known with great accuracy. As stated in art. 17, benzene forms about 80 per cent. of the commercial product benzol.

goes through the point X and nearly through the point W, and lies roughly parallel to the curve

$$\eta = 1-\left(\frac{1}{r}\right)^{0\cdot396}$$

for the ideal air cycle, which is shown at AB.

Proceeding exactly as was done for the correct benzene-air mixture, it is possible to calculate the theoretical efficiency for weak mixtures. This has been done for mixtures 20, 50, and 75 per cent. weak at an expansion ratio of 5:1. The results of these calculations are shown at points Y, Z, and N at 35·71, 38·56, and 41·61 per cent., and the curves EF, GH, and MP are curves drawn through these points by means of equations of similar form to those for ideal air and for the correct mixture.

All these calculated efficiencies are for the benzene-air mixture as it enters the cylinder, undiluted with residual exhaust gas. In other words the cylinder before compression has been assumed to contain only combustible mixture. The methods of Chapter III might, of course, be applied equally well with allowance made for the proportion of residual gas proper to any particular conditions.

In fig. 52 the curve KL shows the highest thermal efficiencies actually observed at compression ratios varying from 4:1 to 7:1. These would all be obtained with fuel-air ratios about 15 per cent. weak. They should therefore be compared with curve EF rather than curve CD. The *relative* efficiencies obtained by such a comparison must not, however, be assumed to be the maximum relative efficiencies for, as will be seen from the petrol curve in fig. 50, the highest relative efficiencies are likely to be associated with somewhat richer fuel-air mixtures. On the other hand, it must be remembered that the engine mixture is diluted with exhaust gases and hence its mixture strength is considerably less than that given by the fuel-air ratio. This effect will tend to be greater at lower compression ratios and smaller at higher compression ratios, and therefore the curve KL not only refers to engine mixtures which are considerably weaker than the measured fuel-air ratio but to engine mixtures which vary with the compression ratio.

A comparison of the curves KL and EF should, however, be useful in giving some indication of the direct heat losses by radiation and conduction to the cylinder walls and any losses due to imperfect combustion, the sum of these losses being represented by the vertical distance between the two curves. An analysis of these losses, and an examination of the possibility of reducing them further, will be found in Chapter VII.

The curve *GH* may be taken as a criterion of efficiency in the case of a gas engine working with the weakest possible fuel-air ratio and with fully advanced ignition.

It is perhaps worth while adding a few words about the construction of fig. 52, which will repay careful study.

Any curve represented by an equation of the form

$$\eta = 1-\left(\frac{1}{r}\right)^n$$

will lie generally parallel to the curves in the figure. One point on the diagram, therefore, through which such a curve is to be drawn, fixes the value of the index n. Besides the ideal air-cycle efficiencies, *AB*, and the curve *KL*, which is merely a fair curve drawn through a series of experimental observations, there are four curves based on theoretical calculations of thermal efficiency when account is taken of the properties and behaviour of the real working substance. These four curves correspond to conditions when the fuel-air mixture is (1) correct, (2) 20 per cent. weak, (3) 50 per cent. weak, (4) 75 per cent. weak. A calculation of efficiency at any one expansion ratio for each of these conditions would be sufficient to locate the curve on the diagram for that fuel-air ratio, but if only four points like *X*, *Y*, *Z*, and *N* had been calculated, it would have remained a pure assumption that when the real working substance was used for calculations in this way, the calculated values for thermal efficiency at other expansion ratios than 5:1 would fall on a curve similar to the air-cycle curve *AB*, and one, therefore, represented by

$$\eta = 1-\left(\frac{1}{r}\right)^n.$$

Besides the four points *X*, *Y*, *Z*, and *N*, therefore, we have calculated the fifth point *W* for an expansion ratio of 10:1, and the fact that a curve for which the index n is settled as being 0·2586 by the point *X* does almost go through the point *W*, justifies us in drawing the other curves by means of a similar formula.

ART. 45. Thermal efficiency at reduced loads. Stratified charge operation

Reduction of power output in a high-speed gas or petrol engine is normally made by throttling the indrawn mixture, for the possibilities of quality governing typical of the Diesel engine are, for the carburetter engine, very limited. There is also the 'hit-and-miss' method commonly employed on small, slow-speed gas engines, and it will be convenient to deal shortly with this before proceeding to the main concern of this article. Quality governing, as exemplified in the Diesel engine, is dealt with in art. 52.

With a hit-and-miss governor the gas inlet valve is left closed for one or more cycles whenever the speed rises above a certain maximum, the air inlet working normally all the time. In this way one or more completely idle, or scavenging, strokes are interposed at intervals in place of working strokes. The result is a large fluctuation of speed which has to be minimized by carrying a heavy fly-wheel but which, even so, puts the method out of court for many purposes.

From the point of view of efficiency the method is to be commended, for such an engine will, under reduced load, show a higher indicated thermal efficiency than its normal full load value. The reason is that on each cycle following a missed, or scavenging, cycle, there is no exhaust gas present to heat the incoming charge of gas and air, which is diluted instead with cool air. This has an appreciable effect, through a lowering of temperature of the cylinder contents, in improving both power and efficiency during the ensuing cycle.

One effect, of course, of employing the ordinary type of quantity governing, or in common parlance 'throttling', with a carburetter engine is that pressures (but not temperatures) are reduced throughout the cycle. Of much more importance, however, is the effect upon combustion of the increased proportion of exhaust gas present during compression.

When considering the effect of a rise of compression ratio on rate of burning (art. 37) it was emphasized that the presence of exhaust gas slows down the rate of combustion. Under throttled conditions the proportion of exhaust gas which mingles with the fresh charge may easily be doubled, and unless ignition advance is increased to counteract the effect of this dilution, indicated thermal efficiency will suffer severely. Brake thermal efficiency must always suffer, through the fall of mechanical efficiency under reduced loads.

The effect of increased dilution and slow burning is very clearly brought out by the two curves shown in fig. 53. These illustrate the results of experiments on a single-cylinder engine running at constant speed (1,500 r.p.m.) in which the load was varied by throttling. The engine was driving an electro-dynamometer which called for a varying torque according to the throttle position. Curve (*a*) shows the effect of throttling upon efficiency when the ignition timing was maintained the same as that for optimum performance at full throttle. The throttle was closed step by step until the power, and therefore the torque, was only 40 per cent. of its full load value, and during the change the efficiency fell away from 31·6 to 26 per cent., a drop of 18 per cent. If, on the other hand, the ignition was advanced step by step as the throttle was closed, so as to give maximum efficiency at each load, curve (*b*) was obtained, and it will be seen that during the first steps of throttling the efficiency actually rose until at 70 per cent. of full load it was 32 per cent. At 40 per cent. torque it had returned to its full load value.

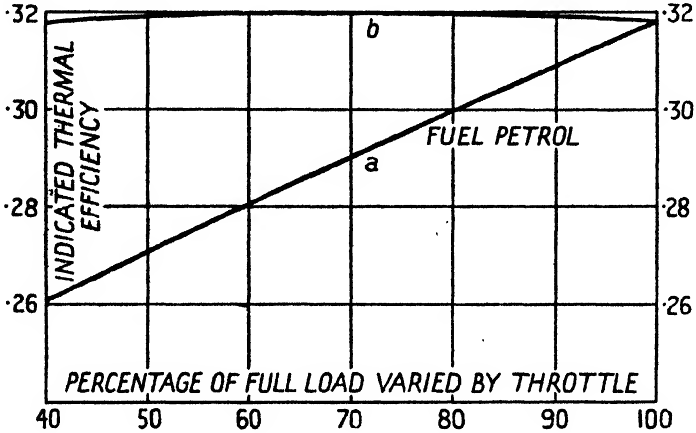

FIG. 53. Thermal efficiency under throttled conditions. Speed 1,500 r.p.m. Compression ratio 5:1. (*a*) Ignition fixed as for maximum power at full throttle, and (*b*) ignition adjusted for maximum power at each throttle setting.

It should be noted that great care must be taken in an experiment of this kind, when indicated power is to be deduced, in order to make certain that the correct allowance is made for the pumping losses under the changing conditions, for they will be different at every throttle setting (see art. 62).

Some further results, obtained on a four-cylinder engine, both on full throttle and also with the load reduced by throttling, are given in art. 56 and illustrated in fig. 76. These results again show the need for an increased ignition advance when running under throttled conditions, if the best power and efficiency are to be maintained.

Combustion under throttled conditions could be speeded up by employing a richer mixture, as indeed is commonly done, but this again involves a drop of efficiency; so that the only way of avoiding the fall of efficiency when a homogeneous mixture is throttled is the provision of adequate ignition advance. Even then, the proper adjustment for each throttle position is a difficult matter in practice, when speeds are not constant; and accordingly it may be said that except in circumstances where it is possible to make very special arrangements, there will always be a loss of indicated, and still more of brake, thermal efficiency at less than full power, when the power is controlled by throttling. This is a serious matter when the engine is for duty such as that in a motor-car, where the greater part of its working life is run under throttled conditions, and it is of interest to examine the possibilities of the alternative method of power control by reducing the fuel charge, that is to say, by quality control.

Fig. 50 of art. 43 illustrates the point which has already been made, that with coal gas as fuel it is possible to maintain a rising efficiency down to 50 per cent. weak, and efficient power control of a gas engine down to about 60 per cent. of the full load torque is therefore possible, provided the ignition timing is suitably advanced for the weak mixtures. With petrol, on the other hand, the efficiency curve in fig. 50 reaches a maximum at 10–15 per cent. weak, and with normal working, in which the fuel-air mixture is homogeneous throughout the cylinder, it is impossible to make an engine work at all if the mixture is much weaker than this. If we can succeed in making the fuel-air mixture remain non-homogeneous, however, right up to the end of compression, so that a fairly rich mixture is collected at those parts in the neighbourhood of the sparking-plugs, leaving pure air at points further removed, then it is possible to maintain rapid and efficient combustion with the fuel charge reduced to a point at which the average fuel-air ratio for the whole charge would be far below the combustion limit, if the fuel were evenly dispersed throughout the cylinder.

Some slight enriching of the mixture near the sparking-plug with a view to throttled running is achieved in very many designs by placing the plug and inlet valve in a pocket to one side of the main cylinder bore, but the result as regards stratification is very slight and is by no means sufficient to achieve the real aim of stratification, which is to produce a rise of efficiency at low loads in a manner equivalent to the

Diesel engine, instead of the fall which is almost inevitable when a homogeneous mixture is throttled.

Fig. 54 is the same as fig. 51 of art. 43 so far as the lines *BB* and *BC* are concerned, but instead of the gas-engine results those from Ricardo's experiments on a petrol engine working with a stratified charge have been shown. The curve *BC*, as in fig. 51, relates to air-benzene mixtures.

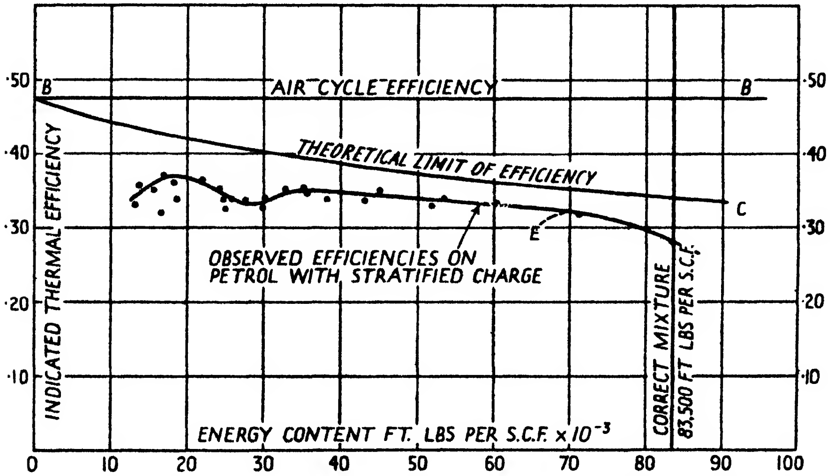

FIG. 54. Thermal efficiencies obtainable at weak petrol-air mixtures with 'stratified charge' operation.

It will be seen that whereas with a homogeneous charge the limit of working on the weak side was as shown at *E*, and the greatest efficiency then obtained was 31·5 per cent., with the stratified charge it was possible to work down to an average mixture strength of 15,000 ft.-lb. per S.C.F., and that the maximum efficiency (at 18,000 ft.-lb. per S.C.F.) had risen to 37 per cent. The curve of experimental points rises steadily as the mixture is weakened, with the exception of a curious kink at a mixture with 28,000 ft.-lb. per S.C.F. This was due to some idiosyncrasy of the engine and its induction system which caused a reduction of volumetric efficiency at this point; and the consequently reduced power is reflected in a reduced efficiency.

Stratification is a difficult, and indeed an almost impossible, thing to effect with a poppet-valve engine which only has valves in the cylinder head. It becomes easy, on the other hand, if the cylinder has either a sleeve valve, or inlet ports in the cylinder wall which are *uncovered by the piston at the end of its stroke.* In either case the ports are directed slightly tangentially so as to give an organized swirl to the cylinder charge which is sufficiently vigorous to be maintained throughout the compression stroke. This swirl about the cylinder axis prevents any general turbulence being set up, and the air remains in well-defined layers along the cylinder axis, which intermingle but little. If arrangements are made for a rich fuel-air mixture to be admitted at one stage of the induction process, followed by pure air, the result is a layer of highly combustible gas close to the cylinder head, shading off rapidly to pure air in contact with the piston. General turbulence is set up at once by combustion, which is quite sufficient, if the mixture is locally over-rich, to effect the necessary mixing with any further air that may be required.

ART. 46. Efficiencies obtained with pure hydrogen as fuel

We have seen that with coal gas as fuel there is no difficulty about employing quality governing down to about 60 per cent. of the full torque, which corresponds to about 50 per cent. of the fuel charge for maximum power (see art. 36). The ability of coal gas to burn fairly rapidly at weak mixtures is derived largely from the 40–50 per cent. of hydrogen which it contains, for this has a rate of flame propagation even in weak mixtures many times greater than that of carbon monoxide and the hydrocarbon vapours.

The behaviour of hydrogen is so unique, and brings out so well the influence of mixture strength and of the average gas temperatures during a cycle upon thermal efficiency, that it is worth while giving some results of experiments with it. In these experiments it was found possible to control the power of the variable-compression engine entirely by quality governing, that is, by regulation of the hydrogen supply, the air throttle remaining full open, from no load up to the richest fuel-air mixture admissible. The same faculty of rapid flame propagation, however, which makes burning possible on the weak side, makes it impossible to work at the rich end of the scale of fuel-air ratios even up as far as the correct mixture. Whenever running was attempted with a hydrogen-air ratio more than about 95 per cent. of that giving complete combustion, violent pre-ignitions set in, accompanied by firing back through the carburetter.

This happened even with the compression ratio lowered to 3·8:1, while at higher ratios smooth running was limited to even weaker mixtures. Fig. 55 shows the efficiencies obtained by Burstall at 7:1 compression and 1,000 r.p.m. with a series of different hydrogen-air mixtures, the mixture strength being expressed in terms of the hydrogen content relative to that of the correct mixture. This method of expressing mixture strength fails to show at a glance the hydrogen content relative to the quantity of hydrogen which *could* be completely burnt by the oxygen content. Burstall found that, at a compression ratio of 7:1, it was impossible to use a mixture with more hydrogen than 80 per cent. of that for complete combustion, for the reasons given above. It will be seen that the highest efficiency was obtained when there was about 41 per cent. of the hydrogen of the correct mixture, and that its value was then 38·3 per cent. Ricardo, working at 1,500 r.p.m. and with a rather weaker mixture still, also at a compression ratio of 7:1, obtained the excellent efficiency of 43 per cent., as compared with the best obtainable with a liquid fuel, at that ratio, of 37 per cent. This exceptional efficiency with hydrogen is due to the possibility of burning very weak mixtures, in which the actual amount of heat generated per cubic foot is small; and also to the fact that there is no CO_2 formed by the combustion. The only triatomic gas present is H_2O, and reference to table 10 on p. 87 shows that the volumetric

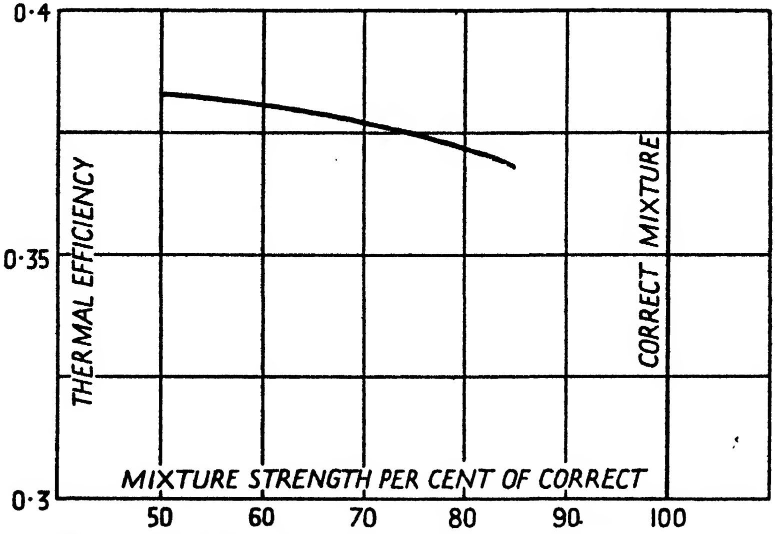

FIG. 55. Thermal efficiencies obtained with hydrogen. Speed 1,000 r.p.m. Compression ratio 7:1.

heat of this gas is considerably lower than that of CO_2. And temperatures in these experiments, as pointed out above, were not high. The low heat production, low maximum temperature, and comparatively small increase of mean volumetric heat during the cycle, mean that the engine can more nearly approach the 'air-cycle' conditions in which there is assumed to be no increase of volumetric heat with temperature, combined with instantaneous heating of the air. The highest efficiency was obtained with a fuel-air mixture developing about 24 C.H.U. per cubic foot, as compared with about 47 or 48 C.H.U. which would be developed in the weakest petrol-air mixture to give efficient combustion. With the low heating value of the hydrogen-air mixture goes, of course, a low power output. The indicated M.E.P. at the maximum efficiency was about 77 lb. per sq. inch, as compared with a maximum obtainable M.E.P. of just about twice this for a non-detonating liquid fuel at the same compression ratio. The requisite ignition advance to bring the peak pressure, when burning hydrogen, to a point 12° after the dead centre, even at the weakest mixture in Burstall's experiments, was only 23°, and it varied steadily down to no more than 2° at the mixture 15 per cent. weak. This was for 7:1 compression. At 5:1 about 40° was required for the mixture 50 per cent. weak, and reference to fig. 31 will show that as compared with this, coal gas under the same conditions required about 70°.

ART. 47. Generalized efficiency referred to fuel or air. Gas engine

It was explained in art. 42 that as a matter of practical expediency it is sometimes advantageous to measure the thermal efficiency of an engine in terms of the air consumed, and that we might take the figure of 713 C.H.U. per standard cubic foot as an approximate calorific value for air. It was emphasized that with rich fuel-air mixtures the heat generated per cubic foot of air would differ, because some carbon would be burned to CO instead of to CO_2.

It is proposed now to carry this idea of the interchangeability of fuel and air much farther, and to examine the relationship between the two thermal efficiencies referred to the one and to the other. We shall begin with a gas engine, for in that it is easy to think of the fuel and air simply as two gaseous partners in the formation of the heated working substance; and except that one of them is present in a smaller proportion than the other (about 1:4 at the correct mixture) there is really no reason why we should regard one as the fuel any more than the other.

It is an important point in what follows, that as a rule about 70 per cent. by volume of coal gas consists of the diatomic gases hydrogen and carbon monoxide, which have the same volumetric heats as air, and that if, therefore, the gas-air ratio be greater than in the correct mixture, the excess of gas does not increase the mean volumetric heat of the mixture to be heated. At and beyond the correct mixture there will be a certain limited amount of CO_2 and H_2O formed, and the bulk of the rest will have the volumetric heat of the diatomic gases, even with a large excess of fuel. The condition of affairs in a petrol engine is very different, because the presence of unburned hydrocarbon vapour rapidly raises the mean volumetric heat of the cylinder contents.

Consider now an engine of compression ratio 5:1. The theoretical efficiency with ideal air as working substance would be 47·1 per cent. In other words this would be the efficiency if the volumetric heat of the working substance never rose above that of air at 15° C. and the conditions of instantaneous combustion and no heat loss were fulfilled. Now the volumetric heat would not increase if the temperature-rise in the cycle were very small. We can imagine, therefore, an engine using ordinary air with only a very small proportion of gas, so that the temperature-rise is trifling; and the engine would have the full air-cycle efficiency of 47·1 per cent. Equally well, we might argue that if the working substance be coal gas, with just a very little air to provide the oxygen necessary for a small rise of temperature, in the limit that engine, too, would achieve air-cycle efficiency.

We have disregarded the practical difficulty of promoting combustion when either gas or air is present in a very small proportion, but that need not upset our discussion of theoretical efficiencies any more than the extravagance of running an engine with a working substance composed almost entirely of hydrogen and carbon monoxide.

Now suppose that we construct a diagram, fig. 56,† of the same type as figs. 51 and 54, but extended so as to include all possible mixture strengths from 100 per cent. air on the left to 100 per cent. gas on the right. For the average coal gas of art. 15 the correct mixture is roughly at 80 per cent. air and 20 per cent. gas.

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† In this and the following article the figures and calculations given in the first edition of the book have not been revised. The discrepancies are slight, and unimportant for the purpose of these articles.

If we regard efficiency from the ordinary standpoint in terms of gas, then the efficiency of the engine will be 47·1 per cent. when the fuel-air mixture is 100 per cent. air; but in terms of air the efficiency will be zero under these conditions, for the engine would be using a great deal of air and doing no work. Similarly, at the extreme right of the diagram the efficiency in terms of gas will be zero, and in terms of air 47·1 per cent. At the correct mixture the efficiencies in terms of gas and air must be identical, because for that condition the calculated heat supply is the same from whichever of the partners in the combustion it is calculated. For the typical coal gas of art. 15 the energy at the correct mixture was 70,500 ft.-lb. per S.C.F. and the calculated efficiency 36·5 per cent., when allowance was made for the rise of the volumetric heat of the working substance with temperature, and for dissociation. If allowance had been made only for the rise of volumetric heats and not for dissociation, the calculated efficiency at the correct mixture would have been 38 per cent.

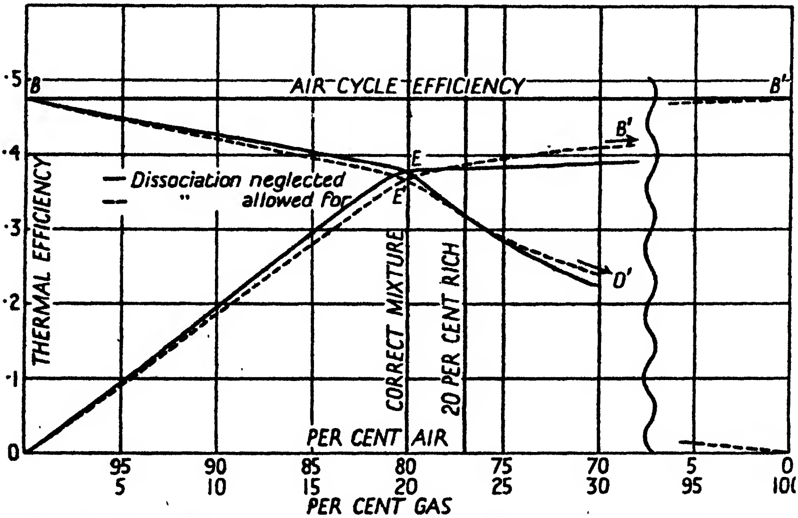

FIG. 56. Generalized efficiency curves referred to gas and to air for gas-air mixtures, from zero gas to 100 per cent. gas. Compression ratio 5:1.

Since the lines representing efficiencies in terms of gas and air must intersect at the correct mixture and must start from 47·1 per cent. and zero respectively, at 100 per cent. air, their general form must be as *BE* and *OE* in fig. 56, or *BE'* and *OE'* when allowance is made for dissociation. The part of the diagram representing gas-air mixtures with more than 50 per cent. excess of gas is of no practical interest, and has been omitted except for the last 5 per cent. on the right-hand side of the diagram; but there is no logical reason against connecting the points *E* and *E′* by smooth curves with *B′* and *O′*, for we have seen that the chemical events in the cylinder will be of just the same nature when there is a large excess of gas, and the volumetric heats about the same. To make the thing a practical possibility, all that would be required would be some means of getting the mixture to ignite, and to complete the combustion, when there is only a very little air present.

The exact shape of the lines must depend upon how the volumetric heats and the amount of dissociation vary with temperature. Calculation shows that the lines *BE* and *BE′* are very nearly straight, and that if one assumes that they are so, then *OE* and *OE′* are slightly convex upwards.† One can argue on grounds of continuity that there can be no sudden change of direction of the lines in going through the points *E* and *E′*, but on the other hand that there will be a point of inflexion at or near these points, where the temperature produced by the reaction reaches a maximum.

It is instructive to follow out exactly what would happen with mixtures richer than the correct, on the assumption that the gas contains only the diatomic gases H_2 and CO, which have the same volumetric heat as air. This will now be done, first on the assumption that no dissociation takes place, and then with allowance made for it.

If there is only the rise of volumetric heat to allow for, then the true thermal efficiency will always be the same when the heat developed per cubic foot is the same.‡ With rich mixtures air would be displaced by gas, and it follows that both heat and efficiency would be the same when the air had been reduced to, say, 80 per cent. of its correct-mixture value as they would be with a weak mixture in which the gas present was 80 per cent. of that in the correct mixture. A point in the line *EB′*, therefore, will be at the same height as a point in *EB*, at corresponding

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† *OE*, for example, will be represented by the equation

$$\text{efficiency} = (0{\cdot}473 - 0{\cdot}093x)\frac{0{\cdot}80x}{(1-0{\cdot}2x)},$$

where x expresses the gas-air mixture as a fraction of the correct mixture (for proof, see Appendix II, p. 289).

‡ In a rich mixture this assumes that selective combustion as between H_2 and CO does not occur. For if, in a rich mixture, all the H_2 is always burnt, at the expense of some of the CO, this will alter the ratio of CO_2 and H_2O in the products of combustion and will produce some alteration in the mean volumetric heat for the same range of temperature.

positions where the quantity of air and the quantity of gas are each the same fraction of the quantities in the correct mixture.

As regards the efficiencies referred to gas in the rich-mixture region, if y is the fraction of the air in a rich gas-air mixture compared with that at the correct mixture, then heat will be reduced in this proportion, and power not quite in the same proportion, owing to a lower maximum temperature and consequent higher efficiency. Although the heat actually developed per cubic foot would be reduced in the proportion y, the total heat value of the fuel supply would be largely increased and the thermal efficiency referred to gas would therefore have fallen right away. Thus, to take a numerical example, the results for mixtures 20 per cent. and 50 per cent. rich in gas would be as in table 24 and have been used in drawing the line EO'.

TABLE 24

Thermal efficiencies referred to gas for rich fuel-air mixtures. Dissociation neglected.

			Fractions of the same quantity at the correct mixture			
Mixture	*Gas*	*Air*	*Heat and approximate power*	*Fuel heat per cubic foot of mixture*	*Thermal efficiency referred to gas*	*Thermal efficiency referred to gas*
	per cent.	per cent.				per cent.
Correct	20	80	1	1	1	38·0
20% rich	23	77	0·96	1·15	0·835	31·7
50% rich	27	73	0·91	1·35	0·675	25·5

The effect of dissociation is, at the correct mixture, to prevent the full heat of combustion of the fuel from being developed and so to lower the maximum pressure, the power developed, and the thermal efficiency; the latter from 38 to 36·5 per cent. for the coal-gas mixture. With rich mixtures, were it not for dissociation, the power must immediately begin to fall off from the correct mixture onwards, as the air present and the heat generated are reduced. The effect of dissociation is to maintain, and even increase, the pressures developed, over and above those at the correct mixture. It was shown in art. 24 and fig. 13 that the calculated temperatures were higher for rich than for correct mixtures when dissociation was allowed for, and in a very beautiful series of closed-vessel explosion experiments with varying mixtures of CO, O_2, and N_2, Fenning and Tizard[8] have shown that when the proportion of O_2 and N_2 was nearly the same as in air, the highest pressures were developed when the proportion of CO was 20 per cent. rich as compared with the complete combustion mixture. The maximum pressure was then 1 per cent. above that obtained at the correct mixture, and even when the excess of CO was 50 per cent., the maximum pressure was only 1 per cent. below that at the correct mixture.

In these experiments the fuel was pure CO with only a trace of hydrogen, but we may assume similarity of behaviour in coal gas with sufficient accuracy for our general discussion, and on this assumption we obtain the figures given in table 25 for the efficiency variation of an engine with rich mixtures when dissociation has been allowed for. It has been assumed that the power obtainable from an engine would show the same variation with mixture strength as the pressures reached in Fenning's experiments. Burstall[32] has actually observed the variation of I.M.E.P. with mixture strength in an engine running upon pure CO as fuel. Fenning's figures have been taken for the calculations, because in closed-vessel experiments there can be no question of any variation of the volumetric or mechanical efficiency of the engine having affected the results; and one can say with confidence that the increase of the maximum pressures with rich fuel-air mixtures was due to dissociation alone, and to nothing else. Burstall's engine experiments confirm Fenning's results very satisfactorily. They show a rise of I.M.E.P. of 1·3 per cent. between the correct mixture and one 10 per cent. rich.

TABLE 25

Thermal efficiencies of rich gas-air mixtures referred (1) *to air and* (2) *to gas, when dissociation has been allowed for.*

			Fractions of the same quantity at the correct mixture					*Thermal efficiency*	
Mixture per cent. of correct	*Gas*	*Air*	*Approximate power (from Fenning's experiments)*	*Heat from air present*	*Thermal efficiency referred to air*	*Heat from gas present*	*Thermal efficiency referred to gas*	*Referred to air*	*Referred to gas*
	per cent.	per cent.							
100	20	80	1·00	1·00	1·00	1·00	1·00	36·5	36·5
120	23	77	1·01	0·96	1·05	1·15	0·88	38·4	32·0
150	27	73	0·99	0·91	1·09	1·35	0·735	40·0	27·0

The figures of table 25 have been used in drawing the lines *E′B′* and *E′O′* in fig. 56. It will be seen that by the time the mixture is 20 per cent. rich the calculated efficiency has already risen above the line *EB′*, which gives the calculated values for no dissociation, and that as compared with the value at the correct mixture it has risen from 36·5 to 38·4 per cent.

These deductions as to a rise of efficiency referred to air, in a gas engine operating with a fuel-rich mixture, are more than confirmed by experiment. In fig. 57 the values obtained by Burstall[20] at compression

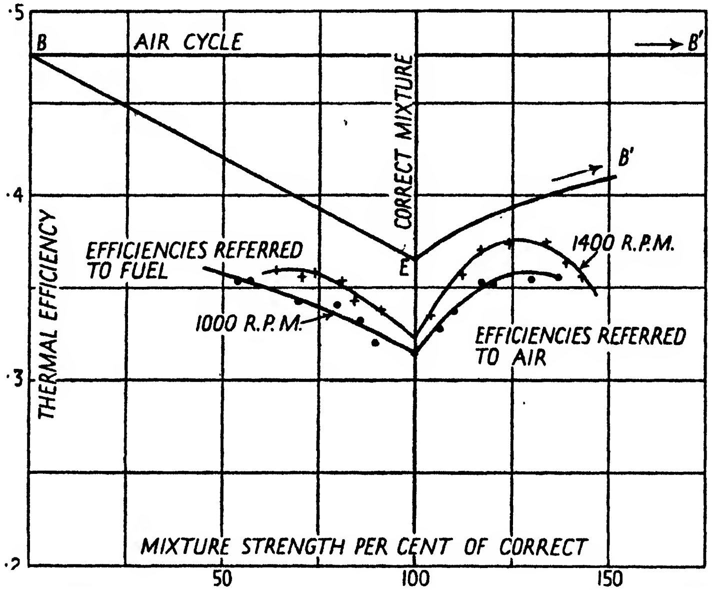

FIG. 57. Observed thermal efficiencies of a gas engine (Burstall) on weak and on rich mixtures, referred to gas and to air.

ratio 5:1 and at speeds of 1,000 and 1,400 r.p.m. are shown by dots and crosses respectively. To the left of the correct-mixture line (weak mixtures) the observed efficiencies are those referred to gas in the usual way; while to the right of the 100 per cent. line they are the figures referred to air. The observed rise of efficiencies on each side of the correct mixture reproduce very nicely the shape of the theoretical line *BEB'*.

The general shape of this line, with its point of minimum efficiency at the correct fuel-air mixture, reflects the rising temperature and increasing amount of dissociation as the correct mixture is approached from either side. As we approach from the left the full heat of the fuel, which forms the denominator in the efficiency calculation, increases in proportion to the fuel present, that is, to the distance from *B*; but, owing to the increase of volumetric heat and of dissociation, a less and less proportion of this total heat can be converted into work, and the efficiency falls. As soon as we pass to the right of the correct mixture along *EB′* the full heat based upon the assumed complete combustion of the *air* present begins to diminish at once, in proportion to the reduced air quantity, but unlike a point on *BE*, near to *E*, there is actually more heat developed, and also a greater volume ratio, at points just to the right of *E*, and both these results of dissociation go to increase the calculated and observed power in spite of a diminution in the heat calculated as available on the basis of the air present. Dissociation would therefore lead us to expect a more rapid rise of efficiency as we move away from the point *E* to the right, than if we move away to the left, through a distance corresponding to the same percentage reduction in the heat calculated as available from the air and the gas respectively. At the mixture defined as 120 per cent. of the correct one, the heat from air present has diminished by 4 per cent. (see table 25) and the efficiency has risen from 36·5 per cent. to 38·4 per cent. (calculated) or from about 32 to 37 per cent. (observed). On the weak-mixture side the heat calculated as available will be the same when the fuel is 0·96 of its correct-mixture proportion, and the efficiencies, calculated and observed, can be seen from the diagram to have risen only from about 36·5 to 37 per cent., and from 32 to 33 per cent. respectively.

The experimental figures show a rise of efficiency on each side of the correct mixture noticeably greater than the calculated figures, an effect which is probably due mainly to the fact that the fuel-air proportions given by Burstall are for mixtures *entering* the cylinder. According as mixtures are weak or rich there will be oxygen or fuel gas left over in the cylinder from the last stroke, and in consequence the actual mixtures in the cylinder will be weaker and richer respectively than the mixtures entering. On the rich-mixture side the increase of efficiency may also be helped, for the reason given below, by selective combustion in favour of the hydrogen present. It was scarcely to be expected, in any case, that the figures of table 25 would give more than a qualitative guide as to what is to be expected in an engine burning coal gas; being based, as they are, upon closed-vessel experiments with pure carbon monoxide as the fuel. In his engine experiments with pure CO Burstall observed a rise of efficiency almost exactly following the calculated curve of fig. 57. His figures are 27·5 per cent. at the correct mixture, rising to 29·5 per cent. with a mixture 20 per cent. rich.

Both theory and experiment, therefore, have brought out the point which was made in art. 42, that the rough constancy of thermal efficiency, referred to air, in a petrol engine with rich mixtures was a chance one, and that in a gas engine this efficiency referred to air would be found to rise when the proportion of gas to air was greater than in the correct mixture. This increase of thermal efficiency referred to air serves also to explain the increase of power observed by Burstall with rich gas-air mixtures; for in a mixture 20 per cent. rich in gas, the air proportion has only dropped to 95 per cent. of its correct mixture value, while the efficiency has increased from 36·5 to 38·4 per cent. An increase of power is therefore to be expected equal to a little over 1 per cent. on the basis of the carbon-monoxide experiments, and, as stated above, the greater increase than this which was actually observed may very possibly be traced to the fact that in mixtures rich in coal gas all the hydrogen is burnt in preference to the carbon monoxide. Reference to the expressions for the dissociated products of combustion in art. 24 will show that this would lead to there being less dissociation at the maximum temperature, and it would also mean a rather lower mean volumetric heat of the products of combustion (see table 10, p. 87).

ART. 48. Generalized efficiency for the petrol engine

The behaviour of the petrol engine, when considered on lines similar to those of the last article, shows up very differently. Imagine as before an engine of compression ratio 5:1, and an efficiency diagram for it, fig. 58, similar in type to fig. 56. We assume that combustion presents no difficulty and that, however extreme the fuel-air ratio may be, the normal chemical reactions proceed between the oxygen and the hydrocarbon according to the quantities of fuel and air present.

To the left of the correct-mixture line the diagram is very much the same as fig. 56, except that the proportion by volume of fuel vapour to air at the correct mixture is about 2 per cent. instead of 20 per cent. The two lines *BE′* and *OE′* will, as with coal gas, intersect at the point where the efficiency is that calculated for the correct mixture strength after allowing for dissociation. As before, we can imagine an engine in which the working substance is all fuel vapour, or, for the sake of being more definite, let us say heptane vapour; this being typical, as regards molecular weight, of an average petrol. As with coal gas, we can argue that when there is very little, and in the limit no, air present, the efficiency referred to the fuel will be zero. In other words, the line

$E'O'$ falls away much as in fig. 56. When we come to the efficiency referred to air, however, there is a difference in the region of rich mixtures. It was argued, in the case of the gas engine, that the efficiency would rise to the air-cycle value of

$$1-\left(\frac{1}{r}\right)^{\gamma-1},$$

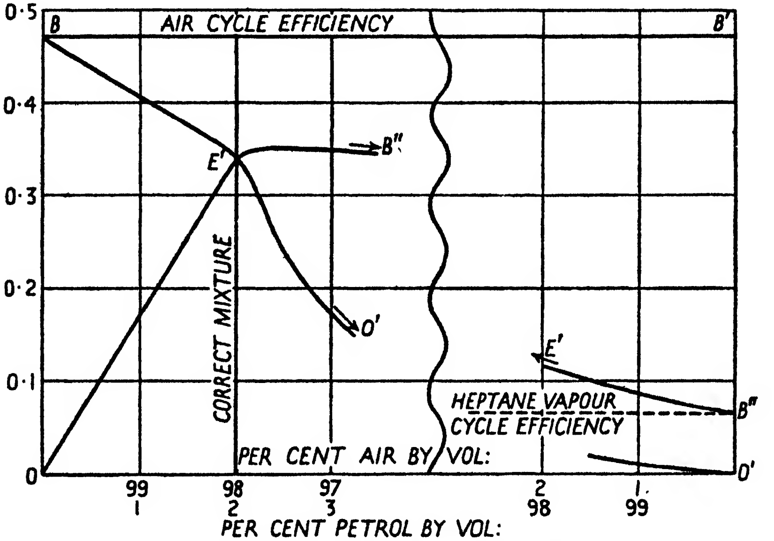

Fig. 58. Generalized efficiency curves referred to fuel and to air for a petrol engine. Compression ratio 5:1.

in the limit when there was no air present and the working substance was all coal gas; but this was because coal gas was largely composed of hydrogen and carbon monoxide, and its volumetric heat could be taken to be the same as that of air. Otherwise the γ in the efficiency formula would have been very different.

When we imagine an engine in which the working substance is, in the limit, entirely heptane vapour, this vapour will have a volumetric heat of 48 C.H.U. per mol as compared with 5·0 C.H.U. per mol for air. The value of γ will therefore be 1·041, and the efficiency of the 'heptane-vapour cycle' will be

$$1-\left(\frac{1}{5}\right)^{0\cdot041} = 0\cdot066.$$

It follows that the line $E'B''$, after rising at first, will fall again later, on account of the rising volumetric heat of the working substance as this gets richer and richer in heptane vapour, until finally, at the extreme right of the diagram, it is at the level of an efficiency 0·066.

The rise of temperature on combustion with rich mixtures due to dissociation, as illustrated in fig. 13, combined with the increase in the volume ratio under these conditions, enables a rough estimate of the increase of power during the first 20 per cent., or 50 per cent., enrichment of the mixture to be made. Figures for this are given for heptane-air mixtures in table 26, together with the heat available as calculated from the air present, and the efficiency referred to air. The heat calculated from the air present is practically constant, because the fuel volume only increases from 2 per cent. to 3 per cent. of the total, even in the mixture 50 per cent. rich.

TABLE 26

Thermal efficiencies referred to air for a petrol engine with rich fuel-air mixtures. Dissociation allowed for.

Mixture per cent. of correct	*Maximum temperature reached*	*Volume ratio* σ	*Fractions of the same quantities at the correct mixture*			*Thermal efficiency referred to air*
			Approximate power (from columns 2 and 3)	*Heat from air present*	*Efficiency referred to air*	
	° C.					
100	2,591	1·085	1·00	1·000	1·00	33·8
120	2,603	1·11	1·03	0·996	1·03	34·9
150	2,366	1·185	1·01	0·99	1·01	34·1

It will be seen from the last column of the table that by the time the mixture is 50 per cent. rich the efficiency has already begun to fall, and from that point it will steadily diminish, a fall in temperature being more than compensated by a rise of volumetric heat of the working substance, on account of admixture with heptane vapour. Finally, as the volumetric heat approaches that of pure heptane vapour, the efficiency falls right away, to the value 6·6 per cent., as indicated by the line from E' which finishes at the point B''.

The rough constancy of air consumption per I.H.P. hour, observed by Ricardo, of which figures were given in table 23, is now seen to correspond to the level portion of the efficiency curve $E'B''$ immediately to the right of the point E'. Reference to table 23 will show that what has been referred to as a rough constancy of air per I.H.P. hour, or thermal efficiency referred to air, was in fact a small rise up to a mixture

20 per cent. rich, followed by a fall at 35 per cent. rich, so that after comparison of the last columns of tables 23 and 26 it may fairly be said that the agreement between experiment and theory is substantially complete.

ART. 49. Thermal efficiency as affected by size and speed

With engines of very different cylinder size the most appropriate design to be adopted may well vary in almost every detail, and it therefore becomes difficult to obtain comparable figures for the effect of size on thermal efficiency; for unless the cylinder details are closely alike, and the speeds nearly so, other factors will certainly enter in and mask any difference of performance due to size alone. The main difference we may expect to find is a reduction of heat loss to the cylinder walls in large sizes on account of a reduction in the surface-volume ratio of the combustion space. It becomes at once clear that the shape of the combustion space must be similar in two engines to be compared, and that it must also be as compact and symmetrical as possible, for otherwise there may be little relation between the heat loss and linear dimensions. Apart from the ratio of surface to volume, the rate of heat loss will depend very much on the amount of scouring action of the hot gases over the cylinder-wall surface, that is, upon their turbulence, which in its turn will increase with the crankshaft speed in similar designs.

An instructive series of tests was carried out by a committee of the Institution of Civil Engineers[23] in the years 1903–6, of which a full account is given by Sir Dugald Clerk in his book.[24] Three gas engines were tested, of which the leading particulars, as well as the experimental results obtained, are given in table 27.

TABLE 27

Engine	*L*	*R*	*X*
Bore and stroke, inches	5½ × 10	9 × 17	14 × 22
r.p.m.	259	204	166
Mean piston speed, feet per min.	430	580	610
Compression ratio	5·67	5·54	5·38
Ratio surface/volume of combustion space . .	1·625	0·975	0·674
Ratio gas/air drawn in	1:9·15	1:9·17	1:8·21
Brake M.E.P.	67	75	74·5
Indicated M.E.P.	80	88	86·5
Gas per B.H.P. hour (in cubic feet at the working temperature)	16·9	15·8	14·9
Indicated thermal efficiency	31·0	32·9	34·8
Ratio to air cycle	0·62	0·66	0·71

The engines were all built by the National Gas Engine Co. and were of the well-known horizontal type. They were all generally similar as regards cylinder design and shape of combustion space, and although there was a range of variation of some 50 per cent. in the revolution speeds, the mean piston speeds were much more on a level, so that gas velocities through the valves, and turbulence in the cylinders, may not have been very different.

With the gas used in these trials, the ratio of gas to air, in a mixture giving complete combustion, was about 1:5, so that all the tests summarized in table 27 were made upon very weak mixtures. This might have been deduced from the low values of the recorded mean effective pressures, and also from the fact that the committee state that the brake horse-powers given were those found to correspond with the most economical result in each engine. We should expect the engines therefore to be working at the left-hand end of the efficiency curve in fig. 50; at about 61 per cent. of the correct mixture, in fact, for engines *L* and *R*, and 68 per cent. for engine *X*.

At these mixture strengths the indicated thermal efficiency actually obtained by Burstall was 36·0 per cent. at 5:1 (see fig. 50) and at the compression ratios of these engines it would have been 37·8 per cent., so that the efficiencies obtained were very low, judged by modern standards. The ignition timing is not stated, and one may surmise it was not the most suitable from the point of view of efficiency. The shape of the combustion chambers, which was by no means compact, as well as the much lower speeds of revolution, must both have tended to exaggerate the heat loss from the combustion space during combustion and expansion. For reasons to be explained on p. 272, however, it requires a large alteration of the heat loss to the cylinder walls to produce any considerable effect upon the thermal efficiency, and the difference between the efficiencies of the *L*, *R*, and *X* engines and those obtained by Burstall must be regarded as too great to be accounted for in this way. We are justified in concluding, therefore, that a more suitable ignition timing would have led to some improvement. If the timing was the same in all three engines, then some of the difference between *L* and *X*, 31·0 to 34·8 per cent., may be accounted for by the fact that the flame would have to travel, in engine *L*, only 40 per cent. of the distance it would have to in engine *X* before reaching the cold cylinder walls, and unnecessary heat loss must have been incurred in engine *L* which could have been avoided by a later ignition timing.

These tests of the Institution of Civil Engineers Committee, although carried out so long ago, are still of great interest and value. They were most accurately made, and the only point on which they are open to criticism is in regard to the mechanical efficiency determinations. The committee fully realized that these were unsatisfactory, but in 1904 the swinging field electro-dynamometer for measuring mechanical losses by motoring had not been developed, and the indicator was the only means available for arriving at indicated power.

Ricardo[25] has given some further comparative results on three more modern petrol engines at high speeds, of which particulars are given in table 28.

TABLE 28

Engine	*A*	*B*	*C*
Bore and stroke, inches	3¼ × 4	4½ × 8	8 × 11
r.p.m.	1,750	1,750	1,250
Mean piston speed, feet per min.	1,166	2,333	2,290
Compression ratio	4·84	4·84	4·84
Ratio surface/volume of combustion space	3·41	2·12	1·05
Brake M.E.P.	114	126	135
Indicated M.E.P.	137·5	146·5	153
Fuel consumption lb. per B.H.P. hour	0·55	0·51	0·48
	per cent.		
Indicated thermal efficiency	29·6	30·8	32·0
Efficiency ratio to air cycle	63·5	66·3	68·9
Efficiency ratio to theoretical limit	86·0	89·0	92·5

The air-cycle efficiency is 46·5 per cent., and the 'theoretical limit' of art. 44 for this ratio, and a mixture 5 per cent. weak, is easily found by proportion between curves *CD* and *EF* of fig. 52 to be 33·7 per cent.

The combustion chamber in each of these engines was as compact as it is possible to have it, with valves in the head, although these were not similar in number and disposition. It will be seen that the piston speeds differ widely, and the ratios of valve area to piston area were so adjusted that, at the speeds stated, the mean gas velocity through the inlet valves was the same for all engines, namely 140 feet per second.

Each engine was calibrated with a fuel-air mixture about 5 per cent. weak and with ignition adjusted for maximum power. Slightly higher thermal efficiencies could therefore have been obtained in each case, but the above tests are quoted as being about as near strictly comparable as it is possible to achieve. It will be seen that there is a definite rise of efficiency with increase of size, which may be put down mainly to the reduction of surface-volume ratio of the combustion space, since turbulence must have been very similar in the three engines. The rise of efficiency is only 2·4 per cent., as against 3·8 per cent. between engines *L* and *X*, although the reduction of surface-volume ratio is considerably greater in the former case, which points to the rise of efficiency between engines *L* and *X* having been due to something more than a simple matter of size.

It is interesting to compare the heat balance-sheet figures, which are available for engines *A* and *B*, with those for the engines of table 27. The comparison may be made from table 29.

TABLE 29†

Comparative heat balance-sheets for the engines of which details are given in Tables 27 and 28.

Engine	*A*	*B*	*L*	*R*	*X*
	per cent.	per cent.	per cent.	per cent.	per cent.
Heat to I.H.P. . .	29·6	30·2	31·8	33·3	34·7
Heat to jacket water .	29·5	27·0	34·1	29·6	25·4
Heat to exhaust, etc. .	40·9	42·8	34·1	37·1	39·9

The compression ratio of the gas engines was higher, and they were working on a 40 per cent. weak mixture. The temperatures throughout their cycles must therefore have been much lower, and the proportion of the total heat supply which goes to I.H.P. (the indicated thermal efficiency) is higher. In spite, however, of the very much higher temperatures in the small, fast-running, petrol engines, and of their far higher degree of turbulence, it will be seen that the proportion of the whole heat which goes to the jackets is less than in all but the very large gas engine *X*. This, of course, is a reflection of their much higher speeds of revolution.

With a change in revolution speed of the order of 1:10, as we have between these two sets of engines, there must of course be some reduction of the proportionate heat flow to the cylinder walls during combustion and expansion; but when we pass from such generalities to a more careful analysis of the effect of speed upon thermal efficiency in engines of a similar type, we are immediately faced by a host of difficulties It is impossible to make a direct experimental comparison showing the effect of speed, in which this is the only variable, for the optimum performance at any one speed depends so much on the adjustment of ignition timing, mixture strength, and valve setting, for that particular

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† In connexion with the form of tables 29 and 30, see p. 271.

speed, that the results obtained by varying the speed alone would be quite meaningless. Variations of power and therefore of efficiency due to inappropriate ignition timing alone would quite mask any differences of heat loss due to the variation of speed. Ignition timing, it is true, could be adjusted for maximum power at each speed, but not so the valve setting, which is only slightly less important, or the mixture strength. And apart from these difficulties, the effect of a change of speed upon turbulence, and heat loss from the combustion chamber, would be different for different engines, so that no results of general applicability would be obtained.

Fortunately, owing to the opposite effects of a reduction of time, and of an increase of turbulence upon the heat loss, the effect of speed upon thermal efficiency seems to be much smaller than was formerly supposed, and accurate analysis is not important.

Some figures given by Ricardo in his book are quoted in table 30,

TABLE 30

Showing the effect of a change of speed upon the heat balance-sheet for a petrol engine.

Bore and stroke, $4\frac{1}{2} \times 8$. Compression ratio, 3·8:1.

r.p.m.	975	1,500	1,700
Mean piston speed	1,300	2,000	2,266
Heat to I.H.P. per cent.	25·9	26·1	26·1
Heat to cooling water	30·4	28·0	27·0
Heat to exhaust, radiation, etc. . . .	43·7	45·9	46·9

from which it will be seen that the improvement of thermal efficiency for an increase of speed of 75 per cent. is almost negligible. Burstall[20] has given the curve shown in fig. 59 for the variation of thermal efficiency of the same design of engine, running on coal gas, from which it will be seen that a variation from 20 per cent. below to 20 per cent. above the design speed of 1,500 r.p.m. produces a change of 0·75 per cent. in the thermal efficiency. The falling away at speeds below 1,000 r.p.m. must not be taken seriously, for at those speeds the valve setting would have become unsuitable. Burstall's curve was only arrived at after a number of corrections had been applied to his observations to make it represent comparable conditions, and some of these corrections, he points out, are open to criticism.

There is a subsidiary effect of increased turbulence at high speeds which may become of importance in a petrol engine and is worth mentioning. That is, the effect of increased scouring during the compression stroke, in removing unevaporated fuel which may be hanging about on, or near, the cool cylinder walls. With fuels of low volatility the effect may be appreciable. In a gas engine of course it will be non-existent.

The figures of table 30 show a decrease of heat loss to the jackets, over the range of speed represented, amounting to 3·4 per cent. of the whole heat input. It must be remembered, however, that the only heat loss which affects thermal efficiency is that during combustion and expansion. This may be taken as about 12–15 per cent. of the total heat supply (see art. 63) as compared with the 30·4 to 27 per cent. to the jackets of table 30. A reduction of the heat to the jackets in the proportion observed means a reduction of the loss during combustion and expansion of only

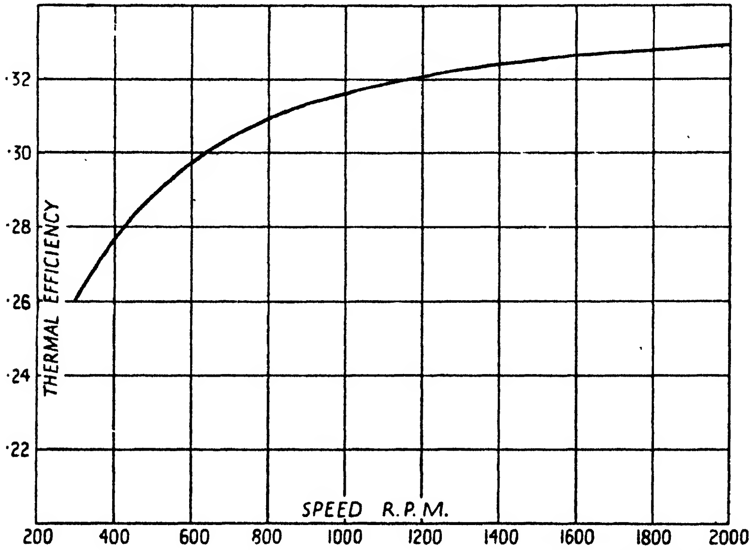

Fig. 59. Curve showing the effect of speed variation upon thermal efficiency in an engine of normal speed 1,500 r.p.m. Compression ratio 5 : 1. Correct mixture (Burstall).

$$3{\cdot}4 \times \frac{12}{30{\cdot}4} = 1{\cdot}12 \text{ per cent.}$$

This proportion of the total heat supply becomes a gain in the heat available for conversion into work, and since the average efficiency of conversion is about 0·3, the percentage gain of indicated work (that is, the gain of thermal efficiency) will be

$$1{\cdot}12 \times 0{\cdot}3 = 0{\cdot}33 \text{ per cent.}$$

A 3·4 per cent. reduction of heat loss to the jackets, therefore, gives an increase of only 0·33 per cent. in the thermal efficiency; which is in reasonably good agreement with the observed figure of 0·2 per cent. in table 30, and confirms Burstall's result in showing how small an effect on the efficiency is to be expected from a large change of speed.

As a general conclusion, therefore, upon the effect of increased speed on thermal efficiency, it may be said that, although with the large changes of speed represented by the difference between a 'slow-running' and a 'high-speed' engine, the reduced time available for heat loss to the cylinder is important, on the other hand, an increase or decrease of speed as much as 20 per cent. above or below the normal speed of an engine will have a very small effect on efficiency, and that such effect as there may be is brought about by a balance of factors incidental to a change of speed, rather than by any influence due to a simple change of speed by itself.

Art. 50. Thermal efficiency as affected by the type of fuel

We exclude from this article all fuels of the 'heavy-oil' type, which are always employed in compression-ignition engines. Among the volatile liquid fuels for spark-ignition engines, the most important distinguishing factor is that of liability to detonate. Apart from this, one may say that, with one exception, there is nothing to choose between them all from the point of view of efficiency. Burstall showed that slightly higher thermal efficiencies could be obtained with coal gas than with petrol on the same engine under comparable conditions, probably for a reason referred to in the last article, namely, that with a gaseous fuel the more homogeneous mixing of the fuel and air prevents any waste through incomplete combustion in the neighbourhood of the cylinder walls. The difference, however, is not large enough to be of any importance in a choice between the two fuels, compared with other factors which must necessarily enter in. Of all the liquid fuels tested by Ricardo the only one which can be differentiated from the rest is alcohol.

Fig. 60 shows observations made upon a variety of fuels at different compression ratios plotted about a mean curve. The differences observed were scarcely outside the margin of experimental error.

The exceptional behaviour of the engine with alcohol was not due to any special combustion characteristics of the fuel, but solely to its exceptionally high latent heat of evaporation. Reference to table 39 on p. 284 will show that the latent heat of ethyl alcohol is about two and a half times that of an average petrol, and in consequence the liquid, although highly volatile (its boiling-point is 78° C.), does not become fully evaporated before the inlet valve closes. The evaporation process is completed during compression, and the amount of heat absorbed is so large as to produce a general lowering of temperatures throughout the cycle. This is equivalent to working with a weak mixture in so far as the lower temperatures of the cycle reduce the detrimental effect of increasing volumetric heats upon the thermal efficiency. When running upon alcohol at full load it is noticeable that exhaust valves remain far cooler than on petrol. They continue to look black under conditions when with other fuels they would be glowing a good cherry red.

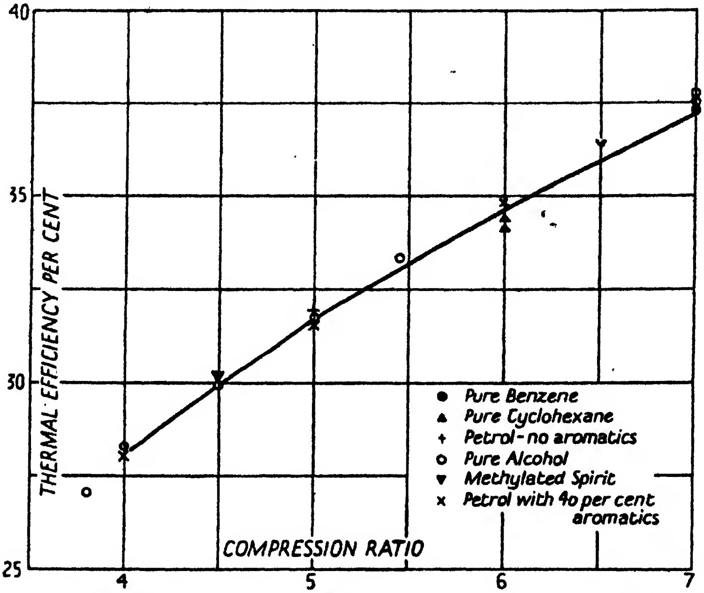

FIG. 60. Observed thermal efficiencies with a variety of liquid fuels at various compression ratios. Speed 1,500 r.p.m.

If compared at the same compression ratio of 5:1, the best efficiency obtainable with alcohol is about 2 per cent. higher than the best with an average petrol, on account of the lower cycle temperatures. But alcohol shares with coal gas the distinction of being a non-detonating fuel. If advantage is taken of this, so that thermal efficiencies with an average petrol and with alcohol are compared at the highest compression ratio which each will stand, taking this as 7·5:1 for alcohol, then the advantage of the latter fuel may be of the order of 20 per cent. Its advantage over benzol on the same basis would not be more than about 8 per cent., for benzol can be used without detonation up to very high compression ratios. Benzol, however, has shown itself liable to dangerous pre-ignitions under conditions in which an engine using coal gas or alcohol would run safely and steadily.

The great drawbacks to alcohol as a fuel are its low calorific value and high price. The latter could no doubt be reduced by large-scale production, but reference to table 39 will show that its calorific value is no more than about two-thirds of that of petrol and as a consequence, in spite of the very high thermal efficiency obtainable, the weight of fuel consumed per horse-power is very large. Even when full advantage is taken of the high compression ratio at which it can be used, the fuel consumption in a single-cylinder engine is not less than about 0·56 lb. per indicated horse-power hour as compared with 0·40 for a good petrol at its highest useful compression ratio.

ART. 51. Thermal efficiency as affected by detonation

We have seen in the last article that all volatile liquid fuels with the exception of alcohol, if they are tested at the same compression ratio, yield about the same thermal efficiency. Quite a number of fuels, however, cannot be tested at all above compression ratios of 5:1 or even 4·5:1 on account of detonation, and any comparison which leaves out of account this question of liability to detonate is ignoring what should be the main distinguishing factor in settling the value of a fuel.

Since compression ratio is the chief factor which influences both power output and efficiency, the basis on which to compare two fuels must be the performance of an engine in which this ratio is adjusted to get the optimum performance with each. The method of H.U.C.R. values was developed by Ricardo to express the quality of fuels from the point of view of detonation. As explained in art. 29 the compression ratio at which a fuel begins to detonate depends not only upon the engine employed for the test, but also on the speed, the fuel-air ratio, the ignition timing, and the temperature of the ingoing mixture. The H.U.C.R., therefore, can be no more than a relative figure which is useful for comparing fuels under similar conditions. Ricardo has standardized the conditions of the test with his 'E. 35' variable-compression engine as being 1,500 r.p.m., with heat added to the ingoing air to the amount of 36 C.H.U. per minute. The reason for this latter condition is to facilitate evaporation of the less volatile fuels. H.U.C.R. is then defined as the highest ratio at which the engine will run on full throttle without detonation, when the fuel-air ratio and ignition timing are adjusted for maximum power output.

The importance of comparing fuels by testing each at its H.U.C.R. was brought out in the last article in connexion with alcohol, and to illustrate the principle more widely there are given in table 31 the thermal efficiencies and power outputs, expressed as indicated mean effective pressure, for a selection of the fuels given by Ricardo, firstly when all are tested at 5:1 compression ratio (columns 2 and 3) and secondly (columns 5 and 6) when each is tested at its H.U.C.R.

TABLE 31

Power outputs and thermal efficiencies obtained with different fuels when tested (a) all at compression ratio 5:1 and (b) each at its H.U.C.R.

Fuel	*Tested at compression ratio 5:1*			*Tested at H.U.C.R.*	
	Maximum I.M.E.P.	*Thermal efficiency*	*H.U.C.R.*	*Maximum I.M.E.P.*	*Thermal efficiency*
Petrol *D*	131·2	31·9	5·35	134	33·1
Petrol *G*	131·0†	31·7†	4·7	128·6	30·7
Benzene	131·6	31·8	6·9	146·5	37·2
Alcohol (95% pure) .	142	32·5	>7·5	161·5 (at 7·5:1)	40·5

† These values could not be obtained experimentally at 5:1 owing to detonation, but have been deduced from experiments at lower ratios in accordance with the known rate at which power and efficiency are affected by the change.

The two petrols '*D*' and '*G*' show a difference, when tested at their H.U.C.R.s, of 4·2 per cent. in power and 7·8 per cent. in efficiency, while if they had each been tested at the H.U.C.R. of '*G*', namely 4·7:1, no difference would have shown up at all. The advantage exhibited by benzene and alcohol tested at their H.U.C.R.s is much greater, and puts them in a class by themselves as compared with an average petrol.

The apparent reduction of power of an engine when running upon a badly detonating fuel is in practice much greater than appears from table 31; for in climbing a hill, low speed is often combined with a fully open throttle, and in these circumstances the onset of violent detonation with an inferior fuel is much accelerated. Every motorist knows that when his engine begins to knock, this must be stopped by partially closing the throttle. But any closing of the throttle means a lower mean effective pressure and a large reduction of power, unless speed can at the same time be increased by a gear change.

The effect of detonation upon the power which an engine can give may be very clearly brought out by an experiment with the variable-compression engine, of which typical results are illustrated in fig. 61.

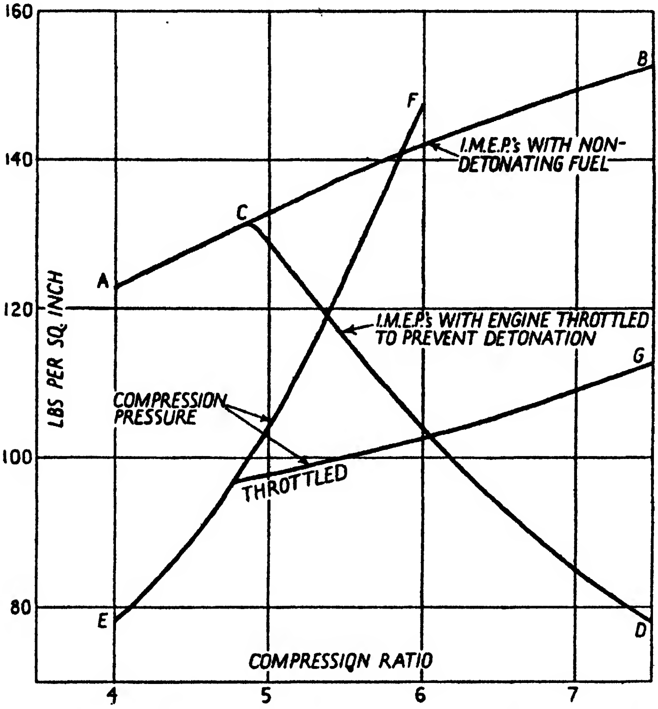

FIG. 61. Indicated mean effective pressure obtainable at different compression ratios with a non-detonating fuel, and with detonation prevented by throttling.

The engine is run throughout at constant speed, and let us suppose that at first a non-detonating fuel such as benzol is used. A series of runs at full throttle are made with compression ratios from 4:1 upwards, and the corresponding M.E.P.s obtained are plotted on the curve *AB*. Next, a fuel is used which begins to detonate at 4·8:1. At this point, *C* on the diagram, the engine is giving an I.M.E.P. of 132 lb. per sq. inch. If now the ratio is raised further, in order to prevent detonation we shall have to close the throttle more and more, so that as the ratio gets higher, the power obtained falls rapidly away along the curve *CD*. At 6:1 for example the engine must be throttled until its I.M.E.P. is only 104 lb. per sq. inch, and at 7:1 it is only 85 lb. per sq. inch.

The curves *EF* and *EG* are interesting as showing that when the throttle is closed to prevent detonation, the quantity of fuel-air mixture is reduced to a degree which keeps the compression pressure, as shown by *EG*, pretty nearly constant as the ratio increases. *EF* shows the rapid increase of compression pressure under full-throttle conditions.

Since the throttle was closed always to a degree just sufficient to prevent detonation, one might be tempted to conclude, from the rough constancy of the compression pressure, that this was a dominating factor in promoting detonation. There is, however, another result of raising the compression ratio which is now recognized to be of much more importance than a change of pressure or temperature. As the ratio is increased from 4:1 to 7:1 the proportion of exhaust gas, containing CO_2 and H_2O of high specific heat, which is left to mix with the incoming charge, drops from 10 per cent. to under 5 per cent. (see table 20), and the change in the volumetric heat imparted to the cylinder contents by this varying admixture of exhaust gas has a large effect on the flame temperature produced by combustion: a change of 1 per cent. by weight in the quantity of exhaust diluent will raise or lower the flame temperature by as much as 20–25° C., and reference to table 20 (p. 143) shows that this is equal to the change of temperature produced by an alteration in the compression of a whole ratio from 4:1 to 5:1. The same table shows that a rise from 4:1 to 5:1 in compression ratio reduces the amount of exhaust diluent from 10 to 7½ per cent. The effect of a rise of this amount in the ratio, therefore, acting through a change in the exhaust diluent, will affect the flame temperature to an extent 2½ times greater than any direct effect through a rise of compression temperature or pressure.

When detonation was prevented by throttling during the rise of compression ratio from 5:1 to 6:1, in the experiment illustrated in fig. 61, the quantity of residual exhaust gas diminished, according to table 20, from 7·5 to 6 per cent. by weight of what the ingoing charge would have been at full throttle; but under throttled conditions the I.M.E.P., according to fig. 61, fell from 132 to 104 lb. per sq. inch, which is very nearly the same ratio of reduction as the quantity of residual exhaust. To a first approximation I.M.E.P. must have been proportional to the quantity of fresh mixture drawn in, and it follows that the conditions at the two ratios, in each of which detonation was just beginning, corresponded to equal proportions of residual exhaust products in the cylinder contents.

Ricardo also demonstrated the influence upon detonation of artificially adding cooled exhaust gas through the carburetter, to a detonating fuel-air mixture, as the compression ratio was raised.

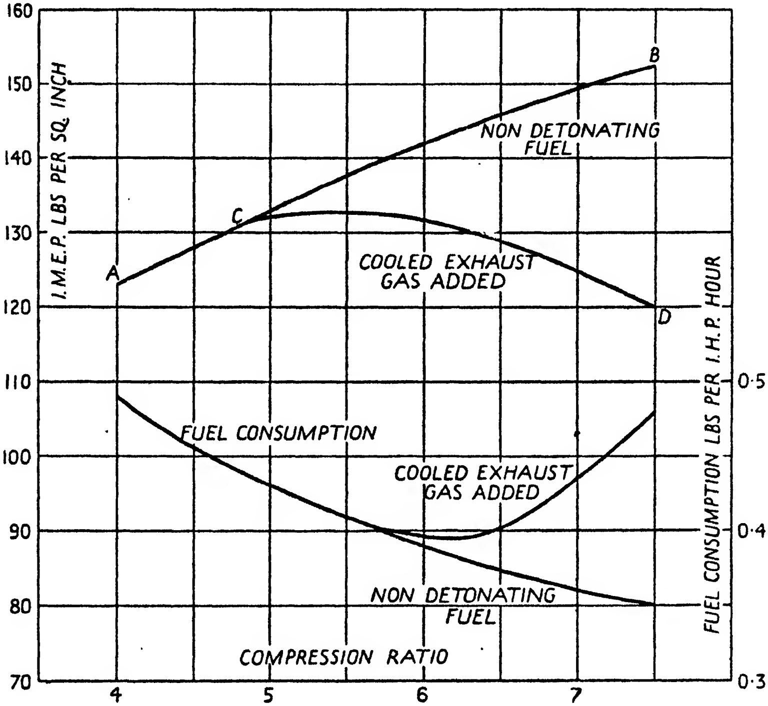

FIG. 62. Indicated mean effective pressures and specific fuel consumptions obtainable at different compression ratios with a non-detonating fuel, and with detonation prevented by adding cooled exhaust gas to the air entering the cylinder.

The curve *AB* in fig. 62 is the same as in fig. 61 and under normal conditions detonation started, as before, at 4·8 : 1. When cooled exhaust gas was added, however, in just sufficient quantity to prevent detonation at each ratio, it was possible to obtain the I.M.E.P.s shown by the curve *CD*. These are lower than those with a non-detonating fuel, because an increasing amount of effective fuel-air mixture is displaced by the diluent. Nevertheless the mean pressures obtained are much higher than those of fig. 61 when detonation was suppressed by throttling: 132 lb. per sq. inch as against 104 at 6:1, and 125 as against 85 at 7:1.

The addition of the exhaust diluent enables the compression ratio to be raised at once, with the detonating fuel, from 4·8:1 to 6:1 without any loss of power whatever; and at the same time brings about a reduction of fuel consumption from 0·44 to 0·39 lb. per I.H.P. hour, giving a gain of thermal efficiency of nearly 11 per cent.

At the higher ratios the quantity of exhaust diluent necessary to suppress detonation becomes so large that good combustion, and hence thermal efficiency, suffer, and the curve of fuel consumption per horsepower rises more and more as the mean effective pressure falls off.

ART. 52. Thermal efficiency of the compression-ignition engine

The problem of achieving a high efficiency in the compression-ignition engine is very different from that in a petrol engine. A high compression ratio in a petrol engine is not essential, but is desirable from the points of view both of power and economy, within the limits imposed by detonation. In the compression-ignition engine a ratio of at least 10:1 is essential to ensure ignition, and the problem becomes that of so controlling the injection, mixing, and combustion of the fuel as to achieve the power and economy proper to the high compression ratios employed.

The curve KL of fig. 52, which was drawn through the plotted results of experiments with volatile fuels, represents the best thermal-efficiency values obtainable in practice on a single-cylinder petrol engine at compression ratios from 4:1 to 7:1. In fig. 63 the same curve has been continued, as at KLM, up to 17:1 by means of the formula

$$\text{efficiency} = 1-\left(\frac{1}{r}\right)^{0\cdot236},$$

which fits well with the experimental figures, and from this extension we may form a rough estimate of what is to be expected at the higher ratios, ranging from 10:1 to 16:1, which are employed in the compression-ignition engine. Two new factors enter into the operation of this type of engine, however, which affect both the power and economy obtainable. The first is the extreme difficulty of achieving perfect mixing of the injected fuel with the air, so that up to the present time it has never been found possible to burn efficiently more than about 75 per cent. of the air in the cylinder; the second factor is the necessity for working with a controlled maximum pressure in the cylinder.

The curve *EF* of fig. 52 giving the calculated efficiencies for a fuel-air mixture in which 80 per cent. of the air is burnt, has also been added to fig. 63 as the curve *CD* and extended by an appropriate formula to the higher ratios. It may be taken as indicating with sufficient accuracy what might be expected in the way of calculated efficiencies at the higher ratios, if all combustion could be assumed to take place at constant volume. This assumption, however, would involve excessive maximum pressures of about 1,100 lb. per sq. inch at 10:1 and 1,800 lb. per sq. inch at 15:1, if the heat developed corresponded to 75 per cent. of the air being burnt. Excessive maximum pressures are avoided by beginning the injection of the fuel rather later, that is, a shorter time before the dead centre. Although a sharp peak pressure

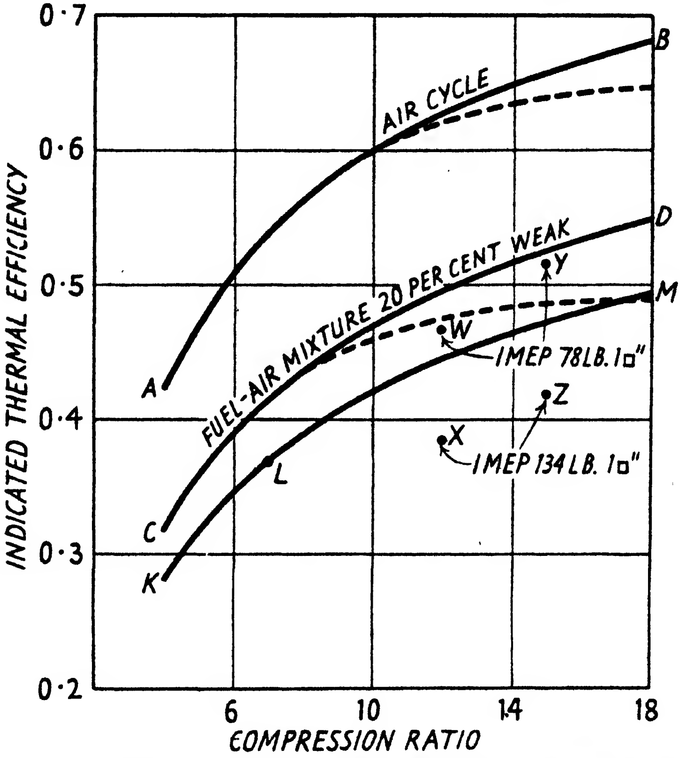

FIG. 63. Thermal efficiency of Diesel engines at different compression ratios.

is still obtained, this peak, at the agreed maximum pressure of perhaps 900 lb. per sq. inch, occurs at a point when the piston has moved forward appreciably on the expansion stroke, and when, therefore, the full ratio of expansion is no longer available. At reduced loads, when less fuel is injected and constant volume combustion can with safety be more nearly approached, very high efficiencies can be achieved; and the problem which faces the research engineer and designer is that of extending these high efficiencies up into the region of mean effective pressures comparable to those customary in a petrol engine at full load. The combustion problem in the compression-ignition engine, therefore, resolves itself into one of making use of the largest possible fraction of the air in the cylinder; for, immensely valuable as is the faculty of yielding very high efficiencies at light loads, it is equally important to obtain a high output per unit of cylinder volume, and this cannot be done without rapid combustion of a large proportion of the air present. As stated already, no high-speed engine of the type has so far succeeded in burning more than about 75 per cent. of the air, and under these conditions the maximum I.M.E.P., consistent with maintaining a good efficiency, was about 135 lb. per sq. inch. If the injected fuel were any further increased, in an attempt to burn more of the air, and get a higher mean pressure, this merely resulted in the production of black smoke without any appreciable increase of power.

To fig. 63 there have been added four experimental values of thermal efficiency obtained on single-cylinder compression-ignition engines. The values 46·6 per cent. and 38·3 per cent. marked *W* and *X*, at 12:1 compression ratio, are taken from a paper by Taylor,[26] and those marked *Y* and *Z*, 51·6 per cent. and 41·9 per cent., from one by Ricardo.[22] The engine used by Taylor had a poppet-valve cylinder of bore and stroke 8 × 11 inches, with directed-spray injection, and a normal speed of 1,000–1,200 r.p.m. Ricardo's engine had a sleeve-valve cylinder of 5½ × 7 inches, in which mixing and combustion were achieved by organized air swirl. The speed was 1,300 r.p.m. The efficiency figures given above are indicated values, and in order that there may be no suspicion of exaggeration through an underestimate of the mechanical efficiency, it is worth while adding that the corresponding brake thermal efficiencies obtained from direct observation, which correspond with the points *W* and *Y*, were 38·7 per cent. and 40 per cent. respectively. These figures were obtained at mean piston speeds of 1,830 and 1,520 feet per minute in the two engines, and are the highest thermal efficiencies recorded. They correspond to a consumption of about 37 per cent. of the air present in the cylinder. The values given at *X* and *Z* are the efficiencies obtained at or near the maximum M.E.P.s at which efficient combustion could be maintained and under these conditions about 75 per cent. of the air in the cylinder was being utilized. Equally good indicated thermal efficiencies have been obtained by Ricardo at moderate M.E.P.s up to 2,570 feet per minute piston speed; but at this speed it has not so far been found possible to maintain high efficiencies up to the M.E.P.s corresponding to the points *X* and *Z*.

Comparisons of experimental values with theoretical limits of efficiency are, for the Diesel engine, of doubtful value; for if a limit is placed upon the maximum cylinder pressure the appropriate theoretical standard depends very much upon the percentage of the total air which is burnt and, therefore, upon the M.E.P. at which we are endeavouring to work. As against this, it may be argued that the curve *KLM* of fig. 63 is extrapolated from an experimental curve for a petrol-air mixture about 15 to 20 per cent. weak (this being the maximum economy mixture) when burnt at constant volume; that is to say, it corresponds to a fuel-air ratio just outside the proportion with which it has been found possible to work efficiently in a Diesel engine, and with which the points *X* and *Z* were obtained. We may, therefore, regard the curve *KM* as a practical limit towards which future full-load efficiencies should strive and which they might be expected to achieve, were it not for the limitation placed upon the maximum cylinder pressures.

The dependence of some of the best Diesel engine performances upon M.E.P., and upon the maximum pressures allowed, and at the same time their relationship to compression ratios, are brought out in fig. 64 in which three sets of Taylor's experiments and one of Ricardo's are plotted against I.M.E.P. It will be seen that the higher efficiencies, with 15:1 compression ratio, were obtained with a maximum pressure of 750 lb. per sq. inch, as against 800 throughout the curve giving the best of the performances at 12:1 ratio. So that the tests at the lower ratio were at any rate at no disadvantage as regards the maximum pressure allowed, and the comparison affords good grounds for concluding that there is some advantage to be gained, as regards economy, in working with the higher ratio. How far this advantage may be expected to extend when a rise of ratio is accompanied by a limited maximum pressure, is a question to be dealt with in the next article. The conclusion reached is that for full power conditions, when we must aim at burning at least 75 per cent. of the air, there is no useful margin to be gained by increasing the ratio above 15:1 unless at the same time we are prepared to face a maximum pressure well above 800 lb. per sq. inch. Apart from questions of power and economy, there are, too, certain practical considerations which have to be taken into account. Pre-ignitions, for example, do occur from time to time in the best regulated engines, and are much more serious at the very high ratios; and it is found, also, that under these conditions the engine performance becomes inconveniently sensitive to an exactly correct timing of the fuel injection.

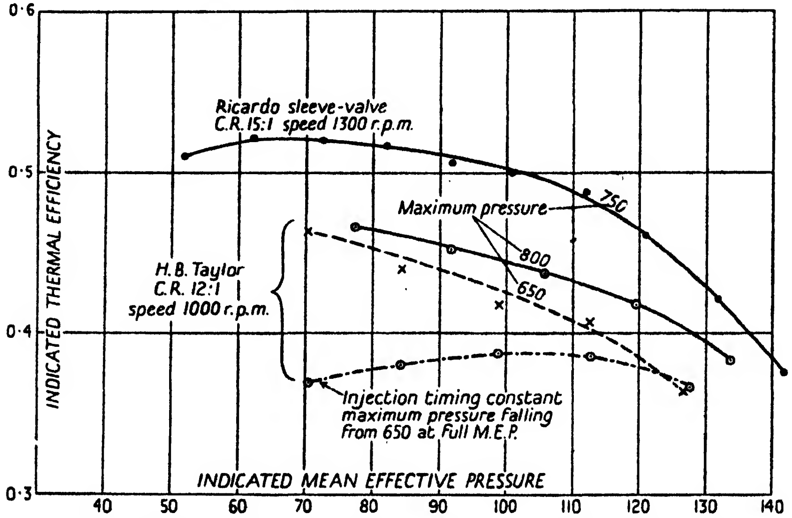

FIG. 64. Variation of thermal efficiency with the power output in Diesel engines, showing the influence of fuel injection timing and of the maximum pressure allowed.

Returning now to fig. 64, the two uppermost of the three curves for the poppet-valve engine of 12:1 ratio give the results of two series of experiments under comparable conditions, but with different maximum pressures, 650 and 800 lb. per sq. inch, allowed in the cylinder. The higher maximum pressures are obtained by allowing an earlier fuel injection at each corresponding value of the I.M.E.P. Combustion is thereby completed earlier, and the effective ratio of expansion is increased. During each series of experiments the moment of commencing injection was made earlier and earlier as the load (and therefore the quantity of fuel injected) was reduced. During the series with a maximum pressure of 650 lb. per sq. inch the timing varied from 12 to 29·5 degrees before the dead centre; and with 800 maximum, from 19 to 37 degrees.

The lowest of the three curves shows what happens if the load, and therefore the duration of fuel injection, is reduced, without at the same time beginning injection earlier. Under these conditions no advantage was taken of the lower cycle temperatures which would have permitted approximation more and more towards constant volume combustion at light loads, without exceeding the maximum pressure limit. The effective expansion ratio was not increased by having an earlier maximum pressure, and the result is that instead of rising, the efficiency at reduced loads remains nearly constant.

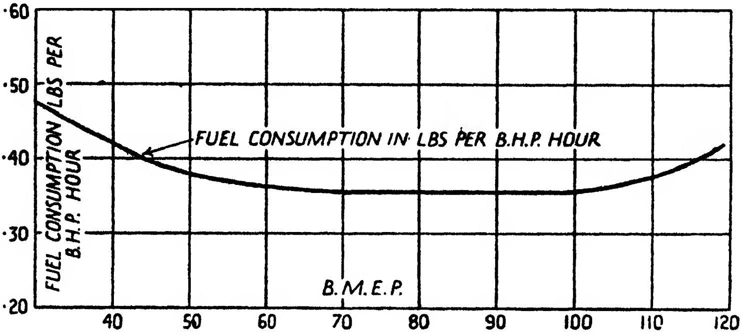

FIG. 65. Ricardo sleeve-valve single-cylinder engine. Variation of specific fuel consumption with power output at constant speed, 1,300 r.p.m. Cylinder size, 5½ × 7 in. bore × stroke. Compression ratio, 15 : 1.

Fig. 65 shows the fuel consumption in pounds per B.H.P. hour (the important figure in practice) for the same series of tests upon the 15 : 1 ratio engine as are represented by the uppermost curve of fig. 64. The curve is typical of what is to be expected from a well-tuned compression-ignition engine. It shows the low fuel consumption which can be obtained over a wide range up to nearly full-power B.M.E.P., and then the rapid increase of fuel consumption when a certain limiting M.E.P. is exceeded. Between 70 and 100 lb. per sq. inch the consumption was constant at 0·355 lb. per B.H.P. hour. On a high-speed engine at 2,200 r.p.m. the only difference, apart from a lower mechanical efficiency, was that the range of economical running was narrower; instead of extending from a B.M.E.P. of 55 up to 110 lb. per sq. inch, it was only from 55 up to 95. As a result of the lower mechanical efficiency the minimum fuel consumption per B.H.P. hour was 0·39 lb. but on an indicated basis the fuel consumptions were identical, which shows that the limit of satisfactorily rapid and complete combustion to give high efficiencies in a compression-ignition engine has not been reached below 2,200 r.p.m.

The same poppet-valve engine for which the results are given above when operating with a compression ratio of 12:1 had previously been calibrated as a petrol engine, with electric ignition and a compression ratio 4·84:1. It is therefore of some interest to give a direct comparison under the two conditions. Taking it first as a Diesel engine, at the maximum I.M.E.P., 133·7 lb. per sq. inch, at which load the indicated thermal efficiency was 38·3 per cent. (see fig. 64), the maximum B.M.E.P. was 120·6 lb. per sq. inch and the fuel consumption 0·40 lb. per B.H.P. hour. When running as a carburetter engine on petrol, with its compression ratio lowered to 4·84, the engine developed 134 lb. per sq. inch B.M.E.P. at the same speed, and with a fuel consumption of 0·49 lb. per B.H.P. hour.

While the change-over from spark- to compression-ignition, therefore, lowered the B.M.E.P. by 10 per cent. from 134 to 120·6, fuel consumption was lowered by 18 per cent. from 0·49 to 0·40.

A further important difference, in regard to economy, between the compression-ignition and the petrol engine is that whereas in the latter, as we have seen, it is never possible for a multi-cylinder engine to achieve the economy of the single-cylinder unit owing to inequality of fuel distribution, with the compression-ignition engine the reverse should be the case, at any rate on a brake horse-power basis. For with the fuel separately injected to each cylinder, the indicated thermal efficiency should be the same for the whole engine as for one cylinder of it, while the mechanical efficiency of the multi-cylinder should show some advantage over the single.

Art. 53. The useful limit of compression ratio in Diesel engines

The place of compression ratio in the functioning of a petrol engine is now well understood. It is recognized that a high compression ratio is advantageous, both from the point of view of power and economy, but that the tendency of fuels to detonate, and ultimately to pre-ignite, places a practical limit upon the ratio which can be employed. The limit may be taken as about 7½:1 with the highest grade of fuel, and under these conditions the maximum cylinder pressures will be just under 1,000 lb. per sq. inch, provided there is no detonation. Engines of the injected fuel type require a compression ratio of not less than 10:1 in order that the compression temperature may be high enough to ensure ignition of the fuel, but it cannot be said that there is any general consensus of opinion as to the limits of compression ratio in this type of engine. The optimum ratio for any particular engine must always depend to some extent upon design details which influence, for example, the shape of the combustion chamber; but it is a matter of some importance to assess the arguments for and against putting up the compression ratio above the minimum necessary to secure combustion, in an injection engine, and to decide where the useful limit is likely to be.

Engines are in operation with ratios varying between 10:1 and 16:1. At 20:1 the compression pressure would be over 900 lb. per sq. inch, that is to say, very near to the present accepted limit of maximum cylinder pressure. Since all combustion must, at this ratio, therefore, take place during the expansion stroke, with a consequent sacrifice of effective expansion ratio, we may safely conclude that, so long as the maximum cylinder pressure is limited to about 1,000 lb. per sq. inch, the useful limit of compression ratio must be somewhere well below 20:1.

On the basis of a comparison between experimental results obtained at 12:1 and 15:1 compression ratio it was shown in the last article that some advantage was to be gained, as regards thermal efficiency, by working at the higher ratio. The question to be answered is whether anything is likely to be gained by making the compression ratio higher still.

In Chapter II the cycle efficiencies, using ideal air as the working substance, were given for the composite cycle of fig. 5 in which heat was added partly at constant volume and partly at constant pressure, in such a way as to maintain a constant maximum pressure. The effect of a limitation to 1,000 lb. per sq. inch maximum pressure was to make the rise of theoretical efficiency with compression ratio follow the dotted line which leaves the air-cycle curve of fig. 63 at ratio 10. The rise of efficiency is much less rapid than for the air-cycle, because the expansion ratio increases only slightly for a large increase in the compression.

It will now be shown how a limitation of maximum pressure may be expected to affect the relationship between efficiency and compression ratio when the real properties of the working substance are taken into account. A convenient method of doing this is illustrated by the dotted portion of fig. 66, in which the adiabatic compression of the composite cycle 1233′4 has been continued until the maximum allowable pressure has been reached by compression alone.

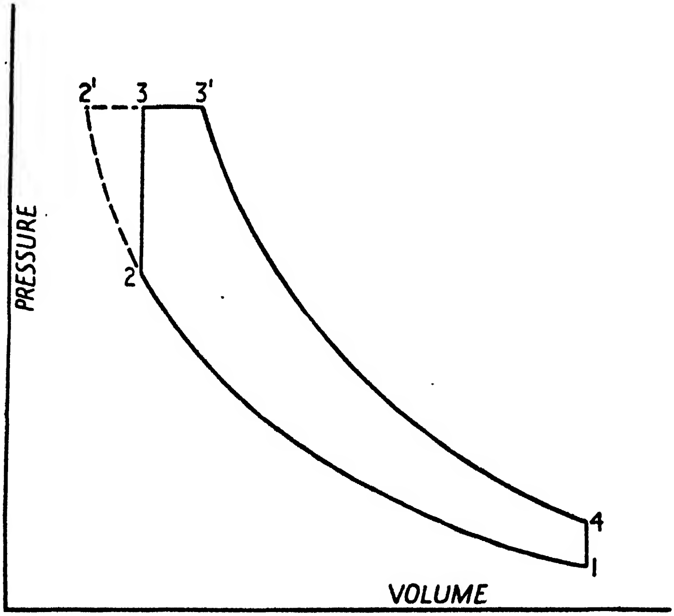

Fig. 66. 'Constant pressure' and composite cycles.

Suppose the cycle 1233′4 represents the form of the cycle at compression ratio 10:1. More work would be done per unit of working substance if the compression were continued up to some higher ratio, so that the cycle became 12′3′4, and more heat would be added also. The efficiencies of the two cycles can be compared by estimating the increase of work and the increase of heat added, as follows:

Extra work of compression from 2 to 2′ = $C_v(T_{2'}-T_2)$, where C_v is the mean volumetric heat of air between T_2 and $T_{2'}$.

Work done during the process 2′–3

$$= P_{2'}(V_3-V_{2'}) = R(eT_3-T_{2'}).$$

$\therefore$ net increase of work $= R(eT_3-T_{2'})-C_v(T_{2'}-T_2)$, (1)

where e has a value slightly greater than 1, and is to allow for an increase in the effective value of the gas constant R over its value during compression, owing to introduction of the fuel and to the increase in the number of molecules during the combustion process 2′–3. For heavy oils, on complete combustion, this increase might amount to 9 per cent., but for some fuels would be only about 4 per cent. In the cycle examined, even at the point 3′ only 80 per cent. of the air, at the most, is supposed to be burnt, and at the point 3 it will be less. The value of e only affects the result to a small degree and we may assume a value of 1·03 for the present, with ample accuracy.

The difference in heat input

$$= e[K_p(T_3-T_{2'})-K_v(T_3-T_2)],$$

where K_p and K_v are the mean volumetric heats of the mixture of the products of combustion and unburnt air.

The difference between the compression temperatures T_2 and $T_{2'}$, even for the full range between 10:1 and 20:1 compression, is only about 250° C., and to a first approximation, therefore, we may take K_v as the same for the range of temperature $(T_3-T_{2'})$ as for (T_3-T_2).

∴ difference in heat input

$$= e[(R+K_v)(T_3-T_{2'})-K_v(T_3-T_2)]$$

$$= eR(T_3-T_{2'})-eK_v(T_{2'}-T_2), \tag{2}$$

K_v not being for the range $(T_{2'}-T_2)$, however, but for $(T_3-T_{2'})$ and (T_3-T_2).

By a comparison of expression (2) with (1), remembering that K_v is greater than C_v, and that e is greater than 1, it appears that the increase of heat input is actually less than the extra work done, and hence that it must pay to work with the higher ratio even when the maximum pressure is kept the same throughout.

In order to estimate the magnitude of the difference two cases will be taken:

$$(a)\quad \frac{V_1}{V_2} = 10 \quad\text{and}\quad \frac{V_1}{V_{2'}} = 20$$

$$(b)\quad \frac{V_1}{V_2} = 15 \quad\text{and}\quad \frac{V_1}{V_{2'}} = 20.$$

For these high ratios of compression the increase of volumetric heat of air has an appreciable effect in reducing the average value of the index γ in PV^γ during compression. In the calculations below, the value of γ has been taken which corresponds to the average value of C_v between 100° C. and the compression temperature.

We have, then, for case (*a*)

$T_1 = 373°$ abs. $p_1 = 14{\cdot}7$ lb. per sq. inch.

$T_2 = 872°$ abs. $p_2 = 14{\cdot}7 \times 10 \times \frac{872}{373} = 344$ lb. per sq. inch.

$T_{2'} = 1091°$ abs. $p_{2'} = 14{\cdot}7 \times 20 \times \frac{1091}{373} = 860$ lb. per sq. inch.

$$T_3 = \frac{872}{1{\cdot}03} \times \frac{860}{344} = 2116° \text{ abs.}$$

The constant pressure cycle 12′3′4, with 20:1 compression, gives the convenient practical maximum pressure of about 850 lb. per sq. inch.

Mean C_v for air between 872° and 1091° abs. = 22·95 ft.-lb. per S.C.F.

Mean K_v for products between 872° and 2116° abs. = 26·58 ft.-lb. per S.C.F.

Mean K_v for products between 1091° and 2116° abs. = 26·99ft.-lb. per S.C.F.

$R = 1{\cdot}985 \times 3{\cdot}90 = 7{\cdot}74.$

Net gain of work $= 7{\cdot}74(1{\cdot}03 \times 2116 - 1091) - 22{\cdot}95(1091 - 872)$
$= 8428 - 5026 = 3402$ ft.-lb. per S.C.F.

Increase in heat input $= 1{\cdot}03(35{,}595 - 33{,}065) = 2606$ ft.-lb. per S.C.F.

If the total heat input is that produced by burning 80 per cent. of the air present, this is about 65,400 ft.-lb. per S.C.F. and would produce a rise of temperature—allowing for dissociation and nitric oxide formation—of about 2,000° C., say 1,988° C. At the compression ratio of 10 to 1 this temperature rise would give a maximum pressure of nearly 1,200 lb. per sq. inch. The maximum, however, is to be 860, and therefore in the composite cycle of 10 to 1 compression, only about 52 per cent. of the available heat is to be generated during the stage 2–3. The comparison, therefore, is between a cycle of 10 to 1 ratio in which about 52 per cent. of the available heat is generated at constant volume and one of 20:1 ratio, wholly at constant pressure.

Now the theoretical efficiency of the 10:1 cycle for a 20 per cent. weak mixture, in which dissociation is allowed for and all combustion takes place at constant volume, is about 0·47 (see middle full line curve, fig. 63). This efficiency is reduced to 0·46 when the pressure is limited to 860 lb. per sq. inch., and the amount of dissociation corresponding to the new condition is recalculated.

Hence comparing the two cycles 1233′4 and 12′3′4, the efficiency of the 20:1 constant pressure cycle will be

$$\frac{65{,}400\times 0{\cdot}46+3{,}402}{65{,}400+2{,}606}=0{\cdot}492,$$

which is an increase of about 7 per cent. for the rise of compression ratio from 10 to 20 with a maximum pressure of 860 lb. per sq. inch.

Now taking case (*b*) in which we compare the 20:1 constant pressure cycle with the composite cycle of 15:1. In this latter cycle only about 15 per cent. of the fuel is burnt at constant volume, when the point (3) is reached, and an appropriate value for *e* will be 1·01 instead of 1·03. We have

$$T_2 = 996^\circ \text{ abs.} \qquad p_2 = 14{\cdot}7\times 15\times\frac{996}{373} = 589 \text{ lb. per sq. inch.}$$

$$T_{2'} = 1091^\circ \text{ as before.} \qquad p_{2'} = 860.$$

$$T_3 = \frac{996}{1{\cdot}01}\times\frac{860}{589} = 1440^\circ \text{ abs.}$$

whence, after allowing for the appropriate volumetric heats corresponding to the different temperature ranges we obtain:

Net loss of work = 605 ft.-lb. per S.C.F.

Decrease in heat input = 422 ,, ,,

It has been found above that the efficiency of the 20:1 constant pressure cycle is 0·492, and therefore the efficiency of the 15:1 composite cycle

$$=\frac{65{,}400\times 0{\cdot}492-605}{65{,}400-422}=0{\cdot}486.$$

Hence it appears that, under the assumed conditions of a maximum pressure limited to 860 lb. per sq. inch, while an increase in compression ratio from 10:1 to 15:1 gives an increase of theoretical efficiency from 0·46 to 0·486, a further increase of the ratio to 20:1 produces a further rise of efficiency only to 0·492.

To sum up, therefore, at a compression ratio of 10:1 it is possible to burn at constant volume rather more than half the air present without exceeding a maximum pressure of 860 lb. per sq. in. and the theoretical efficiency, if not more than 80 per cent. of the air is finally burnt, will be 0·46. If the compression ratio is raised to 15:1, still with the same maximum pressure, the theoretical efficiency is increased to 0·486, but raising the ratio further to 20:1 gives only a very slight additional increase of efficiency, to 0·492. The dotted curve in fig. 63 (p. 215), which breaks away from the 20 per cent. weak curve at the point where it is just possible to have all combustion at constant volume without exceeding the maximum pressure of 860 lb. per sq. in., illustrates the variation of theoretical efficiency, and leads to the conclusion that for full power conditions, when we must aim at burning at least 75 per cent. of the air, there is no useful margin to be gained by increasing the compression ratio above 15:1 unless we are prepared for an increase of maximum pressure well above 860 lb. per sq. inch.

In order to illustrate further the dependence of efficiency upon maximum pressure, it may be added that the reduction of efficiency at 15:1 through limitation of the maximum pressure to 860 lb. per sq. inch is seen from the curves to be from 0·525 to 0·486. If the heat from burning 80 per cent. of the air had all been added at constant volume, the maximum pressure would have been about double, 1,760 lb. per sq. inch at this ratio. In other words the price of a rise from 0·486 to 0·525 in the calculated efficiency is an increase of about 900 lb. per sq. inch in the maximum pressure.

VII

THE TESTING OF ENGINES

Art. 54. The aims of practical testing

Engine testing to the student and to the practical engineer mean two very different things. To the student an engine test is a piece of practical experience, during which the engine does what it is expected to do, and he himself verifies facts about it which he has probably been told in a lecture or has read in a book. To the engineer, a test is a proof, firstly, that the engine will do what it was designed to do, and secondly, that it is capable of continuing to function without breakdown for a proper length of time. The first of these two aspects differs only from the trial made for demonstration purposes in the attitude towards the engine of those conducting the test; but the second aspect, the one of endurance, would probably have been eliminated by the time an engine had reached the dignified respectability of being a piece of equipment in a students' laboratory.

It is all the more important for the student to realize that in practical life no engine can be said to have got through its early troubles, and be a fully proved design, until it has done many hours of continuous running at full power without needing attention. It is these hours of endurance running that bring out the weak points of a design, in a way which no short time test can possibly do. Endurance testing will be mentioned again in a subsequent volume, in connexion with certain specially vital points of design and frequent sources of weakness. In the present chapter we shall be concerned only with tests to ascertain the limits of what an engine can do.

The very varied information which is required from an engine test may be grouped under certain headings. The main points are:

(*a*) What is the power delivered by the engine, and what adjustments have to be made to obtain the best performance from it.

(*b*) What is its rate of fuel consumption, and hence its fuel consumption per horse-power.

(*c*) How the losses, mechanical and thermal, are made up; and hence how much better the engine might be, and in what ways.

The measurement of B.H.P. under any one set of steady conditions involves only the direct measurement of the speed, and of the torque available at the shaft, either by a swinging field electrical, or a water, dynamometer, and these measurements are easily obtained with great accuracy. No difficulty in obtaining steady readings, and figures repeatable to 0·5 per cent. should be encountered with a well-tuned engine. Since attention, except in endurance tests, is always concentrated upon cylinder performance, and since a test is always run at a constant speed, it is customary and convenient always to speak of the torque available in terms of a mean effective pressure per working stroke in the cylinders, to which it is proportional. The Brake Mean Effective Pressure, or B.M.E.P. represents no actual gas pressure which exists anywhere in the engine, but bears the same relation to the Indicated Mean Effective Pressure, which might be measured by the perfect indicator, if one existed, as the B.H.P. bears to the I.H.P. In other words

$$\text{B.M.E.P.} = \text{I.M.E.P.} \times \text{mechanical efficiency}$$

and

$$\text{mechanical efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P.}}.$$

Direct determination of the mechanical efficiency by the simultaneous measurement of B.H.P. and I.H.P. becomes quite unsatisfactory with internal combustion engines at high speeds.† Any indicator diagram drawn out to a base of piston displacement for the measurement of the I.M.E.P., must be so accurately 'phased' if large errors are not to come in, that the method becomes impracticable. By correct phasing is meant the requirement that the pressure, indicated, for example, as occurring at half-stroke, must represent the pressure when the piston is *exactly* at this point of its own stroke. It can easily be shown that an error of no more than 1° of crank angle between the relative phase of the real piston and of the indicator motion, will introduce an error of as much as 5 per cent. in the measured I.M.E.P. The method is therefore quite ruled out for accurate work, and it becomes necessary to arrive at the I.M.E.P. from measurements of the B.H.P. and of the mechanical losses when the engine is motored round.

For the determination of motoring losses some form of swinging field electro-dynamometer is invaluable, for it enables a measurement of the torque necessary to turn the engine to be made immediately

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† In this connexion one may refer to some very complete tests by Rosencrans and Felbeck[37] in which great precautions were taken to obtain an accurate record of pressure variations in the cylinder. In spite of this, the trials exhibit seemingly impossible variations of mechanical efficiency in the engine.

firing has ceased, and while temperature and other conditions are therefore as nearly as possible the same as while running under power. Since about 60 per cent. of the power required to motor an engine is on account of piston friction, any change of viscosity or condition of the lubricating oil on the cylinder walls will soon introduce errors into the mechanical loss determination. For the testing of high-power engines it is convenient to have both an electric and a water dynamometer driven by the engine, the former large enough to provide the power for determining the motoring losses, and the latter to absorb the major part of the power output when the engine is running under load.

Art. 55. The use of the indicator

Great use has been made in earlier chapters of a type of indicator diagram in which a pressure record is obtained on a base of equal time intervals, or crank-angle degrees. Such a record can, of course, be replotted to a base of piston displacement; but the same difficulty of correct phasing comes in as was referred to in the last article, for we have to determine which point on the time scale corresponds exactly, for example, with the dead centre. This can be done quite well within a degree or so, but that is useless where accurate figures for I.M.E.P. are required.

There is no need, after the use which has been made of the diagrams, to emphasize the value of the pressure-time record in studying what goes on in an engine cylinder during the 45° on each side of the dead centre. Such a record might be obtained in various ways, one of them being to employ an ordinary piston indicator in which the phase of the motion has been displaced by 90° from that of the piston, and then to convert the diagram obtained to a base of equal time intervals. This would be laborious, and, moreover, any indicator in which the tracing point endeavours to follow the rise and fall of pressure throughout the stroke of a high-speed internal combustion engine is subject to two drawbacks. It is very difficult, in the first place, to design an instrument with so little inertia that it *can* follow the extremely rapid pressure changes accurately; and secondly, the pressure changes often do not repeat exactly from cycle to cycle, and what one is really concerned with is the average behaviour taken over a large number of cycles.

The type of instrument with which all the diagrams in this book have been taken fulfils exactly the conditions required. It gives a pressure record direct to a base of equal time intervals, and the record obtained can be made the average of a very large or a smaller number of cycles according to choice. A full account of the 'Farnborough' indicator has been given by Wood,[27] and only sufficient description will be given here to explain another essential virtue which it possesses over any piston indicator.

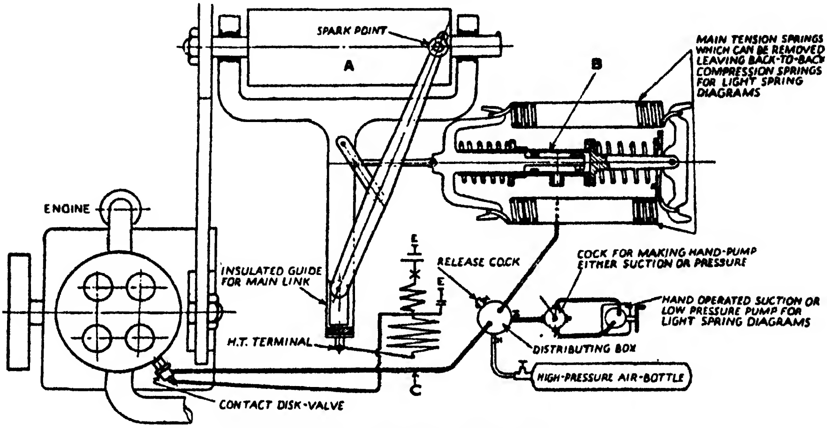

FIG. 67. The 'Farnborough' indicator.

The paper sheet on which the record is obtained is wrapped round the horizontal metal drum *A* in fig. 67, which rotates at crankshaft speed. A sparking point can move horizontally, parallel to the axis of the drum, and close to its surface, its movement being controlled by springs, and so made proportional to the pressure in the forcing chamber at *B*. The pressure is supplied from a high-pressure air bottle, and is allowed to rise gradually, the sparking point moving from end to end along the drum surface while it does so. There is at the engine end of the pipe *C*, in which the pressure is the same as that in the forcing cylinder, a little disk valve with some free movement between two circular seats, one on each side of it. This disk valve can be mounted so as to be almost flush with the wall of the cylinder to be indicated, and is held in contact by gas pressure with one or other of its circular seats. The pressure on the outer side of this disk valve being either constant or slowly rising, there will be two occasions in every cycle when the disk valve will flick over from one seat to the other, unless the external pressure is greater than any which occurs at any time during the cyclical changes of pressure inside the cylinder. Whenever this flick-over happens, a spark passes from the spark point to the metal drum and leaves a fine puncture through the paper wrapped round it. As the pressure outside the disk valve gradually rises, and the sparking point moves along correspondingly against the calibrated springs, the point in the crank revolution, and equally in that of the indicator drum, when the disk valve flicks over will alter. And a record will be obtained which relates the pressure in the engine cylinder, through its causing the flick, to a particular instant of time, and to a particular pressure, which is fixed by the position of the sparking-point in its travel.

The peculiar virtue of this type of indicator is that the pressure is recorded at the wall of the engine cylinder. Every one who has worked with an indicator in which a pipe connects the inside of the engine cylinder with the moving element of the indicator, be it piston or diaphragm, knows the troubles inherent in such a connexion. Apart from the time taken for pressure variations to travel up the pipe, which can be more or less allowed for, there is the likelihood of combustion and sometimes detonations occurring in the pipe, which will set up pressures at the indicator end which bear no relation to the pressures in the engine cylinder that we desire to measure. In the Farnborough indicator all this is eliminated. The disk valve, moreover, is extremely light, and its movement practically instantaneous, so that the only time error which must be allowed for is electrical lag between the movement of the disk and the passing of the spark; but this is a quantity which will be constant, and is easily measured and checked from time to time.

A reproduction of an actual record taken with the indicator is given in fig. 68. The record is from a critical paper[33] dealing with the instrument, and shows with what a high degree of accuracy a fair curve could be drawn through the line of punctures, when circumstances are favourable. A somewhat different form of instrument using the same type of recording mechanism has been described by Taylor and Draper.[60]

For much experimental work upon what goes on in an engine cylinder, in the way of variations in the rate of pressure-rise during combustion, this indicator is an invaluable instrument. The pressure-time curve is an accurate record of the succession of events, and for the study of combustion this is what we require: their absolute location within an accuracy of 1°, in relation to the crank position, is immaterial except when a measure of the indicated power is desired.

An indicator of the Farnborough type, however, can only give a record of the average pressure variation with time over a large number of cycles, and until the invention of the cathode-ray tube there was no form of indicator sufficiently devoid of inertia to follow accurately the rapid variations of gas pressure which may occur within a single cycle of a high-speed internal combustion engine. The absolute magnitudes of these pressure fluctuations are not often of practical importance, but it has been mentioned in art. 30 that pressure waves of a characteristic frequency are set up when detonation occurs, and the employment of the cathode-ray tube for detecting the existence of such high-frequency pressure variation is a possible development of the highest practical importance. It promises to afford a positive and certain indication of the onset of detonation at an earlier stage than any other method, and to be applicable to engines of any type.

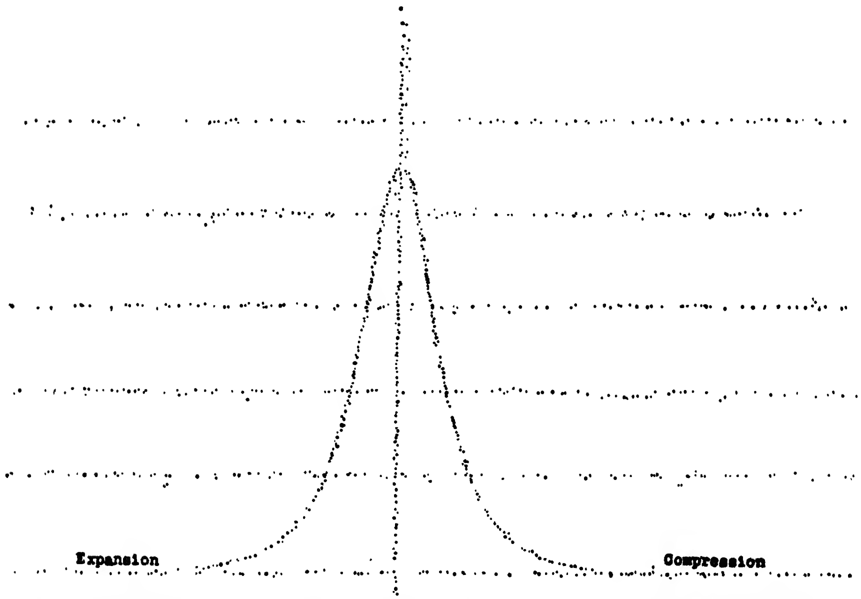

FIG. 68. Specimen record from the Farnborough indicator. Engine motored at 1,600 r.p.m., compression ratio, 13·5 : 1.

The application of the cathode-ray tube as an engine indicator has been made possible by the development of the technique of electrical valve amplification. There are several forms of indicator which differ in their electrical stages, but in each type the first stage is not essentially different from the old-fashioned indicator: the gas pressure in the cylinder produces deflexion of an elastic member, usually a steel diaphragm, and the movement is employed either to generate electromagnetically an infinitesimal current which is subsequently amplified or else to produce minute variations of current by alteration of a capacity or resistance in the electric circuit. The important novelty of the modern technique is that, with all the amplification necessary to produce an easily visible record carried out electrically, infinitesimal deflexions of the elastic member are sufficient, and it can therefore be made very stiff and of a natural frequency of vibration so high —40,000 to 60,000 per sec.—as to follow with ease the individual pressure fluctuations in the cylinder. In the absence of any comprehensive critical publication dealing with the various types of cathode-ray indicator, the reader is referred to descriptions of certain individual instruments with illustrations of the types of records obtained.[58]

ART. 56. **Power at full throttle. The speed-torque relationship**

The power obtainable from an engine will depend upon the mean effective pressure in the cylinders, and the speed. The limit of speed is usually imposed by mechanical considerations such as inertia loading on the bearings and satisfactory valve operation. Mechanical limitations will not be considered here, and we shall simply assume that every engine has some maximum safe speed of operation, and a rather lower normal full speed at which it is designed to run. If full use is to be made of its speed, the engine's 'breathing' arrangements must be designed to enable it to get a full cylinder charge up to its designed speed. In other words its valve area must be sufficient to get the charge in, in the time available, without an undue amount of throttling. This will depend not only upon the average area of the passage through the valves, but upon their opening and closing times. How sensitive an engine's performance can be to its valve timing will be illustrated presently. If we assume that the optimum valve timing has been found for an engine designed to run, say, at 3,000 r.p.m., then the power, B.M.E.P., and mean effective pressure which is the equivalent of the friction and 'pumping' losses, will be related to the speed as in the typical curves given in fig. 69. It will be observed that the maximum power is developed at just over 3,000 r.p.m., but that the B.M.E.P. begins to fall off at about 2,000 r.p.m. owing to throttling at the valves and the increasing friction losses.

An engine with its valve timing arranged so as to develop its maximum power at 3,000 r.p.m. may be quite unsuited for running at lower speeds, and vice versa. The point is illustrated in fig. 70 in which the two curves *A* and *B* show the I.M.E.P. for the same engine over a range of speeds from 800 to 2,000 r.p.m. with two different sets of valve timings.

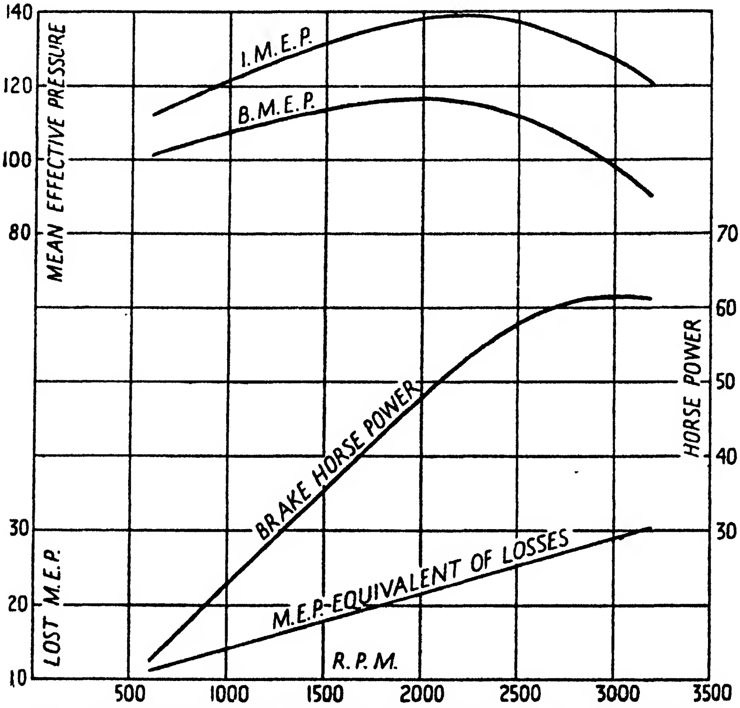

FIG. 69. Typical test results from a 4-cylinder high-speed engine.

The effect of the change of valve timing between tests *A* and *B* was to bring the opening and closing of the valves more nearly at the ends of the piston stroke. The result is a higher mean effective pressure below 1,650 r.p.m., but above that speed there is a rapid falling away in the pressures obtainable with setting *B* as compared with setting *A*. The comparison is instructive, as showing that a curve of power against speed does not necessarily exhibit the merits or demerits of an engine, but only the suitability of the adjustments for any particular condition of running. Such tests are therefore of no great interest in a book concerned with principles of general application, except in so far as it is important to understand the factors which influence the form of the curves which the tests yield. At every speed it is, of course,

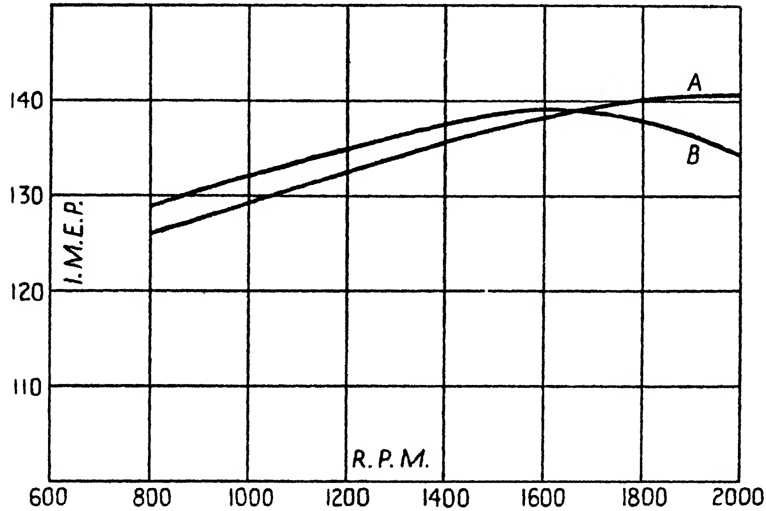

Fig. 70. Maximum indicated mean effective pressures obtainable with different valve timing.

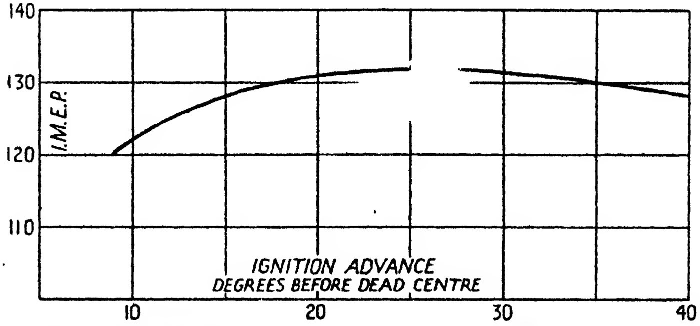

Fig. 71. The effect of ignition timing on power. Speed, 1,800 r.p.m. Fuel, petrol.

essential that ignition timing should be adjusted to suit. Fig. 71 gives a quantitative illustration of the importance of correct ignition timing for maximum power. Throughout these tests the speed was 1,800 r.p.m., and all other conditions, such as mixture strength and all relevant temperatures, were maintained constant while the ignition timing was varied from 9° to 40° before the dead centre. Maximum power was obtained with ignition advance about 26° early, and fell away about 10 per cent. when this was reduced to 9° early. Further advance beyond 26° caused a slight loss of power owing to an increase in the amount of negative work done by the piston. When ignition is too early there is an appreciable increase of pressure during the last part of the compression stroke, due to early combustion.

Apart from adjustments of valve timing and ignition to suit the normal running speed, a controlling influence upon the power output is the volumetric efficiency. This is a factor of fundamental importance, and one which depends both upon the original design of the engine and upon the conditions of running.

ART. 57. **Volumetric efficiency**

There are two different ways in which the volumetric efficiency of an engine may be reckoned. We may define it as

(*a*) The ratio between the volume of combustible mixture drawn in per stroke, reckoned at the pressure and temperature of the surrounding air, and the swept volume of the piston; or

(*b*) The ratio between the quantity of combustible mixture drawn in per stroke and the quantity which would fill the swept volume at N.T.P.

The second definition will be accepted in what follows, as giving a quantity somewhat easier to grasp clearly; although on this definition it would be more correctly described as a gravimetric efficiency, for it is really the ratio of two masses. The numerator is the mass of combustible mixture—or of air in a Diesel engine—drawn in per stroke; and the denominator the mass which would occupy the swept volume of the piston at N.T.P.

If the inlet valve closed exactly at the end of the suction stroke, and if the pressure in the cylinder at that instant were 14·7 lb. per sq. inch, then the volumetric efficiency on the second definition would be

$$\frac{273}{\text{Temperature in degrees C. abs. of the indrawn charge, before mixing with the exhaust gas in the clearance space}}$$

If we take the temperature of the indrawn charge under these conditions as 60° C., as suggested in art. 37, we obtain a volumetric efficiency of 82 per cent. In practice it is usually less than this, because the pressure at the end of the suction stroke is rather less than atmospheric. The time necessary to open and close the valve makes it necessary to begin the closure some 35° before the end of the stroke, which introduces some throttling, and it is further necessary to delay completing the closure until some 25° after the end of the stroke, so that the effective swept volume is slightly less than would be given by the full movement of the piston.

Volumetric efficiency depends upon so many variables that a good average figure is difficult to give. It may be said that it should lie between 72 and 78 per cent. for a normally aspirated engine, at its designed speed, but may be higher in special circumstances, such as on an exceptionally cold day with a high barometer. Apart from valve timing, form of induction system, and day to day conditions of external temperature and pressure, the volumetric efficiency depends upon the breathing capacity of the engine, that is to say, upon the mean area of the valve openings in relation to the area and mean speed of the piston; upon the mixture strength, and latent heat of evaporation of the fuel; upon the heat received by the ingoing charge during its passage through the induction system; upon the cylinder wall temperature; and upon the compression ratio.

Taking first the question of breathing capacity, it is more important to provide ample area for the inlet than for the exhaust valves. When the exhaust valve opens the gas in the cylinder will be at a pressure of something over 50 lb. per sq. inch, and the momentum attained by the first rush of gas into the exhaust system is of great assistance in helping the withdrawal of the remainder. Like everything else in engine design, the decision as to the size of valves is a question of compromise. In order to keep down the power used up in getting the gas into and out of the cylinder, the 'pumping loss', it is clearly an advantage to have large valves and keep the mean gas velocity through the ports as low as possible. Large valves, however, are more difficult to keep cool; and we have learnt already the importance of turbulence, which depends upon maintaining a high gas velocity through the inlet valves. With poppet-valve engines which have valves in the cylinder head, a further factor is the limitation of the space into which they must be crowded; and the shaping of the passages before and after the valve seat becomes of great importance if the throat area available is to be used efficiently.

Experience has shown that the best compromise is attained if the mean gas velocity through the inlet valve lies between 120 and 140 feet per second, and that for good volumetric efficiency it should not be greater than 160 feet per second at the normal speed of the engine. Gas speeds through the exhaust valve, for the reason given above, can safely be allowed to be 50 per cent. greater than through the inlet valve. These figures are expressed in accordance with a simplifying convention which imagines the valves to be fully open during their entire opening period, and the figures are therefore considerably below the real mean gas velocity.

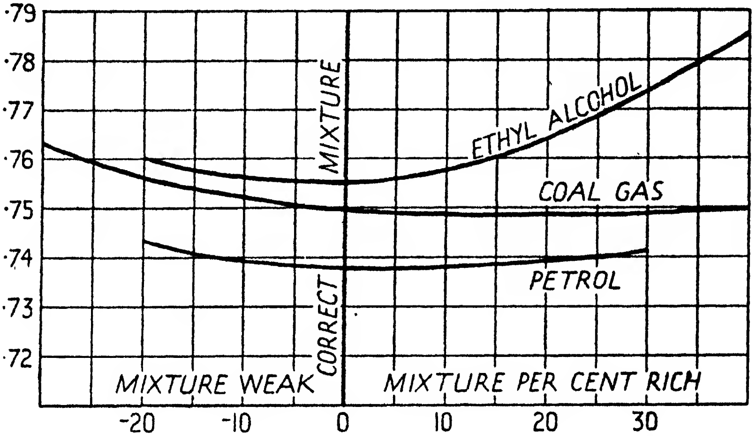

FIG. 72. Variation of volumetric efficiency with fuel-air ratio.

As regards the influence of the mixture strength with liquid fuels upon volumetric efficiency, it has been pointed out in art. 22 that the latent heat of evaporation of an average petrol would lower the temperature of the indrawn air, at the correct mixture strength, if no extra heat were added, by about 28° C. For benzene the figure is 33° C., and for ethyl alcohol about 103° C. for the pure spirit, with an additional 3·5° C. for every 1 per cent. of water it may contain; all figures being on the assumption that the fuel is entirely evaporated by the time the inlet valve closes. With alcohol, however, this is not so. The heat required for evaporation is so great that, in spite of its high volatility, the liquid is unable to pick up sufficient heat in the time available to give complete evaporation before the valve closes.

Volumetric efficiency is affected by mixture strength, partly because of a difference in the amount of cooling by evaporation, with a liquid fuel, and partly because of the alteration in the cylinder-wall temperatures. Fig. 72 shows the variation of volumetric efficiency with different mixture strengths for the E. 35 engine at 5:1 compression ratio using petrol, alcohol, and coal-gas. When running on coal-gas no heat was added to the ingoing air, but with the liquid fuels the air was heated electrically to the extent of 0·024 C.H.U. per revolution, otherwise the volumetric efficiencies would have been about 4 per cent. higher (see fig. 73). Although the absolute values of the volumetric efficiency on coal-gas cannot usefully be compared with the values where the liquid fuels are concerned, owing to differences in the carburetter and heat input conditions, yet the difference between the ways in which the volumetric efficiency is influenced by the mixture strength, as shown by the three curves of fig. 72, is most instructive. There is a slight diminution from the weak to the correct mixture in all three, followed in the case of petrol by a slight, and with alcohol a much more pronounced, increase. With coal-gas there is some indication of a dropping tendency until 20 per cent. rich, after which the value is substantially constant.

The diminution of volumetric efficiency (measured in terms of air only) as more fuel is added to the weak mixtures, which is barely 1 per cent., may be due in part, for the liquid fuels, to an additional displacement of air by fuel vapour; but the more important cause is most probably a rise in the surface temperature of the walls of the cylinder head and piston, as the flame temperature increases with the richer mixtures. Beyond the correct mixture an excess of the liquid fuels will increase the average volumetric heat of the cylinder contents and so keep down the flame temperature and that of piston and cylinder walls. Besides this, evaporation of the extra fuel will have a direct cooling effect upon the cylinder wall surface, and there is therefore a double influence tending to increase the volumetric efficiency. With alcohol, because of its great latent heat, the direct cooling effect is exaggerated and the consequent increase of volumetric efficiency much more marked. With coal-gas the flame temperature will continue to increase up to the maximum power mixture, 20 per cent. rich, and beyond that point, in the absence of any cooling by additional evaporation, the curve remains nearly horizontal.

It should, perhaps, be explained that with the liquid fuels the volumetric efficiency was measured on the basis of the air only which was drawn in. In the coal-gas experiments, of course, it was the total volume of gas-air mixture which was measured.

A comparison of fig. 72 with the earlier fig. 33 in which I.M.E.P.s were shown for different mixture strengths again brings out the point that, while the increase of volumetric efficiency between the correct and the maximum power mixture strength is only about 0·5 per cent., the increase of power is 3–5 per cent., so that this increase must be mainly attributed to dissociation, which has the twofold effect of increasing the maximum temperature and also the volume ratio σ, with the rich mixtures.

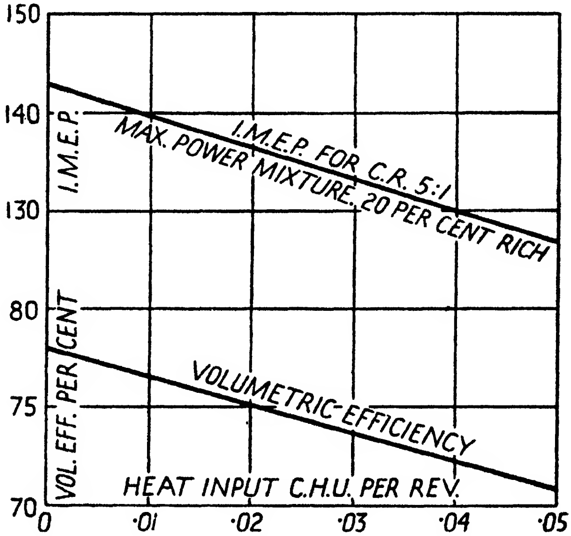

Fig. 73. The variation of volumetric efficiency and maximum I.M.E.P. with the amount of heat given to the indrawn mixture. Speed, 1,500 r.p.m. Compression ratio, 5:1. Fuel, petrol.

It has been mentioned that if no heat had been given to the ingoing air the volumetric efficiencies for the liquid fuels of fig. 72 would have been about 4 per cent. higher. The relationship is linear within the range of practical heat additions, and is given in fig. 73 for petrol, at the maximum power mixture strength, that is, 20 per cent. rich. The corresponding I.M.E.P.s for 5:1 compression ratio are shown in the same figure, and are proportional to the volumetric efficiencies. For the correct mixture strength the volumetric efficiencies would be 0·5 per cent., and the I.M.E.P.s about 4 per cent., lower throughout.

A lowering of cylinder cooling water temperature from 100° C. to 40° C. produced, in the same engine, an increase of 2 per cent. in the volumetric efficiency on account of reduced heating of the ingoing charge. The values of fig. 73 were all obtained with a jacket water temperature of 60° C., so that the maximum variation on this account from the values given would be ± 1 per cent.

There is at first sight no reason why an increase in compression ratio should cause a fall off in the volumetric efficiency, and the fact that it was found to do so to a quite considerable extent, as illustrated in fig. 74, was very puzzling. The mixing of the ingoing charge with less hot residual gas at the higher ratios would have no effect if volumetric heats were the same at all temperatures, but increasing as they do, the effect of a reduction in the residuals should tend towards an increase of volumetric efficiency. During the experiments at different ratios great precautions were taken to maintain the mixture strength, jacket temperature, and heat input constant, so that there could be no doubt the change was a true result of compression ratio only.

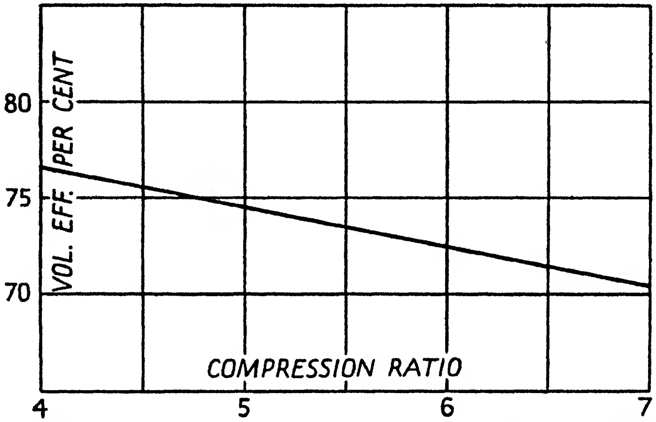

FIG. 74. The variation of the volumetric efficiency with compression ratio in Ricardo 'E. 35' single-cylinder engine.

An explanation of the phenomenon suggested by Alcock[7] is as follows. While the piston is almost stationary for an appreciable time at the end of the exhaust stroke, very hot gas remains in the clearance space, for no fresh mixture will so far have entered through the inlet valve to cool it. During this period the hot residual exhaust gas will rapidly lose heat to the cylinder walls, and any contraction, due to this loss of heat, which occurs after the closing of the exhaust valve will have the result of allowing an extra quantity of fresh charge to enter through the inlet valve—in other words, it will increase the volumetric efficiency. Now at high compression ratios the quantity of residual exhaust gas is less, and also it is cooler, so that on both accounts the amount of its contraction due to cooling will be less, and so also will the amount of fresh charge which can enter to make up for the contraction.

This explanation is perfectly logical, and satisfactory in a qualitative sense. When examined quantitatively, however, it is not easy to see the whole observed increase of volumetric efficiency, from 70·5 to 76·5 per cent. between 7:1 and 4:1, as being accounted for in this way.

The observed increase amounts to 8·5 per cent. of the air drawn in at the 7:1 ratio, so that if we call the swept volume V_s, and the clearance volumes at the two ratios $V_s/6$ and $V_s/3$, and suppose that the hot gas left in the clearance space contracts in each case, by cooling, to $1/n$ of the volume it had when the exhaust valve closed, then

$$(n-1)\frac{V_s}{n}\left[\frac{1}{3}-\frac{1}{6}\right] = 0{\cdot}085V_s,$$

whence
$$\frac{1}{n} = 0{\cdot}49.$$

The absolute temperature of the residual gas, therefore, would have to be about halved by loss of heat to the cylinder walls to account for the observed change of volumetric efficiency.

The magnitude of the effect, if Alcock's explanation be correct, will clearly be very much affected by the valve timing. In most high-speed engines there is a certain amount of 'valve overlap', by which is meant that the exhaust valve does not close until some degrees of crankshaft revolution after the inlet valve has opened. In Ricardo's engine, on the other hand, there was, for special reasons, a 'negative overlap' of about 6 degrees, in the sense that there was a period of this duration at the end of the exhaust stroke during which the cylinder contents were isolated completely, the exhaust valve having closed and the inlet valve being not yet open. These conditions are clearly those in which any effect caused by rapid cooling and contraction of very hot residual exhaust gas would be exaggerated. A significant fact is that Burstall,[20] working with the same design of engine but a slightly different valve timing, and with a weak mixture of coal-gas and air, found no change of volumetric efficiency at all, between 4:1 and 7:1. This he accepts as being due to the weak mixture and lower gas temperatures. In some other experiments with rich mixtures a slight change was found, but very much less than with petrol.

With lower cycle temperatures the effect might be expected to be less, but that it should disappear altogether is scarcely consistent with its being caused solely by an interchange of heat with the cylinder walls. It seems likely, therefore, that the large reduction of volumetric efficiency with increase of compression ratio shown in fig. 74 is an effect depending very much upon the valve timing, and one which may very possibly be non-existent in an engine with any considerable amount of valve overlap.

ART. 58. **Fuel consumption measurements**

It was explained in the last chapter that we have to think of the thermal efficiency, and the fuel consumption, of an engine as two separate things, which give different and equally important kinds of information about it. An engine's thermal efficiency figure expresses the best it *can* do in the way of turning heat into work; its fuel consumption tells us what it is doing at the moment. Its thermal efficiency is a quality of the engine itself; but a single test of the rate at which it is consuming fuel may be no more than a test of the suitability of its carburetter adjustment.

Nevertheless, the fuel consumption per B.H.P. is the figure of practical importance, and engine economy will now be approached from this point of view. Provided proper precautions are taken to ensure that the lowest possible fuel consumption per horse-power has been found, then this gives us all the information we require about the engine's economy, and incidentally enables the best thermal efficiency actually attainable to be calculated, if we wish to do so.

We have seen that over a large range of mixture strength on the rich side the power obtainable from an engine at constant speed varies little, but is a maximum at about 20 per cent. rich. It follows, therefore, that if an engine is running on the maximum power mixture strength, or a richer one, and steps are then taken to cut down the supply of fuel while running, by means of an adjustable jet in the carburetter, we shall at first reduce the fuel consumption with little alteration of power, and there will be a proportionate reduction in the fuel consumption per horse-power. A point will come presently when further fuel reduction causes a more and more rapid fall in power, and the fuel consumption per horse-power, after reaching a minimum, begins to increase again. A typical 'consumption loop', which represents the results of an experiment of the kind described, on a single-cylinder engine, is shown in fig. 75. The progress of the experiment as described above while the fuel is cut down may be followed round the curve from *A* to *B* and *C*. At first there is a rapid drop of fuel consumption, with at first a slight increase, followed by a slight decrease of I.M.E.P.; then the more rapid fall off in power as we turn the corner at *B*; and finally there comes a time when the power drops roughly in proportion to the fuel reduction, giving a period *BC* of nearly constant fuel consumption per horse-power, with a minimum point easily obtained by drawing a horizontal tangent. In the same way the maximum power is easily obtained by drawing a vertical tangent to the curve at the right-hand side. The point *B*, where the fuel consumption per horse-power is about 20 per cent. below that giving maximum power, would correspond to the correct fuel-air mixture, giving just complete combustion. At each reading throughout the experiment it will be necessary, of course, to adjust the ignition advance so as to get maximum power; for although the speed is constant, the rate of burning will vary with the fuel-air ratio.

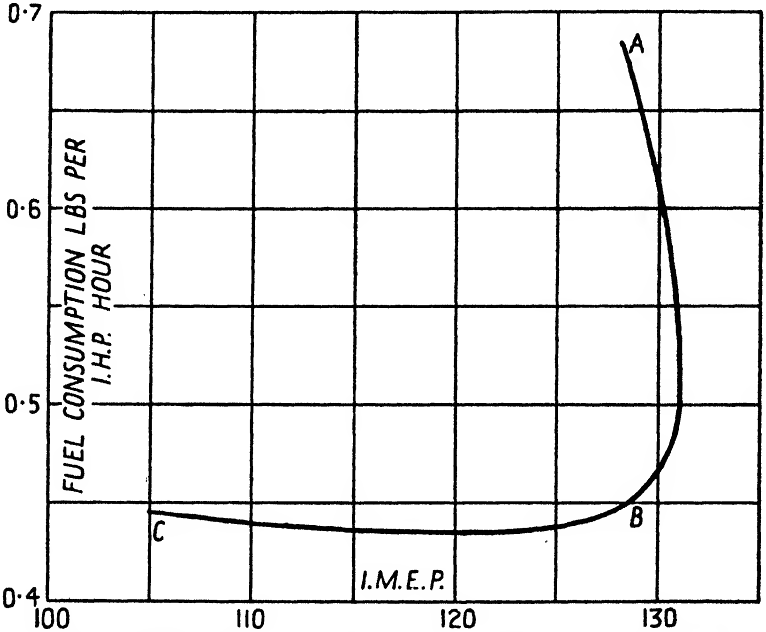

FIG. 75. Typical 'consumption loop'. Speed, 1,500 r.p.m. Compression ratio, 5:1. Fuel, petrol.

It will now be clear that, unless we know at what point we are working on this consumption loop in any particular experiment, a mere measurement of horse-power and fuel consumption may tell little or nothing about the economy of the engine. If the carburetter adjustment were such that the experiment was represented by the point *A*, it would mean that a lot of fuel was being wasted and that we were not even getting all the power we might have done with a less rich mixture. The excessive richness of the mixture would have been evident to an experienced tester from the yellow colour of the flames from the exhaust ports, if they could be seen, and men get very skilful at judging carburetter adjustment from the look of the exhaust gas; but, nevertheless, the plotting of a consumption loop, as the result of a series of observations, is the only way of getting full and satisfactory information about the fuel economy of an engine.

The length of the approximately horizontal portion *BC* of the consumption loop is an important characteristic of the engine and of the fuel used, for it tells the range of power over which a good economy can be maintained before the limit is reached, at *C*, because the fuel-air ratio is too weak for rapid and regular combustion. The curve of fig. 75 is for a single-cylinder engine. In a multi-cylinder engine the economical range of fuel-air ratio on the weak side is always narrow in comparison because, as explained in arts. 41 and 42, there is always some inequality of fuel distribution, and so soon as the first cylinder becomes starved of fuel the limit of steady running for the engine will have been reached; even though the majority of the cylinders may still be receiving uneconomical fuel-air mixtures. The curves *A*, *B*, and *C* in fig. 76 show a series of consumption loops plotted against B.M.E.P. for a four-cylinder engine, all taken at the same speed of 1,000 r.p.m. but at different throttle openings, so that the range of B.M.E.P. is different for each loop. In curve *A* there is scarcely any range of constant fuel consumption per horse-power, corresponding to the horizontal portion of the curve in fig. 75, but for 10 lb. per sq. inch above and below 100, at which the minimum point occurs, the increase in fuel consumption is not more than 5 per cent. The curves *B* and *C*, for the partially throttled conditions, show progressively less range of power control by fuel-air ratio, owing to the increasing proportion of residual gas to fresh charge when the engine is throttled. This slows down the combustion too much to allow of the power being maintained down to the weaker mixtures.

This effect of dilution with exhaust gas under throttled conditions is brought out by the curve through the points *D*, *E*, and *F*, which shows the ignition advance necessary to give maximum power at the correct fuel-air ratio, at each throttle position. It increases, on account of the slower burning, from 14° at full throttle to 30° when the engine was throttled so as to give about one-third of its maximum output.

Measurements of minimum fuel consumption per horse-power, by themselves, tell nothing about the thermal efficiency, and it is only because the calorific values per pound of nearly all the liquid fuels are substantially the same, that for practical purposes we can use the fuel consumption figure to express the heat economy of an engine. This point is well brought out in fig. 77, in which fuel consumption loops are given for the single-cylinder engine using alcohol and petrol at the same compression ratio of 5:1. The curve for pure benzene has also been added, in order to show that with the two fuels, petrol and benzene, which give identical thermal efficiencies (see fig. 60), the minimum consumption expressed in pounds per horse-power is some 4 per cent. greater on the benzene, on account of a difference in the calorific value. It may be mentioned that the petrol was itself rich in aromatics, and the difference in calorific values, therefore, was not the full difference between paraffins and aromatics when pure. If the consumption had been expressed in pints, instead of pounds, per horse-power, the consumption of benzene would have been 7 per cent. less

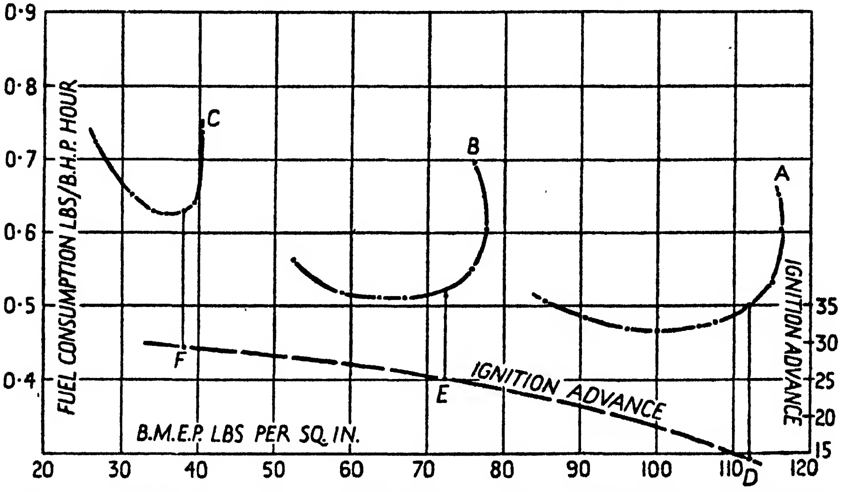

FIG. 76. Consumption loops at three different throttle openings. Four-cylinder petrol engine. Constant speed, 1,000 r.p.m.

instead of 4 per cent. greater, for the specific gravity was 0·88 as against 0·78 for the petrol.

It is very necessary, therefore, to recognize the importance of the units in which fuel consumption is expressed, and the nature of the fuel consumed. Although fuel consumption per horse-power is a useful and readily apprehended figure, the only way of giving unequivocal data about the performance of an engine itself is to state its thermal efficiency.

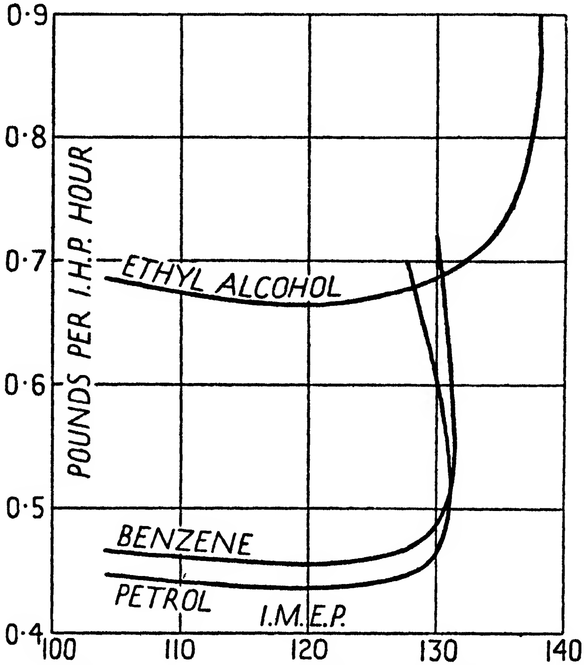

FIG. 77. Consumption loops with different fuels. Speed, 1,500 r.p.m. Compression ratio, 5:1.

In spite of the fact that the lowest consumption of alcohol at the 5:1 ratio, in the tests of fig. 77, was 0·67 lb. per I.H.P. hour as against 0·43 for petrol, the thermal efficiency was actually rather higher, being 33·5 as against 32 per cent. The difference in thermal efficiency is due to the lower temperatures which prevail throughout the alcohol cycle, on account of the large amount of cooling during its early stages, caused by evaporation of this fuel with an exceptionally high latent heat. The effect on volumetric efficiency of this cooling, which was noted in the last article, is reflected in the fact that the maximum I.M.E.P. using alcohol was 138, and increased up to the richest mixtures, whereas with petrol the I.M.E.P. shows a maximum of 131·5 at a consumption of 0·5 lb. per I.H.P. hour.

In spite of the higher power and higher thermal efficiency with alcohol, its low calorific value and high rate of consumption must prevent its competing with petrol unless sold at about two-thirds of the price, if it were to be used in the same engine. Alcohol, however, has the compensating advantage that it is practically a non-detonating fuel. At any rate it will burn quite satisfactorily at 7·5:1 compression, and it is of some interest to see how nearly alcohol as a fuel could compete with ordinary petrol on a basis of weight consumed per horse-power, if full advantage be taken of the possibility of using it in a high-compression engine. In fig. 78 the consumption loop for petrol at 4:1 is given side by side with that for alcohol at 7:1, and it will be seen that the minimum consumptions per horse-power are 0·50 and 0·60 respectively, while the maximum power with alcohol is nearly 50 per cent. greater for the same size of engine, the maximum I.M.E.P.s being 122 and 170 lb. per sq. inch. In a comparison of consumptions on a volume basis of gallons per I.H.P. hour the alcohol would approach even

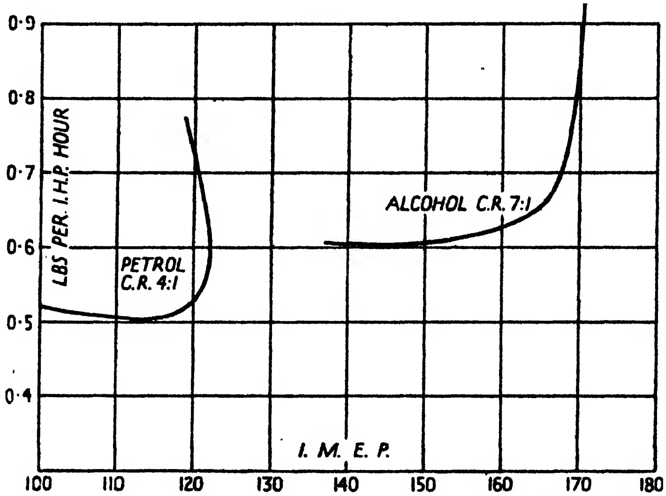

FIG. 78. Comparative consumption loops with petrol and alcohol at low and high compression ratios respectively.

closer to the petrol, for its specific gravity was 0·815† as compared with 0·78 for the petrol.

As a final example of the consumption loop type of diagram there is reproduced in fig. 79 a series of four curves for benzene at compression ratios 4, 5, 6, and 7 to 1, from which the variations of both maximum power and efficiency with change of ratio can be read off. From the positions of the horizontal tangents to each curve, namely at 0·53, 0·46, 0·42, and 0·39 lb. per I.H.P. hour, and the effective heating value of the benzene, 9,735 C.H.U. per lb., the maximum thermal efficiencies of the engine at each ratio can be calculated and are as given in fig. 52.

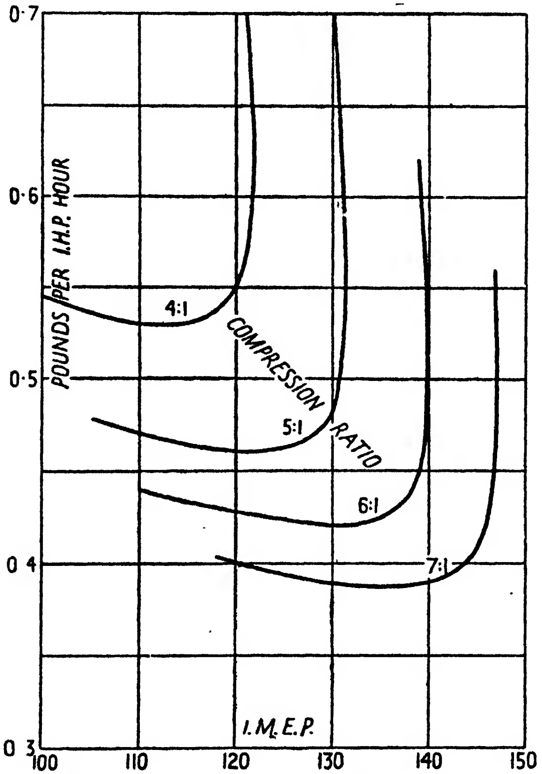

Fig. 79. Consumption loops at four different compression ratios. Speed, 1,500 r.p.m. Fuel, pure benzene.

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† The alcohol contained 5 per cent. of water.

ART. 59. Detonation and the 'Octane Number' scale

It has been emphasized already that perhaps the most important characteristic of a volatile liquid fuel is its behaviour in regard to detonation. Any statement of quality of a fuel must define its behaviour in this respect. Hitherto in this book we have done so in terms of H.U.C.R. values, as defined in art. 29, and all the stated values have been those found in the Ricardo E. 35 engine under certain standard conditions.

H.U.C.R. values found in this way enable us to place a range of fuels in a certain order of merit, but it by no means follows that any other engine, under different conditions, would operate satisfactorily with a certain spirit because its compression ratio was the same as the H.U.C.R. of the spirit as found in the E. 35. The compression ratio to avoid detonation would depend very much upon the speed, as well as upon a host of other factors. It does not even follow that the behaviour of a series of spirits in some other engine would reproduce the order of merit of their H.U.C.R.s as found in the variable compression engine.

It is very important, therefore, to devise some way of defining the quality of any fuel as regards detonation which does not relate only to one particular engine. It is important, too, that the checking of fuels should be able to be made in a test engine as simple and as cheap as possible, by comparatively unskilled testers, instead of one's being dependent upon an elaborate and costly piece of research apparatus like the E. 35 variable compression engine.

The first stage is to decide upon a *scale* of detonation which is, so far as possible, independent of any particular engine. Such a scale has been adopted in terms of two pure hydrocarbons, one of which is known to be very good as regards detonation, and the other very bad. The two substances chosen are 'Iso-octane' and *n*-Heptane, first suggested by Dr. Graham Edgar,[38] as the good and the bad partners respectively. These both comply with the first essential, namely, that they should be obtainable in a state of high purity, and they have the further important advantage that the volatilities of mixtures of the two are not very different from those of the petrols to be matched against them.

Mixtures containing from zero to 100 per cent. of iso-octane in *n*-heptane will vary from something having very low anti-knock qualities to what is almost a non-detonating fuel. The anti-knock quality of any fuel, when compared with the standard, can then be defined by its 'Octane Number', that is to say, the percentage of iso-octane in the iso-octane-heptane mixture which exactly matches the fuel under test.

This does not, of course, make the scale of detonation independent either of an engine or of the testing conditions, for what the Octane Number of the standard mixture may be when it is just matching some fuel under test will depend upon the design of engine used and the conditions of the test in regard to speed, cylinder temperature, air temperature, ignition advance, and mixture strength.

It is, therefore, necessary, in conjunction with the scale of Octane Numbers to agree upon a standard test engine and conditions for the test. The 'C.F.R.' test engine[59] has been widely adopted by international agreement and except for some of the highest grades of aviation fuel it has proved very satisfactory. The best conditions to standardize for the test have been the subject of much discussion, and there is some difference of opinion in different countries; but this is a mere matter of detail which need not concern us here—it will be referred to again in the next article. It is of interest, however, to give the correlation between the Octane Number scale and H.U.C.R. values as determined in the E. 35 engine. Table 32 gives the H.U.C.R. values of a series of iso-octane-heptane blends determined in the standard manner, and the position in the scale of certain typical classes of fuels.

TABLE 32

Correlation between Octane Numbers and H.U.C.R. values in the E. 35 engine under standard conditions

Fuel	*Octane Number*	*H.U.C.R. in E. 35 engine*
	0	3·75
Inferior Automobile Spirit	30	4·5
	40	4·75
	50	5·0
Average Automobile Spirit	60	5·3
	70	5·7
Aviation spirit	75	5·95
	80	6·2
High duty aviation spirit	85	6·5
	90	6·8

The above correlation was obtained by direct observation upon iso-octane-heptane blends in the E. 35 engine. It does not follow that a fuel of which the Octane Number was, say, 75, when matched against the standard *fuel in a different test* engine, would necessarily show an H.U.C.R. of 5·95 when tested in the E. 35. It must be understood that the correspondence of Octane Numbers and H.U.C.R. values will be affected by the chemical and physical nature of the fuel, and by the conditions of test. That is why it is so important to have one standard type of engine and standard conditions of test, according to which a uniform scale of fuel rating may become general.

For a description of the C.F.R. test engine and discussion of the conditions of test reference may be made to the original sources.[59]

ART. 60. The effect of testing conditions on detonation. Methods of detection

It is not intended to deal in detail with the testing conditions laid down for the grading of fuels, but only to set out certain characteristics of fuel behaviour and engine conditions which affect detonation and have to be allowed for when settling the test conditions, details of which can be found in the references given at the end of the last article.

Since the influence of test conditions on detonation is most easily apprehended in terms of a change of H.U.C.R., this method will continue to be used. As stated earlier, in art. 29, the figures for H.U.C.R. must always relate only to the particular engine on which the test was made. The following results refer in all cases to the Ricardo E. 35 variable compression ratio engine.

The compression ratio at which a fuel will begin to detonate will depend upon the speed of the engine, the fuel-air ratio, the ignition timing, the number and size of the valves, the design of the combustion space, and the amount of pre-heating given to the ingoing charge. Ricardo has given figures for a number of different fuels, as obtained on his E. 35 engine, under certain conditions. A selection from these values, and from figures obtained by Callendar and his associates, has been given in table 12. All Ricardo's detonation tests were made at 1,500 r.p.m., with mixture strength and ignition advance adjusted so as to give maximum power, this being also, for most fuels, the mixture strength which shows the greatest readiness to detonate. A standard outlet temperature of 60° C. for the cylinder-cooling water was also adopted, and an amount of heat input to the carburetter equal to 36 C.H.U. per minute of 0·024 C.H.U. per revolution. The standard conditions adopted by Callendar and his associates,[12] in their work on detonation, using the same design of engine, were similar, except that in order to eliminate one variable and so to simplify the technique of the tests, they adopted a constant ignition advance of 30° before the dead centre. Since all tests were at the same speed, the difference in the conditions as compared with Ricardo's was unimportant.

The question of mixture strength is important. This is illustrated in fig. 80, where the line *AB* shows how the H.U.C.R of a typical petrol varies according to the mixture strength. The greatest readiness to detonate occurs at the maximum power mixture, 20 per cent. rich. Fuels containing metallic dopes, and the naphthene fuel cyclohexane, exhibit the peculiarity of detonating most readily on weaker mixtures. The variation of H.U.C.R. for cyclohexane is shown by the curve *CD*, and at *EF* is the relationship for the same petrol as that of the curve *AB*, but with 0·2 per cent. by volume of ethyl fluid added (9 c.c. per gallon).

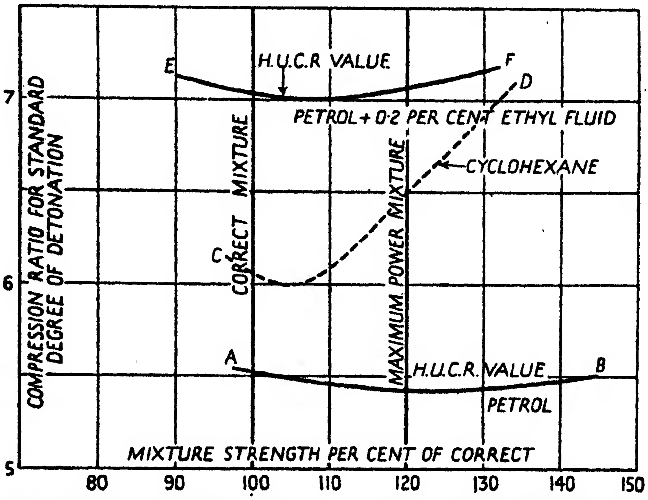

FIG. 80. The variation of H.U.C.R. with mixture strength for three different fuels.

Apart from the cathode-ray indicator (referred to in arts. 30 and 55) and from simply listening for the characteristic metallic knock, three methods of detecting detonation have been developed. These are the 'bouncing pin',[17] the maximum pressure indicator, and the 'temperature plug'.

It was pointed out in Chapter IV that when detonation occurs, pressures may be developed locally far in excess of the normal maximum pressures. These sharp rises of pressure are made use of in the bouncing pin apparatus of Midgeley and Boyd to throw a heavy pin off its seat, so as to make contact at its other end with an easily deflected spring. The duration of contact with this spring is a rough measure of the original momentum of the pin, and this, in its turn, of the violence of the blow from the detonation wave which gave it the momentum. A record of the number of contacts with the spring, and of the length of their duration, is obtained by the amount of gas produced in an electrolytic circuit which is closed by the contact. The apparatus works very well for comparing the violence and persistence of the detonation with two different fuels; its weakness lies in the fact that it is not satisfactory for detecting incipient detonation, because a certain violence of blow is necessary to displace the bouncing pin. It is very difficult, moreover, to maintain the adjustment of the various parts of the apparatus so that it reacts equally, from day to day, to similar degrees of detonation as judged by other methods.

It is unquestionably important sometimes to have a more positive method of maintaining the same standard between one observer and another, as to what is 'incipient detonation', than is possible when this is left to personal judgement of the noise of the characteristic knock, more especially when high-power and noisy aero-engines are in question. With these, incipient detonation may sometimes be accompanied by no clearly definable noise, and may only be detectable by what may be termed harsh running. A more satisfactory way of getting over the difficulty than by relying on auditory observation is by the use of an adaptation of the Farnborough indicator which was described in art. 55. If the indicator be connected in the normal way to the engine cylinder, and the balancing pressure is set to one, two, or three hundred pounds per sq. inch above the normal maximum pressure, then the incipient detonation will be marked by an occasional record of an excessive pressure, the frequency depending upon how much the balancing pressure exceeds the normal. A convenient form of this detonation indicator has been worked out in which, as the balancing pressure is gradually raised, its magnitude is indicated on the usual type of 'dial and pointer' pressure gauge. Every time the cylinder pressure exceeds the balancing pressure a spark passes from the tip of the pressure gauge pointer on to the scale behind it. A standard of detonation is established in terms of a certain number of sparks per minute at some agreed value of the balancing pressure.

The third, temperature plug, method of detecting detonation depends upon the fact that as soon as detonation begins there is a rise of temperature at the surface of the cylinder head. If a metal plug incorporating a platinum resistance thermometer or a thermocouple element is screwed into a sparking-plug hole, the temperature-rise indicative of incipient detonation can be observed before any knock can be heard.

Opinions differ as to the convenience and reliability of the various methods, and, indeed, these differ according to circumstances. With the Ricardo E. 35 engine, which has been largely used for comparing H.U.C.R. values through its having been first in the field, the bouncing pin cannot be used for reasons of design; and, although the maximum pressure indicator could be used, auditory methods have been found more satisfactory. A skilful observer can repeat observations of H.U.C.R. with fair certainty by auditory methods to 0·05 of a ratio. An examination of fig. 19 on p. 112 will show that this corresponds to an addition of about 2 per cent. of benzol, or 0·26 c.c. of tetra-ethyl-lead per gallon.

A weakness of all three methods referred to above, and one from which the cathode-ray indicator does not suffer, is that the amount of the effect which is relied upon for indication, whether it is a rise of pressure or of temperature, depends very much upon the position of the indicating point in the cylinder. The sharp rises of pressure and temperature associated with detonation are very local, and, if the indicating point is remote from the centre of disturbance, it may be impossible to get any useful indication at all. The cathode-ray indicator, giving as it does a true record of the pressure wave system which is typical of detonation, travelling to and fro across the cylinder head, should not be subject to the same drawback, and promises to provide a really satisfactory indicator of detonation when developed for this purpose.

Art. 61. Combustion chamber efficiency

It has already been explained in art. 41 that the real efficiency of a multi-cylinder engine, as a machine for turning heat into work, may be obscured by irregularity of fuel distribution to the cylinders, if the efficiency is calculated from the minimum fuel consumption per horse-power. A rough figure for the real efficiency of an engine, what we may call its combustion chamber efficiency, can be found from measurements of the air consumption per horse-power, as explained in art. 42, owing to the fortunate circumstance that the complete combustion of one pound of air with the appropriate amount of any hydrocarbon fuel, yields very closely the same amount of heat. If an engine uses the air which it takes in economically, it is an efficient engine so far as its cylinders are concerned, although these may have the misfortune to be tied on to an inefficient induction system; but if the cylinders do not use the air efficiently, no amount of skill or good luck with the induction system can make the engine efficient.

The efficiency of a combustion chamber will depend first and foremost upon its characteristics in regard to detonation. It was stated in Chapter IV that detonation, while primarily a fuel characteristic, was also affected by combustion chamber design. Two engines may behave very differently, even when using the same fuel at the same compression ratio, and the efficient design is that which allows the highest ratio to be used with any given fuel. Efficiency from this point of view will go with a compact shape, combined with sufficient but not too much turbulence; and this in its turn will depend upon the size and arrangement of the inlet valves.

The term 'turbulence' has been used repeatedly in this book, and the reader should by now have a fairly clear idea of what is meant, although no definition has been given. It is scarcely possible, unfortunately, to frame a satisfactory definition, because one cannot measure turbulence as such, but only the effects it produces. When dealing with organized air swirl in the Ricardo type of sleeve-valve compression-ignition engine, in art. 39, figures were given for the rate of air swirl about the cylinder axis in terms of engine r.p.m. These figures were obtained from the readings of an anemometer placed in the cylinder head when the engine was motored round at various speeds. Admittedly, the anemometer readings would not give true speeds of air movement in the cylinder, but at least they provided figures which could be correlated consistently with the rate of burning when fuel was introduced into the cylinder.

In a poppet-valve engine, or with a sleeve-valve when the air ports are not arranged tangentially, there must clearly be a great deal of irregular eddying motion of the gases in the cylinder at the end of the induction stroke, which dies down during compression. This eddying motion, or turbulence, is set up by, and must be roughly proportional to, the entering speed of the air charge through the inlet valves. In a given engine this gas velocity of entry will be proportional to engine speed. Since the major part of the heat lost to the cylinder walls is by convection and conduction, the rate of loss of heat will be roughly proportional to gas turbulence, and therefore to engine speed.

Compactness of general shape in the combustion chamber is important for the avoidance of pockets in which stagnant gas may collect and set up detonation; next in importance is adequate cooling of the exhaust valve seats, and so of the valves; and, finally, a placing of the sparking-plug or plugs so as to take advantage of the valve arrangement, and of the turbulence, peculiar to each design. A certain degree of turbulence is essential for promoting rapid combustion, scouring unburnt fuel from on or near the walls, and avoiding detonation due to stagnant pockets of gas. Experiments show that as turbulence is increased within certain limits, power output, thermal efficiency, and heat flow to the cylinder walls all increase simultaneously. That thermal efficiency and heat loss should increase at the same time sounds contradictory; but the explanation is that the greater turbulence ensures the rapid burning of a larger proportion of the fuel. With a low degree of turbulence quite an appreciable proportion of the fuel-air mixture may burn so slowly that its heat may only be developed late in the expansion stroke, and so may add merely to the heat carried away to waste through the exhaust valve.

It is possible, on the other hand, to have too much turbulence, which will be detrimental to efficiency through its promoting heat loss to the cylinder walls and, probably more important, through its effect upon combustion. Excessive turbulence may produce harsh running, and it has even been found possible, with turbulence of the organized swirl type, to prevent combustion altogether. In fig. 81 are given two diagrams from the same engine with very different degrees of turbulence, the one normal, the other excessive. Under conditions of normal turbulence, and with ignition 28° before the dead centre, the pressure rises steadily at the rate of about 17 lb. per sq. inch per degree of crankshaft revolution. With excessive turbulence, although ignition was retarded until 12° before the dead centre, the rate of pressure rise was over 50 lb. per sq. inch per degree, and a very rough running engine of lower power output was the result. What, precisely, is meant by a 'rough running' engine is as difficult to define as the meaning of turbulence. It was said in art. 39, when discussing combustion in the Diesel engine, that roughness is more easily apprehended from experience of an engine on the test-bed than described in words, and this is equally true of the petrol engine. A rough running engine is noisy, without necessarily giving out any of the characteristic knock of detonation; there is also a certain amount of general vibration, unless the engine is very solidly mounted on the test-bed; and if the roughness be allowed to continue for long both the connecting-rod and main bearings will probably suffer.

The desired smoothness of running and its relation to turbulence is a point of great importance. Apart from the average rate of pressure-rise, smoothness appears to depend particularly, in some designs at any rate, upon the rate of rise at the very beginning of combustion. Fig. 82 shows a cross-section, and a plan view, of the Ricardo design of 'turbulent head'. The aim of this design is to get over the difficulty that in a poppet-valve engine, when side-by-side valves are used, it is impossible to obtain enough turbulence at the end of the compression stroke, if the turbulence be derived from the entering velocity of the gases alone. In the Ricardo head there is a restricted passage between the swept volume of the piston and the combustion space proper, which surrounds the valves. During the compression stroke, and especially during the last stages when ignition occurs, air is forced through the restricted passage from the cylinder barrel into the combustion space, where it creates a vigorous turbulence just at the moment when it is required. In some interesting experiments[26] with one of these turbulent heads, the effect was explored of varying the clearance, '*d*' in fig. 82

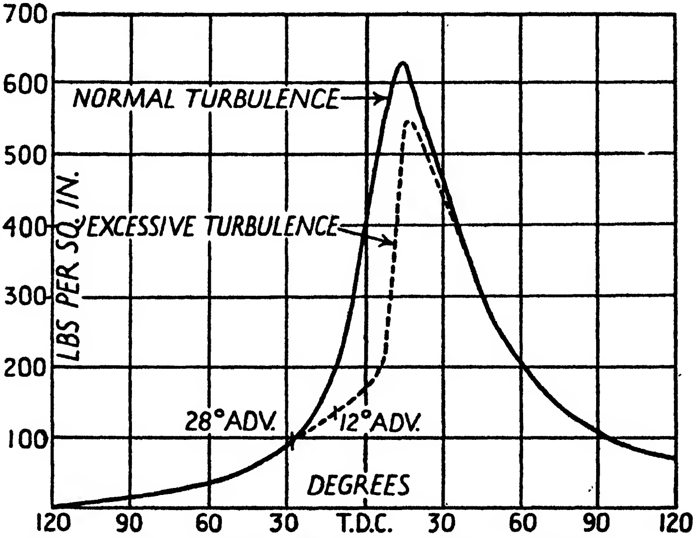

FIG. 81. Indicator diagrams showing the influence of air turbulence in the cylinder upon rate of combustion in a petrol engine.

between the cylinder head and the top of the piston when at the dead centre. The sparking-plug was over the piston in the position indicated. In fig. 83 are shown the two indicator diagrams obtained with maximum and minimum values of the clearance *d*, the compression ratio being arranged to be the same in each case. It will be noticed that although the rate of pressure-rise above 200 lb. per sq. inch is identical in the two diagrams, the way in which it gets going is very different. The beginning of the pressure-rise is longer delayed and much more sudden with the small clearance. This is, no doubt, because in these conditions the flame spreads at first less rapidly from the sparking-plug, owing to the proximity of the comparatively cool cylinder head, and the thin layer of burning gas. During the last tenth of an inch of movement of the piston, a fan of burning gas is projected at high velocity into the rest of the combustion space. The consequent very rapid initiation of combustion throughout the bulk of the mixture produced a very rough running engine; while with the larger head clearance of 0·272 inch the engine was noticeably smooth, although the compression ratio and maximum pressure were the same as before.

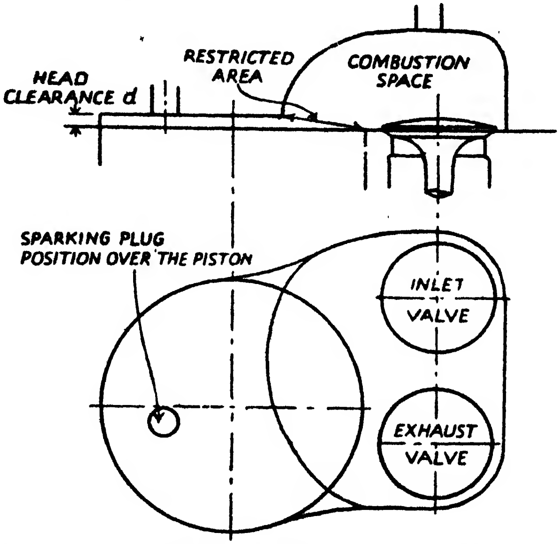

Fig. 82. Elevation and plan of Ricardo 'turbulent head'.

The results of these experiments upon a special type of cylinder head are of limited applicability, but have been quoted as illustrating the important practical point that roughness of running is a feature which depends upon the *rate* of pressure-rise, and more especially upon the rate of rise just at the beginning, before or just after the dead centre. Roughness of running can be controlled, therefore, when all the factors controlling the rate of combustion are properly understood.

In a compact combustion chamber, without the peculiar features of those illustrated in fig. 82, turbulence during combustion will, apart from engine speed, depend upon the size of the inlet valves and their relation to the shape of the cylinder head. An arrangement of valves which gives to the air upon entry an organized swirl, as distinct from miscellaneous eddying motion, will enable the air movement to persist throughout the compression period and assist combustion after passage of the spark. Whether or not such an organized air movement is achieved, a well-designed cylinder head in which air movement is not impeded, and which has a low surface-to-volume ratio, will give improved efficiency through not allowing unburnt mixture to hang about near the cool walls and so escape combustion.

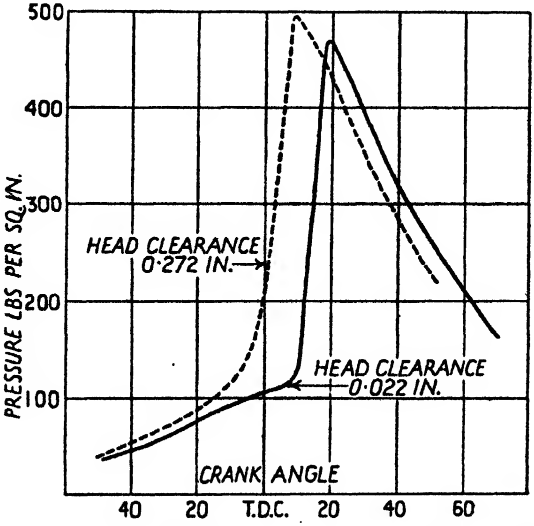

FIG. 83. Indicator diagrams showing the effect of a variation of the head clearance in a Ricardo turbulent head.

Increase of speed will have the dual effect upon turbulence of increasing the initial gas velocity through the valves, and reducing the time during which the initial turbulence can die down. The rapid increase of detonation at low speeds, which is common to all engines, is partly due to the corresponding reduction of turbulence, for then the conditions outlined above are reversed. At low speeds there is more likelihood that stagnant pockets of gas may collect near the exhaust valves, and thus provide the very conditions to promote detonation, more especially if the exhaust valve is placed in a side pocket of the combustion space, and is not well cooled.

Art. 62. Analysis of mechanical and pumping losses

It has been stated in art. 55 that the indicator cannot be regarded as an accurate instrument for determining indicated horse-power. We are driven back, therefore, upon a measurement of B.H.P. and of lost horse-power,† and upon adding the two together, to find the I.H.P. Since it is impossible to measure the losses while the engine is running under load, it becomes important to estimate how nearly the horse-power needed to motor the engine round gives an accurate measure of the losses, both at full load and at less than full load.

It is essential, in the first place, to make the motoring test immediately after running under power. Of the total lost power, nearly 60 per cent. is due to piston friction and any alteration, either of cylinder wall temperature or of the condition of the lubricating oil, from those which obtain under normal working conditions may introduce appreciable errors. When running under power the oily surface on the cylinder walls becomes contaminated, and it is found that after a few minutes motoring the clean oil thrown up from the connecting-rod big-ends leads to a drop in the friction losses below the full power value. Apart from this gradual change of piston lubrication, there will be a reduction in the friction loss at the piston immediately an engine stops firing, on account of the reduced gas pressure and side thrust. Fortunately, this reduction in the friction loss is just about compensated by a simultaneous increase in the pumping loss, by which is meant the power required to get the working substance into, and out of, the cylinder. This pumping loss will be different when an engine is being motored from when running under its own power, because of the very different pressure in the cylinder at the moment the exhaust valve opens. When it is being motored there will be none of the momentum effect referred

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† The term 'lost' horse-power is here used to represent the difference between the brake horse-power and the indicated horse-power, as this would be measured by the perfect indicator if such a thing existed.

to in arts. 11 and 57. In fig. 84 are shown two light spring diagrams for the exhaust and suction strokes taken from the same engine running at the same speed of 2,000 r.p.m. The dotted line diagram shows the pressures when running under its own power at a B.M.E.P. of 90 lb. per sq. inch, and the full line diagram the pressure when it was being motored. The depression of the cylinder pressure below atmospheric during the suction stroke depended only upon the mean gas velocity through the inlet valves, which in this engine was 125 feet per second at 2,000 r.p.m. The depression was therefore the same under the two conditions of the diagrams. The average excess of pressure in the cylinder during the exhaust stroke, however, which is a measure of the work done in ejecting the used gases from the cylinder, is, when under its own power, reduced practically to zero, as compared with 3·5 lb. per sq. inch when it was being motored round. The reason for this, as explained earlier, is that under the pressure of 50 lb. per sq. inch or so in the cylinder when the exhaust valve opens, the first blast of escaping gases acquires considerable momentum which assists the outflow of the remainder. During the middle part of the stroke, while the valve was fully open, the pressure in the cylinder dropped to well below atmospheric, and the diagram shows that exhaust gas was therefore

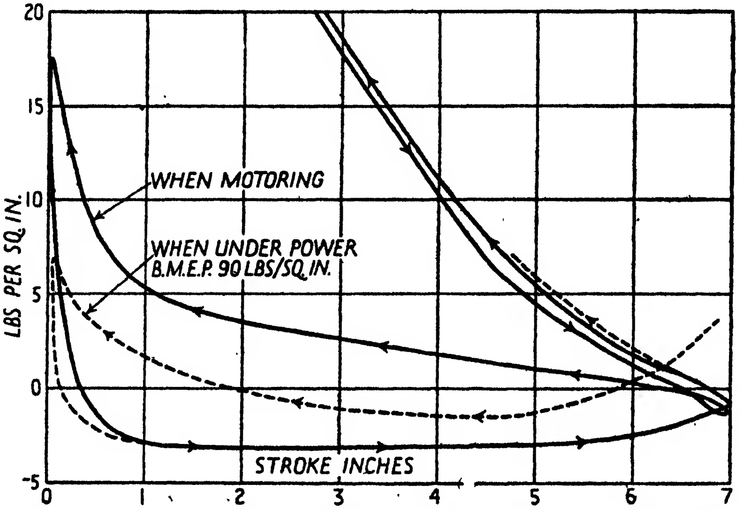

FIG. 84. Light spring indicator diagrams from a single-cylinder Diesel engine, exhaust and suction strokes, when motored and when under power and giving a B.M.E.P. 90 lb. per sq. inch.

being sucked out of the cylinder. When motoring, on the other hand, the ejection of the exhaust gas depends wholly upon the motion of the piston, and the pressure given by the full line curve is the result.

Since any one test will be done at constant speed, it is convenient to express all the various losses in terms of a mean effective pressure per working stroke of the piston. From fig. 84 the combined pumping loss during the exhaust and suction strokes can be directly measured as 7·5 lb. per sq. inch when motoring, and 4·0 lb. per sq. inch under power.

A minimum figure for the mechanical friction loss of an engine may be obtained by motoring it with cylinder heads removed, if that is possible. Owing to the gas loading during compression the friction under normal conditions of motoring will be slightly greater than the minimum so found, although itself less than the friction when under power. The difference between the total motoring loss, and the sum of the pumping and the minimum friction loss, is made up of the extra friction loss due to gas pressure when motoring, and the leakage and heat loss during compression and expansion. Leakage may take place past the piston or past the valves. It may be serious in an engine due for overhaul, but in a well-tuned, high-speed engine it should be negligible. The effect of leakage and heat loss is, of course, to make the work recovered by expansion less than the work of the compression stroke. The amount of the heat loss becomes important in certain designs of high-compression injection engines, but for compression ratios up to 7:1 it is small and, as already stated, the increase of the friction loss under load is just about compensated for by the decrease of the pumping loss, so that it may be taken that the power required to motor a petrol engine immediately after running at full load gives, when added to the B.H.P., as accurate a figure for the I.H.P. as can at present be obtained.

The truth of this generalization, and the proportions in which the various factors contribute to the observed power required for motoring, must necessarily depend to some extent upon the particular engine and the speed; for the pumping loss will depend upon the mean valve areas, as well as upon speed, and the friction will depend upon the weight and design of the pistons. It may, however, be useful to give some typical figures, first for a petrol, and then for a high-compression injection engine.

Taking as an example of the former the kind of figures to be expected from a high-speed petrol engine of compression ratio about 5·5:1, this should give a B.M.E.P. of 125 lb. per sq. inch at full load, and at the normal speed of, say, 2,000 r.p.m.; and when motored under the same conditions of speed, temperature, etc., the power absorbed would be equivalent to about 18 lb. per sq. inch M.E.P. Fig. 85 shows the probable subdivision of the losses at full load between speeds of 1,000 and 2,000 r.p.m., expressed as M.E.P. on the pistons. When motoring, the pumping loss at 2,000 r.p.m. was probably 2 or 3 lb. per sq. inch greater, and the friction loss less by the same amount.

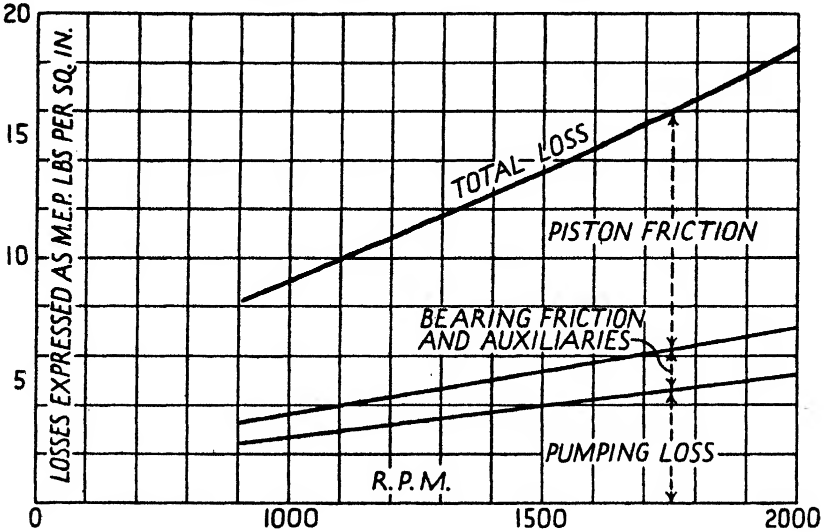

FIG. 85. Approximate subdivision of losses in a high duty petrol engine.

In vol. II of this book a large amount of data upon friction and pumping losses in various types of aero-engine has been collected together, and an analysis made of the dependence of the losses upon speed, throttle opening, and other factors. For further information reference should therefore be made to vol. II.

In the high compression ratio injection engine conditions are different from the petrol engine in two respects. In the first place, the heat loss during compression and expansion at the high ratios may make the power taken in a motoring test considerably greater than the lost horse-power at full load; and secondly, under reduced loads the air may not be throttled and the pumping loss may remain therefore substantially constant. There may be a slight increase of pumping loss on the exhaust stroke, owing to less momentum in the exhaust gases, but this increase is not more than 10 per cent. of the full load pumping loss even when running light. Friction loss will decrease with the load, and since there may be no compensating increase in the pumping loss, as there is in a throttled engine, the total lost horse-power, instead of increasing, may be expected to decrease somewhat with the load.

There appears to be a substantial difference, in regard to power absorbed while motoring, between high-compression engines of the sleeve-valve, and those of the poppet-valve types. As a quantitative example of a sleeve-valve engine we may quote an analysis made by Ricardo of the losses in the single-cylinder engine, of compression ratio 14:1, upon which the diagrams of fig. 84 were obtained.

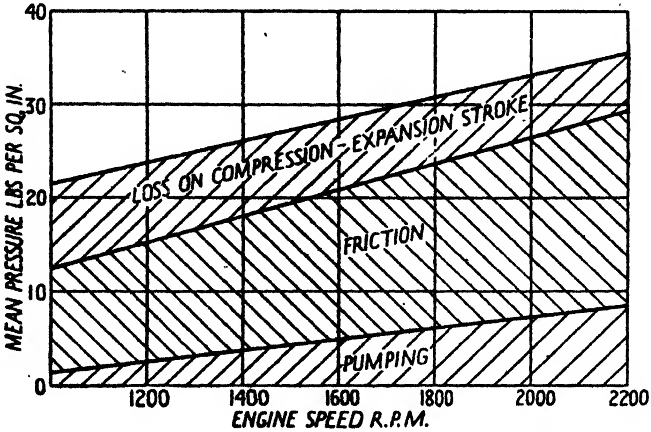

FIG. 86. Subdivision of the power required to motor round a single-cylinder sleeve-valve Diesel engine at different speeds. Compression ratio, 14:1.

For that engine fig. 86 gives the actually measured motoring loss between speeds of 1,000 and 2,200 r.p.m., with the estimated subdivision into compression and expansion loss, friction, and pumping. At 2,000 r.p.m. the total motoring loss was equivalent to 33 lb. per sq. inch M.E.P. Of this Ricardo estimates that 6·5 lb. per sq. inch corresponded to heat loss on compression-expansion, and should not therefore be counted as part of that which has to be added to the B.M.E.P. to give the I.M.E.P. The remaining 26·5 lb. per sq. inch was made up, while motoring, as follows:

Pumping	7·5 lb. per sq. inch.
Friction	17·5+1·5 = 19 lb. per sq. inch.

Of the two parts of the friction loss the first was the actually measured friction loss with no gas pressure, and the second the estimated increase due to compression pressure. The pumping loss of 7·5 lb. per sq. inch was measured, while motoring, by means of light spring diagrams; and a similar measurement made under power conditions showed that it was then reduced to 4·0 lb. per sq. inch by elimination of the exhaust stroke part. An allowance for extra friction under power conditions, due to higher gas pressures, has to be made, and this Ricardo fixes at 1 lb. per sq. inch. We therefore have the figures of table 33 for the probable subdivision of the lost power for the two conditions of motoring and running under full power, both at 2,000 r.p.m.

TABLE 33

Probable subdivision of motoring and full load losses for single-cylinder Diesel engine of compression ratio 14:1 *at a speed of* 2,000 *r.p.m.*

	Motoring	*Under power*
	lb. per sq. inch	lb. per sq. inch
Pumping	7·5 (measured)	4·0 (measured)
Friction without gas pressure	17·5 (measured)	17·5 (measured)
Allowance for gas pressure	1·5 (estimated)	2·5 (estimated)
Compression-expansion loss	6·5 (by difference)	—
Total	33·0 (measured)	24·0 (estimated)

According to these figures of table 33, one estimates the motoring loss in a sleeve-valve engine, expressed as mean effective pressure, to be greater by 9·lb. per sq. inch at 2,000 r.p.m. than the difference between the B.M.E.P. and I.M.E.P. at full load at the same speed. As contrasted with this result, figures obtained when motoring a multi-cylinder poppet-valve engine, first as a petrol engine of compression ratio about 5:1, and then with the compression raised to about 14:1 showed an increase in the lost M.E.P. at 2,000 r.p.m. of only about 1 lb. per sq. inch. This result is confirmed, moreover, by others obtained upon the single-cylinder poppet-valve engine referred to in art. 52. At the end of that article it was mentioned that the engine had also been calibrated as a petrol engine of compression ratio 4·84:1. The motoring losses of that engine at its high compression ratio, 12:1, were found to show an increase of only 0·5 lb. per sq. inch at the normal full speed, 1,200 r.p.m., over the losses at the low compression ratio.

The small effect of raising the compression ratio upon the motoring losses in these two poppet-valve engines, as compared with the large difference observed by Ricardo on sleeve-valve engines, must be put down to heat loss during compression and expansion which is, no doubt, largely accelerated by the organized air swirl as well as by the form of the cylinder head. Nevertheless, the difference observed between the two types is surprisingly large.

Under reduced loads the pumping loss in either type of engine, for the reason already noticed, will be constant or will show a very slight increase. In the sleeve-valve engine it rose at 2,000 r.p.m. from 4·0 to 4·3 lb. per sq. inch between 90 and zero B.M.E.P. The friction loss when the B.M.E.P. is zero, that is, when the engine is running light, must drop to something not very different from its value under motoring conditions. The difference of friction between full load and motoring has been given above as 1 lb. per sq. inch, and therefore, from full load (90 B.M.E.P.) to no load at 2,000 r.p.m. the decrease of lost power may be taken to be of the same order. At other speeds the probable subdivision of losses can be estimated from fig. 86 and table 33. Actual figures can do no more, in any case, than illustrate the general principles which should govern an estimate of the losses in any particular engine. As already pointed out, the actual figures for an individual engine must always be governed by the design.

Art. 63. Analysis of heat losses

We have hitherto regarded a high thermal efficiency as being a desirable feature in an engine only because it means a low fuel consumption per horse-power. There is, however, another and even more important reason why it is a good thing. An engine of a given size and speed requires a definite maximum quantity of air in order to develop its full power, and from the combustion of this air a certain amount of heat is developed. The engine turns a fraction of this heat into work, and all the heat that it fails to convert is, as Ricardo has expressively put it, 'left over to make trouble'. It is certainly true to say that the majority of breakdowns among high-duty engines can be traced, directly or indirectly, to a failure to meet the very severe conditions of heat flow which are created by the necessity of getting rid of the waste heat which the engine has failed to convert into work. Failures traceable directly to excessive heat flow are seized and burnt pistons, cracked cylinders, pitted and burnt valves, and distorted valve seats; while most failures of lubrication may be traced indirectly to excessive temperatures which lead to contamination of the oil by sludge and carbon, and so to choked oil ways and bearing failures. These failures are all caused by the 70 to 80 per cent. of waste heat which is not converted into work. It might, at first sight, be supposed that whether this waste amounted to 70 per cent. or 80 per cent. of the whole heat supply would not make all the difference between a reliable and an unreliable engine. But consider two engines of the same size which are receiving the same amount of heat per minute. If the brake thermal efficiency of one is 20 per cent. and of the other 30 per cent., not only is the waste heat of the efficient one less by 12½ per cent., but its full power output is 50 per cent. greater. It follows therefore that to get the same power output from these two engines of the same size, we need only to run the efficient one at two-thirds of its full power and therefore with $\frac{2}{3} \times 87{\cdot}5 = 58$ per cent. of the waste heat. In other words, for the same power output, the inefficient engine has nearly twice the waste heat to be safely got rid of, and a vastly greater liability to breakdown. Ricardo has stated that, of the very many engines which pass through his hands during a year for test, it is in every case the efficient engines which stand up successfully to arduous endurance tests, and that all the common troubles of dirty oil, gummed up piston rings, and burnt-out valves, are consistently associated with the inefficient engines. It cannot be too strongly emphasized that the value of a high efficiency is not that of saving a few shillings' worth of fuel, so much as saving many pounds' worth of repairs or redesign; for the whole reliability and ultimately the cost of maintenance of an engine is bound up with the question of its inherent efficiency.

In order to understand the problem of dealing with the waste heat it is necessary to follow the history of the heat supply from the moment of its generation by combustion in the cylinder. The portions of this which can be directly measured are the portion which is converted into useful work on the piston, and that which is carried away by the cylinder jacket water. For this reason a first, simple, analysis of the heat supply is often given in the form shown in table 34, of which the figures are typical for a single-cylinder engine of compression ratio 5:1, and a speed 1,500 r.p.m.

The figures of table 34 are typical for a single-cylinder test engine at a particular speed, fuel-air mixture, and compression ratio; they must not be read as being generally applicable. The proportion of heat passing to the cylinder jackets in any particular engine will depend very much upon how much water-cooled surface there is

TABLE 34

Typical heat balance-sheet for an engine of compression ratio 5:1.

Heat to I.H.P.	31 5 per cent.
„ jacket cooling water	28·5 „
„ exhaust, radiation, etc.	40·0 „
Total	100·0

beyond the exhaust valve, and upon the speed; and it will be different at different fuel-air mixtures and compression ratios. In a multi-cylinder engine it will, in general, be less than in a single-cylinder test engine under the same conditions. This is due to the close grouping of the cylinders and their exhaust systems, which may lead to a smaller area of water-cooled surface per cylinder being beyond the exhaust valves.

Gibson[34] has given figures for the fraction of the total heat generated which finds its way to the cooling-water jackets in a number of different designs of aero-engines, and in some others. In these results the heat to the jackets per minute has been expressed as a fraction of the brake horse-power. At 1,500 r.p m. and a compression ratio of 5:1 the fraction varied between 0·8 and 0·9 in the different designs. In only one engine did the heat to the jackets substantially exceed the B.H.P. This was not an aero-engine but a low compression automobile engine of ratio 4·2:1, with an L-shaped cylinder head. In that engine at 1,500 r.p.m. the ratio to the B.H.P. was about 1·5. Gibson also gives figures from two air-cooled single aero-engine-cylinder test engines, of compression ratio 4·6:1, in which the ratio to the B.H.P. of the heat passing to the cylinder walls was 1·1.

In all the multi-cylinder aero-engines the I.H.P. must have been 10–15 per cent. greater than the B.H.P., so that the heat to the jackets, never more than about 0·9 of the B.H.P., was not more than 0·8 of the I.H.P. The engines tested by Gibson date between 1916 and 1919, and in a modern high-speed, high-compression engine the heat to the cooling water would be substantially less. Further data on this point are given in vol. II. As explained in art. 49, and in somewhat more detail below, these differences in heat loss to the cylinder walls may be expected to have only a negligible effect upon the thermal efficiency. The heat which does not reach the cooling water will nearly all go to swell the proportion of waste heat discharged with the exhaust gas.

In all aero-engines, and to a less extent in automobiles, there is a fair amount of heat carried off direct from the engine by the air stream over it, which does not therefore have to be dissipated by the radiator. An engine which passed an amount of heat to the cooling water equal to 60 per cent. of the B.H.P., when running on the test-bed and well enclosed, would only need a radiator capable of dealing with about 52 per cent. of the B.H.P. if the engine were tested in a wind of 60 miles per hour; and for less or greater wind speeds, roughly in proportion.

Turning now to the dependence of jacket heat loss upon speed and compression ratio, Gibson's figures show a small but consistent reduction of the ratio of this loss to the B.H.P. with increase of speed. The diminution is very nearly 1·5 per cent. per 100 r.p.m. Thus in an 8-cylinder V-type aero-engine the ratio dropped from 0·93 to 0·85 for an increase of speed from 1,000 to 1,600 r.p.m., corresponding to an increase of B.H.P. from 107 to 168. As previously explained, this comparatively slight decrease in heat lost from the working fluid per cycle follows from the fact that the shortened duration of each cycle is compensated for by the increased gas turbulence, and *rate* of heat loss; because gas velocities increase in proportion to engine speed.

As regards the effect of compression ratio, Gibson's results show a quite regular drop in the same fraction as before from 0·83 to 0·69 for a series of ratios 4·5, 5·0, 5·5, and 6 to 1, all the tests being at 1,600 r.p.m. The reduction of heat to the cooling water at the higher ratios will be chiefly in that part lost during the exhaust stroke. The higher the ratio of expansion, the more efficient will be the cycle, and therefore the cooler the working fluid when the exhaust valve opens.

It is the higher temperature of the exhaust gases, also, which may cause some increase in the proportion of heat to the cooling water when the mixture is excessively weak. The burning becomes slow and irregular when the fuel-air ratio is less than 80 per cent. of the correct one; so that combustion may continue throughout the expansion stroke, thus leading to a very inefficient cycle and to too much heat remaining unconverted into mechanical work when the exhaust valve opens.

The same conditions lead to overheated exhaust valves, and for the same reason; and when mixture weakness becomes excessive, so that burning continues even during the exhaust stroke until the inlet valve opens, ignition of the incoming charge leads to the familiar 'pop-back' through the carburetter.

For the practical problem of designing an adequate radiator for a given size of engine the useful figure is that given above, which expresses the heat to be dealt with per minute as a fraction of the B.H.P. For a thermodynamic analysis of engine performance, on the other hand, it is rather the indicated horse-power we should consider, for we are concerned only with the conversion and loss of heat in the cylinders themselves. The cylinders might be extremely efficient, and yet the whole engine might show a very poor brake thermal efficiency if the power actually available at the shaft were reduced by large friction losses in bearings, and in driving auxiliaries such as oil and water pumps. The existence of these losses would, of course, be shown up in the motoring test (see art. 54) and would be allowed for in arriving at the I.H.P.

In pursuing our discussion of what becomes of the heat generated by combustion every cycle we shall therefore start from the simple form of analysis given in table 34. The last line of that table, showing the quantity of heat wasted with the exhaust and by radiation, was obtained by subtraction of the other two items from the total, and this makes the table as it stands incorrect, for a reason now to be explained. Suppose the first line had shown the heat equivalent of the B.H.P. instead of the I.H.P., then the figure obtained by subtraction for the exhaust and radiation loss would have been increased by the difference between the indicated and the brake horse-power, although in point of fact most of the piston friction loss takes the form of heat carried away by the jacket water and must have been already included, therefore, in item 2 of the table.

If we put down the I.H.P. as an item in the heat balance-sheet, then, since part of the difference between the I.H.P. and B.H.P. leaves the engine as heat, the sum of the quantities in table 34 ought to add up to more than the total heat supply from the fuel per minute, by the heat equivalent of the power absorbed in piston friction.

About 40 per cent. of the total lost power is either pumping loss, or is radiated as friction heat from the bearings, and cannot be measured in any case. The only part of it for which a correction is important is the piston friction, lest this should be included twice over in the balance-sheet. To avoid this we ought, for strict accuracy when using the I.H.P., to subtract from the measured heat to the jackets that part of the lost power which is derived from piston friction, because in the mechanically perfect engine from which the whole I.H.P. would be available at the shaft the heat to the jackets would be so reduced.

In practice we are only interested in the I.H.P. and in the heat which actually passes to the jackets. We are also interested in the relationship of these two to the total heat supply, and to one another, but we are not really interested in the last item of table 34, and the error explained above is not important provided its existence and magnitude are understood. Piston friction loss accounts for about 60 per cent. of the whole mechanical loss in an average engine. It amounts, therefore, to about 6 per cent. of the I.H.P. or, say, 2 per cent. of the whole heat supply per minute.

Since we are vitally interested only in the first two items of table 34 it will be simpler and more convenient to retain the analysis in that form, while recognizing its slight inconsistency. The important thing to grasp about the relationship between I.H.P. and heat loss to the cylinder walls is the point already made in art. 49, namely, that even if it were possible to run an engine under truly adiabatic conditions in which this loss to the cylinder walls was entirely suppressed, that would only mean an increase in the heat to I.H.P. of some 4 per cent. of the total. The reason for this may be considered under two separate headings.

(1) About half of the 28·5 per cent. loss to the cooling water shown in table 34 (13·5 per cent. according to table 37) is given up to the cylinder walls and exhaust ports by the hot gases after the exhaust valve has opened, and the suppression of the whole of this would merely have added to the exhaust loss without affecting the power in any way.

(2) Of the heat which is lost to the cylinder walls before the opening of the exhaust valve, something like 10 per cent. of the total heat is lost during expansion, much of it during the later stages when the area of exposed surface is large. Only a very small part of this heat which is lost late on the expansion stroke could in any case have been converted into work, since the available expansion ratio remaining would have been so small.

Of the remaining 5 per cent., which may be taken as lost before any appreciable expansion has occurred, and which would, therefore, have been all available for conversion at the efficiency corresponding to an expansion in the ratio 5:1, the work equivalent would have been $0{\cdot}4 \times 5 = 2{\cdot}0$ per cent. of the total heat; 40 per cent. being the efficiency of expansion alone in the ratio 5:1, when nothing has to be subtracted on account of negative work done during compression.

If we assume that the effective efficiency of conversion of the heat lost during expansion would have been, on the average, half as much

TABLE 35

Heat balance-sheet for an engine of compression ratio 5:1 *if all heat loss were suppressed*

Heat to I.H.P. as given in table 34 with heat losses	31·5	
addition for 5 per cent. saved during combustion and converted at 40 per cent. efficiency	2·0	
addition for 10 per cent. saved during expansion and converted at an average efficiency 20 per cent.	2·0	
Total heat to I.H.P.		35·5 per cent.
Heat to jacket water		nil
Heat to exhaust		64·5 per cent.

as this, or 20 per cent., then the new heat balance-sheet for adiabatic working may be drawn up as in table 35, which shows, as anticipated above, an increase of only 4 per cent. in the heat to I.H.P. over that of table 34.

It has been stated that in an engine of compression ratio 5:1 and indicated thermal efficiency 31·5 per cent., it may be taken that 5 per cent. of the fuel heat is lost to the cylinder walls and piston during combustion, 10 per cent. during expansion, and 13·5 per cent. during the exhaust period; and something must be added in justification of these figures.

The total time occupied by combustion, expansion, and exhaust may be fixed with fair accuracy as 380° of crank revolution. This period is taken as extending from 10° before the dead centre, when combustion is well started, to 10° after the end of the exhaust stroke, by which time the exhaust valve, with a normal valve timing, would be closed. We may, somewhat arbitrarily but with all the accuracy required, divide up the period into 30° for the combustion at nearly constant volume, 130° for expansion, and 220° for the exhaust period. During the combustion period the gases are at a very high pressure and temperature, and in a state of violent turbulence. The rate of loss of heat by convection per unit area of surface will be high, and will be aided by radiation loss. The area of exposed metal surface, however, for an engine of 5:1 compression ratio will be only about one-third of the area at the end of the stroke, or one-half of the average area during the expansion and exhaust periods.

No reliable measurements of the heat loss by radiation from the hot gases in an engine cylinder while in operation have so far been made, but one can argue indirectly that the loss from this cause cannot be important. Such a loss would all be concentrated at the beginning of the expansion stroke (being proportional to the fourth power of the absolute temperature) and would be inversely proportional to engine speed. Referring to table 30 (p. 205), a loss of 4 per cent. of the fuel heat at 975 r.p.m. would be reduced to 2·3 per cent. at 1,700, and this would mean an increase of heat to I.H.P. of $1{\cdot}7 \times 0{\cdot}35 = 0{\cdot}6$ per cent., since the whole of the saving would be available for conversion at about 35 per cent. efficiency by expansion in the ratio 3·8 : 1, with no deduction for negative work of compression. According to table 30, however, the increase of heat to I.H.P. for this increase of speed was only 0·2 per cent., so that if we assume the whole of the increase to have been due to reduced radiation loss, this leads to an approximate figure of 1 per cent. of the total fuel heat as the radiation loss at 1,000 r.p.m., dropping to 0·5 per cent. at 2,000 r.p.m.

According to Burstall's results for an engine of compression ratio 5 : 1, given in fig. 59, an increase of speed from 1,200 to 1,800 r.p.m. gave an increase of thermal efficiency from 0·32 to 0·328. If we again assume this to have been due entirely to a reduction of radiation loss we arrive at $0{\cdot}008 \div 0{\cdot}4 = 0{\cdot}02$ as the increase in the percentage of total heat supply per cycle which was available for conversion at the higher speed. These experiments were made at the correct coal gas-air mixture, of ratio 1 : 4·1, for which the maximum temperature and radiation loss would be less than in a petrol engine, but the figures confirm that the loss by radiation must be comparatively insignificant.

During the exhaust period the gases are at a comparatively low temperature, say 800° C. to 1,000° C., but they are again in a state of vigorous turbulence. More especially is this so in the neighbourhood of the exhaust valve, where there will also be a vigorous scrubbing action and rapid loss of heat to some metal surfaces beyond the valve, which are always included within the water-jacketed portion of the cylinder.

Some interesting experiments have been made by Ricardo upon an engine in which separate cooling water was supplied to the cylinder head and cylinder barrel. Since it was a sleeve-valve cylinder, the exhaust valves and ports were located round the cylinder barrel, and heat flow to the cylinder head, therefore, was not complicated by heat given up by the hot exhaust gas during efflux from the cylinder.

On the other hand, the cylinder head was in the form of a deep cone, and very well cooled, the water outlet temperature being kept at 50° C. throughout. The proportion of heat carried away from the head was therefore probably rather larger than would normally be the case, apart from that beyond the exhaust valve.

In these experiments the distribution of heat was as shown in table 36, the heat to exhaust, radiation, etc. being found by difference.

TABLE 36

Distribution of heat supply in a sleeve-valve engine of compression ratio 4·3:1 *at* 1,500 *r.p.m. Separate cooling water supply to cylinder head and barrel*

Heat to I.H.P.	29·4 per cent.
,, cylinder head	4·8 ,,
,, cylinder barrel	16·7 ,,
,, exhaust (by difference) . .	49·1 ,,
Total	100·0

The compression ratio was 4·3:1 and the thermal efficiency was therefore rather lower than that given in table 34 as being typical for an engine of ratio 5:1. In this special engine the length of water-cooled passage outside the exhaust ports was reduced to an absolute minimum, and the diminished amount of heat given up to the jackets by the issuing exhaust gas is reflected in the reduced proportion of heat to the cooling water, 21·5 per cent. only, in spite of the low water temperature.

There must still have been some heat given up by the hot gases to the cooling water beyond the exhaust valve, but less than in the normal design of poppet-valve engine with valves in the cylinder head. In this latter type the amount of heat given up beyond the valve must vary within wide limits according to the design of engine, and one can do no more than suggest an average figure: 4 per cent. of the total heat supply appears to be a reasonable figure.

A suggested subdivision of the heat to the cooling water, 28·5 per cent., given in table 34 may now be made, and is set out in table 37;

TABLE 37

Approximate distribution of heat supply in a poppet-valve engine of compression ratio 5:1

Heat to I.H.P.		31·5 per cent.
Heat to cooling water		
(*a*) from cylinder head during combustion	4·0	
(*b*) ,, ,, ,, ,, expansion	4·0	
(*c*) ,, ,, ,, ,, exhaust	3·5	
(*d*) ,, ,, beyond exhaust valve	4·0	
(*e*) ,, ,, barrel during expansion	7·0	
(*f*) ,, ,, ,, ,, exhaust	6·0	
	28·5	28·5 per cent.
Heat to exhaust, radiation, etc.		40·0 per cent.

the figures being based upon what is said above and upon the following rough generalizations:

(1) Of the whole heat lost during combustion, one-quarter goes to the piston, and thence via the cylinder barrel to the cooling water. The piston supplies about one-third of the surface of the combustion space, in an engine of compression ratio 5 : 1, but to compensate for this, its surface temperature will be higher than that of the rest of the cylinder head unless it is provided with an independent cooling system.

(2) The rate of loss of heat to the cylinder head per unit area during combustion is about 4 times the average rate during the rest of the expansion; the latter period being regarded as lasting from 20° to 150° after the dead centre, at which point the exhaust valve opens. The expansion period is therefore about 4 times as long as the combustion period.

(3) The average exposed surface of cylinder barrel, during expansion and exhaust, is about equal to that of the cylinder head.

(4) The average temperature difference between the gases and the cooled metal surfaces, during the expansion, is twice that during the exhaust period.

As regards the subdivision of heat to the cooling water in this table, it should be noted that items (*a*) and (*b*) refer to the cylinder head only. The total loss to cylinder head and piston during combustion would be 5⅓ per cent. in accordance with assumption (1) above. Furthermore, the heat passing to the piston reappears again as heat from the cylinder barrel in item (*e*). Items (*a*) and (*b*) are made equal in accordance with assumption (2).

During the exhaust period the rate of heat loss has been taken as about half that during expansion, but the duration of the period is 1·7 times that of expansion. This fixes items (*c*) and (*f*) as 0·85 times items (*b*) and (*e*) respectively.

Item (*e*) is made up of a direct heat loss to the barrel from the working fluid of 4⅓ per cent., about equal to item (*b*) since the average exposed surface is the same, together with the heat loss to the piston, 2⅔ per cent., corresponding to the losses to the cylinder head under items (*a*) and (*b*).

The total heat lost by the working fluid works out, from the figures suggested, as 5⅓ per cent. during combustion, 9⅔ per cent. during expansion, and 13½ per cent. during the exhaust period, of which 4 per cent.

is given up to the metal surfaces beyond the exhaust valve. It used to be thought that as much as 10 per cent. or more of the total heat supply was lost during combustion, but there can be little doubt that this was a gross overestimate. If anything, the proportion of heat allocated to the exhaust period in table 37 is an underestimate, and in order to preserve a logical proportion with the other periods when their relative duration and exposed surfaces are considered, about 5 per cent. during the combustion period is all that can be allowed. It will be noticed that in order to keep it as large as this in table 37 one has to assume a rate of heat loss per sq. foot 4 times as great as the average during the next 130° of crank revolution. This comparatively small heat loss during combustion fits in well with the margin found experimentally between the best observed thermal efficiencies and those calculated for the ideal engine with full allowance made for the properties of the real working fluid.

The subdivision of the heat supply into useful and harmful portions has so far been made only in relation to full throttle, and nearly full power, conditions. These are the important ones, for it is under these conditions that inability to deal with the harmful, or waste heat, portion, is likely to lead to breakdown. Many petrol engines, however, spend the greater part of their working life under throttled conditions, and it is useful therefore to examine how the heat distribution is affected at reduced loads. Fig. 53 on p. 185 showed how the thermal efficiency of an engine could be maintained practically constant while the engine was throttled, at constant speed, down to 40 per cent. of its full load torque, provided that the ignition timing was suitably advanced to allow for dilution with exhaust gas, and for the consequent slower burning. In fig. 87 the same curve of thermal efficiency is reproduced, this time as the percentage of the total heat which is converted to I.H.P., and there have been added also the two curves showing the percentages of the reduced heat supply under throttled conditions which may be expected to pass ultimately to the cooling water and to the exhaust.

The dotted curves indicate what might have been expected to happen if the ignition advance had not been adjusted as the throttle was closed. The heat to I.H.P., instead of remaining practically constant, falls off to 26 per cent. at 40 per cent. of full load. Heat to the cooling water increases from 36 to 38 per cent. owing to higher temperatures during the exhaust and later stages of expansion. And the waste heat to the exhaust increases from 32 to 38 per cent.

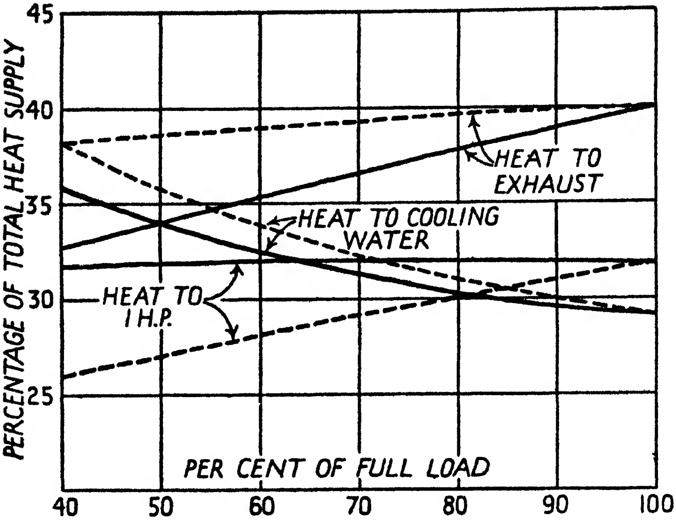

FIG. 87. Distribution of heat in a petrol engine under different conditions of load, when varied by throttling Full lines, with optimum ignition timing. Dotted lines, with ignition fixed as for full throttle.

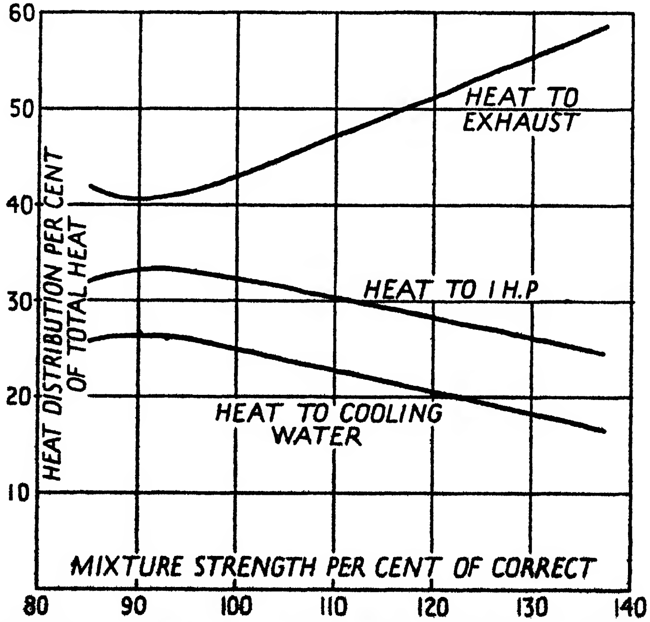

FIG. 88. Distribution of heat as affected by change of mixture strength and expressed as a percentage of the total heat supply. Speed, 1,500 r.p.m. Compression ratio, 5·45 : 1. Fuel, petrol.

These and the other results given in this article, except those quoted from Gibson's paper, have all been based upon experiments in which the fuel-air ratio was that giving maximum economy. That is to say, the fuel-air ratio must have been about 10 per cent. weaker than the mixture giving just complete combustion of both fuel and air. And it will be instructive, in conclusion, to examine how the heat distribution may be affected by a change of fuel-air ratio, and also by the fuel, if in its thermal properties it differs widely from an average petrol, as alcohol does.

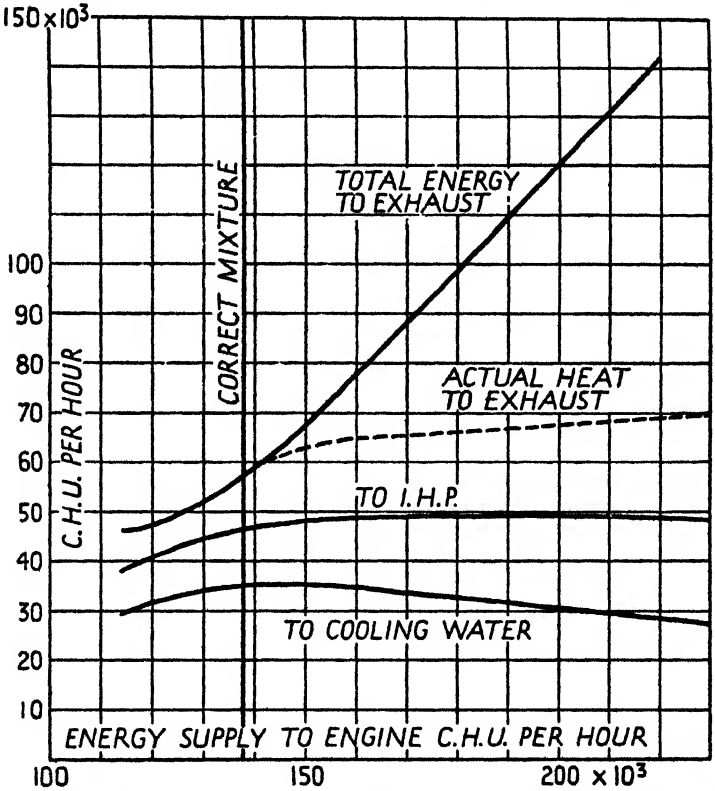

FIG. 89. Distribution of energy as affected by mixture strength and expressed as actual energy quantities, for varying supplies of potential energy per hour in the fuel.

The effect of a change of fuel-air ratio is illustrated in fig. 88. The shape of the curve of heat to I.H.P. is similar to that of any typical curve of efficiency against fuel-air ratio for a liquid fuel. It shows a maximum at a point 10 per cent. on the weak side of the correct mixture. Heat to the cooling water is in this diagram shown as a percentage of the increasing total heat supply, so that the point is not brought out that, as the fuel-air ratio increases, so too does the actual amount of heat to the cooling water, up to the maximum power mixture, about 20 per cent. rich; for the rate of heat loss follows the changes of mean temperature of the gases throughout a cycle. The actual heat to I.H.P. and to the cooling water per hour is shown in fig. 89, in which these quantities are plotted against total heat supply to the engine. It should be noted that this total heat supply is calculated from the total fuel present in the rich mixtures, and does not therefore represent a number of heat units actually developed within the engine. The uppermost curve in each of figs. 88 and 89 shows the heat, actual or potential, which is carried away to waste through the exhaust valve. And the dotted curve in fig. 89 shows what part of this waste is in the form of heat, assuming that the heat which can be developed in the

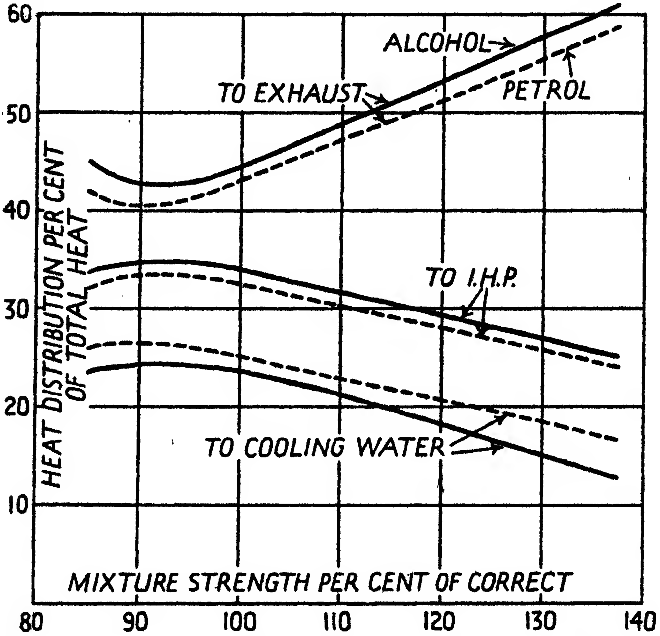

FIG. 90. Distribution of heat at different fuel-air ratios, with alcohol as fuel. The dotted curves give the distribution with petrol for comparison (same as fig. 88). Speed, 1,500 r.p.m. Compression ratio, 5·45 : 1.

cylinder reaches a maximum at the maximum power mixture strength. With richer mixtures there is a slight decline in the heat to the cooling water, owing to lower gas temperatures, and to direct cooling of the cylinder walls by evaporation of excess fuel. There is a corresponding increase in the heat to the exhaust. When employing a fuel like alcohol, the heat to the cooling water is noticeably less than with petrol at the same compression ratio, although the power output may be quite considerably greater; the effect being again due to lower gas temperatures and to direct cooling of the cylinder walls by fuel evaporation. The latent heat of evaporation of alcohol, at the correct mixture strength, would produce a drop of temperature in the induction system of about 112° C.,† as compared with 28° C. for petrol, and in addition the maximum rise of temperature on combustion is about 100° C. less, so that the maximum flame temperature may be expected to be about 165° C. lower than with petrol. This difference is reflected both in the higher efficiency and the lower heat loss to the jackets exhibited by alcohol, as shown in fig. 90, where the dotted curves for petrol, similar to those of fig. 88, have been added for comparison.

---

† Alcohol with 2·5 per cent. of water. With 5 per cent. of water there would be a further 9° C. drop.

TABLE 38

Atomic Weights

Name.	*Symbol*	O = 16
Aluminium	Al	26·97
Antimony	Sb	121·77
Argon	A	39·91
Arsenic	As	74·96
Barium	Ba	137·37
Beryllium	Be	9·02
Bismuth	Bi	209·0
Boron	B	10·82
Bromine	Br	79·916
Cadmium	Cd	112·41
Caesium	Cs	132·81
Calcium	Ca	40·07
Carbon	C	12·000
Cerium	Ce	140·25
Chlorine	Cl	35·457
Chromium	Cr	52·01
Cobalt	Co	58·94
Copper	Cu	63·57
Dysprosium	Dy	162·52
Erbium	Er	167·7
Europium	Eu	152·0
Fluorine	F	19·00
Gadolinium	Gd	157·26
Gallium	Ga	69·72
Germanium	Ge	72·60
Gold	Au	197·2
Hafnium	Ha	178·6
Helium	He	4·00
Holmium	Ho	163·4
Hydrogen	H	1·008
Indium	In	114·8
Iodine	I	126·932
Iridium	Ir	193·1
Iron	Fe	55·84
Krypton	Kr	82·9
Lanthanum	La	138·90
Lead	Pb	207·20
Lithium	Li	6·94
Lutecium	Lu	175·0
Magnesium	Mg	24·32
Manganese	Mn	54·93
Mercury	Hg	200·61
Molybdenum	Mo	96·0
Neodymium	Nd	144·27
Neon	Ne	20·2
Nickel	Ni	58·68
Niobium	Nm	93·1
Nitrogen	N	14·008
Osmium	Os	190·8
Oxygen	O	16·00

Atomic Weights—continued

Name	*Symbol*	O = 16
Palladium	Pd	106·7
Phosphorus	P	31·027
Platinum	Pt	195·23
Potassium	K	39·096
Praseodymium	Pr	140·92
Radium	Ra	225·95
Radon	Rn	222·4
Rhodium	Rh	102·91
Rubidium	Rb	85·44
Ruthenium	Ru	101·7
Samarium	Sa	150·43
Scandium	Sc	45·10
Selenium	Se	79·2
Silicon	Si	28·06
Silver	Ag	107·88
Sodium	Na	22·997
Strontium	Sr	87·63
Sulphur	S	32·064
Tantalum	Ta	181·5
Tellurium	Te	127·5
Terbium	Tb	159·2
Thallium	Tl	204·39
Thorium	Th	232·15
Thulium	Tm	169·4
Tin	Sn	118·70
Titanium	Ti	48·1
Tungsten	W	184·0
Uranium	U	238·17
Vanadium	V	50·96
Xenon	X	130·2
Ytterbium	Yb	173·6
Yttrium	Yt	88·9
Zinc	Zn	65·38
Zirconium	Zr	91·3

TABLE 39

Data concerning selected liquid fuels

Fuel	*Specific gravity at 20° C.*	*Heat of Combustion at room temperature and at a constant pressure of 1 atm. gm.-cal.*$_{15°}$ *per gm.-mol. or C.H.U. per lb.-mol.*	*Latent heat of evaporation of fuel from and at 25° C. C.H.U. per lb.*	*True† lower calorific value of fuel-vapour at constant volume and room temperature in C.H.U.*		*Ratio of air to fuel by weight in correct mixture*	*Heat of combustion at constant volume and 100° C. per S.C.F. of correct mixture of air and fuel-vapour C.H.U.*
				per lb.	*per gallon*		
n-Pentane . . . C_5H_{12}	0·626	vapour 845,270 at 25° C. (R)	86	10,867	68,020	15·30	55·82
n-Hexane . . . C_6H_{14}	0·660	vapour 1,002,400 at 25° C. (R)	86	10,804	71,300	15·22	56·05
n-Heptane . . . C_7H_{16}	0·684	vapour 1,159,400 at 25° C. (R)	90	10,759	73,590	15·16	56·21
n-Octane . . . C_8H_{18}	0·703	vapour 1,316,400 at 25° C. (R)	83	10,724	75,380	15·11	56·32
Benzene . . . C_6H_6	0·879	liquid 783,370	103	9,737	85,590	13·25	57·67
Toluene . . . $C_6H_5CH_3$	0·866	liquid 940,000 (approx.)	102	9,860	85,390	13·48	57·67
Xylene . . . $C_6H_4(CH_3)_2$	0·880	liquid 1,091,300	93	9,890	87,040	13·65	57·32
Cyclohexane . . C_6H_{12}	0·778	liquid 938,300	93	10,513	81,790	14·76	56·12
Methyl Alcohol . CH_3OH	0·791	vapour 182,580 at 25° C. (R)	279	5,053	39,970	6·46	55·35
Ethyl Alcohol . . C_2H_5OH	0·789	vapour 336,780 at 25° C. (R)	220	6,642	52,410	8·99	55·74

Values marked (R) are due to Rossini. Landolt-Börnstein-Roth-Scheel *Tabellen* are the source of much of the other basic data.

† See end of art. 25.

TABLE 40

Data concerning certain gaseous fuels

Fuel	*Formula*	*Approx. molecular weight*	*Heat of combustion at room temperature and at a constant pressure of 1 atm. gm.-cal.$_{15°}$ per gm.-mol. or C.H.U. per lb.-mol.*	*'True'† lower calorific value at constant volume and at room temperature*		*Volume ratio air/gas in correct mixture*	*Heat of combustion at constant volume and 100° C. of 1 S.C.F. of correct mixture C.H.U.*
				C.H.U. per S.C.F.	*C.H.U. per mol.*		
Coal gas (average)				256·6	92,140	4·13	50·02
Hydrogen	H_2	2	68,313 at 25° C. (R)	160·2	57,520	2·38	47·52
Carbon monoxide	CO	28	67,623 at 25° C. (R)	187·5	67,320	2·38	55·50
Methane	CH_4	16	212,790 at 25° C. (R)	534·1	191,785	9·52	50·75
Ethane	C_2H_6	30	372,810 at 25° C. (R)	951·4	341,595	16·66	53·86
Propane	C_3H_8	44	530,570 at 25° C. (R)	1,362	489,140	23·80	54·93
***n*-Butane**	C_4H_{10}	58	687,940 at 25° C. (R)	1,772	636,305	30·94	55·47
Ethylene	C_2H_4	28	345,800	904·2	324,685	14·28	59·17
Propylene	C_3H_6	42	494,900	1,291	463,520	21·42	57·58
Butylene	C_4H_8	56	653,900	1,705	612,245	28·56	57·69
Acetylene	C_2H_2	26	312,400	839·8	301,550	11·90	65·14

Values marked (R) are due to Rossini, whereas the other heats of combustion in column 4 are taken from Landolt-Börnstein *Tabellen*.

† See end of art. 25.

TABLE 41

Air-Benzene Mixtures—C.R. 5:1

Degree of richness	*Composition of cylinder contents at T_e*							*Volume before combustion*	*Volume after combustion*	*Volume ratio σ*	*Heat of combustion plus compression per mol of mixture*	*Mean volumetric heat per mol of mixture*	*Maximum temperature T_e °C.*	*Compression temperature °C.*	*Rise of temperature on explosion*	*Product of σ and rise of temperature on explosion*
α	CO_2	CO	H_2O	H_2	O_2	NO	N_2									
NEGLECTING DISSOCIATION AND NITRIC OXIDE FORMATION																
—0·5	6·00	..	3·00	..	7·50	..	56·4	72·4	72·9	1·007	..	..	1,917	390	1,527	..
—0·2	9·60	..	4·80	..	3·00	..	56·4	73·0	73·8	1·011	..	..	2,618	379	2,239	..
0	12·00	..	6·00	..	..	..	56·4	73·4	74·4	1·014	..	.	3,028	372	2,656	..
0·1	10·431	2·769	6·369	0·231	..	..	56·4	73·6	76·2	1·035	..	..	2,936	369	2,567	..
0·2	8·964	5·436	6·636	0·564	..	..	56·4	73·8	78·0	1·057	..	.	2,842	366	2,476	..
0·5	5·365	12·635	6·635	2·365	..	..	56·4	74·4	83·4	1·121	..	.	2,573	357	2,216	..
TAKING ACCOUNT OF DISSOCIATION																
—0·5	5·980	0·020	2·998	0·002	7·511	..	56·40	72·4	72·911	1·007	12,056	6·644	1,914	390	1,524	1,535
—0·35	7·592	0·208	3·881	0·019	5·364	..	56·40	72·7	73·464	1·011	14,942	6·935	2,254	384	1,870	1,891
—0·2	8·665	0·935	4·721	0·079	3·507	..	56·40	73·0	74·307	1·018	17,275	7·163	2,511	379	2,132	2,170
0	9·009	2·991	5·723	0·277	1·634	..	56·40	73·4	76·034	1·036	19,264	7·394	2,705	372	2,333	2,417
0·1	8·754	4·446	6·150	0·450	0·948	..	56·40	73·6	77·148	1·048	19,802	7·474	2,749	369	2,380	2,495
0·2	8·209	6·191	6·496	0·704	0·447	..	56·40	73·8	78·447	1·063	20,010	7·536	2,755	366	2,389	2,539
0·3	7·367	8·233	6·712	1·088	0·160	..	56·40	74·0	79·900	1·080	19,843	7·574	2,720	363	2,357	2,546
0·4	6·352	10·448	6·757	1·643	0·046	..	56·40	74·2	81·646	1·100	19,390	7·596	2,652	360	2,292	2,521
0·5	5·350	12·650	6·626	2·374	0·012	..	56·40	74·4	83·412	1·121	18,831	7·612	2,571	357	2,214	2,482
TAKING ACCOUNT OF DISSOCIATION AND THE FORMATION OF NITRIC OXIDE																
—0·5	5·983	0·017	2·998	0·002	7·203	0·612	56·094	72·4	72·909	1·007	11,878	6·632	1,891	390	1,501	1,512
—0·35	7·614	0·186	3·884	0·016	4·861	0·980	55·910	72·7	73·451	1·010	14,676	6·921	2,220	384	1,836	1,854
—0·2	8·711	0·889	4·724	0·076	2·914	1·137	55·832	73·0	74·283	1·018	16,984	7·151	2,475	379	2,096	2,134
0	8·962	3·038	5·715	0·285	1·190	0·942	55·929	73·4	76·061	1·036	18,938	7·379	2,666	372	2,294	2,377
0·1	8·621	4·579	6·127	0·473	0·657	0·738	56·031	73·6	77·226	1·049	19,446	7·456	2,708	369	2,339	2,454
0·2	8·022	6·378	6·455	0·745	0·307	0·509	56·146	73·8	78·562	1·064	19,659	7·516	2,716	366	2,350	2,501
0·3	7·196	8·404	6·662	1·138	0·118	0·306	56·247	74·0	80·071	1·082	19,558	7·559	2,688	363	2,325	2,516
0·4	6·248	10·552	6·715	1·685	0·038	0·161	56·320	74·2	81·719	1·101	19,216	7·587	2,633	360	2,273	2,503
0·5	5·302	12·698	6·599	2·401	0·010	0·078	56·361	74·4	83·449	1·122	18,725	7·606	2,561	357	2,204	2,473

APPENDIX I

To find the efficiency of an ideal cycle in which some heat is communicated to the working substance at constant volume, and the rest at constant pressure. The cycle is that represented in fig. 5.

Let $$\frac{P_3}{P_2} = \alpha \quad \frac{V_{3'}}{V_3} = \beta \quad \text{and} \quad \frac{V_1}{V_2} = r.$$

Then we have

$$\begin{aligned} \text{heat supplied } 2\text{–}3 &= K_v(T_3 - T_2), \\ \text{,,} \qquad \text{,,} \qquad 3\text{–}3' &= K_p(T_{3'} - T_3), \\ \text{,, rejected } 4\text{–}1 &= K_v(T_4 - T_1). \end{aligned}$$

Hence $$\text{efficiency} = 1 - \frac{K_v(T_4 - T_1)}{K_v(T_3 - T_2) + K_p(T_{3'} - T_3)}.$$

Now $$\frac{P_3}{P_2} = \alpha = \frac{T_3}{T_2} \quad \text{and} \quad K_p = \gamma K_v.$$

$$\begin{aligned} T_2 &= T_1 r^{\gamma-1}, \\ T_3 &= T_2\alpha = T_1\alpha r^{\gamma-1}, \\ T_{3'} &= T_3\beta = T_1\alpha\beta r^{\gamma-1}, \\ T_4 &= T_{3'}\left(\frac{\beta}{r}\right)^{\gamma-1} = T_1\left(\frac{\beta}{r}\right)^{\gamma-1}\alpha\beta r^{\gamma-1}. \end{aligned}$$

After substituting γK_v for K_p, and replacing each temperature in the efficiency expression by its equivalent in terms of T_1, as given above, the equation for efficiency reduces to

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{\alpha\beta^\gamma - 1}{(\alpha-1) + \gamma\alpha(\beta-1)}\right].$$

If $\beta = 1$ ($V_{3'} = V_3$), the cycle becomes the ordinary constant volume cycle of fig. 2, and the efficiency formula reduces to

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1}.$$

If $\alpha = 1$, we have a cycle with all combustion at constant pressure, but with the toe at the right of the diagram in fig. 3 cut off. The efficiency formula becomes

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{\beta^\gamma - 1}{\gamma(\beta-1)}\right].$$

This might stand as the ideal cycle for the Diesel engine as it was first practically developed, but for reasons given in Chapter II it is doubtful whether there is now any need for such a cycle. If anything more closely akin to the practical high-compression injection cycle is required than the plain constant volume cycle of fig. 2, then one should go straight to the composite cycle of fig. 5; for in any high-speed, high-efficiency engine it is probable that injection will have to start before the dead centre, and that a fair proportion of constant volume combustion must always take place.

APPENDIX II

The horse-power of an engine will be equal to

$$\text{(a constant)} \times \text{(thermal efficiency)} \times \text{(fuel used per hour)} \times \text{(calorific value of fuel)}.$$

We may treat either gas or air as the fuel and equate the corresponding expressions for the horse-power; therefore,

therm. efficy. referred to air

$$= \text{therm. efficy. referred to gas} \times \frac{\text{gas used per hour}}{\text{air used per hour}} \times \frac{\text{cal. value of gas}}{\text{cal. value of air}}.$$

Since the calorific value of air is taken as the heat per cubic ft. of air in the correct mixture, assumed to be 1:4, the calorific value of gas is 4 times that of air.

At any fraction x of the correct mixture, the ratio of gas to air per hour

$$= \frac{0{\cdot}2x}{1-0{\cdot}2x}.$$

At this same mixture the thermal efficiency referred to gas

$$= 0{\cdot}473-0{\cdot}093x,$$

since, when $x = 1$ the efficiency $= 0{\cdot}38$.

Hence at mixture x, thermal efficiency referred to air

$$= (0{\cdot}473-0{\cdot}093x)\frac{0{\cdot}8x}{1-0{\cdot}2x}.$$

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