

# THE HIGH-SPEED INTERNAL-COMBUSTION ENGINE

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## PREFACE TO NEW EDITION

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The present volume is a revised and somewhat extended edition of Vol. II of *The Internal-Combustion Engine* published in 1923.

During the seven years which have elapsed since the above volume was published progress has been considerable; it has, however, for the most part taken the form of improving and consolidating existing orthodox design, and the only important new development which has arisen during this period is the achievement of the high-speed Diesel.

Since progress has, to a large extent, followed along the lines foreshadowed in the original volume, and since also this volume dealt mainly with general principles rather than specific designs, there has been little need for actual revision. The present volume therefore contains additions rather than alterations, and a new chapter has been added dealing with the high-speed Diesel engine.

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# THE HIGH-SPEED INTERNAL-COMBUSTION ENGINE

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## INTRODUCTION

Until the advent of the Great War, most of the scientific talent in this country which had interested itself in the development of the internal-combustion engine was devoted to research upon, and development of, the relatively heavy slow-speed stationary type, a type which, it now appears, has but a limited scope, for in the really large powers it can hardly compete with the modern steam turbine, while in the smaller powers, its field is being narrowed daily by the gradual spread of electricity.

Very soon after hostilities began, it became apparent that the light mobile high-speed type of internal-combustion engine applied to transport, aircraft, and later to tanks, was destined to play a very important, if not decisive, part in the conduct of the War. Every effort was then made to concentrate all the available scientific talent on the development of the high-speed engine. Independent scientists and investigators, and such national institutions as the Royal Aircraft Establishment and the National Physical Laboratory were requested to turn their attention to this subject, and every facility was lavished on them. They were invited to co-operate with the manufacturers and were asked to make a careful theoretical study of both the mechanical and thermodynamic problems involved, and to recommend how and in what direction the general efficiency of these light engines might be maintained and improved. The campaign of intensive research which resulted from this sudden influx of scientific talent, accompanied by almost inexhaustible funds for research, has resulted in the production of light high-speed engines which, besides giving what a few years ago would have been considered an almost incredible power



output in relation to their size and weight, can show as high an efficiency as that of the largest slow-speed type. What is perhaps more important still, the basic principles, both mechanical and thermo-dynamic, upon which the performance of such an engine depends, have been investigated in so complete and comprehensive a manner that the performance of any engine can now be gauged with accuracy from a study of the design alone; or conversely, an engine can be designed to fulfil any specific requirement as to power output or efficiency, with the same precision as in the case of a steam-engine.

That the internal-combustion engine has found its ultimate sphere in the light mobile high-speed type is now evidenced by the fact that, whereas in the years immediately before the War the annual output in horse-power of both the light and heavy type in this country was about equal, to-day the aggregate annual power output of the light high-speed type is at least ten times that of all other types, and in numbers probably nearer twenty times.

To-day far more is known about both the possibilities and the limitations of the high-speed internal-combustion engine than was the case before the War, and it seems fairly evident that its rôle lies in the propulsion of all forms of transport where its light weight and low fuel consumption render it supremely valuable. It is in the author's opinion extremely doubtful whether it will ever attain to more than its present very uncertain footing for stationary purposes, where neither its own light weight nor that of the fuel it consumes can be of much assistance to it in the struggle for existence.

Already, and in an incredibly short space of time, the internal-combustion engine has gained practically undisputed sway over all forms of road transport, and in doing so has developed and even almost revolutionized this previously decaying system. In a few short years it has both opened up the possibility of aerial transport and made it a powerful factor, in war at all events; it will almost certainly extend to rail transport also, beginning, no doubt, in countries where long distances have to be traversed and where fuel and water are scarce. It is already ousting the steam-engine from the smaller classes of shipping and is extending very gradually to the larger vessels, but here its progress will probably be slow, for the steam-turbine shows to particular advantage as a marine engine, because in this field alone it can always get that upon which its efficiency so largely depends, namely, an unlimited supply of cold water. Also the large steamship, alone of all forms of transport, requires a very

high-powered installation, and it is in units of high power output that steam retains its supremacy.

With but few exceptions, all light mobile engines consume volatile liquid fuels.

Up to the present the only two fuels available in bulk are petrol—a generic term covering any low-boiling distillate from crude petroleum—and benzol, a distillate from coal-tar, consisting of benzene with a small percentage of toluene.

The supply of both these fuels, and more particularly the latter, will soon become unequal to the demand, with the result that a critical situation is bound to arise in the not very distant future. Civilization is now so deeply committed to the use of internal-combustion engines for all road transport and for many other purposes, that it is a matter of absolute necessity to find an alternative fuel. Fortunately such a fuel is in sight in the form of alcohol; this is a vegetable product whose consumption involves no drain on the world's storage and which, in tropical countries at all events, can ultimately be produced in quantities, sufficient to meet the world's demand, at all events at the present rate of consumption.

By the use of a fuel derived from vegetation, mankind is adapting the sun's heat to the development of motive power, as it becomes available from day to day; by using mineral fuels, he is consuming a legacy—and a limited legacy at that—of heat stored away many thousands of years ago. In the one case he is, as it were, living within his income, in the other he is squandering his capital.

The mobile internal-combustion engine is now no longer a luxury; it has become one of the prime necessities of peaceful civilization and the prime necessity in time of war; therefore, the assurance of its fuel supply should be considered a matter of national importance. It is perfectly well known that alcohol is an excellent fuel, and there is little doubt but that sufficient supplies could be produced within the tropical regions of the British Empire, yet little or nothing is being done to encourage its development. Judging from past experience, no active steps will be taken until a serious crisis has arisen, and since it must take at least ten years to create the necessary organization and machinery for the production of alcohol on the scale which will be required, the crisis may be a serious and prolonged one.

In the author's opinion it is unlikely that any crisis in the fuel situation will have an adverse effect on the development of the

internal-combustion engine, for the simple reason that it has become a necessity, but it will probably have the effect of increasing the cost of transport and with it the cost of living generally. It will also, of course, have the effect of forcing designers of engines to concentrate more attention on the attainment of high efficiency and fuel economy, which is all to the good.

# CHAPTER I

## VOLATILE LIQUID FUEL FOR INTERNAL-COMBUSTION ENGINES

The volatile liquid fuels available in bulk at the present day, or likely to be available in the near future, consist of petrol, benzol, kerosene, and alcohol.

Petrol, as is well known, is a distillate from crude petroleum; it consists of a heterogeneous mixture of all those hydrocarbon fractions which boil between the limits of 140° F. and about 400° F.

These fractions belong to three different series :

	General Formula.				
The Paraffins ...	...	...	...	...	$C_nH_{2n+2}$
The Naphthenes ...	...	...	...	...	$C_nH_{2n}$
and the Aromatics ...	...	...	...	...	$C_nH_{2n-6}$

In addition to these three leading groups there are also present a small proportion of members of the olefine series, though the proportion found in "natural" as opposed to "cracked" petrols is usually so small as to be almost insignificant.

The individual members of the paraffin series present in petrol are :

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Hexane ...	$C_6H_{14}$	156	0.663
Heptane ...	$C_7H_{16}$	209	0.691
Octane ...	$C_8H_{18}$	258	0.709
Nonane ...	$C_9H_{20}$	302	0.723
Decane ...	$C_{10}H_{22}$	343	0.735
Undecane ...	$C_{11}H_{24}$	383	0.746

Those of the naphthene series are :

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Cyclohexane ... ..	$C_6H_{12}$	178	0.780
Hexahydrotoluene ... ..	$C_7H_{14}$	212	0.770
Hexahydroxylene ... ..	$C_8H_{16}$	246.2	0.756

and those of the aromatic series :

Fuel.	Formula.	Boiling Point 0° F.	Specific Gravity at 60° F.
Benzene ... ..	$C_6H_6$	176	0.884
Toluene ... ..	$C_7H_8$	230	0.870
Xylene ... ..	$C_8H_{10}$	284	0.862

Although, in most examples of commercial petrol, the members of the paraffin series predominate, yet this is by no means always the case, and it is the exception rather than the rule, for the paraffin content of a petrol to exceed 60 per cent of the whole. Generally speaking, paraffins are commonest in the Western oilfields, naphthenes in the near Eastern, and aromatics in the far Eastern oilfields. This, however, is only a rough generalization, for there are many exceptions.

The following table gives the analysis of seven typical samples of petrol drawn from widely different parts of the world and illustrates how greatly the composition may vary. In this table the presence of small traces of other complex substances such as thiophene, &c., which play no perceptible part in the behaviour of the fuel, has been ignored.

Sample.	Approximate Composition by Weight per cent.			Specific Gravity.
	Paraffins.	Naphthenes.	Aromatics.	
A	26.0	35.0	39.0	0.782
B	62.0	23.0	15.0	0.723
C	61.0	30.5	8.5	0.727
D	38.0	47.0	15.0	0.760
E	68.0	20.0	12.0	0.719
F	80.0	15.2	4.8	0.704
H	10.0	85.0	5.0	0.767
Average	49.3	36.5	14.2	0.740

It is evident from the above that the specific gravity as a measure either of the composition or of the volatility of a fuel is quite meaningless. If, as is sometimes erroneously supposed, petrol consisted entirely of members of the paraffin series, then, since the gravity, the molecular weight, and the boiling point of the members of this series rise together, the gravity would be a measure of the volatility. The presence, however, of even a very small proportion of aromatics whose specific gravity ranges from 0.860 to about 0.885 will, of course, entirely upset any deductions which could be drawn from considerations of specific gravity alone. As a specific illustration it may be mentioned that of the fuels tabulated above, sample B sp. gr. 0.723, a special very low-boiling aircraft spirit prepared for the Cross-Atlantic flight, is by far the most volatile, yet its gravity is not the lowest by any means. The best fuels, i.e. those from which the highest power and efficiency could be obtained, were samples A, D, and H, which have the highest specific gravity of all, but which are rich in aromatics or naphthenes or both, while the worst without question is sample E sp. gr. 0.719.

It will be shown later that, of the three leading groups, the presence of the aromatics is the most of all to be desired from every point of view, that of the naphthenes next, while the paraffins are highly objectionable and the smaller the proportion present the better.

Though the phenomenon of detonation will be discussed later it may be stated at this stage that it is by far the most important factor in determining the quality of a fuel, and it is one which depends primarily upon its chemical composition. The paraffin series are the worst of all from this point of view, and they become progressively worse as their molecular weight and gravity increase; thus, for example, hexane is much better than heptane, and so on. The naphthenes are very much better, while the aromatics are the best of all as regards detonation.

Commercial benzol is a coal-tar distillate consisting primarily of pure benzene  $C_6H_6$ , with a little toluene and a trace of xylene. These are all aromatics. Its specific gravity ranges from 0.875 to 0.882, depending on the proportion of toluene present.

This fuel has many advantages over petrol, but fully to realize these, it is necessary to work with a much higher compression ratio.

The available members of the alcohol group consist of methyl, ethyl, and butyl alcohol. These are not true hydrocarbons, since

each contains oxygen in the molecule. In consequence of this and of their higher latent heat of evaporation, the behaviour of these fuels is somewhat different to that of the true hydrocarbons. So far as their tendency to detonate is concerned they are even better than the aromatics, though when used under a very high compression methyl alcohol in particular is liable to pre-ignite without warning. Owing to their high latent heat and low flame temperature the whole temperature of the cycle is lower, while owing again to their high latent heat of evaporation and therefore to the reduced suction temperature, the volumetric efficiency of an engine using alcohol is considerably higher than when using petrol or benzol. The result of the lower flame temperature is that the engine operates at a higher thermal efficiency, while the increase in volumetric efficiency much more than balances their lower internal energy, so that the maximum power output on alcohol is considerably greater, while the heat flow to the engine cylinder is lower than on petrol or benzol.

The properties of a fuel which determine its value for use in an internal-combustion engine are :

- (i.) Tendency to detonate.
- (ii.) Latent heat.
- (iii.) Volatility.
- (iv.) Calorific value of the fuel.
- (v.) Heat value of the mixture.

All volatile liquid fuels when vaporized and mixed with air in the proportion required to give complete combustion have, within very close limits, the same heat value per standard cubic inch of mixture, hence they all give the same power and the same thermal efficiency when used under the same conditions. It is only by varying the compression ratio or by altering the degree of vaporization in the carburettor or induction pipe that any variation in power output or efficiency can be obtained.

**Tendency of Fuels to detonate.**—The phenomena of detonation as apart from its relation to the nature of the fuel will be dealt with later, but for the present it is sufficient to state that the limit to which the compression ratio can be raised, and therefore the limit of power output and efficiency, is governed by the conditions which control detonation and pre-ignition.

With all known petroleum spirits, detonation precedes and subsequently produces pre-ignition, but in the case of certain fuels such

as ether, carbon disulphide (at a low compression ratio), also the pure aromatics and alcohol (at a high compression ratio), pre-ignition is liable to occur without preliminary detonation.

Of the constituents of petrol the paraffins are by far the worst offenders as regards detonation, while the aromatics are the best. It is found also that all mixtures of these bodies obey the ordinary proportional laws and that there is practically a straight line relation between the mixture proportion of two or more hydrocarbons and the compression ratio at which detonation occurs.

In fig. 1 is shown as a full line the observed relation between

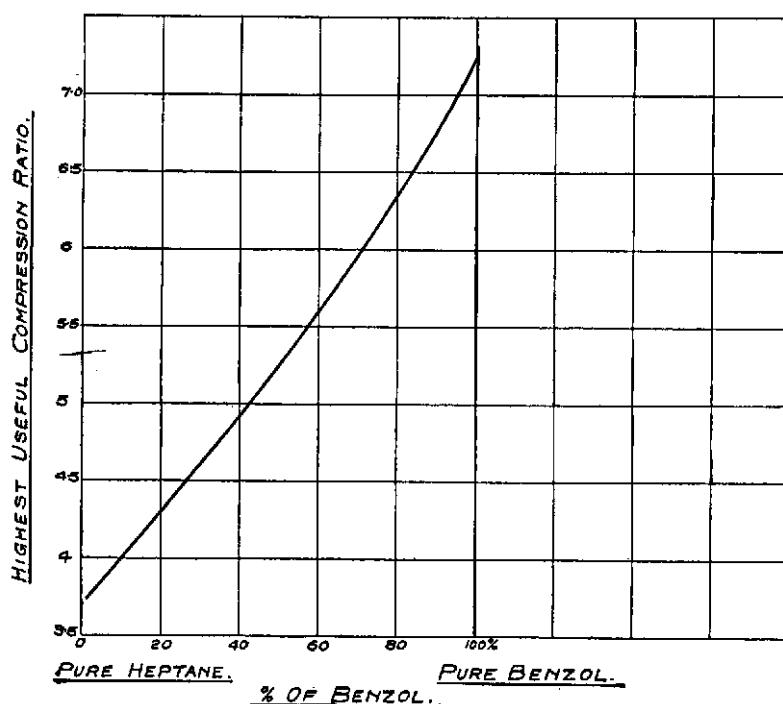


Fig. 1.—Curve showing Compression Ratio at which Detonation occurs

the compression ratio and the point at which detonation occurs when to pure heptane varying proportions of benzene (benzol) are added. Owing to the influence of combustion-chamber design, and to other factors which will be considered later, it is not possible to lay down a hard and fast relation between the fuel and the highest compression ratio at which it may be used in any type of engine, but it is possible in the light of present knowledge to give relative values, though there is some difficulty in selecting substances to be taken as standards. In the investigations which the author's firm carried



out on behalf of the Asiatic Petroleum Co., a sample of petrol, consisting mainly of paraffins from which nearly all the aromatics had been removed by sulphonation, was taken as zero, and at the other end of the scale pure toluene was chosen; the relative tendency of different fuels to detonate was then expressed in terms of their "toluene value," i.e. the equivalent proportion of toluene which it would be necessary to mix with the standard aromatic free petrol in order to give it the same tendency to detonate as that of the sample under examination. Later investigation showed that the standard "aromatic free" petrol which contained about 35 per cent of naphthenes and the lighter members of the paraffin series, was by no means the worst offender as regards detonation, and that in fact several samples of commercial petrol were actually considerably worse. Also it was found that toluene was not so effective in resisting detonation as ethyl alcohol. Since, however, the expression "toluene value" has become rather widely used, it is probably better to retain the term.

Table I gives the toluene values and the highest useful compression for various fuels. The highest useful compression ratio may be defined as the highest ratio at which a particularly efficient engine used for the purpose of investigating the behaviour of fuels could be operated without detonation at any mixture strength or with any ignition timing, with a standard amount of preheating to the carburettor, and at a speed of 1500 R.P.M.

It is therefore purely a relative term; that is to say, its absolute value applies only to one particular type of engine operated under one given set of conditions, but its relative value is applicable to any type of engine and under any conditions, as will be shown later.

In this connection reference may be made to the common belief that the rate of burning of the fuel, though one of the factors controlling detonation, forms a limit to the speed at which an engine can run.

The normal rate of burning (as distinct from the detonation rate) of any stagnant fuel/air mixture is so low as to be practically useless so far as any internal-combustion engine is concerned. We must look, therefore, entirely to turbulence or the mechanical distribution of the flame to spread combustion throughout the whole mass of the working fluid, and, since this is the case, it follows that the normal rate of burning of any fuel is practically without influence on the speed at which an engine will run.

It has been found that a fuel with a low normal rate of burning,

TABLE I

Fuel.	Highest Useful Compression Ratio in variable Com- pression Engine. Total Volume Clearance Volume	Toluene Value. Toluene=100 %, Aromatic Free Petrol=0 %.
Aromatic Free Petrol	4.85	0
" A " Petrol	6.0	38.0
" B " "	5.7	28.0
" C " "	5.25	13.5
" D " "	5.35	16.5
" E " "	4.7	-5.0
" F " "	5.05	6.5
" G " "	4.55	-10.0
" H " "	5.9	35.0
" I " "	4.3	-20.0
<i>Heavy Fuels</i>		
Heavy Aromatics	6.5	55.0
Kerosene	4.2	-22.0
<i>Paraffin Series</i>		
Pentane (Normal)	5.85	33.0
Hexane (80 % pure)	5.1	8.0
Heptane (97 % pure)	3.75	-37.0
<i>Aromatic Series</i>		
Benzene (pure)	6.9 †	67.0
Toluene (99 % pure)	>7.0	100.0
Xylene (91 % pure)	>7.0	85.0
<i>Naphthene Series</i>		
Cyclohexane (93 % pure)	5.9 †	35.0
Hexahydrotoluene (80 %)	5.8	31.5
Hexahydroxylene (60 %)	4.9	1.5
<i>Olefines</i>		
Cracked Spirit (53 % unsat.)	5.55	23.5
<i>Alcohol Group, &amp;c.</i>		
Ethyl Alcohol (98 %)	>7.5	>88.0
" " (95 vol. %)	>7.5	>88.0
Methyl Alcohol (Wood Naphtha)	5.2 †	..
Methylated Spirits	6.5 †	..
Butyl Alcohol (Coml.)	7.3	80.0
Ether (50 % in petrol)	3.9	(-32.0)
Carbon Disulph. (50 %)	5.15 †	(9.0)

NOTE.—This sign (†) indicates that pre-ignition occurred before audible detonation.

such, for example, as ethyl alcohol, will operate just as efficiently in a high-speed, low-compression, engine as will hexane or petrol, and that the relative efficiency and power obtained is exactly the same throughout the whole speed range. The normal rate of burning of a fuel has, therefore, no connection with the speed at which an engine may be run.

In fact, since a slow-burning fuel is generally less prone to detonation, it is much more desirable than one which burns more rapidly.

It is only when using excessively weak and therefore slow-burning mixtures, or when the normal turbulence within the cylinder is unduly low, that the influence of a naturally slow-burning fuel can be felt at all, and, even so, this can always be counteracted by a small advance of the ignition timing. Under all normal working conditions the difference in respect to engine speed between a normally slow or rapid burning volatile liquid fuel is quite imperceptible.

**Latent Heat.**—The influence of the latent heat of evaporation of the fuel is a very important factor, but one which is usually ignored. This factor, coupled with the mean volatility, determines the density of the charge taken into the cylinder. It is, of course, clear that the weight of charge taken into the cylinder will, in any given case, be inversely proportional to its absolute temperature at the moment when the inlet valve closes. There is definite evidence from experimental results that, with the exception of alcohol and the other members of its group, all fuels boiling below about 400° F. are completely evaporated before the commencement of the compression stroke by contact with the hot walls and by admixture with the highly heated residual exhaust products in the cylinder; excepting only a very small proportion which may enter the cylinder in coarse drops, and so not only escape evaporation, but, even to a large extent, combustion also. This proportion is, however, quite insignificant, and has no influence, in so far as power output is concerned.

The absolute temperature at the commencement of the compression stroke is dependent upon (a) the amount of external heating applied, and (b) the latent heat of evaporation. It is largely independent of the temperature of the mixture during its entry to the cylinder. In fact, the final absolute temperature, and, therefore, the weight of the charge taken into the cylinder, are dependent upon the quantity and latent heat of the fuel, and upon the amount of heat added to it, external to the cylinder.

For example, a highly volatile fuel entering the cylinder at 40° F., and a fuel of low-vapour tension entering at 80° F., will both have the same final absolute temperature at the commencement of compression, if the latent heat of both is the same, and if both receive the same amount of pre-heating. In the former case most of the evaporation has taken place outside the cylinder, and the added heat has been absorbed by the latent heat of evaporation; in the latter case, little or no evaporation has taken place outside the cylinder, and the added heat has therefore raised the temperature of the air and of the, still liquid, fuel. In both cases contact and admixture with the highly heated exhaust products in the cylinder will complete evaporation, and in both cases the final temperature will be the same, hence the weight of working fluid (which is inversely proportional to the absolute temperature at the end of the suction stroke), and therefore the power output, will be the same in both cases.

From the above considerations it will be seen that, with any given amount of pre-heating (provided it is not excessive), the volumetric efficiency, and therefore the power output, will increase with the latent heat of the fuel.

TABLE II

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Total Energy liberated by Combustion. Ft. lb. per standard cub. in.	Relative Power, Output allowing for Increase in Density due to Evaporation. Octane=100.
<i>Paraffin Series</i>			
Hexane ... ..	156	48.33	100.2
Heptane ... ..	133	48.64	100.1
Octane ... ..	128	48.73	100.0
Nonane ... ..	..	48.78	..
Decane ... ..	108	48.82	99.4
<i>Aromatic Series</i>			
Benzene ... ..	172	47.51	100.1
Toluene ... ..	151	47.98	100.0
Xylene ... ..	145	48.26	100.6
<i>Naphthene Series</i>			
Cyclohexane ... ..	156	48.11	100.0
Hexahydrotoluene ... ..	138	48.32	99.8
Hexahydroxylene ... ..	133	48.49	99.8

Some calculations illustrating this point more fully will be found in Chapter II.

The above conclusions may be summed up as follows:—

(1) The power output is inversely proportional to the absolute temperature of the working fluid at the end of the suction stroke—since it is the temperature at this point which controls the weight of charge, and therefore the volumetric efficiency.

(2) Other things being equal, the final suction temperature is controlled by the amount of external heating, on the one hand, and the latent heat of the fuel, on the other; it is nearly independent of the temperature of the entering charge and of its volatility.

Apart from the alcohol group, the variation in latent heat is not very large, and does not exercise any important influence. It is interesting to note, however, that in cases where the total internal energy is lower, the latent heat is generally slightly higher; consequently a slightly greater weight of charge is taken into the cylinder, sufficient in most cases to compensate for the lower internal energy, and thus bring the actual power output to substantially the same in all cases.

This point is well illustrated by the instance of benzene, as will be seen later, or by reference to Table II. The energy liberated by the combustion of a cubic inch (at standard temperature and pressure) of benzene-air mixture is appreciably lower than that of the hydrocarbons forming the greater proportion of petrols. On the other hand, the latent heat of benzene is considerably greater, and as a result the power output obtainable under similar conditions from benzene is the same as that from petrol to within less than one half of 1 per cent.

The following Table III gives the latent heat of evaporation of a number of hydrocarbons and other substances enumerated in the previous tables. The air-to-fuel ratio by weight, also the drop in temperature of the mixture due to evaporation of the liquid, are shown for each fuel. The calculations are made for mixtures giving complete combustion, but without excess of air.

In the case of alcohol, owing to the very much higher latent heat and to the fact that the proportion of fuel to air is also much greater, the latent heat of evaporation plays a supremely important part, and results in a really marked increase in power as compared with other fuels, although the total internal energy of unit mass of mixture is lower than that of either petrol or benzol. Moreover, there is introduced a feature which is not observed to any marked

TABLE III

Name of Fuel.	Latent Heat of Evaporation, B.Th.U.s per lb.	Air-to-fuel Ratio (by Weight) for just complete Combustion.	Fall in Temperature of Mixture due to Latent Heat of Evaporation, ° F.
<i>Paraffin Series</i>			
Hexane... ..	156	15.2	37.8
Heptane... ..	133	15.1	32.4
Octane... ..	128	15.05	29.0
Nonane... ..	..	15.0	..
Decane... ..	108	15.0	20.1
<i>Aromatic Series</i>			
Benzene... ..	172	13.2	46.8
Toluene... ..	151	13.4	40.5
Xylene... ..	145	13.6	38.7
<i>Naphthene Series</i>			
Cyclohexane... ..	156	14.7	38.7
Hexahydrotoluene... ..	138	14.7	34.2
Hexahydroxylene... ..	133	14.7	32.4
<i>Olefine Series</i>			
Heptylene... ..	167 (app.)	14.7	41.4
Decylene... ..	..	14.7	..
<i>Alcoholic Group</i>			
Ethyl Alcohol... ..	397	8.95	148.8
Methyl Alcohol... ..	512	6.44	252.0
<i>Miscellaneous</i>			
Ether... ..	158	11.14	49.5
Carbon Disulph... ..	153	9.35	55.8
Acetylene... ..	Gas	13.2	..
Carbon Monox... ..	Gas	2.45	..
Hydrogen... ..	Gas	34.3	..

The last column is calculated on the assumption that the specific heat of the fuel vapour is constant for all at 0.5.

extent with other fuels—namely, that the power output increases very considerably when an over-rich mixture is used, because more fuel is then evaporated, the temperature of the charge is lowered, and the gain in weight of charge considerably more than outweighs the loss due to the greater specific heat of the products of combustion.

In fig. 2 are shown the actual measured volumetric efficiency as

found in the author's fuel research engine when using petrol and ethyl alcohol under precisely similar conditions as to temperature, &c., and at a compression ratio of 5:1. In both cases a careful series of measurements was made at mixture strengths ranging from 20 per cent weak to 25 per cent over-rich.

In the case of fuels whose volatility is very low, such as kerosene, butyl alcohol, &c., advantage cannot be taken of the latent heat of evaporation, because it then becomes necessary to add an excessive amount of heat before entry to the cylinder, in order to prevent condensation in the induction system. For this reason alone the power output obtainable from kerosene is actually some 15 per cent

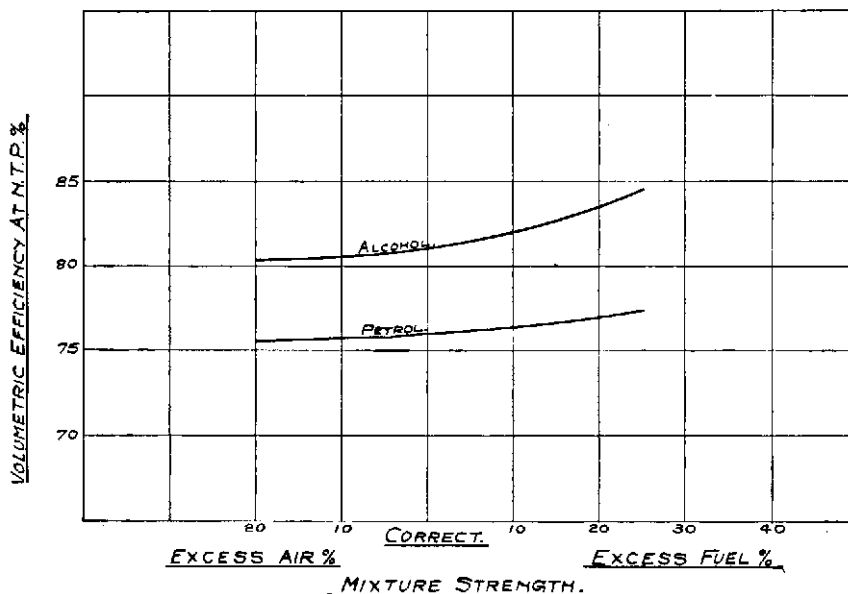


Fig. 2.—Observed Volumetric Efficiency on Petrol and Alcohol at different Mixture Strengths

lower than from petrol or other volatile hydrocarbons at the same compression ratio.

**Volatility.**—The mean volatility of a fuel is of importance since this determines the amount of pre-heating required to give reasonably uniform distribution. The amount of pre-heating governs, in its turn, the use which may be made of the latent heat of the liquid fuel.

In single-cylinder engines volatility is, between wide limits, of comparatively little consequence since the exposed surface of the induction pipe is relatively small, but as the number of cylinders is

TABLE IV

Fuel.	Rise or Fall of Temp. in Induction Pipe (indicating app. Mean Volatility). Heat 65 B.Th.U. Min. Deg. F.	Fall in Temperature (calculated) of Air Mixture Fuel due to Latent Heat of Evaporation. Deg. F.
Aromatic Free Petrol ... ..	+ 18	32.4
"A" Petrol ... ..	+ 19.8	36.0
"B" " " " " " " " " " "	+ 5.4	34.2
"C" " " " " " " " " " "	+ 12.6	33.3
"D" " " " " " " " " " "	+ 19.8	33.0
"E" " " " " " " " " " "	+ 18.9	32.8
"F" " " " " " " " " " "	+ 9.5	32.8
"G" " " " " " " " " " "	+ 32.4	...
"H" " " " " " " " " " "	+ 21.6	36.0
"I" " " " " " " " " " "	+ 25.2	...
<i>Heavy Fuels</i>		
Heavy Aromatics ... ..	+ 50.5	35.7
Kerosene ... ..	+ 56	26.1
<i>Paraffin Series</i>		
Pentane (Normal) ... ..	...	37.8
Hexane (80 % pure) ... ..	0	37.8
Heptane (97 % pure) ... ..	+ 10	32.4
<i>Aromatic Series</i>		
Benzene (pure) ... ..	- 13.1	46.8
Toluene (99 % pure) ... ..	+ 14.4	40.5
Xylene (91 % pure) ... ..	+ 32.4	38.7
<i>Naphthene Series</i>		
Cyclohexane (93 % pure) ... ..	- 5.4	38.7
Hexahydrotoluene (80 %) ... ..	+ 5.4	34.2
Hexahydroxylene (60 %) ... ..	+ 24.3	32.4
<i>Olefines</i>		
Cracked Spirit (53 % unsat.) ... ..	+ 19	37.5
<i>Alcohol Group, &amp;c.</i>		
Ethyl Alcohol (98 %) ... ..	+ 2.7	153.0
" " (95 vol. %) ... ..	- 3.6	176.0
Methyl Alcohol (Wood Naphtha) ... ..	- 15.3	252.0
Methylated Spirits ... ..	- 1.8	198.0
Butyl Alcohol (Coml.) ... ..	+ 18	...
Ether (50 % in petrol) ... ..	+ 1.8	39.6
Carbon Disulph. (50 %) ... ..	- 12.6	48.6



multiplied and the length and surface of the induction system increased, so does the importance of volatility increase. A rough approximation (it is no more) of the relative volatility of different fuels can be obtained by measuring the rise or fall of temperature in the induction pipe of an engine, when a known weight of fuel and air are passing into it and when the amount of heat supplied to the carburettor can be recorded accurately.

The figures in Table IV (p. 17) give some clue as to the relative volatility of the different fuels—all were tested under exactly the same conditions as regards speed, temperature, heat input, &c. The temperatures measured in the induction pipe and recorded in the table are, in each instance, those found when the mixture strength was that giving complete combustion.

The last column gives the calculated drop in temperature assuming no heat input and that the whole of the fuel be evaporated before entry to the cylinder.

**Final Boiling Point.**—It is always desirable to keep the final boiling point of any fuel as low as possible, because the higher boiling fractions are liable to condense on the cylinder walls and so to pass down into the crankcase, where they foul the lubricating oil.

As a general rule, so long as the final boiling point does not exceed say 400° F. no serious trouble need be feared on this score, for, if any fuel does condense on the walls, it will evaporate off in the crankcase, but in the case of kerosene and other high boiling fuels condensation on the cylinder walls and in the crankcase is one of the most serious difficulties with which designers of engines using these fuels have to contend.

**Starting.**—The readiness of a fuel to start from cold depends upon the proportion of low boiling-point fractions present in the fuel.

With nearly all commercial petrols the full vapour pressure at normal atmospheric temperature is only reached when at least 3 per cent of the volume of the vessel is occupied with liquid.

In an actual engine, starting is required with a minimum of about one-thirtieth of this, and even then the mixture strength would be about nine times richer than the normal running mixture. It follows, therefore, that unless a fuel has an abnormally high vapour tension it is necessary always to provide a very large excess for starting, e.g. by flooding or by the use of a special pilot jet, &c., as in the Zenith carburettor or by other means.

**Calorific Value.**—The heat liberated by the combination of the

fuel and air is usually determined by burning the fuel in some form of calorimeter. The heat value so found includes the latent heat of the water formed, because in any form of calorimeter the temperature necessarily falls below the boiling point of water. Since, however, it is quite out of the question in any internal-combustion engine to make use of the latent heat of the water, it is customary to deduct from the total heat liberated that due to the condensation of the water formed. The heat value found after this deduction has been made, is termed the lower calorific value of the fuel and is generally accepted as the basis upon which to calculate the thermal efficiency of an engine. In the case of internal-combustion engines using volatile liquid fuels, such a value for the available heat is not quite correct because, when a fuel is burnt in a bomb or other calorimeter, some of the heat of combustion is devoted to evaporating the remainder of the liquid fuel and is therefore absorbed. Now when used in an engine the whole of the liquid is evaporated before combustion takes place, and the heat required for its evaporation is supplied by the waste heat from the cycle or by the available heat already present in the air. In either case it is supplied by heat other than the useful heat of combustion of the fuel. If, therefore, it be accepted as correct that the latent heat of evaporation of the water formed should be deducted from the total heat of combustion as determined by calorimetric measurement, because this heat cannot be utilized, then it is equally right and proper that the latent heat of evaporation of the liquid fuel itself should be added to the observed calorimetric determination because its equivalent value in the heat of combustion can be and is used in any internal-combustion engine in which the fuel is evaporated before combustion starts, i.e. in any but Diesel engines; strictly speaking, there should be added the latent heat at constant volume, which is less than that at constant pressure by an amount equal to the work done on the atmosphere if the liquid is evaporated when unenclosed. In the following Table V is given the heat of combustion of various fuels in terms of B.Th.U.s per lb. and per gallon, the first two columns being the usually accepted lower calorific value and the second the revised value corrected to include the latent heat of evaporation of the liquid fuel itself. This latter corrected value is used throughout this volume for all determinations of thermal efficiency.

It cannot be too strongly emphasized that the heat value of a volatile liquid fuel bears no relation whatever to the power output

TABLE V

Fuel.	Calorific (lower) Value (exclusive of Latent Heat).		Calorific (lower) Value (including Latent Heat at Constant Volume).		Latent Heat of Evapora- tion of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.
	B.Th.U. per lb.	B.Th.U. per gallon.	B.Th.U. per lb.	B.Th.U. per gallon.	
Aromatic Free Petrol ...	19,080	137,000	19,200	136,200	133.0
"A" Petrol ...	18,450	144,300	18,580	145,200	142.0
"B" " ...	18,890	136,600	19,020	137,500	140.0
"C" " ...	19,000	138,100	19,120	137,000	135.0
"D" " ...	18,770	142,600	18,890	143,500	132.0
"E" " ...	18,970	136,400	19,090	137,100	132.0
"F" " ...	19,130	134,700	19,250	135,500	134.0
"G" " ...	...	...	...	...	...
"H" " ...	18,790	144,100	18,920	145,000	145.0
"I" " ...	...	...	...	...	...
<i>Heavy Fuels</i>					
Heavy Aromatics ...	17,900 (App.)	158,500 (App.)	18,030	159,600	136.0
Kerosene ...	19,000 (App.)	154,400 (App.)	19,100	155,200	108.0
<i>Paraffin Series</i>					
Pentane (Normal) ...	19,600	122,300	19,740	123,100	154.0
Hexane (80 % pure) ...	19,250	131,900	19,390	132,900	156.0
Heptane (97 % pure) ...	19,300	132,800	19,420	134,100	133.0
<i>Aromatic Series</i>					
Benzene (pure) ...	17,302	152,950	17,460	154,200	172.0
Toluene (99 % pure) ...	17,522	152,500	17,660	153,600	151.0
Xylene (91 % pure) ...	17,800	153,500	17,930	154,500	145.0
<i>Naphthene Series</i>					
Cyclohexane (93 % pure) ...	18,800	147,800	18,940	149,000	156.0
Hexahydrotoluene (80 %) ...	18,760	146,200	18,890	147,200	138.0
Hexahydroxylene (60 %) ...	18,770 (App.)	139,700 (App.)	18,890	140,600	133.0
<i>Olefines</i>					
Cracked Spirit (53 % unsat.) ...	18,400 (App.)	139,400 (App.)	18,540	140,200	150.0 (App.)
<i>Alcohol Group, &amp;c.</i>					
Ethyl Alcohol (98 %) ...	11,480	91,600	11,840	94,500	406.0
" " (95 vol. %) ...	10,790	88,000	11,130	92,000	442.0
Methyl Alcohol (Wood Naphtha) ...	9,630	79,900	10,030	83,300	500.0 (App.)
Methylated Spirits ...	10,200	83,700	10,580	86,900	450.0 (App.)
Butyl Alcohol (Coml.) ...	...	...	...	...	...
Ether (50 % in petrol) ...	16,700 (App.)	121,300 (App.)	16,830	122,500	146.0 (App.)
Carbon Disulph. (50 %) ...	10,600	105,400	10,730	106,600	146.0

obtainable from that fuel. In the case of gaseous fuels when the calorific value is very low, the apparent relationship is due largely to the presence in the gas of a considerable proportion of inert diluents such as nitrogen or carbon dioxide, in which case, of course, the power output is reduced because these diluents displace some of the available oxygen. In the case of volatile liquid fuels, however, the vapour contains no diluents at all, and the power output available is, therefore, entirely independent of the calorific value of the fuel.

The heat value of a fuel is, on the other hand, a direct measure of the quantity of fuel required, the lower the heat value the greater being the quantity needed to do the same work.

**Heat Value of Mixture.**—It is upon the heat value of the mixture of fuel and air, in the proportion required to give complete combustion, that the power output of an engine depends, and in this connection we find that all hydrocarbon fuels give within very close limits the same heat value per standard cubic inch of correct mixture. When allowance is made for the increase or decrease in specific volume after combustion, the variation becomes even less.

The following Table VI gives :—

- |          |   |
|----------|---|
| Col. (1) | The corrected calorific value of various petrols and other fuels.   |
| Col. (2) | The ratio of air/fuel by weight for complete combustion.  |
| Col. (3) | The increase or decrease in specific volume after combustion.   |
| Col. (4) | The ft.-lb. of energy liberated by the combustion of one standard cubic inch of mixture giving complete combustion, i.e. the total available internal energy. |

The heat value of the “correct” mixture is usually termed the total “internal energy” of the working fluid, and this term will in future be used in order to distinguish it from the calorific value of the fuel, which latter has no influence on the power output.

**Thermal Efficiency obtainable from Different Fuels.**—

Provided that the fuel is reasonably volatile, the thermal efficiency obtainable at any given compression ratio is substantially the same for all hydrocarbon fuels, irrespective of their chemical composition or of any other factor. In the case of the alcohol group, however, a somewhat higher thermal efficiency is obtained because, owing in part to their higher latent heat, and in part to their lower flame temperature, both the mean and the maximum temperatures of the cycle are lower, and the losses are therefore somewhat less. The range of burning on the weak side, which, by controlling the flame temperature, would control also the efficiency, happens to be almost

TABLE VI

Fuel.	(1) Calorific (lower) Value (including Latent Heat at Constant Volume).		(2) Air-to- Fuel Ratio by Weight for Complete Com- bustion.	(3) Spec. Volume after Com- bustion — that be- fore Com- bustion or “Volume Ratio.”	(4) Total Energy liberated by Combustion per cub. in. at n.t.p. of Mixture giving complete Combustion foot-lb.
	B.Th.U. per lb.	B.Th.U. per gall.			
Aromatic Free Petrol ...	19,200	136,200	15.05	1.053	48.5
“ A ” Petrol ...	18,580	145,200	14.3	1.038	48.15
“ B ” „ ...	19,020	137,500	14.7	1.049	48.45
“ C ” „ ...	19,120	137,000	14.8	1.052	48.53
“ D ” „ ...	18,890	143,500	14.6	1.047	48.35
“ E ” „ ...	19,090	137,100	14.9	1.051	48.51
“ F ” „ ...	19,250	135,500	15.0	1.053	48.54
“ G ” „ ...	...	...	...	...	...
“ H ” „ ...	18,920	145,000	14.7	1.048	48.31
“ I ” „ ...	...	...	...	...	...
<i>Heavy Fuels</i>					
Heavy Aromatics ...	18,030	159,600	13.8	1.04	48.52
Kerosene ...	19,100	155,200	15.0	1.06	48.91
<i>Paraffin Series</i>					
Pentane (Normal) ..	19,740	123,100	15.25	1.051	48.7
Hexane (80 % pure) ...	19,390	132,900	15.2	1.051	48.35
Heptane (97 % pure) ..	19,420	134,100	15.1	1.056	48.64
<i>Aromatic Series</i>					
Benzene (pure) ...	17,460	154,200	13.2	1.013	47.51
Toluene (99 % pure) ...	17,660	153,600	13.4	1.023	47.98
Xylene (91 % pure) ...	17,930	154,500	13.6	1.03	48.1
<i>Naphthene Series</i>					
Cyclohexane (93 % pure) ...	18,940	149,000	14.7	1.044	48.11
Hexahydrotoluene (80 %) ...	18,890	147,200	14.7	1.047	48.2
Hexahydroxylene (60 %) ...	18,890	146,600	14.8	1.054	48.59
<i>Olefines</i>					
Cracked Spirit (53 % unsat.) ...	18,540	140,200	14.8 (App.)	1.054	49.54
<i>Alcohol Group, &amp;c.</i>					
Ethyl Alcohol (98.5 %) ...	11,840	94,500	8.9	1.065	47.39
“ ” (95 vol. %) ...	11,130	92,000	8.4	1.065	46.86
Methyl Alcohol (Wood Naphtha)	10,030	83,300	6.5	1.06 (App.)	48.2 (App.)
Methylated Spirit ...	10,580	86,900	8.0 (App.)	1.064	48.82
Butyl Alcohol (Coml.) ...	...	...	...	...	...
Ether (50 % in petrol)...	16,830	122,500	13.0	1.06	49.2
Carbon Disulph. (50 %) ...	10,730	106,600	10.8	0.98	39.4

exactly the same in the case of all volatile liquid fuels yet examined with the exception of ether, and, in all cases, the maximum thermal efficiency is obtained when the mixture is 15 per cent weak. If any attempt be made to weaken the mixture beyond this point, combustion becomes unduly slow and incomplete, and the efficiency falls away in consequence. Theoretically, of course, the efficiency should rise steadily as the mixture strength is reduced and should follow an almost straight line law until at the point when the mixture is infinitely weak the efficiency should be equal to the air standard. In practice the extent to which the mixture can be weakened with increase of efficiency depends to some small extent, in the case of a single-cylinder engine, upon the position of the sparking plug and the intensity of the spark, while in a multi-cylinder engine it depends to a much larger extent upon the uniformity of distribution, but in both cases it is very limited.

It is perhaps rather remarkable that the maximum thermal efficiency obtainable with two fuels so widely different as hexane  $C_6H_{14}$  and benzene  $C_6H_6$  should be the same, but the explanation lies in the fact that while for  $CO_2$  the dissociation is great at high temperatures, yet the increase in specific heat is small. On the other hand, for  $H_2O$  the dissociation is small but the increase in specific heat is great. Curiously enough, these effects almost exactly balance one another, so that the sum of the losses from each source is practically the same.

The following Table VII gives :—

- Col. (1) The lower calorific value of the fuel corrected for the latent heat of evaporation.
- Col. (2) The lowest fuel consumption in terms of lbs. and pints per I.H.P. hour at a compression ratio of 5 : 1.
- Col. (3) The corresponding thermal efficiency.

The above figures relate to the thermal efficiency obtainable when all fuels are used at the same compression ratio.

It has, however, been stated previously that the highest compression ratio at which a fuel may be used depends upon its tendency to detonate, and it has been shown that this varies widely.

The following Table VIII shows the fuel consumption in terms both of pints and lbs. per I.H.P. hour when each fuel is used at its highest useful compression ratio.

It will be noted that in the case of kerosene and one or two other examples which are not very volatile, the observed thermal efficiency is considerably lower, due to the fact that a substantial

TABLE VII

Fuel.	(1) Calorific (lower) Value (including Latent Heat at Constant Volume).		(2) Minimum Con- sumption at Com- pression Ratio of 5:1 per I.H.P. Hour.		(3) Thermal Efficiency at Com- pression Ratio of 5:1 per cent.
	B.Th.U. per lb.	B.Th.U. per gallon.	lb.	Pints.	
Aromatic Free Petrol ...	19,200	136,200	0.415*	0.462*	31.9*
"A" Petrol ...	18,580	145,200	0.432	0.442	31.7
"B" " ...	19,020	137,500	0.423	0.468	31.7
"C" " ...	19,120	137,000	0.421	0.463	31.6
"D" " ...	18,890	143,500	0.422	0.445	31.9
"E" " ...	19,090	137,100	0.421*	0.469*	31.7*
"F" " ...	19,250	135,500	0.414	0.471	31.9
"G" " ...	...	...	0.426	0.454	...
"H" " ...	18,920	145,000	0.425	0.443	31.7
"I" " ...	...	...	0.418	0.460	...
<i>Heavy Fuels</i>					
Heavy Aromatics ...	18,030	159,600	0.510	0.461	27.6
Kerosene ...	19,100	155,200	0.523*	0.515*	25.4*
<i>Paraffin Series</i>					
Pentane (Normal) ...	19,740	123,100	...	...	...
Hexane (80 % pure) ...	19,390	132,900	0.411	0.480	32.0
Heptane (97 % pure) ...	19,420	134,100	0.410*	0.475*	31.9*
<i>Aromatic Series</i>					
Benzene (pure) ...	17,460	154,200	0.458	0.415	31.8
Toluene (99 % pure) ...	17,660	153,600	0.455	0.418	31.7
Xylene (91 % pure) ...	17,930	154,500	0.452	0.420	31.4
<i>Naphthene Series</i>					
Cyclohexane (93 % pure) ...	18,940	149,000	0.420	0.427	31.9
Hexahydrotoluene (80 %) ...	18,890	147,200	0.425	0.430	31.7
Hexahydroxylene (60 %) ...	18,890	140,600	0.424*	0.456*	31.8*
<i>Olefines</i>					
Cracked Spirit (53 % unsat.) ...	18,540	140,200	0.429	0.453	32.0
<i>Alcohol Group, &amp;c.</i>					
Ethyl Alcohol (98 %) ...	11,840	94,500	0.663	0.665	32.4
" " (95 vol. %) ...	11,130	92,000	0.705	0.692	32.5
Methyl Alcohol (Wood Naphtha)	10,030	83,300	0.777	0.750	32.7
Methylated Spirits ...	10,580	86,900	0.740	0.721	32.5
Butyl Alcohol (Coml.) ...	...	...	0.566	0.550	...
Ether (50 % in petrol) ...	16,830	122,500	...	...	...
Carbon Disulph. (50 %) ...	10,730	106,600	...	...	...

\* This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5:1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.

TABLE VIII

Fuel.	Minimum Consumption at Highest Useful Compression per I.H.P. Hour.	
	lb.	Pints.
Aromatic Free Petrol ... ..	0.422	0.471
"A" Petrol ... ..	0.393	0.402
"B" " " ... ..	0.393	0.435
"C" " " ... ..	0.410	0.451
"D" " " ... ..	0.407	0.428
"E" " " ... ..	0.435	0.484
"F" " " ... ..	0.412	0.469
"G" " " ... ..	0.449	0.478
"H" " " ... ..	0.389	0.405
"I" " " ... ..	0.457	0.503
<i>Heavy Fuels</i>		
Heavy Aromatics ... ..	0.447	0.404
Kerosene ... ..	0.581	0.571
<i>Paraffin Series</i>		
Pentane (Normal) ... ..	..	..
Hexane (80 % pure) ... ..	0.405	0.473
Heptane (97 % pure) .. ..	0.491	0.568
<i>Aromatic Series</i>		
Benzene (pure) ... ..	0.392	0.355
Toluene (99 % pure) ... ..	0.385	0.354
Xylene (91 % pure) ... ..	0.381	0.354
<i>Naphthene Series</i>		
Cyclohexane (93 % pure)... ..	0.385	0.392
Hexahydrotoluene (80 %) ... ..	0.394	0.404
Hexahydroxylene (60 %) ... ..	0.429	0.461
<i>Olefines</i>		
Cracked Spirit (53 % unsat.) .. ..	0.405	0.428
<i>Alcohol Group, &amp;c.</i>		
Ethyl Alcohol (98 %) ... ..	0.532	0.533
" " (95 vol. %) ... ..	0.565	0.555
Methyl Alcohol (Wood Naphtha) ... ..	0.725	0.700
Methylated Spirits... ..	0.625	0.609
Butyl Alcohol (Coml.) ... ..	0.472	0.459
Ether (50 % in petrol) ... ..	..	..
Carbon Disulph. (50 %) ... ..	..	..

proportion of the liquid fuel is deposited on the walls of the induction piping and cylinder and escapes combustion. This proportion



could be reduced by further pre-heating, but all the tests were carried out, for comparative purposes, at exactly the same heat input to the carburettor.

By additional pre-heating a slightly higher thermal efficiency can be obtained, but the power output is reduced and the tendency to detonate increased thereby.

**The Maximum Power Output.**—The maximum power output obtainable from any fuel depends upon the internal energy of the working fluid and upon the latent heat of evaporation of the liquid.

The former varies very little, the latter considerably as between different fuels—generally speaking as regards the true hydrocarbon fuels the variations in internal energy and latent heat just about balance, with the result that the maximum power output is the same for all. For example, the total internal energy of benzene is about 1.5 per cent less than that of hexane, on the other hand the latent heat of benzene is considerably greater, and a greater weight of mixture is therefore retained in the cylinder, with the result that under identical temperature and other conditions both give, at the same compression ratio, the same power output to within less than half of 1 per cent.

In the case of alcohol, although the total internal energy of the mixture is appreciably lower, yet the latent heat is so much greater that a much denser charge is retained in the cylinder and the power output is some 5 per cent greater despite the lower internal energy.

Table IX shows :

- Col. (1) The total internal energy.
- Col. (2) The latent heat of evaporation.
- Col. (3) The observed indicated mean pressure (power output) at a compression ratio of 5 : 1.
- Col. (4) The observed indicated mean pressure at the highest useful compression ratio.

**Fuels for Aircraft.**—For all commercial purposes other than aircraft, fuel is supplied by bulk, not weight, and it is therefore the heat value per gallon and not per lb. which need be considered. In the particular case of aircraft, however, the weight and not the bulk of the fuel becomes the primary consideration. Other things being equal, therefore, the fuel with the highest calorific value per lb. will be the most efficient. From this point of view alone, the paraffin series would appear the most desirable. Unfortunately, however, owing to their tendency to detonate, the members of this series cannot be used in a high compression and therefore in an efficient engine.

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TABLE IX

Fuel.	Total Energy liberated by Combustion per cub. in. of Mixture giving complete Combustion foot-lb.	Latent Heat of Evaporation of Fuel (at Constant Pressure atmospheric) B.Th.U. per lb.	Max. Ind. Mean Effective Pressure at Compression Ratio of 5:1. Heat 65 B.Th.U. Min. lb. per sq. in.	Max. Ind. Mean Effective Pressure at Highest Useful Compression. Heat 65 B.Th.U. Min. lb. per sq. in.
Aromatic Free Petrol ...	48.5	133.0	131.3*	130.0
" A " Petrol ...	48.15	142.0	131.2	140.1
" B " " ...	48.45	140.0	131.5	137.5
" C " " ...	48.53	135.0	131.0	133.9
" D " " ...	48.35	132.0	131.2	134.9
" E " " ...	48.51	132.0	131.0*	128.6
" F " " ...	48.54	134.0	131.8	132.7
" G " " ...	..	..	131.5*	127.4
" H " " ...	48.31	145.0	131.0	139.5
" I " " ...	..	..	131.7*	125.1
<i>Heavy Fuels</i>				
Heavy Aromatics ...	48.52	136.0	131.1	142.5
Kerosene... ..	48.91	108.0	130.7	123.0
<i>Paraffin Series</i>				
Pentane (Normal) ...	48.7	154.0	131.3	139.0
Hexane (80 % pure) ...	48.35	156.0	132.3	133.1
Heptane (97 % pure) ...	48.64	133.0	131.2*	119.5
<i>Aromatic Series</i>				
Benzene (pure) ...	47.51	172.0	131.6	146.5
Toluene (99 % pure) ...	47.98	151.0	131.5	147.0
Xylene (91 % pure) ...	48.1	145.0	131.5	146.8
<i>Naphthene Series</i>				
Cyclohexane (93 % pure) ...	48.11	156.0	131.3	139.0
Hexahydrotoluene (80 %) ...	48.2	138.0	131.0	137.9
Hexahydroxylene (60 %) ...	48.59	133.0	130.8*	130.0
<i>Olefines</i>				
Cracked Spirit (53 % unsat.) ...	49.54	150.0 (App.)	131.6	136.0
<i>Alcohol Group, &amp;c.</i>				
Ethyl Alcohol (98 %) ...	47.39	406.0	137.8	156.5
" " (95 vol. %)	46.86	442.0	142.0	161.2
Methyl Alcohol (Wood Naphtha)	48.2 (App.)	500.0 (App.)	144.8	146.6
Methylated Spirits ...	46.82	450.0 (App.)	144.5	155.5
Butyl Alcohol (Coml.) ...	..	..	138.0	156.0
Ether (50 % in petrol)...	49.2	146.0 (App.)	136.0*	125.0
Carbon Disulph. (50 %) ...	39.4	146.0	124.5	125.7

\* This sign indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5:1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same compression.

If the compression were adjusted to suit the fuel, then the greatest distance could be flown on that fuel which gives the highest value for the product thermal efficiency  $\times$  heat value per lb., and it is interesting to note that of all the fuels examined the highest figures are :

			Heat Value.
			Efficiency B.Th.U.
			per cent. per lb. Product.
(1) Xylene ...	...	...	$0.373 \times 17,930 = 6700$
(2) Cyclohexane ...	...	...	$0.349 \times 18,940 = 6600$
(3) Petrol (sample B) ...	...	...	$0.341 \times 19,020 = 6500$

Although xylene comes first on the list, it is not a practicable fuel on account of its high boiling point and low volatility, also it is only efficient when used at a compression ratio so high (namely, above 6.75 : 1) as to render the engine unduly heavy owing to the very high maximum pressures to be withstood. Cyclohexane, one of the lighter members of the naphthene series, would actually give the best possible results from every point of view, but is not obtainable in bulk. The next on the list, viz. the petrol sample (B), is a very light and highly volatile fuel which was prepared by the Asiatic Petroleum Co. for, and used in, the cross-Atlantic flight.

In aircraft engines it is particularly desirable to use a very volatile fuel on account of distribution, and in order to be able to reduce to the minimum the amount of pre-heating required, and so to make the maximum possible use of the latent heat of evaporation to increase the power output.

For present-day (1928) aircraft engines which have an average compression ratio of about 5.4:1 the fuel used should have a toluene value of not less than +20, that is to say, it should contain not less than about 25 per cent of aromatics or their equivalent in naphthenes, in order to eliminate detonation and so enable the engine to develop its full power at ground level, as is most necessary when starting heavily laden. Since, however, detonation is largely a function of pressure, once an aeroplane has attained a reasonable altitude and is in air at a lower density, the tendency to detonate will disappear and a fuel of lower toluene value may be used. From an idealist point of view, therefore, it would seem desirable to employ a fuel of high toluene value at or near ground level, and to change over to a nearly pure paraffin petrol so soon as a sufficient altitude has been obtained.

By the use of alcohol, or of a fuel with high latent heat, or of one which has had its latent heat of evaporation augmented by the

introduction of water in solution with the fuel, a very substantial advantage may be gained, because, as shown previously, the high latent heat will increase the available power output very considerably—a most important consideration, especially in the case of aeroplanes when taking off with heavy loads.

#### SUMMARY

The results of a lengthy investigation of the various volatile liquid fuels carried out on behalf of the Asiatic Petroleum Co. may be summarized as follows :—

(1) It has been proved that the tendency of a fuel to detonate is the one outstanding factor in determining its value for use in a constant-volume internal-combustion engine. Compared with this, most other considerations are of secondary importance.

(2) There appears to be little doubt as to the correctness of the view, now generally accepted, that detonation is largely dependent upon the normal rate of burning of the fuel and is less the lower the rate of burning.

(3) In all cases it seems that a low rate of burning is advantageous. No fuel has yet been found whose rate of burning was too low to permit of maximum efficiency being obtained in the highest speed engine yet tested.

(4) Fuels capable of standing a very high compression will operate in a low compression engine, equally as efficiently as those whose normal rate of burning is high—provided that there is a reasonable degree of turbulence in the combustion chamber.

(5) Apart from the limitations introduced by detonation, the power output obtainable from all volatile liquid fuels, with the exception of the alcohol group, is the same at the same compression to within less than 2 per cent. Such variations as occur, within this range, are due rather to variations in the latent heat of evaporation than to any other circumstance.

(6) Owing to the high latent heat and low boiling point of alcohol and certain other bodies, the weight of charge per cycle is greater and a higher power output is obtained in consequence.

(7) The efficiency with which all volatile fuels, other than alcohol, &c., are burnt is practically the same, at the same compression ratio, irrespective of rate of burning, provided the compression is low enough to avoid detonation under any circumstances. In the case of alcohol, the efficiency is slightly higher, on account of the lower flame temperature.

(8) The useful range of burning is, to all intents and purposes, the same for all volatile liquid fuels.

(9) The unavoidable losses due to the combined influence of dissociation and change of specific heat at high temperatures are substantially the same in all cases.

(10) All the experimental results indicate that the performance of any combination of hydrocarbons as regards detonation, and therefore the power output and efficiency obtainable, is the mean performance of each of the components. The performance of any complex fuel such as petrol can therefore be predicted, once the nature and proportion of its constituents are determined, or conversely, a fuel can be prepared to give any required performance, within the limits available.

(11) The highest useful compression ratio for, and therefore the power output obtainable from, any petrol is governed by the relative proportions of aromatics, naphthenes, and paraffins it contains—the smaller the proportion of the latter the better from almost every point of view.

(12) To judge of the quality of a fuel by its specific gravity is entirely misleading. If naphthene and aromatic fractions are present in any large proportion (as is frequently the case), then a high specific gravity is a substantial advantage.

(13) Owing to the very rich mixture delivered normally by pilot jets, and the still further enrichment effected by flooding, only a relatively small proportion of highly volatile constituents appears to be required for starting.

Table X (facing p. 30) and Table XI (facing p. 32) give a general summary of the above investigations and test results which were carried out on behalf of the Asiatic Petroleum Co. That the author is permitted to publish them without reservation of any kind is due to the generosity and public spirit of this most enterprising company. The following particulars of the single-cylinder research engine and its accessories which formed the principal piece of apparatus used for the research, may be of some interest :—

In figs. 3-6 are shown drawings and photographs of the variable compression engine referred to in connection with the preceding tests.

In the design of this engine the following considerations were taken into account :—

(1) In view of the prolonged and extensive nature of the tests, not only were durability and reliability regarded as matters of



primary importance, but every effort was made to ensure mechanical consistency.

(2) Every expedient known at the time, viz. 1919, was adopted to attain the highest possible thermal efficiency and power output, and to ensure that all losses, whether thermal or frictional, were reduced to the absolute minimum, and maintained as nearly constant as possible, under all conditions.

(3) The engine was designed to run when required at a piston speed in excess of that of contemporary engines.

(4) Means are provided for varying the compression of the engine over any range from 3·7 : 1 up to 8 : 1 while running at full power, and without disturbing any temperature, frictional, mechanical, or other conditions.

(5) The combustion chamber is so designed that its general form and ratio of surface to volume undergo the minimum of alteration when the compression is varied, and to this end a very long stroke is employed.

(6) Special means are adopted to render the engine as little sensitive to changes in the temperature of the lubricant as possible. Ball bearings are used wherever possible, in order to reduce variation in friction with different oil temperatures, and the water jacketing round the barrel of the cylinder is stagnant, and therefore quickly attains a constant temperature, independent of the temperature of the supply. This ensures that the piston friction, which is dependent upon the temperature of the oil on the cylinder walls, reaches a minimum in the course of a few minutes, and thereafter remains constant. The importance of retaining as nearly as possible the same general form of combustion chamber under all conditions of compression cannot be overestimated. Very misleading results have been obtained when the compression ratio has been varied by fitting different pistons, some with concave, others with convex crowns. In one series of experiments with different compression ratios which the writer examined, and which were obviously carried out with the most scrupulous care, the results were entirely vitiated because the whole character and efficiency of the combustion chamber were completely changed, as between the low compression and the high, with the result that a certain optimum compression ratio was claimed to have been found, after which any further increase in compression resulted in loss of power and efficiency. A careful scrutiny of the results showed that at or near the so-called optimum compression ratio the efficiency of the combustion chamber

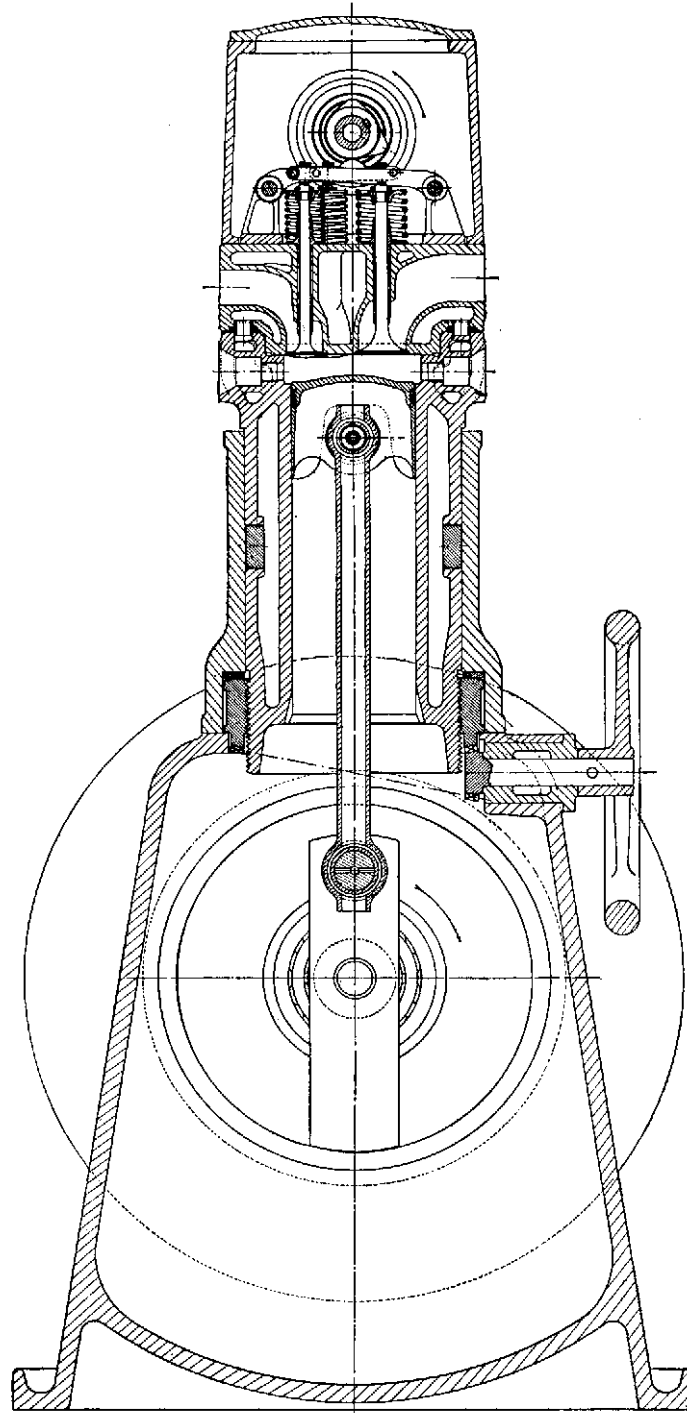


Fig. 3



TABLE XI.—EXPERIMENTAL RESULTS FOR VARIABLE COMPRESSION ENGINE.

A.	B.	C.	D.	E.		F.		H.	I.	J.	K.	L.	M.	
				lb.	Pints.	lb.	Pints.						At the same Compression Ratio per cent.	Relative Thermal Efficiency (compared with that obtained with Toluene).
Highest Useful Compression Ratio in Variable Compression Engine.	Pressure at Highest Useful Compression (calculated). lb. per sq. in. (gauge).	Compression Temp. at Highest Useful Compression (calculated). Deg. F.	Toluene Value. Volume = 100%. Aromatic Free. Petrol = 0%.	Minimum Consumption at Compression Ratio of 5:1 per I.H.P. Hour.	Minimum Consumption at Highest Useful Compression per I.H.P. Hour.	Thermal Efficiency at Compression Ratio of 5:1 per cent.	Thermal Efficiency at Highest Useful Compression Ratio of 5:1 per cent.	Thermal Efficiency at Highest Useful Compression per cent.	Max. Ind. Mean Effective Pressure at Compression Ratio of 5:1. Heat 65 B.Th.U./Min. lb. per sq. in.	Max. Ind. Mean Effective Pressure at Highest Useful Compression. Heat 65 B.Th.U./Min. lb. per sq. in.	Max. Ind. Mean Effective Pressure at Highest Useful Compression. No Heat. lb. per sq. in.	Rise or Fall of Temp. in Induction Pipe (indicating app. Mean Volatility). Heat 65 B.Th.U./Min. Deg. F.	At the same Compression Ratio per cent.	At Highest Useful Compression per cent.
4.85	105.5	738.0	0	0.415*	0.462*	0.492	0.471	31.4	131.3*	130.0	138.1	+18.0	100.0 (App.)	83.7
6.0	148.5	808.0	38.0	0.432	0.432	0.393	0.402	34.9	131.2	140.1	148.5	+10.8	100.0 (App.)	93.0
5.7	132.5	792.0	28.0	0.423	0.468	0.393	0.435	34.1	131.5	137.5	146.0	+5.2	100.0 (App.)	91.0
5.25	118.0	765.0	13.5	0.421	0.468	0.410	0.431	33.5	131.0	133.5	142.4	+12.6	100.0 (App.)	86.7
5.35	121.5	770.0	16.5	0.422	0.445	0.407	0.425	33.1	131.2	134.9	142.9	+19.8	100.0 (App.)	88.3
4.7	100.5	720.0	-6.5	0.421*	0.469*	0.435	0.484	30.7	131.0*	128.6	136.7	+19.8	100.0 (App.)	81.8
5.05	111.5	752.0	-10.0	0.414	0.471	0.412	0.469	32.1	131.8	132.7	140.5	+9.5	100.0 (App.)	85.8
5.55	130.0	718.0	-35.0	0.426*	0.454*	0.449	0.478	—	131.5*	127.4	135.4	+16.6	100.0 (App.)	92.2
5.9	149.5	804.0	33.0	0.425	0.443	0.389	0.405	34.6	131.0	139.5	147.8	+21.6	—	—
4.5	89.0	704.0	-20.0	0.418*	0.460*	0.457	0.503	—	131.7*	125.1	132.5	+25.2	—	—
6.5	163.5	820.0	55.0	0.510	0.461	0.447	0.404	31.5	131.1	142.5	—	+50.5	86.0	84.0
4.2	86.0	696.0	-22.0	0.523*	0.515*	0.581	0.571	22.9	130.7*	123.0	—	+56.0	80.0	61.2
5.85	138.5	800.0	33.0	—	—	—	—	—	131.3	139.0	—	—	—	—
5.1	113.5	754.0	8.0	0.411	0.480	0.405	0.473	32.4	132.3	133.1	141.2	0	100.0 (App.)	86.4
3.75	72.0	667.0	-37.0	0.410*	0.475*	0.491	0.568	26.7	131.2*	119.5 at 9.8:1	125.5	+10.0	100.0 (App.)	71.2
6.9†	179.0	842.0	67.0	0.458	0.415	0.392	0.355	37.2	131.6	146.5	156.0	-13.1	100.0 (App.)	99.2
>7.0	>183.0	>846.0	100.0	0.455	0.418	0.385	0.354	37.5	131.5	147.0 at 7.0:1	156.3	+14.4	100.0 (App.)	100.0
>7.0	>183.0	>846.0	85.0	0.452	0.420	0.381	0.354	37.3	131.5	146.8 at 7.0:1	156.1	+32.4	100.0 (App.)	99.5
5.9†	140.5	802.0	35.0	0.420	0.427	0.385	0.392	34.0	131.3	139.0	148.0	-5.4	100.0 (App.)	93.0
5.8	136.5	798.0	31.5	0.425	0.436	0.394	0.404	34.3	131.0	137.9	146.5	+5.4	100.0 (App.)	91.5
4.9	107.0	741.0	1.5	0.424*	0.456*	0.429	0.461	31.5	130.8*	130.0	138.1	+24.3	100.0 (App.)	84.0
5.55	128.0	782.0	23.5	0.429	0.453	0.405	0.428	33.0	131.6	136.0	145.4	+19.0	100.0 (App.)	84.0
>7.5	>204.0	>796.0	>88.0	0.663	0.665	0.532	0.533	40.4	137.8	156.5 at 7.5:1	165.5	+2.7	102.0	107.9
>7.5	>204.0	>796.0	>88.0	0.705	0.692	0.565	0.565	40.5	142.0	161.2 at 7.5:1	170.0	+3.6	102.5	108.1
5.9†	116.5	648.0*	—	0.777	0.750	0.725	0.700	35.1	144.8	148.6	153.9	-13.3	103.0	93.5
5.5†	>183.0	720.0*	—	0.740	0.721	0.625	0.609	38.5	144.5	155.5	165.0	-1.8	102.5	102.7
7.3	193.0	—	80.0	0.566	0.550	0.472	0.459	—	138.0	156.0	164.5	+18.0	—	—
3.9	77.0	662.0	(-32.0)	—	—	—	—	—	136.0*	125.0	132.5	+1.8	—	—
5.15†	115.0	734.0	(9.4)	—	—	—	—	—	124.5	125.7	136.3	-12.6	—	—

This sign (†) indicates that pre-ignition occurred before audible detonation.

This sign (\*) indicates that the values are only calculated, since these fuels could not be tested at a compression ratio of 5:1 owing to detonation. The values have been inserted to show the efficiency and power obtained relatively to the other fuels if used at the same conditions.

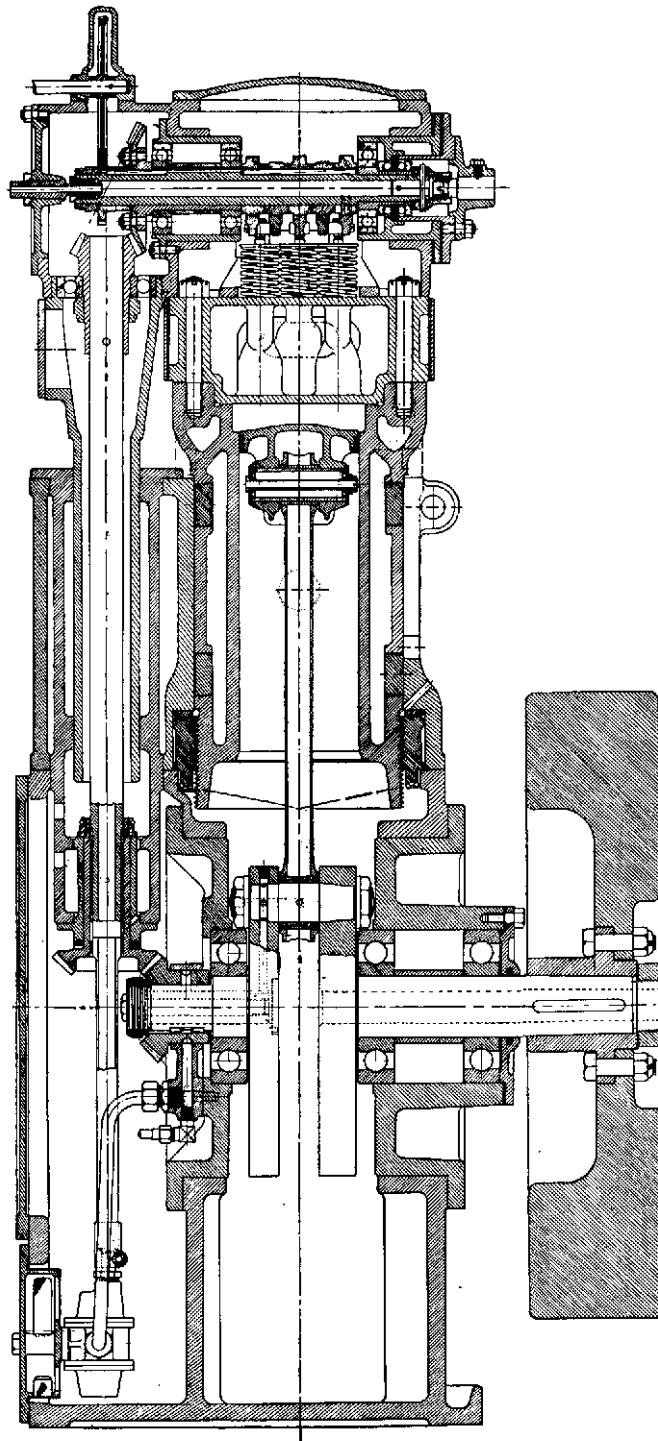


Fig. 4.—Variable Compression Engine, Sectional Side Elevation

(E 246)

was at a maximum, and that at the higher compressions it fell away very rapidly, and, indeed, became quite exceptionally inefficient.

In the variable compression engine designed for the purpose of these tests, the efficiency of the combustion chamber undergoes very little change between the lowest compression ratio and the highest, with the result that the efficiency increases with increase of compression at a perfectly steady rate throughout the whole range.

As will be seen from the sectional drawings, figs. 3 and 4, the compression ratio is varied by raising or lowering the whole cylinder,

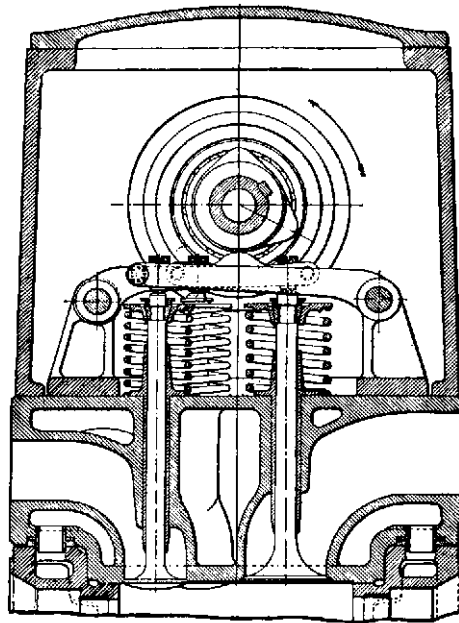


Fig. 5.—Section through Cylinder Head showing Valve Gear

together with the carburettor, camshaft, and valve gear (fig. 5); by this means the compression ratio can be varied over any range in the course of a few seconds, and without disturbing any of the temperature conditions or any adjustments.

To measure and record the compression ratio in use, a micrometer is provided as shown in fig. 6. This is arranged to operate electrical contacts, and controls a pilot lamp, which lights up immediately the desired compression is reached, so that the operator can adjust the micrometer screw at his leisure to the compression ratio he requires before making any

alteration, and can then see at a glance by the lighting up of the lamp that this compression has been reached.

For the ignition of the charge four sparking plugs are fitted equidistant round the circumference of the combustion chamber, each of which is connected to a Remy high-tension coil. The low-tension circuit of all the coils is operated by a single Remy contact breaker driven directly from one end of the camshaft. The object of using this arrangement in preference to magnetos was twofold :

- (1) To ensure that the passage of the spark across all four plugs should be absolutely synchronous.
- (2) To ensure that the intensity of the spark should be the same at all settings.

In practice, it was found that the use of two sparking plugs on opposite sides of the combustion space gave equally good results, and all tests were therefore run under these conditions. To measure accurately both the power and friction losses the engine is direct coupled to a balanced swinging field electric dynamometer, one arm of which carries a dead weight of 40 lb., which is slightly in excess

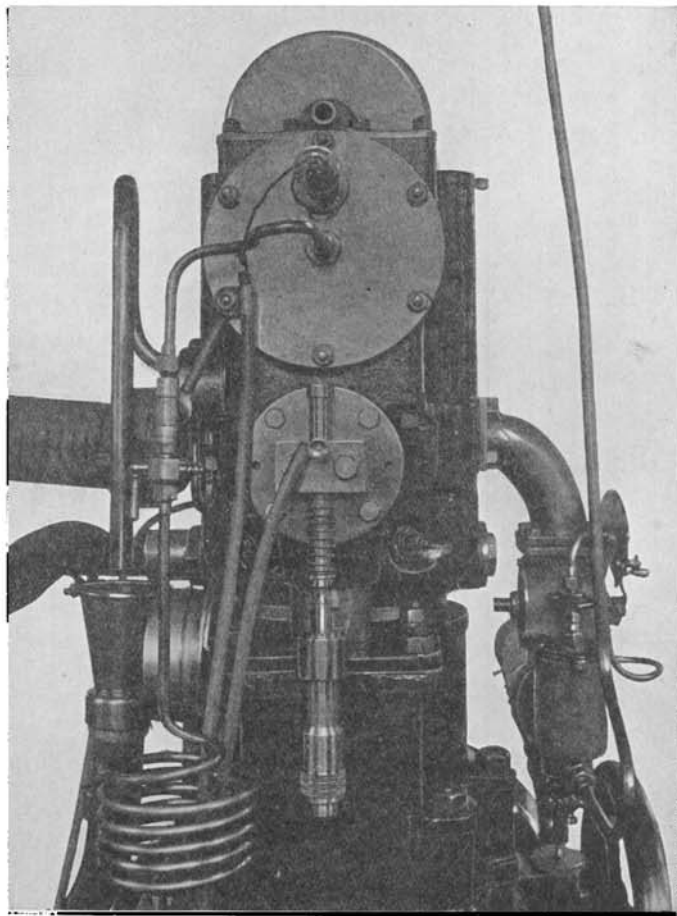


Fig. 6.—Photo of upper Part of Engine showing Micrometer for measuring Compression Ratio

of the maximum torque of the engine—a light open-scale spring balance is used to record the difference in torque between the dead weight and that developed by the engine—this arrangement permits of exceedingly accurate determinations, since a very small variation in torque corresponds with a wide range on the spring balance. The mean torque on the dynamometer arm is in the neighbourhood of 35 lb., and the difference can be read off at a glance to within less

than one-tenth of a pound. The steadiness of the dynamometer is such that the needle of the spring balance does not vibrate or oscillate through a range of more than  $\pm 0.1$  lb. Generally speaking, all readings of torque may be taken as being accurate to within one-third of 1 per cent, while the standard of accuracy of the average of

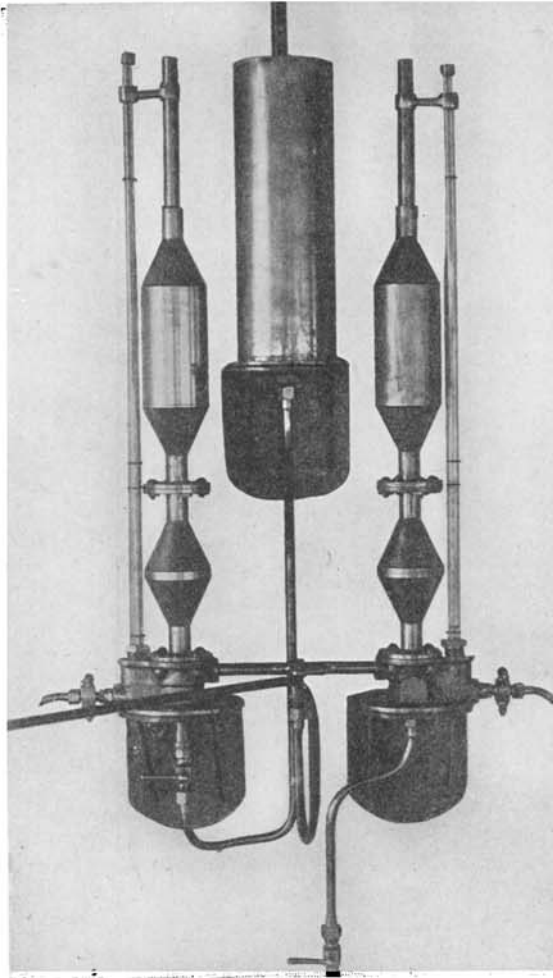


Fig. 7.—Fuel-measuring Device

several readings is, of course, considerably higher. The load is controlled by varying the field excitation of the dynamometer. For this purpose, two rheostats are provided in the field circuit, one of which gives coarse graduations, and the other, a continuous coil resistance, affords continuous range, and is used for fine adjustments.

The fuel-measuring device consists of two vessels, each comprising

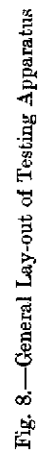
two conical-ended chambers connected together at either end by a narrow throat, as shown in fig. 7. The upper chamber has a capacity of exactly one pint, and the lower of  $\frac{1}{4}$  pint. A gauge glass is fitted to each vessel. The rate of fall of liquid in the glass is very rapid when passing the narrow throats, so that its passage past the marks on the gauge glass can be clocked with extreme accuracy. Geared off one end of the camshaft is a revolution counter operated by means of a magnetic clutch, and so arranged that the counter is thrown into operation as the liquid in the gauge glass passes the first mark, and is thrown out, and a brake applied to its spindle as the liquid passes the second mark; thus the actual number of revolutions during the consumption of either one pint or  $\frac{1}{4}$  pint of fuel is automatically recorded.

The carburettor is a standard Claudel-Hobson aircraft type, but fitted with a fine adjustment needle valve controlling the jet, so that the mixture can be varied between close limits. An electrical heater is fitted in the carburettor air intake passage, and the exact amount of heat supplied can be read off from instruments on the switchboard. A thermometer fitted in an insulated pocket, and projecting into the inlet valve port, is provided to record, very approximately, the temperature of the working fluid during its entry to the cylinder. From the known amount of heat supplied, and from the measured difference in the temperature of the air before and after its entry to the carburettor, it is possible to determine at least a relative measure of the mean volatility of the fuel used. The readings of the thermometer in the induction passage are, however, of relative value only. Owing to the variations in the temperature of the thermometer pocket due to the deposition of liquid fuel upon it, this thermometer behaves as a wet bulb instrument, and even the relative values of its readings cannot be relied upon as between fuels of widely different latent heats of evaporation.

A general lay-out of the apparatus is shown in fig. 8.

A small calibrated gas-holder is also provided, from which the engine can draw its supply of air when running on a liquid fuel.

The fall of the gas-holder controls an electrical contact mechanism which in turn operates a magnetic counter on the observer's desk; this counter being inter-connected, electrically, with the revolution counter on the engine. By this means the air consumption per revolution can be measured with great accuracy.



## CHAPTER II

### DETONATION

The phenomenon of detonation appears to be the setting up in the cylinder of an explosion wave. This occurs when the rapidity of combustion of that portion of the working fluid first ignited is such that, by its expansion, it compresses before it the unburnt portion beyond a certain rate. When the rate of temperature rise due to compression by the burning portion of the charge exceeds that at which it can get rid of its heat by conduction, convection, &c., by a certain margin, the remaining portion ignites spontaneously and nearly simultaneously throughout its whole bulk, thus setting up an explosion wave which strikes the walls of the cylinder with a hammer-like blow and, reacting in its turn, compresses afresh the portion first ignited. This further raises the temperature of that portion, and with it the temperature of any isolated or partially insulated objects in its vicinity, thus ultimately giving rise to pre-ignition. It would appear, therefore, pretty certain that detonation depends primarily upon the rate of burning of that portion of the charge first ignited, and it remains to discover what actually controls this rate.

#### **Influence of the Nature of the Fuel upon Detonation.—**

This subject has already been touched on in Chapter I. Broadly speaking, it would appear both from actual engine tests and from Tizard's researches, that two factors determine whether or not a fuel will detonate:

- (1) The self-ignition temperature of the fuel/air mixture.
- (2) The rate of acceleration of burning as the ignition temperature is exceeded.

Both the true self-ignition temperature, if indeed such a term can be used, and the rate of acceleration of burning appear to depend primarily upon the chemical composition of the fuel.

Reference has been made to the self-ignition temperature of a fuel/air mixture. Strictly speaking there is no such thing, for any fuel/air mixture will actually combine in time at almost any temperature. In the case of hydrocarbon fuels the rate of combination

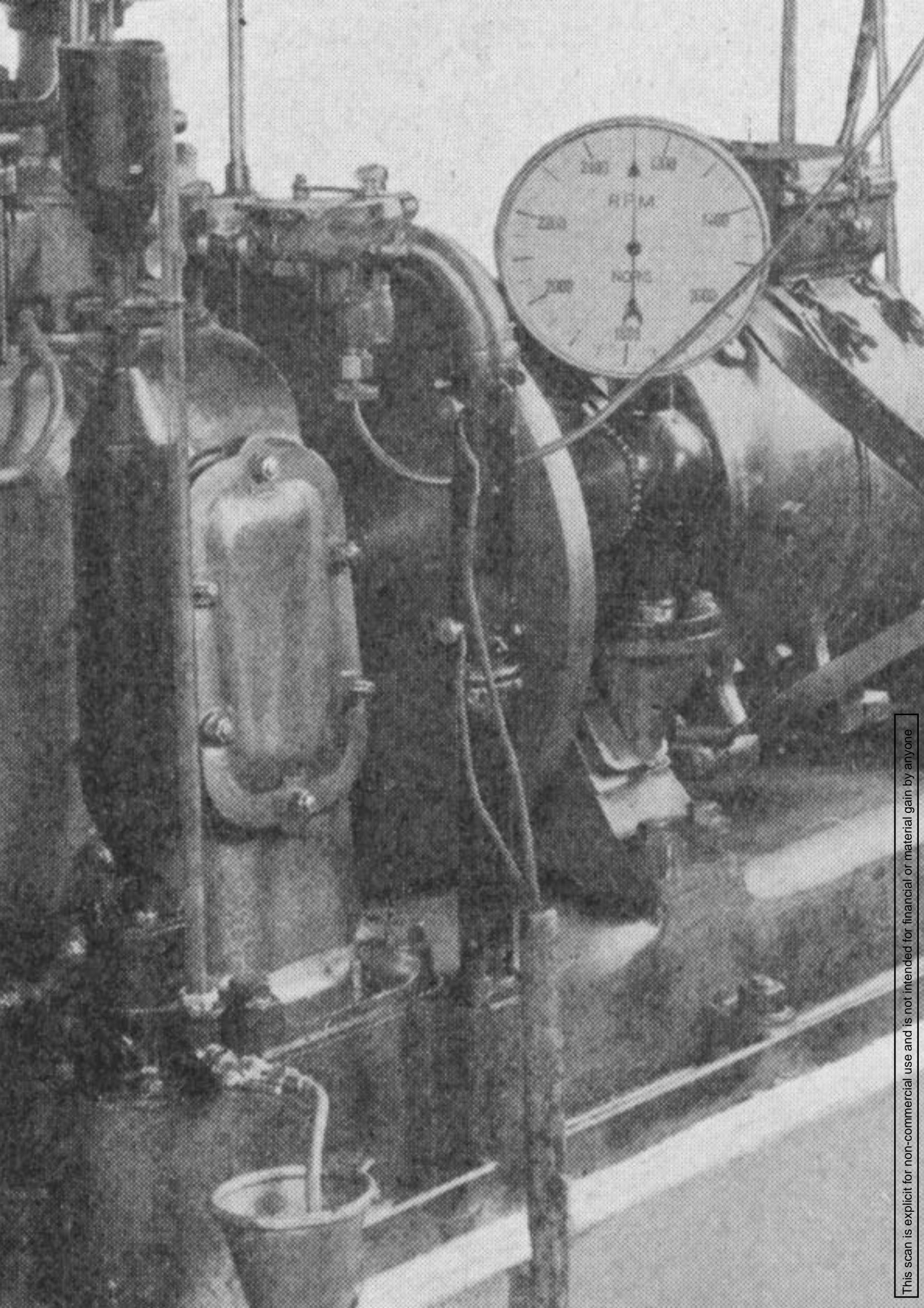


is multiplied about three times for every 3 per cent increase in temperature, so that, in practice, there is a comparatively narrow range of temperature over which the rate of burning is reasonable. In this connection the case of carbon disulphide is particularly interesting—this fuel has an exceedingly low ignition temperature, so low that when used neat in an engine it will pre-ignite at once even at a compression ratio of 3·6 : 1, yet, when mixed with petrol, carbon disulphide is quite effective in preventing detonation and will allow of the use of a higher compression ratio than is normally possible with petrol. On investigation it has been found that, in the particular case of carbon disulphide, the rate of acceleration of burning is very much slower and that it trebles its rate for every 7 per cent rise in temperature as against about every 3 per cent for all hydrocarbon fuels.

Apart from the variations in the composition of the fuel, various design factors enter also into the question and exert a powerful influence. Of these the most important appears to be the absolute distance from the point of ignition to the most remote corner of the combustion chamber. This, however, is not the whole story, for the tendency to detonation is influenced also by the degree of turbulence of the gases in the remote corner and by the temperature of the boundary surfaces, both of which factors determine the facility with which the entrapped gases can dispose of the heat generated by local compression in front of the advancing inflammation. Thus the tendency to detonation will be greatest when the corner most remote from the sparking plug contains relatively stagnant gas or is bounded by some hot surface, such as the head of the exhaust valve.

Since the absolute length of flame path appears to be the main determining factor, it follows that, given similarity of design, the smaller the cylinder, the higher the ratio of compression which can be employed with any given fuel. From this consideration it appears that the efficiency of the petrol engine will increase as the size of the cylinder is reduced. Against this, however, both the heat loss to the cylinder walls and the loss due to the proportion of chilled and incompletely burned gas increases as the cylinder size is reduced.

**Effect on Engine Efficiency and Power.**—The net result of these various factors is that the thermal efficiency of a petrol engine is, within wide limits, nearly independent of its size. The specific power output of a small engine is greater than that of a large one, since the cycle can be performed more frequently. The reason for this difference is purely a mechanical one, the limits being due to the inertia stresses in the moving parts.



exhaust) described in Chapter I. Ignition is by two synchronised sparks occurring at diametrically opposite sides of the combustion chamber, the distance between the two sparking plugs being 5 in. exactly. With a certain fuel and with the mixture strength and

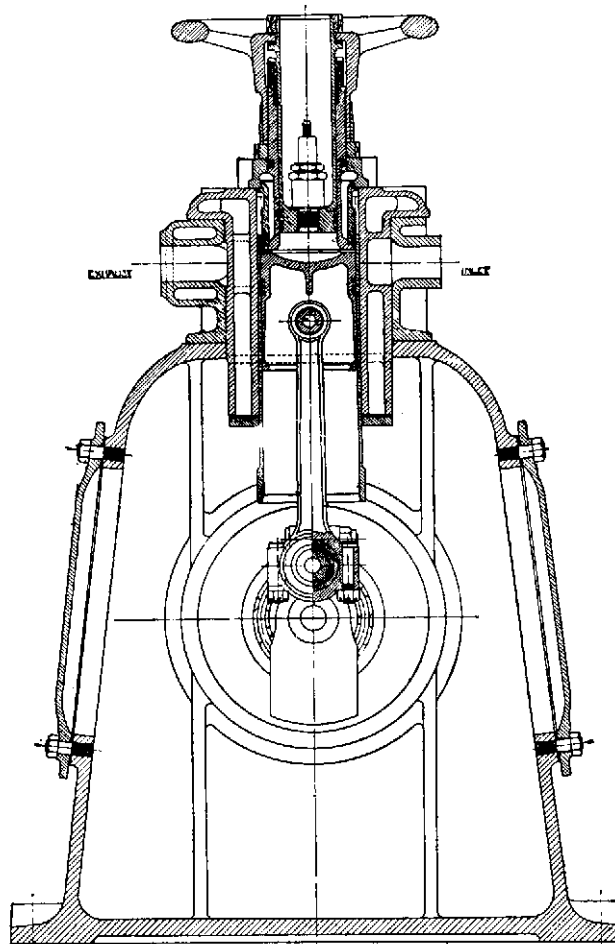


Fig. 10.—Cross Sectional Arrangement of Single-sleeve-valve and Research-unit Type—E.5-1

ignition timing adjusted to give maximum power, the highest useful compression ratio (i.e. the highest useful compression as limited by the incidence of detonation) is found to be 5·3. The ratio corresponds with an air standard efficiency of 48·5 per cent.

The second engine, "E.5", figs. 9, 10, and 11, has a cylinder bore of 2·75 in. and a stroke of 3·25 in., with a single sleeve valve. Ignition is by a single plug placed centrally in the combustion

chamber, which is of slightly conical form, the extreme distance from the ignition point to the furthest point in the combustion chamber being 1.7 in. With the same fuel and under the same conditions as to mixture strength and ignition advance, the highest useful compression, as limited by detonation, is found to be 6.8.

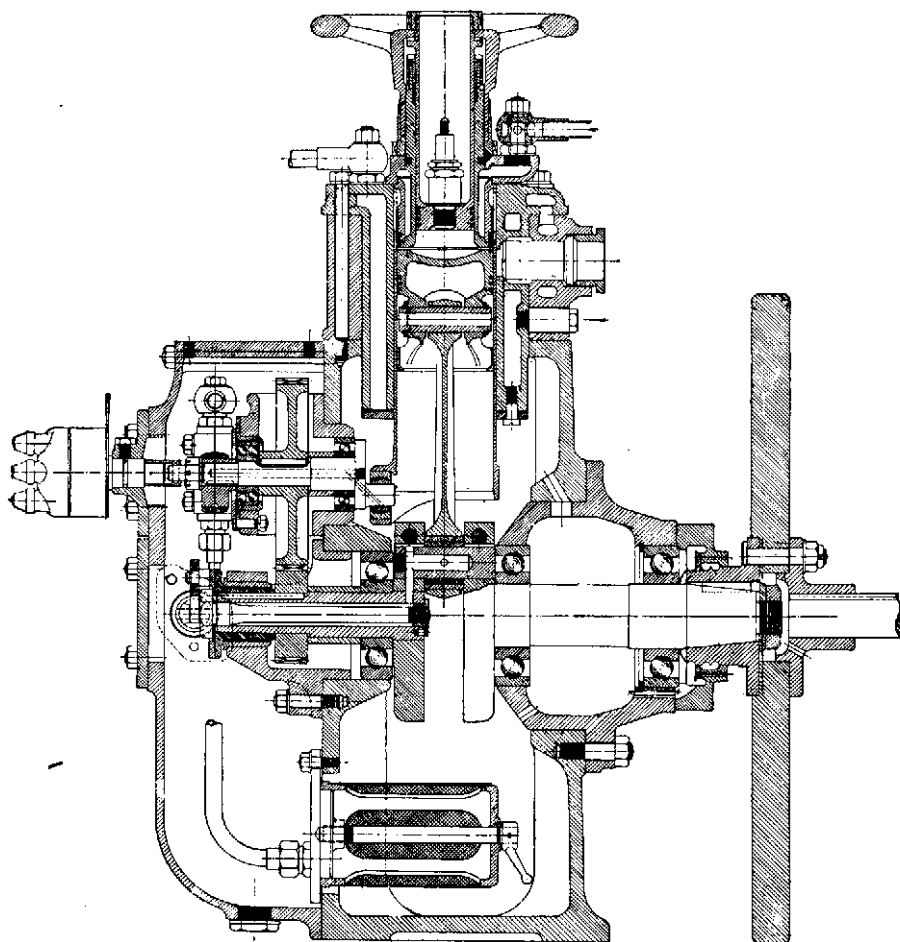


Fig. 11.—Single-sleeve-valve Research-unit Type—E.1-5

This ratio corresponds to an air standard efficiency of 53.5 per cent. Thus, the difference between these two engines due to the combined influence of design factors and scale effects is no less than 1.5 ratios of compression, corresponding with an increase in thermal efficiency of 10 per cent. Here the difference in favour of "E.5" is attributable primarily to:—

(a) The smaller cylinder and therefore reduced length of flame travel.

(b) To the absence of hot exhaust valve heads.

Taking another extreme variation in design, a "T-headed" poppet-valve engine of 4.375-in. bore and 6-in. stroke, in which the sparking-plug was placed directly over the inlet valve and therefore at one side, detonated heavily on the same fuel at a compression ratio of only 3.7.

As showing the effect of cylinder size, apart from differences of general design, may be quoted the results of tests made with four

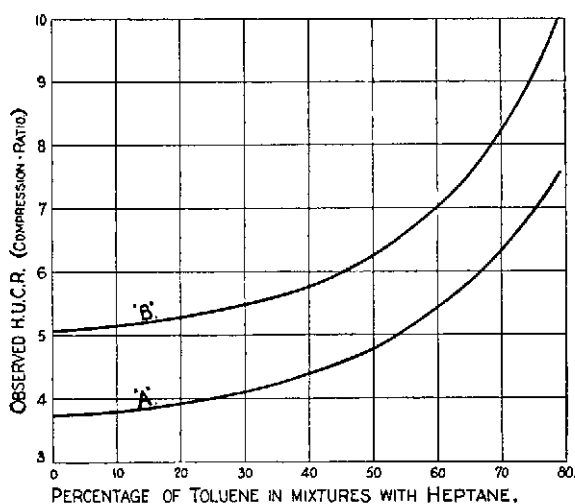


Fig. 12.—Curve A "E.35" Engine  
,, B "E.5" ,,

experimental single-cylinder single-sleeve-valve engines. The bores of these engines are 2.75, 3.5, 5.5, and 8.5 in. respectively. All have very nearly the same stroke/bore ratio and the same form of combustion chamber, namely, a cone with the sparking-plug at the apex. They are, in every respect, as nearly similar as is possible over so large a range of sizes. In each case also the maximum length of

flame path was approximately 25 per cent greater than the cylinder radius.

The highest useful compression ratio as determined by detonation for each of these engines was very carefully observed and may be expressed for the same fuel (20 per cent/80 per cent benzol-petrol blend) as follows:—

Diameter of cylinder (in.)	2.75	3.5	5.5	8.5.
H.U.C. Ratio	7.9	7.5	6.2	5.4.

In each case the engine speed was adjusted to that giving maximum torque, and the temperature conditions, mixture strength, and all other relevant factors were made as nearly as possible the same for each engine.

The observations, when plotted against cylinder size, will be

found to lie on a reasonably smooth curve, except that the 5.5-in. cylinder is relatively rather low in H.U.C.R. This is probably due to some difference in the degree or nature of the turbulence of the charge prior to ignition.

**Example of the Effect of Fuel.**—Despite these wide differences between engines, due, in part, to design factors and, in part, to scale effect, the relative tendency to detonation as between different fuels is nearly the same for the different engines. Fig. 12 shows the H.U.C.R., or highest useful compression ratio for the two engines “E.35” and “E.5” with different fuels consisting of varying mixtures of pure *n*-heptane and pure toluene.

The two curves emphasize that although the engine design factor exercises a great effect on the tendency to detonation, it has little or no influence on the relative H.U.C.R. of different fuels.

**Example of the Effect of Position of the Point of Ignition.**—

As showing the influence of sparking-plug position on detonation, the following very carefully conducted series of tests may be cited as generally representative. The engine used for the purpose of the tests was the “E.35” variable compression engine. This engine is provided with four radial plug positions and special means exist to ensure that any two plugs can be fired simultaneously with perfect synchronism. The position of the plugs and their relation to that of the valves is indicated in diagram fig. 13. Throughout the full range of observations the following conditions were maintained constant:—

Speed 1500 R.P.M.

Mixture strength 15 per cent rich (maximum power setting).

Jacket Temperature 50° C.

Heat input to the induction system 1350 watts.

Fuel, “Texas gasoline”.

Upon analysis these results show clearly that the ignition advance required and the H.U.C.R. are both dependent very greatly upon the length of flame path, from the sparking-plug to the farthest point reached by the flame. The relative positions of the

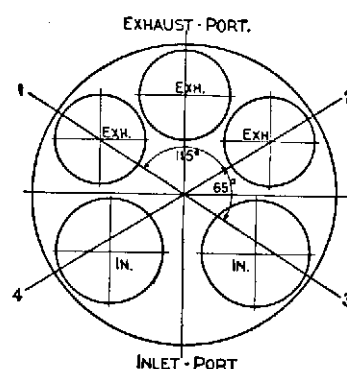


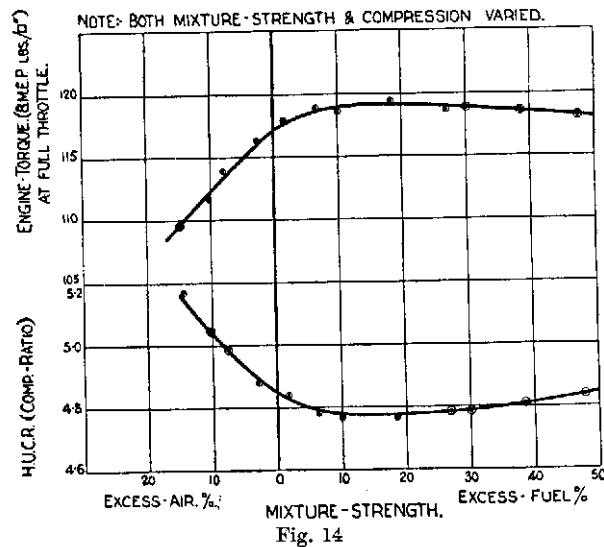
Fig. 13.—Sketch showing Valve and Sparking-plug Positions. Positions 1, 2, 3, and 4.

TABLE I

Plug Position.				Ignition Timing giving Max. Torque.	H.U.C.R.	Brake Mean Effective Press. lb. per sq. in.
1 and 3	...	...	...	30° early	5.3	138.0
2 and 4	...	...	...	30° "	5.25	136.6
3 and 4	...	...	...	34° "	4.98	134.5
1 and 2	...	...	...	32° "	4.9	133.8
1 and 4	...	...	...	32° "	4.8	132.1
2 and 3	...	...	...	32° "	4.84	132.8
1 only	...	...	...	39° "	4.95	133.5
2 "	...	...	...	39° "	4.9	133.1
3 "	...	...	...	40° "	4.84	132.8
4 "	...	...	...	42° "	4.85	132.8

active sparking-plugs and of the various valves are shown to exert a secondary influence as well.

**Examples of the Effects of Mixture Strength and Timing of Ignition.**—The influence of mixture strength on



H.U.C.R. and engine output in the case of the single-cylinder engine in which the issue is not confused by errors in uniformity of distribution is shown graphically in fig. 14. In fig. 15 is shown the influence of mixture strength on the ignition advance required

to give maximum output. In this test, the compression ratio was fixed at a point below that at which detonation could intrude.

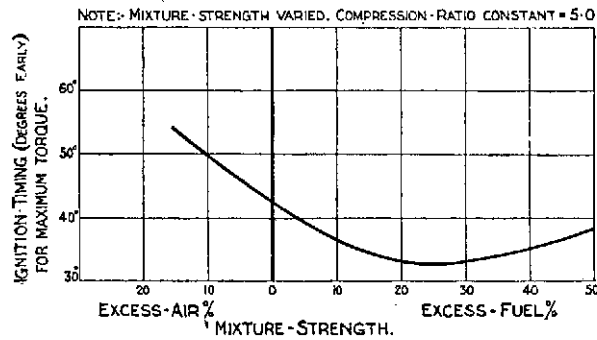


Fig. 15.—“E.35” Engine

In fig. 16 is shown the ignition advance required and the engine output obtained as compression ratio was varied, all other conditions being as nearly constant as possible.

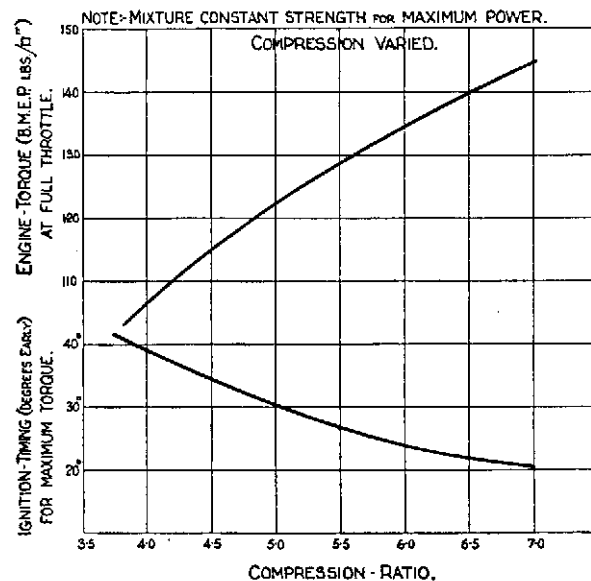


Fig. 16.—“E.35” Engine

The results plotted in these three groups were obtained with “E.35”, the overhead-valve type variable compression engine.

**Metallic “Anti-knocks”.**—Only of recent years has it been known that certain metals have the power, if introduced in a



proper state, of suppressing detonation in an engine in a most remarkable manner. It was first shown by Midgley in 1922 that lead tetraethyl possessed this power. The effectiveness of this "anti-knock" is best illustrated by stating that one molecule of lead tetraethyl in 50,000 molecules of mixture (of fuel and air) may raise the H.U.C.R. by one ratio.

It is not proposed to discuss fully the many fantastic theories that have been proposed to account for the action of metallic "anti-knocks". Callender's admirably reasoned explanation and the theory proposed by Egerton must, however, be referred to.

Briefly, Callender's theory suggests that the active agents in promoting detonation are the unstable peroxides which are formed as intermediate products during the process of combustion of hydrocarbons. The presence of certain readily oxidizable metals in a finely-divided state provides an alternative and more attractive partner for the oxygen and thus prevents the formation of peroxides as products of partial combustion. According to Callender's theory, the progress of events is briefly as follows:—The metal compound, which is volatile and soluble in petrol, is carried into the cylinder as a vapour along with the fuel and air. Under the conditions of temperature and pressure existing within the cylinder during the early stages of compression, the unstable metallic compound dissociates, setting free the metal in a very finely divided and highly active state. It is assumed, then, that at least a portion of the fuel during compression is still in a liquid but finely-divided state, and that each droplet forms a nucleus for oxidation. It is at this stage, during the formation of organic oxides or peroxides that the metal probably intervenes, to reduce these compounds. It is presumed that, owing to the instability of such organic peroxides in very small proportions, their presence increases greatly the speed of subsequent combustion and, therefore, the consequent detonation. The theory would explain the fact that so little metal is needed to suppress detonation. Moreover, this theory will serve to explain other anomalies as between undoped fuels of different physical characteristics.

Still more recently Egerton has proposed that the action of "anti-knocks" is somewhat different. He was led to this conclusion by finding that potassium produced an effect similar to that of lead in suppressing detonation in an engine. Since potassium is so rapidly oxidized, it must be supposed that no uncombined metal would have time to reach the fuel. When, in the author's variable

compression engine, lead vapour was produced by an electric arc and was carried with the air and fuel into the cylinder, it was found to reduce detonation. When thallium was used in the arc the effect was very much greater. Whether in either metal the arc was struck in an atmosphere of air or of nitrogen, the suppression of detonation was equally marked. Since thallium and probably also lead vapour under such conditions would be almost instantly oxidized in air, it could not be doubted that these metals even in the form of oxides could serve to suppress detonation if in a sufficiently

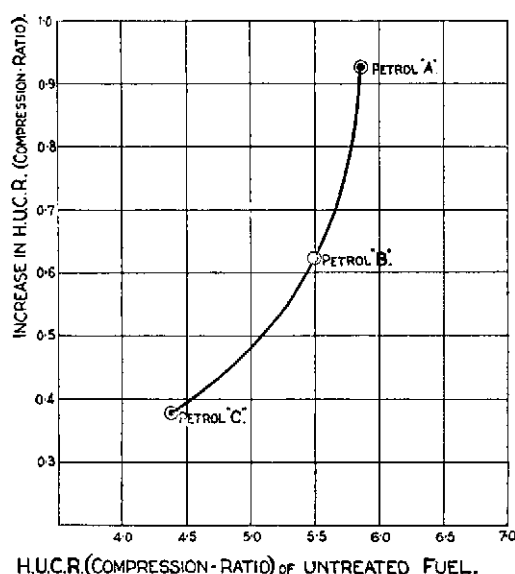


Fig. 17

divided state. It was observed, furthermore, that the metals which are effective as "anti-knocks" are capable of forming two oxides, and that the temperatures of transitions of these oxides were found to be within the range of temperature which occurs during compression in an engine.

In Egerton's conception of the processes of combustion he supposes the existence of a chain of intermediate products. By its interference with this chain of reactions the active metallic oxide is believed to work. To account for the effectiveness of so small a concentration, it is suggested that the higher oxide of the metal is reduced to the lower, thereby oxidizing the intermediate organic product which would otherwise give rise to detonation. On next meeting an oxygen molecule the now lower metallic oxide is

reconverted to the highest form. By such means the molecule is rejuvenated with enormous frequency and is so enabled to work over and over again.

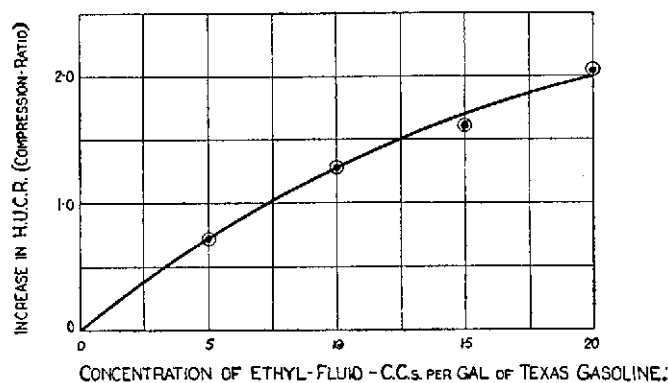


Fig. 18.—"E 35" Engine

It has also been observed that the influence of a given concentration of lead tetraethyl depends upon the chemical composition of the hydrocarbon fuel. In general, the effectiveness in suppress-

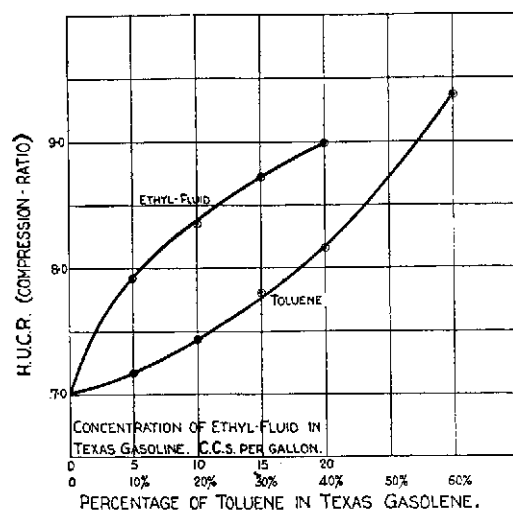


Fig. 19

ing detonation is greater in a fuel of high initial H.U.C.R. than in a fuel which, when untreated, detonates at a lower compression.

**Example of Influence of Fuel on the Effectiveness of an "Anti-knock".**—In fig. 17 is summarized in curve form the

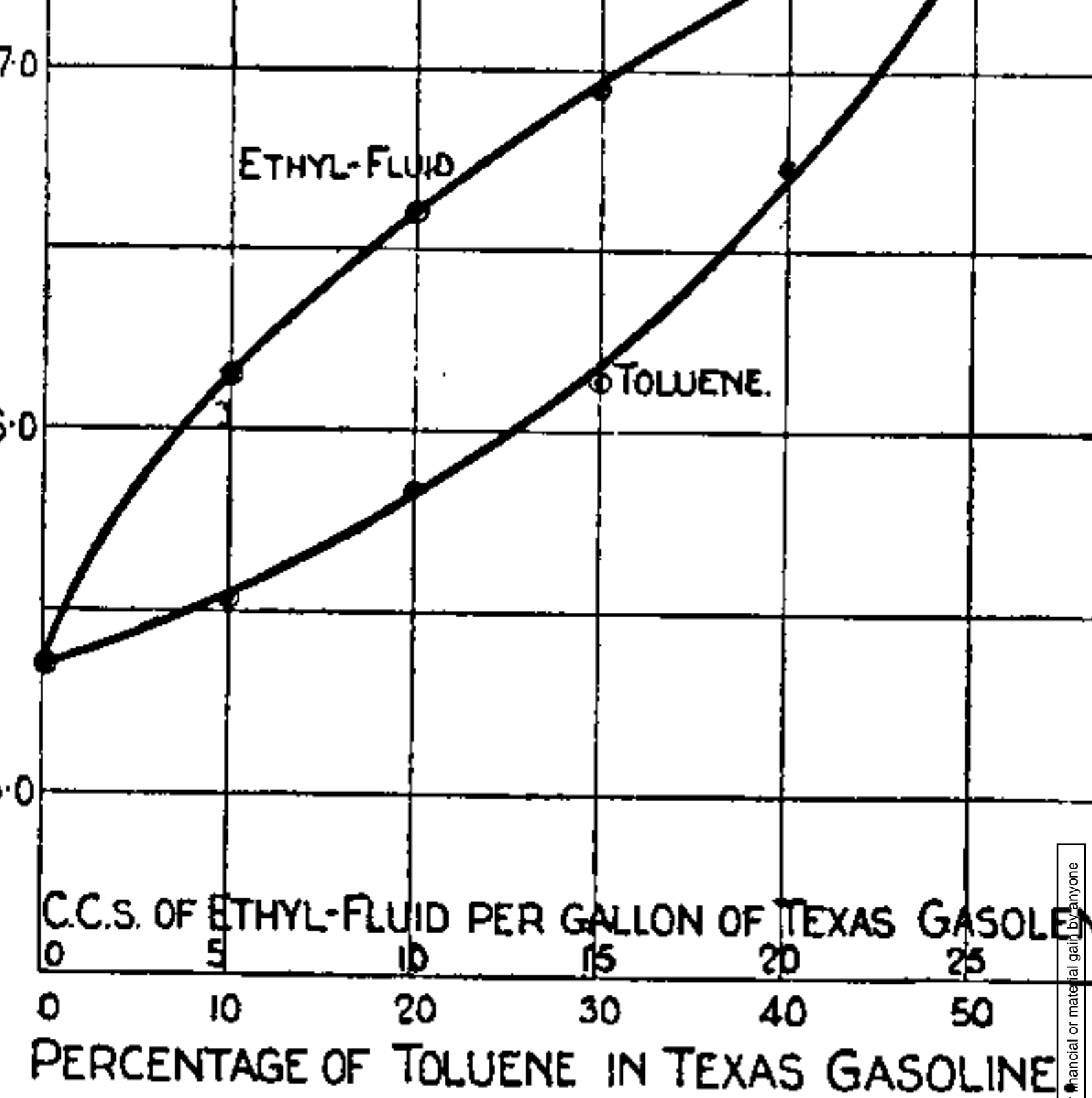


Fig. 20.—“E.35” Engine

that the steepness of the curve is increased. Even up to the point where the curve

Fluid" in terms of the percentage of toluene giving the same increase of H.U.C.R. From this graph, since the two curves are on the same scales, the effects of various concentrations are readily comparable for the two very different types and sizes of engine. It is interesting to observe that the curves are so nearly parallel and that the difference in effectiveness at no point is very great, when it is considered that the variations in design factors are almost the widest possible.

**Conclusions on Usefulness of "Anti-knocks".**—Unfortunately, owing to the residues left behind in the form of lead compounds and the adverse effect of these deposits upon the engine

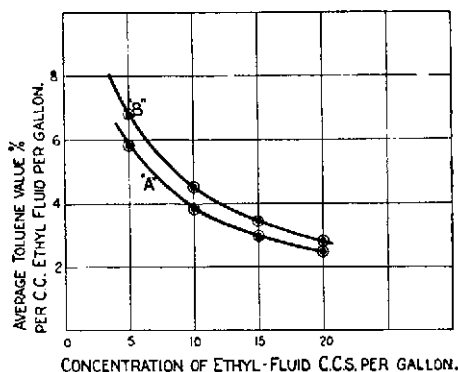


Fig. 21.—Curve "A" "E.35" Engine  
" "B" "E.5" "

generally and particularly upon the insulation of the sparking plugs, only very small concentrations of lead tetraethyl can generally be employed.

For use in road vehicle engines it has been found that up to only 3.5 c. c. of "Ethyl Fluid" per gallon of fuel can be employed without harm. From the graph, fig. 17, it will be seen that when applied to a poor petrol, this concentration will raise the H.U.C.R. approximately 0.3 ratios, when applied to a "good" petrol the increase will be greater, but, on the other hand, the need is correspondingly less.

Summarizing the usefulness of metallic "anti-knocks" so far as their development has proceeded up to the present time, it may be said:—

(a) For road vehicles, owing to the limitations set by the deposits of residue, the advantages both in power and economy are relatively small. Far more scope remains for improved efficiency

by further attention to engine design. Probably almost equal scope is available from improvements in the methods of "cracking".

(b) For engines where a very high duty is required for comparatively short periods such as racing cars or aeroplanes, the use of metallic "anti-knocks" is of considerable value.

**Range of Burning.**—So long as a fuel contains no free hydrogen, the available range of burning differs but little between the different volatile liquid fuels and is, compared with illuminating or other gaseous fuels, very narrow indeed. It is clear that so far as efficiency is concerned, it is only the range of burning on the weak side that need be taken into consideration; the range of burning on the rich side, that is up to the point at which the mixture fails to ignite owing to over-richness, is of comparatively little practical interest. It has already been shown that any range on the weak side is of the utmost importance, for so long as the fuel will burn completely, the weaker the mixture, the lower the flame temperature, and therefore the higher the efficiency. If combustion were complete the flame temperature would diminish very nearly in proportion to the mixture strength. Unfortunately, incomplete and delayed combustion become apparent so soon as the mixture strength is reduced by more than about 15 per cent below that giving complete combustion; from 12 per cent to 18 per cent weak, the loss by delayed and incomplete combustion and the gain due to the lower flame temperature, just about balance each other, while beyond about 18 per cent weak the net efficiency begins to fall away rapidly, and the rate of burning becomes so slow that it continues throughout the exhaust stroke and so ignites the fresh charge on its entry to the cylinder, causing the familiar back-fire into the induction system. This can, however, be obviated to a limited extent by advancing the time of ignition, and the range on the weak side can be extended slightly in consequence, but the angle of ignition advance required, efficiently to burn a mixture only 20 per cent weak, is so excessive as, in practice, to be almost impracticable. In fig. 22 is shown a typical curve showing the relation between thermal efficiency, mean pressure and mixture strength with fixed ignition timing. The mixture strength is plotted in terms of percentage excess of fuel or air. In fig. 23 is shown a similar curve, but with the ignition advanced a further 30° between 10 per cent and 20 per cent weak; in both cases the mean pressure and efficiency were obtained by direct measurement, and represent the mean of a large number of tests on various fuels. The point at

which combustion is complete, that is, at which there is no excess either of fuel or air, is shown by a vertical line, to the right of which

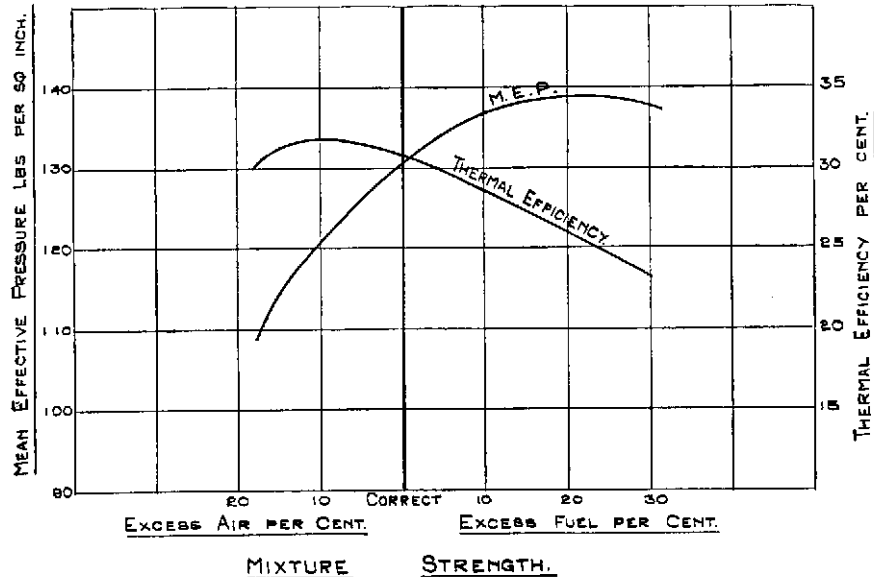


Fig. 22.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with fixed Ignition Timing. Fuel, Petrol

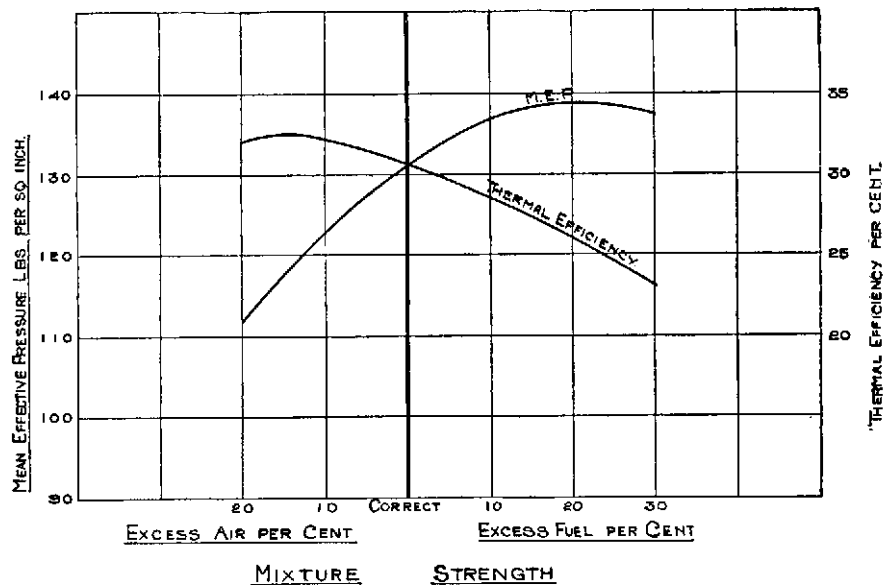


Fig. 23.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Petrol

the mixture is over rich and to the left it is weak. In fig. 24 is shown a similar curve taken with alcohol. It will be observed that while,

in each case, the M.E.P. increases as the mixture is enriched beyond the point of complete combustion, it is only in the case of alcohol that there is any increase beyond 20 per cent excess of fuel. The increase in mean pressure with excess fuel depends upon the inter-relation of a number of factors; on the one hand are the increase in specific volume and the increase in volumetric efficiency due to the latent heat of the fuel, both of which tend to increase the power output with further increase of mixture strength. Against these must be offset the higher specific heat of the products of combustion, when the fuel is only partially burnt. In the

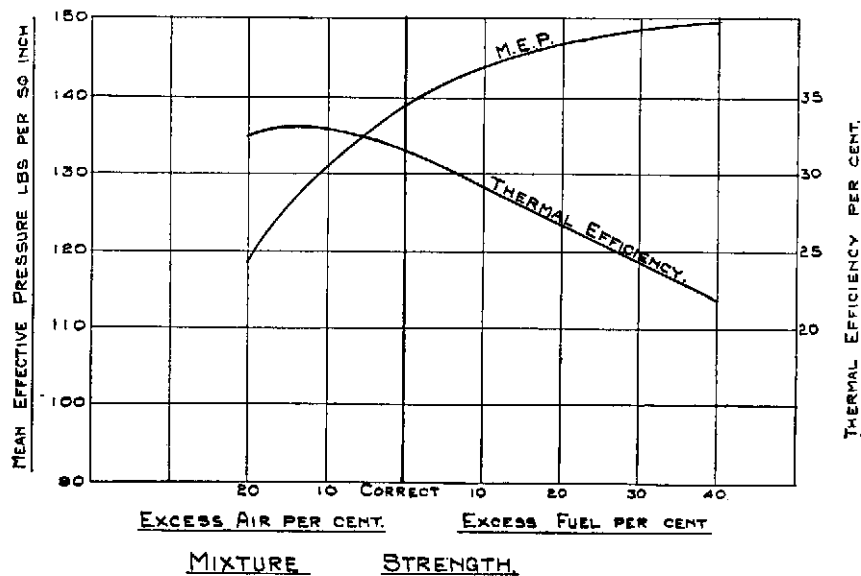


Fig. 24.—Indicated Mean Pressure and Thermal Efficiency at different Mixture Strengths with Ignition Timing adjusted for each Change in Mixture Strength. Fuel, Alcohol

case of petrol and benzol the increase is small, because the latent heat is low and the several factors more or less balance one another. The latent heat of benzol is higher than petrol, and moreover more use can be made of it, because the fuel is homogeneous, but, on the other hand, the change in specific volume is smaller. In the case of alcohol, the latent heat is very much higher and the change in specific volume also is greater, consequently with this fuel the mean pressure increases as the mixture is enriched to a far greater extent than with either of the other two.

The range of burning on the weak side is substantially the same in all three cases and indeed in the case of all known volatile liquid fuels.



It will be observed that the efficiency is at a maximum when the excess of air is between 10 and 18 per cent; with an excess of 20 per cent the process of combustion becomes so slow that the running of the engine is unstable, and both misfiring and back-firing through the inlet valve are liable to occur. The tests on which these results are based were all carried out on single-cylinder engines. In the case of a multi-cylinder engine, however well designed the distribution system may be, it is practically impossible to maintain the mixture strength to all cylinders uniform to within closer limits than about 10 to 20 per cent. If the mean strength supplied by the carburettor is, say, 15 per cent weak, it follows that one or more cylinders will receive mixture as much as 20 to 25 per cent weak which will result in misfiring and back-firing, through the carburettor. To allow therefore for inequalities in distribution, it is necessary, in practice, even in the very best examples to supply a mixture containing not more than about 10 per cent excess of air. With such a mixture the supply to some cylinders will be about that giving complete combustion to others about 20 per cent weak, and the maximum efficiency obtainable will be about 1.5 per cent lower than could be obtained from a single-cylinder engine. It will be seen, therefore, that in order to legislate for the weakest cylinder and prevent back-firing, one or more of the remainder must receive a mixture which is too rich for maximum economy, and the indicated thermal efficiency of a multi-cylinder engine will, on this account, always be lower than that of a single cylinder. Further, the greater the number of cylinders fed from any one source of supply, the lower the efficiency.

To sum up, (1) the available range of mixture strength on the weak side, with all volatile hydrocarbon fuels, is very narrow, far narrower than with most of the gaseous fuels.

(2) Owing to the narrow range of available mixture strength and to the inevitable inequalities in distribution, multi-cylinder engines cannot run with the most economical mixture strength, hence their fuel consumption per H.P. hour must always be slightly higher than that of a single cylinder—how much higher, depends upon the number of cylinders supplied by any one carburettor and, of course, on the efficiency of the distribution system.

(3) With all fuels, slightly more power is developed with an over-rich mixture as compared with the mixture giving complete combustion; with alcohol the increase in power is very marked, and amounts to nearly 10 per cent with very rich mixtures.

(4) In the case of single-cylinder engines running on petrol or benzol, maximum economy is obtained at about 92 to 94 per cent of full power; in the case of multi-cylinder engines, for the reasons given above, the maximum economy on the same fuels is slightly lower, but is obtained at about 96 to 97 per cent of full power.

**The Temperatures of the Cycle.**—From the data and information now available, it is possible to deduce with a fair degree of accuracy, the temperature changes throughout the cycle when liquid hydrocarbon fuels are used.

The first temperature which has to be determined is that at the end of the suction stroke, for from this all the other cycle temperatures can be obtained. It is also of direct importance as determining the volumetric efficiency of the engine.

The temperature at this point is influenced by the final exhaust temperature, owing to the admixture of the fresh charge with the residual exhaust products in the clearance space. It therefore should strictly be evaluated by a "hit-and-miss" type of calculation, by which a residual exhaust temperature is assumed, and the cycle temperatures worked out on that assumption. The assumed exhaust temperature is then modified till it agrees with that which is obtained by calculating round the cycle.

Fortunately, however, it requires a large change in exhaust temperature appreciably to alter the suction-temperature, so that the assumed and calculated temperatures need not check very closely.

It is, in the author's opinion, always preferable to take a concrete example; we will therefore consider the case of a cylinder with a swept volume of 80 cub. in. running at 2000 R.P.M. Let the compression ratio be 5:1, making the total cylinder volume 100 cub. in., and let the following conditions be assumed :—

Mean jacket temperature	...	...	...	...	140° F.
Heat input to charge external to cylinder (by carburettor heating, &c.)	...	...	...	...	0.05 B.Th.U. per cycle.
Absolute pressure in cylinder at end of suction stroke	...	...	...	...	14.0 lb./in.
Absolute pressure in cylinder at end of exhaust stroke	...	...	...	...	14.7 "
Mean temperature of outside air	...	...	...	...	60° F.

These are conditions which may be taken to represent average practice.

The fuel is assumed to be an average high-class petrol of composition say 50 per cent paraffins, 35 per cent naphthenes, and 15

per cent aromatics, while the case of ethyl alcohol and benzene (benzol) are also dealt with.

The relevant properties of these fuels are :

Fuel.	S.G.	Effective Cal. Val. for B.Th.U./lb.	Latent Heat of Evaporation, B.Th.U./lb.	Boiling Point.
Petrol ... ..	0.740	19,000	135	160°-400° F.
Benzene ... ..	0.884	17,460	172	176° F.
Ethyl alcohol ...	0.798	11,840	397	172° F.

Fuel.	Change in Sp. Vol. after Combustion (Correct Mixtures).	Complete Combustion Mixture (by Weight).	Energy liberated per Standard cub. in. of Mixture ft. lb. exclusive of change of Specific Volume.
	per cent.		
Petrol ... ..	+5.0	14.3/1	46.2
Benzene ... ..	+1.3	13.2/1	46.9
Alcohol... ..	+6.5	8.95/1	44.5

Consider first the conditions when running on petrol, starting with the commencement of the suction stroke. The cylinder then contains 20 cub. in. of hot exhaust products at atmospheric pressure. These will be at a temperature of about 2100° F. absolute, as will be shown later. It is desirable in order to arrive at their heat capacity relative to that of the entering charge to reduce them to terms of normal temperature and pressure. The volume of the residual exhaust products under these conditions will therefore be  $20 \times \frac{491}{2100} = 4.68$  cub. in.

The incoming charge consists of air at a temperature of about 60° F. and a small proportion of fuel, entirely or partly vaporized. The air/fuel ratio for complete combustion will, in the case of petrol, be 14.3/1. The latent heat of evaporation of 1 lb. of petrol is 135 B.Th.U.s. This is supplied by 14.3 lb. of air of which the specific heat at constant pressure is 0.237; the drop in temperature of this air will be  $\frac{135}{14.3 \times 0.237} = 40^\circ \text{ F.}$

For the mass of air in question 1° F. change in temperature is produced by .00067 B.Th.U., so that the latent heat of evaporation

will take up  $40 \times 0.00067 = 0.0268$  B.Th.U. From the cylinder walls at a temperature of about  $140^\circ$  F., and from the still hotter valve and piston surfaces, &c., the charge picks up about 0.0005 B.Th.U. per cub. in., or in this case 0.04 B.Th.U. per cycle. The net gain of sensible heat to the charge is  $0.04 + 0.05 - 0.0268 = 0.0632$  B.Th.U. per cycle, which will raise the temperature by about  $95^\circ$  F. to  $155^\circ$  F., assuming that all the fuel is completely evaporated. If the direct heating does not complete evaporation, mixture with the hot residual gases will certainly do so unless the mechanical pulverization of the fuel in the carburettor is very inadequate. It follows therefore that

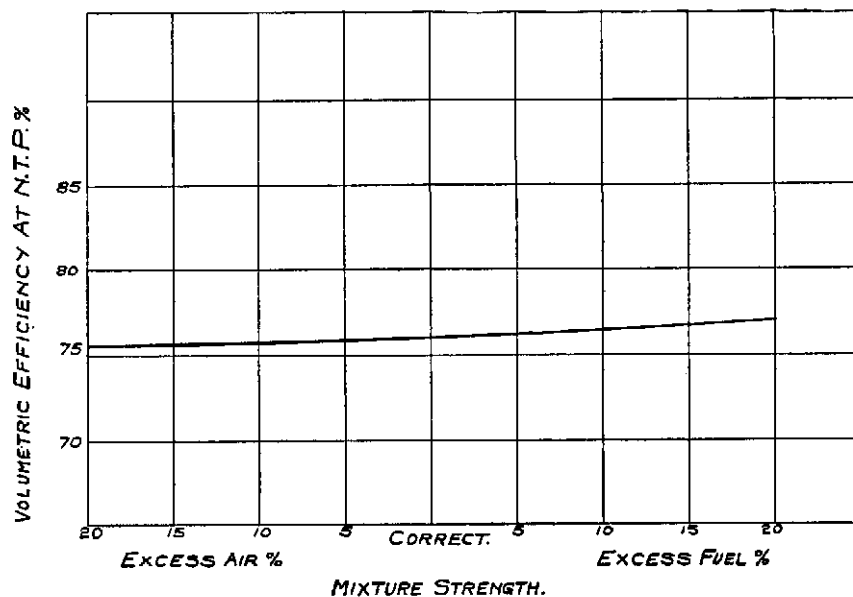


Fig. 25.—Observed Volumetric Efficiency with different Mixture Strengths at a Compression Ratio of 5.0 : 1. Fuel, Petrol

the final suction temperature will be the same at whatever stage the latent heat of evaporation is extracted, provided only that evaporation is complete before compression starts.

We now have 80 cub. in. of fresh charge at  $155^\circ$  F. or  $614^\circ$  F. abs. and 14.0 lb./in. absolute pressure. Reduced to N.T.P. this becomes

$$80 \times \frac{14.0}{14.7} \times \frac{491}{614} = 60.9 \text{ cub. in.}$$

This gives a volumetric efficiency of  $\frac{60.9}{80} = 76.2$  per cent, a figure which agrees very closely with that found in, for example, the author's single-cylinder research engine under similar temperature conditions, as shown in fig. 25, which indicates the observed volumetric efficiency at different mixture

strengths when running with the heat input to the carburettor specified above, and at a piston speed of 2000 ft. per minute.

The volume of the residual exhaust gases at N.T.P. was shown to be 4.68 cub. in., so that the total standard volume of the combined mixture =  $60.9 + 4.68 = 65.58$  cub. in. As this actually fills a volume of 100 cub. in. at a pressure of 14.0 lb./in. its temperature must be

$$\frac{100 \times 14.0}{14.7 \times 65.58} \times 491 = 717^\circ \text{ F. absolute or } 258^\circ \text{ F.}$$

This figure may be taken as the final suction temperature to within about  $\pm 10^\circ \text{ F.}$  in the case considered. The chief possibilities of error lie in

(1) The temperature of the exhaust products. Owing, however, to the fact that as their temperature increases their mass correspondingly decreases, a large error here has but little influence on the suction temperature.

(2) The heat picked up from the cylinder walls, &c. This is certainly rather a doubtful figure, but the author has arrived at it by running an engine with varying jacket temperatures, and adding known quantities of heat by means of an electrical resistance in the induction pipe until the volumetric efficiency, and therefore the suction temperature, was constant. By measuring the difference in external heat necessary to do this with varying jacket temperatures, it was possible to estimate the total amount of heat taken up from this source.

(3) In most cases the amount of pre-heating of the charge before its entry to the cylinder is very uncertain, since in practice the heat is generally supplied either from the exhaust or from the circulating water. For experimental work the author prefers to supply this heat electrically so that it can readily and accurately be measured.

If benzene be used instead of petrol, we find that, owing both to its higher latent heat and to the greater proportion of fuel needed to combine with the air, the final suction temperature is lower, namely, about  $235^\circ \text{ F.}$ , while the volumetric efficiency is correspondingly higher, about 78.5 per cent. The residual exhaust temperature may be taken, as with petrol, as  $2100^\circ \text{ F. absolute.}$

In the third case, that of ethyl alcohol, the extremely high latent heat and the large proportion of fuel in the mixture produce much greater cooling effect, with the result that, arguing on the above premises, the final suction temperature, even after admixture with the residual exhaust products, will only be  $67^\circ \text{ F.}$ ; the corresponding volumetric efficiency should therefore be 104.3 per cent.

At such a temperature, however, and in the short time available, the alcohol would not be evaporated completely by the end of the suction stroke, so that such a calculation is not applicable in this instance. Experiment has shown that the minimum temperature at which evaporation is completed before compression, is in the neighbourhood of  $150^{\circ}\text{F.}$ , and it will be well to consider the case in which sufficient pre-heating is supplied to effect this. In this case the volumetric efficiency works out at 90 per cent, and 37 per cent of the fuel remains to be evaporated by means of additional external heating to the extent of  $\cdot 37 \times 0\cdot 143 = 0\cdot 053$  B.Th.U. per cycle.

In practice it is found that the volumetric efficiency on alcohol is considerably lower than 90 per cent, namely, 82-83 per cent as compared with 76 per cent in the case of petrol. This shows that under normal working conditions only a comparatively small proportion of the alcohol is actually evaporated by the end of the suction stroke. It is mainly for this reason that efficient pulverization, which accelerates the transfer of heat from the air to the fuel particles, is so particularly desirable when using alcohol.

The above conclusions may be summarized as follows. Under the conditions quoted above—and they may be taken as average working conditions, except perhaps in the case of alcohol—the final suction temperature and the volumetric efficiency will be :

Fuel.	Suction Temperature.	Volumetric Efficiency.
	$^{\circ}\text{F.}$	per cent.
Petrol ... ..	258	76.2
Benzene... ..	235	78.5
Alcohol ... ..	150	90.0

With a 20 per cent weak mixture the figures become :

Fuel.	Suction Temperature.	Volumetric Efficiency.
Petrol ... ..	263	75.5
Benzene... ..	242	77.6
Alcohol ... ..	150	89.6

In the case of alcohol extra heat to the amount of  $0\cdot 033$  B.Th.U.

per cycle has to be supplied, and 0.029 B.Th.U. with the weak mixture, if evaporation is to be complete.

It should be noted that the volumetric efficiency refers to the volume of air and fuel vapour, not air alone.

**Compression Temperature.**—During the compression stroke the fuel and air are compressed nearly adiabatically to one-fifth of their original volume. At the commencement of compression the mixture, in the case of petrol or benzene, is slightly below the mean surface temperature of the cylinder, so at first they will absorb heat, but will lose heat later in the stroke. For these fuels the mean value of the exponent during compression for such an engine running at 2000 R.P.M. may be taken as 1.35. With alcohol the conditions are not quite the same, for

(1) The final suction temperature is substantially below the mean wall temperature, so that the mixture is appreciably heated during the earlier part of the stroke.

(2) As the alcohol forms an appreciable fraction of the charge (11 per cent by weight), it will decrease the mean exponent  $\gamma$  for the mixture, since  $\gamma$  for alcohol vapour is only 1.13 as against 1.4 for air.

(3) A considerable portion of the fuel is liquid at the end of the suction stroke, unless extra induction heat is supplied, and some of the heat of compression is applied to its evaporation.

We will ignore for the moment the last factor, and take  $\gamma$  for a "correct" alcohol mixture as being about 1.33.

The absolute temperature at the conclusion of the compression stroke in the case of petrol or benzene may be arrived at by multiplying the final absolute suction temperature by  $5^{(1.35-1)} = 1.755$ ; This gives :

Petrol	...	...	correct mixture	1258° F. abs.
			20 per cent weak mixture	1267° F. abs.
Benzol	...	...	correct mixture	1220° F. abs.
			20 per cent weak mixture	1232° F. abs.

The compression pressure in each case will be  $14.0 \times 5^{(1.35)} = 123$  lb. per square inch absolute or 108.3 lb. per square inch gauge.

With the alcohol mixtures, the final suction temperatures are the same both for the normal and the weak mixtures, so that the compression temperature, uncorrected for the latent heat of the surplus liquid, are also the same, being

$$609 \times 5^{(1.33-1)} = 1035^\circ \text{ F. abs.}$$

With the correct mixture there is, as previously shown, a heat deficit of 0.053 B.Th.U. per cycle, this amount being necessary for the evaporation of the alcohol which remains in a liquid state at the end of the suction stroke. This causes a drop of  $0.053 \times 1270 = 67^\circ \text{ F.}$ , making the actual compression temperature  $968^\circ \text{ F. abs.}$  or  $475^\circ \text{ F.}$

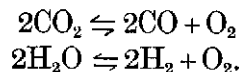
With the weak mixture, the heat deficit is only 0.029 B.Th.U. per cycle, which raises the compression temperature to  $998^\circ \text{ F. abs.}$  or  $505^\circ \text{ F.}$

The uncorrected compression pressure is  $14.0 \times 5^{(1.33)} = 119 \text{ lb.}$  per square inch absolute. The cooling due to evaporation reduces this to 111 lb. per square inch abs. for the normal mixture, and 114.7 lb. per square inch abs. for the weak mixture.

**Combustion Temperatures.**—Since combustion takes place at constant volume, the whole of the chemical energy stored in the mixture, with the exception of that lost to the walls of the combustion chamber, will be devoted to increasing its internal energy. It is well known, however, that the obvious method of finding the temperature rise, that of dividing the heat output by the specific heat at normal temperatures, gives results that are far too high, in fact practically double those that are actually attained. This discrepancy is due to the three following influences :—

(1) The specific heat of the gases, of which the working fluid is composed, increases with increase of temperature.

(2) At high temperatures the products of combustion,  $\text{CO}_2$  and water, dissociate with absorption of heat into carbon monoxide and oxygen, and hydrogen and oxygen respectively, according to the formulæ



(3) Heat is lost by radiation and conduction to the walls of the combustion chamber.

The specific heat of the gases, of which the working fluid is composed, and the amount of dissociation at any temperature, have been found by direct experiment, while the loss to the jackets can only be estimated from actual engine experiments. The amount of energy taken up by the gases as sensible heat can readily be calculated, since the specific heat of a gas at constant volume is unaffected by changes of pressure or by admixture with other gases. Unfortunately the degree of dissociation is influenced by alterations in pressure and, in cases such as the present, where two kinds of



dissociation have a common product (oxygen in this case), by the relative proportions of the gases present. This renders the actual calculation of the energy absorbed in this manner a somewhat complex and extremely wearisome process; it has, however, been dealt with very fully by Tizard and Pye, so that the results alone will be given. The most convenient method is to construct a curve showing the total internal energy of the mixture, at any temperature, plotted against the temperature. Since the amount of internal energy at any stage in the cycle can easily be found, the corresponding temperatures can at once be read off.

From the work of Tizard and Pye<sup>1</sup> which is based on the experimental data of Pier and Bjerrum<sup>2</sup> the curve shown in fig. 26 has been constructed, which also gives a graphical construction developed by one of the author's assistants, Mr. J. F. Alcock, whereby the temperatures at the beginning and end of the expansion stroke can be read off with a fair degree of accuracy. Owing to the influence, previously mentioned, of pressure and charge proportions on dissociation the construction of such a curve to meet a reasonably wide range of conditions renders it necessary to compromise to some extent and to take mean values.

Thus, the energy curve shown applies strictly only to a benzene-air complete-combustion mixture at a compression ratio of 5 to 1. However, the effect of a change in the compression ratio is almost negligible within the limits used on the constant-volume cycle; as is the effect, also, of replacing benzene by any other hydrocarbon fuel. If used for a fuel such as alcohol or ether, however, the error becomes appreciable, owing to the different specific heat of the products of combustion. Strictly speaking, it does not apply to mixtures which are either weaker or richer than the chemically correct mixture, although here again the divergence is very small within the range available and with a homogeneous mixture.

The internal energy curve is plotted in terms of foot-pounds per standard cubic inch on a vertical scale, against the temperature on a horizontal scale. The other full-line curve shows the energy present as heat, so that the difference between the two curves shows the chemical energy stored in the products of dissociation. Zero energy is taken at 100° C. (212° F.), as being an average temperature at the beginning of compression. Variations in this temperature, being relatively small, will have but

<sup>1</sup> *Automobile Engineer*, February and March 1921.

<sup>2</sup> *Zeitschrift für physikalische Chemie*, 1912.

little influence on the combustion and exhaust temperatures. The explanation given below of the use of the diagram is supplemented

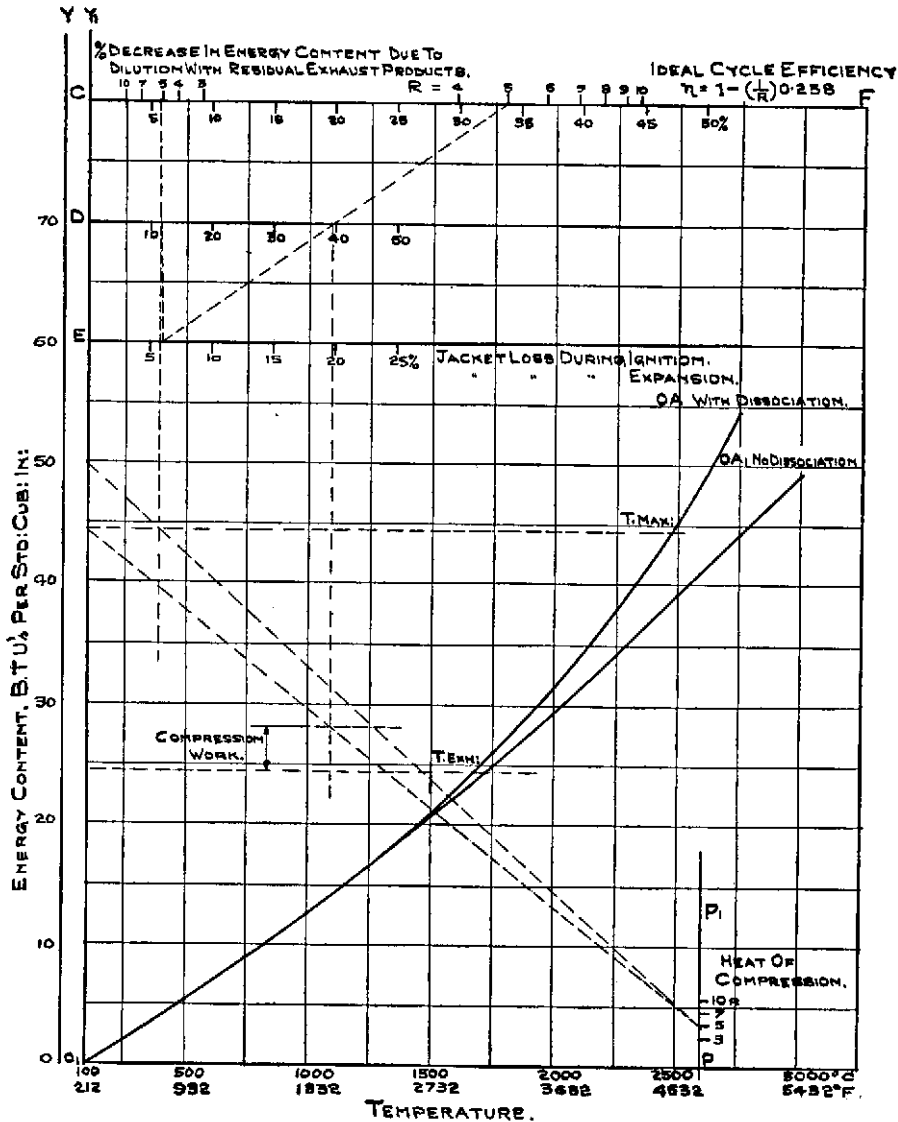


Fig. 26.—Internal Energy Diagram for working fluid of Internal-Combustion Engine running on Volatile Hydrocarbon Fuels

by an example worked out for the following data, the construction lines of the example being shown dotted in the figure.

- |                             |                             |
|-----------------------------|-----------------------------|
| Compression ratio, R        | = 5 to 1                    |
| Energy content              | = 46.2 ft. lb. per cub. in. |
| Heat loss during combustion | = 6 per cent                |
| Heat loss during expansion  | = 6 per cent                |
- (E 246)

The actual value of these losses will be considered in Chap. III.

There are three factors in an actual engine which modify the temperature attained by the combustion of a mixture of any given energy content. They are the

- (1) Heat put into the mixture by compression.
- (2) Loss due to cooling by the walls of the combustion chamber during combustion.
- (3) Effective weakening of the mixture due to dilution with the residual exhaust products.

Factor (1) is allowed for by laying off the heats of compression for various ratios by the marks " $R=5$ ," &c., on the line  $PP_1$  near the bottom of the diagram. The energy content is then marked off above this on the vertical line  $O_1Y_1$  representing the  $100^\circ\text{C}$ . ( $212^\circ\text{F}$ .) starting-point. In the example, the 46.2 ft. lb. energy content is laid off above the 3.6 ft. lb. of compression, making a total of 49.8 ft. lb., this being the gross energy content from which the losses due to factors (2) and (3) must be deducted. These are provided for in the following manner:—

On the horizontal scale C is marked the effective energy loss due to dilution with residual exhaust, assumed to be at  $2100^\circ\text{F}$ . abs. ( $=1165^\circ\text{C}$ . abs.). Scale E shows the percentage loss due to cooling during combustion. This is laid off at any figure which previous experience shows as probable for the type of combustion chamber in question, namely, 6 per cent in the example. A line is then drawn between these two points, and the point of intersection of the line so drawn with the scale D gives the total percentage loss due to these two causes, corresponding to 11.5 per cent in the example chosen.

To transfer this to the diagram, a line is dropped vertically from the above intersection point. Another line is drawn from the point on the line  $O_1Y_1$ , giving the gross British thermal units per cubic inch, to the suitable compression point on the line  $PP_1$ , and representing 100 per cent on scale D; or 49.8 ft. lb. per cub. in. in the example. From the intersection of the above two lines, a horizontal line is run to the energy scale on one side, and to the energy curve on the other. The point on the energy scale shows the net energy available for expansion, or 44.5 ft. lb. per cub. in. in the example. From the energy-temperature curve the actual flame temperature can be read off, this is  $2475^\circ\text{C}$ . ( $4487^\circ\text{F}$ .) in the example.

The drop in temperature during the expansion stroke depends on the two factors of (a) external work done, and (b) heat loss to the

walls. The net power output is given as a percentage of the heat content of the mixture on scale F, the formula used being  $n = 1 - \left(\frac{1}{r}\right)^{0.258}$ . 0.258 is the mean value for the exponent, and covers all dissociation and similar effects, but not wall losses. The wall loss during expansion is laid off on scale E. A line drawn between these points gives their sum on scale D as before. A perpendicular from this point is dropped to meet a line from the net-energy point on  $O_1Y_1$  to the suitable-compression point on the line  $PP_1$ . As the gross work done during expansion is the sum of the net work mentioned above and the compression work, this latter amount, which is 3.6 ft. lb. in the example, must be laid off below the above intersection point, to find the energy content at the end of expansion, which is 24.5 ft. lb. in the example. The corresponding final temperature, 1675° C. (3047° F.) in the example, can then be read off from the energy curve.

Taking an actual example from experiments on the author's variable-compression engine, with a correct mixture of an energy content of 46.2 ft. lb. per cub. in. and a compression ratio of 5 to 1, the actual maximum flame temperature, as obtained from the above diagram in fig. 17, allowing for the additional heat of compression, the wall loss during combustion and the dilution by residual exhaust products, will be 2475° C. (4487° F.), corresponding to an energy content of 44.5 ft. lb. per standard cub. in. At a ratio of 5 to 1, the observed indicated thermal efficiency is 31 per cent; of this, 5 per cent is due to the change in specific volume of the mixture, so that the heat drop is  $46.2 \times 31 \times 100/105 = 13.6$  ft. lb. per cub. in. Add to this the 3.6 ft. lb. of compression work restored during expansion, and the 6 per cent of 46.2 or 2.8 ft. lb. of wall loss during expansion; the total heat-drop during expansion then becomes 20.0 ft. lb. per cub. in., leaving a final energy content of 24.5 ft. lb. per cub. in., which, it will be observed, coincides with the figure found in the example under the same conditions. The corresponding final temperature is 1675° C. (3047° F.)

While affecting the final temperature directly, it should be observed that the loss of heat during expansion has only a slight influence on the actual efficiency; this has been ignored in the construction, because much of it is lost late in the expansion stroke, where its value is less. Another slight error allowed to remain in the construction, for the sake of simplicity, is that a percentage of the *net* heat available during combustion is deducted for the jacket

loss during expansion ; whereas this is given as a proportion of the *total* heat available in the fuel. The error due to this cause is, however, very small, being in the case considered  $2.8 [(46.2 - 44.5) \div 44.5] = 0.11$  ft. lb. per cub. in., and can be ignored safely.

At the end of the expansion stroke the temperature of the gases is about  $3050^{\circ}$  F. or  $3509^{\circ}$  F. absolute, as has been seen, and they are at an absolute pressure of about 70 lb. per square inch. When the exhaust valve opens these gases will expand rapidly down to atmospheric pressure, and their temperature will fall in the ratio  $\left(\frac{14.7}{70}\right)^{\frac{\gamma-1}{\gamma}}$ , where  $\gamma$  is approximately 1.30. This brings their temperature down to  $2450^{\circ}$  F. abs., and the loss of heat during the exhaust stroke further reduces their temperature to about  $2100^{\circ}$  F. abs.

A similar calculation in the case of an ethyl-alcohol mixture makes the residual temperature about  $1950^{\circ}$  F. abs.

When the temperature at any point in the cycle has been found, the corresponding pressure is, of course, easily obtained. Thus, in the case of the petrol given above, the compression pressure is 123 lb. per square inch abs. and the temperature  $1258^{\circ}$  F. abs. As the temperature after ignition is  $4487^{\circ}$  F., or  $4946^{\circ}$  F. abs., and the ratio of the specific volumes before and after combustion 1.05, the pressure at the commencement of expansion is therefore  $123 \times 1.05 \times \frac{4946}{1258} = 508.5$  lb. per square inch abs. or 493.8 lb. per square inch gauge.

At the end of the expansion stroke the volume has increased five times, and the temperature has dropped to  $3509^{\circ}$  F. abs. The pressure is therefore  $\frac{123}{5} \times 1.05 \times \frac{3509}{1258} = 72.2$  lb. per square inch abs., or 57.5 lb. per square inch gauge.

The "explosion" pressure thus calculated is somewhat higher than that actually found in practice due to the finite speed of burning of the mixture, which causes a slight rounding of the peak of the diagram, though not enough appreciably to affect the thermal efficiency.

With benzene, the temperatures at the beginning and end of expansion will be  $2470^{\circ}$  C. ( $4480^{\circ}$  F.) and  $1670^{\circ}$  C. ( $3040^{\circ}$  F.) respectively, the corresponding pressures being 505 lb. per square inch abs. (490.3 lb. gauge) and 71.8 lb. per square inch abs. (57.1 lb. gauge) respectively.

From the above considerations we are able to construct an

indicator diagram giving the pressures and temperatures throughout the cycle under normal working conditions, such as would apply in the case of a small but efficient cylinder of the size quoted, and when running at a speed of 2000 R.P.M. The influence of cylinder size on the performance will be considered later in Chapter III, when it will be seen that it does not play any very important part; neither, indeed, within reasonable limits does the actual speed of revolution, provided always that it be fairly high, i.e. 1000 R.P.M. or over. At speeds below about 1000 R.P.M. and with a cylinder of the size under consideration, the loss of heat to the cylinder walls will begin to make itself felt, and will influence the performance appreciably.

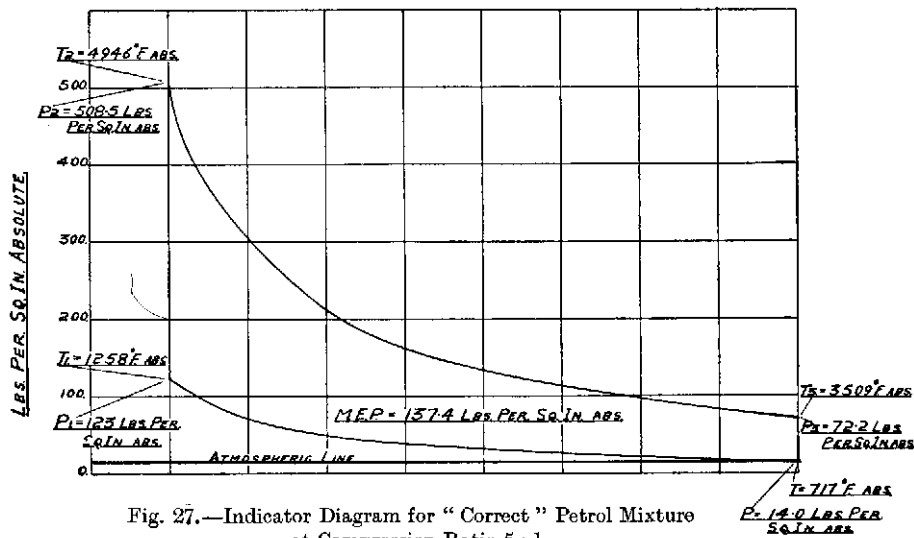


Fig. 27.—Indicator Diagram for "Correct" Petrol Mixture at Compression Ratio 5 : 1.

The indicator diagram shown in fig. 27 applies therefore strictly only to the conditions quoted above, though it would differ but little over a wide range of size or speed provided always that the combustion chamber were of an efficient form and that there is ample turbulence to spread combustion rapidly throughout the whole mass of the working fluid.

Once the thermal efficiency, the volumetric efficiency, and the total internal energy of the fuel-air mixture have been determined, it is possible to arrive also at the mean effective pressure and therefore the power output, by multiplying together the thermal efficiency, the total internal energy (in terms of ft. lb. per cubic inch  $\times 12$ ), and the volumetric efficiency.

In the case under consideration the thermal efficiency with a "correct" mixture, i.e. a mixture giving complete combustion without excess either of fuel or air, is 31 per cent. With a 20 per cent weak mixture it should be considerably higher on account of the lower maximum flame temperature, but this, in practice, is offset by the very much slower rate of burning, with the result that it will be increased only to about 32.5 per cent.

The energy liberated by the combustion of 1 standard cubic inch of petrol-air mixture was given as 46.2 ft. lb., but in so far as the available energy from the point of view of power output is concerned, we must multiply this by the change in specific volume, since, if the products of combustion occupy at the same temperature a greater volume than before, the pressures will be higher, though the temperatures remain unaltered. This factor has been taken into account in determining the maximum and terminal pressures, and it, of course, applies also to the mean effective pressure, which will be increased or decreased according as the specific volume increases or decreases; here petrol scores a marked advantage over benzene (benzol), and alcohol over either. It is convenient to use the term "total internal energy of the mixture" (as distinct from the heat energy liberated by combustion) to include the change in specific volume, i.e. the heat energy liberated, multiplied by the specific volume for a "correct" mixture. The total internal energy for the three fuels under consideration then becomes:

Fuel.	Change in Specific Volume.	Heat Energy liberated per Standard cub. in. ft. lb.	Total Internal Energy per Standard cub. in. ft. lb.
	per cent.		
Petrol ... ..	+5.0	46.2	48.5
Benzene ... ..	+1.3	46.9	47.5
Alcohol ... ..	+6.5	44.5	47.4

With petrol and a correct mixture in the example chosen, the thermal efficiency is 31 per cent, the internal energy 48.5 ft. lb. per standard cubic inch, and the volumetric efficiency 76.2 per cent. The indicated mean effective pressure will therefore be

$$.31 \times (48.5 \times 12) \times .762 = 137.4 \text{ lb. per square inch.}$$

In the case of benzene the same calculation will give

$$.31 \times (47.5 \times 12) \times .785 = 137.8 \text{ lb. per square inch,}$$

or, to all intents and purposes, exactly the same as with petrol, since the lower internal energy balances almost exactly the higher volumetric efficiency.

With alcohol, the same calculation will show that the indicated mean pressure should be 158.3 lb. per square inch, but this is based on a volumetric efficiency of 90 per cent, an efficiency which is not, in practice, realized for reasons already given. If the volumetric efficiency be taken as 82.5 per cent, which corresponds with experimental results under similar conditions, the indicated mean pressure becomes  $158.3 \times \frac{.825}{.900} = 145$  lb. per square inch, which is in very close agreement with observed results.

Again, for a 20 per cent weak mixture we must reduce the total internal energy accordingly, but, on the other hand, the thermal efficiency is increased owing to the lower flame temperature; also the volumetric efficiency is reduced very slightly owing to the smaller influence of the latent heat of evaporation for such a mixture. The figures then become :

Fuel.	Indicated Mean Pressure, lb. per sq. in.		Observed I.M.E.P. (Correct Mixture), lb. per sq. in.
	Correct Mixture.	20 per cent Weak.	
Petrol ... ..	137.4	118.0	136.0
Benzene ... ..	137.8	118.2	136.0
Alcohol ... ..	145.0	124.0	143.6

If, now, the mixture be further enriched by the addition of 20 per cent excess of fuel, although the heat liberated per standard cubic inch of mixture will be no greater, the total internal energy will be increased very slightly owing to the greater increase in specific volume when excess of fuel is present; again, owing to the greater latent heat available the volumetric efficiency will be increased slightly, while, finally, the thermal efficiency, based on that portion of the fuel which is burnt completely, will be slightly higher on account of the lower flame temperature; the net result of all these changes will be that the indicated mean pressure will, in all cases, except that of benzene, be some 4.5 per cent greater than with a correct mixture. In the case of benzene, however, owing to the small change in specific volume, the gain in power by using a



20 per cent rich mixture will be only about 3 per cent. If the mixture be enriched beyond 20 per cent excess of fuel, there will be no further gain in power on petrol or benzene owing to the slower rate of burning, but, on alcohol, there will still be a small further gain on account of its high latent heat and large increase of specific volume.

The indicated horse-power obtainable from the cylinder under consideration will be as follows :—

Fuel.	I.H.P.		
	20 per cent Weak.	Correct.	20 per cent Rich.
Petrol ... ..	15·75	18·35	19·15
Benzene ... ..	15·8	18·4	19·0
Alcohol... ..	16·55	19·35	20·2

## CHAPTER III

### DISTRIBUTION OF HEAT IN A HIGH-SPEED FOUR-CYCLE ENGINE

It is usual to express the distribution of heat in an Internal-Combustion engine in terms of the proportion converted into indicated horse-power, the proportion yielded up to the cylinder walls, and, lastly, that rejected to the exhaust; the latter item being the residue after deducting from the total heat of the fuel the two former categories; it generally includes also the losses by radiation. This form of subdivision is perfectly legitimate so long as it is clearly understood that it is no more than a convenient form in which both to measure and to express the heat distribution; indeed, it is practically the only form in which it can be directly measured. Many engineers, however, still appear to be under the impression that it represents the true disposal of the available heat of the fuel, and, acting on this supposition, are often sadly misled.

The proportion of the total heat of the fuel converted to indicated horse-power can be determined readily enough and quite accurately from the known heat supplied and the known horse-power developed from it.

The heat yielded up to the cylinder walls and carried away by the cooling water can also be determined fairly accurately; it must be understood, however, that it includes:

- (1) The heat given up by radiation, conduction, convection, &c., during the period of combustion.
- (2) The heat given up during the expansion period.
- (3) The heat given up during the exhaust stroke.

It is necessary to examine each of these sources separately.

(1) **Heat lost during Combustion.**—The period of combustion as distinct from expansion is relatively a short one, but it is one during which the ruling temperature in the combustion chamber is very high indeed, i.e. between 4200° F. and 4500° F. in the case of most volatile liquid fuels such as petrol, benzol, &c. Also it is

a period during which the gases within the combustion chamber are in a state of violent commotion so that heat is conveyed very readily by convection, &c.

Now, if by any means the loss of heat to the cylinder walls during this period could be suppressed, such heat could be converted into indicated horse-power at an efficiency corresponding to the efficiency of expansion alone (i.e. exclusive of the negative work done during compression), which in an engine with a compression ratio of 5 : 1 is roughly about 40 per cent. The remaining 60 per cent of the heat so recovered would, in any event, be rejected to the exhaust after expansion.

**(2) Heat lost during Expansion.**—Loss of heat during the expansion stroke may or may not be serious, depending upon the stage in the expansion stroke at which it is lost. Loss of heat at the very commencement of the expansion stroke is almost equally as serious as that lost during the combustion period, because had its loss been suppressed it would have been utilized at an efficiency corresponding to nearly the full ratio of expansion, whereas heat lost during the latter part of the expansion stroke is of very little moment, for even had its loss been suppressed it could have done but little useful work during the remainder of the expansion stroke and nearly the whole of it would have been rejected to the exhaust in any case.

At first sight it would appear that, owing to the higher temperatures and pressures ruling at the beginning of the expansion stroke, the loss of heat will be much greater during the earlier period, but against this it must be remembered that, as the expansion proceeds and the piston descends, an increasing area of cold cylinder barrel is exposed. Also, owing to dissociation and subsequent recombination, the fall in temperature during the expansion stroke is nothing like so great as it might appear; the final temperature, even with a compression ratio of 5 : 1, being still well over 3000° F.

From the above considerations it will be seen that, though it is customary to yoke together the heat lost during combustion and expansion as though its influence during each period were the same, it is most certainly inaccurate and misleading to do so. Of the average heat loss during expansion, probably only about 20 per cent could be converted into useful work and the remaining 80 per cent would be rejected to the exhaust.

**(3) Heat lost during the Exhaust Stroke.**—Although during the exhaust stroke the temperature of the gases is very much lower, yet heat is given up to the cooling water with great rapidity during

this period, for in addition to the normal heat flow to the cylinder walls the hot gases are issuing at an exceedingly high velocity past the exhaust valve and through a short length of exhaust pipe which is always included in the cylinder jacket and cooled by the circulating water, consequently of the total heat carried away by the cooling water at least one-half and often more than half is given up during the exhaust period. Now the whole of the heat taken up during the exhaust stroke, by far the bulk of that taken up during expansion, and about 60 per cent of that taken up during combustion, should have been debited to the exhaust loss account.

In addition, of the total heat carried away by the cooling water, a very substantial proportion is generated by the friction of the piston on the cylinder walls.

It is interesting to take a specific example and to trace out as accurately as it is possible so to do the true gain in efficiency which would be effected if all heat loss to the cylinder walls were completely suppressed. Let us take, as a fair average example, the case of a well-designed and efficient engine with a compression ratio of 5:1 in which

- 32 per cent of the total heat of the fuel is converted into useful work on the piston,
- 28 per cent of the total heat of the fuel is carried away by the cooling water,
- 40 per cent of the total heat of the fuel remains and is accounted as lost to exhaust radiation, &c.

Of the total heat carried away by the cooling water, approximately 6 per cent will be lost to the walls of the cylinder during the combustion period, about 7 per cent will be yielded up during expansion, and the remaining 15 per cent during the exhaust stroke. Of the 6 per cent lost during the combustion period roughly about 40 per cent would appear as useful work or 2.4 per cent of the total heat of the fuel. Of the 7 per cent lost during expansion, somewhere about 20 per cent would be utilized or 1.4 per cent of the total heat of the fuel. Of the 15 per cent lost during the exhaust stroke, no part could have been utilized. We find therefore that, although 28 per cent of the total heat of the fuel has been carried away by the cooling water, only 3.8 per cent could have been converted directly into useful work on the piston, and the efficiency of the engine would be increased from 32 per cent to 35.8 per cent only, a gain of barely 12 per cent. Nor is this all, for, had all loss of heat to the cylinder walls been suppressed, the temperature of the working fluid would necessarily have been correspondingly higher, with the

result that the losses due to the increase both of specific heat and of dissociation at the higher temperatures would be increased substantially, and the net gain would be very small, probably only from 32 per cent to about 34.5 per cent or possibly 35 per cent.

These figures show clearly how relatively small a part the loss of heat to the cylinder walls plays in an Internal-Combustion engine, and how misleading it may be to assess that loss of heat by the heat carried away by the cooling water. As a first approximation it is fairly correct to assume that, of all the heat carried away by the cooling water of a cylinder, little more than 10 per cent could actually be converted directly into useful work.

Tables I., II., and III. show the distribution of heat as found in the author's  $4\frac{1}{2}$  in.  $\times$  8 in. variable-compression research engine shown in figs. 3 and 4 under several different conditions. These figures were all obtained under circumstances which ensure a very high degree of accuracy.

*Group A.*—Distribution of heat at different speeds with wide open throttle. In all cases the mixture strength was approximately 10 per cent weak, the jacket temperature was kept constant at 140° F., and the heat input to the carburettor was maintained at 0.0433 B.Th.U. per revolution.

The three sets of tests in this group were run, one on ethyl alcohol 95 per cent, and the other on petrol, sample (A), at a compression ratio of 3.8 : 1, while the third set was run on ethyl alcohol 95 per cent at a compression ratio of 7 : 1.

It will be observed :

(1) That owing to the lower mean cycle temperature the thermal efficiency obtained with alcohol is substantially higher than that obtained with petrol at the same compression ratio.

(2) For the same reason, the proportion of heat carried away by the cooling water is less.

(3) That the thermal efficiency is affected very little by a wide variation in speed.

(4) That the proportion of the heat taken away by the cooling water falls slightly as the speed is increased.

The results shown in Group B were observed under the following conditions. The engine was run at a constant speed of 1500 R.P.M., corresponding to a piston speed of 2000 ft. per min., and the load was varied by throttling, the mixture strength being kept constant throughout at about 10 per cent weak, while the heat input to the carburettor was maintained at 0.0433 B.Th.U. at full load and

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TABLE I—GROUP A

Fuel, Ethyl Alcohol 95 per cent Compression ratio 3·8 : 1				
R.P.M. ... ..	975	1300	1500	1700
Piston speed ft. per min. ...	1300	1733	2000	2266
Heat to I.H.P., per cent ...	26·9	27·0	26·9	27·0
„ cooling water, per cent	25·1	24·7	24·4	24·2
„ exhaust, radiation, &c.	48·0	48·3	48·7	48·8
Total heat ... ..	100·0	100·0	100·0	100·0
Fuel, Petrol, sample (A) Compression ratio 3·8 : 1				
R.P.M. ... ..	975	...	1500	1700
Piston speed, ft. per min. ...	1300	...	2000	2266
Heat to I.H.P., per cent ...	25·9	...	26·1	26·1
„ cooling water, per cent	30·4	...	28·0	27·0
„ exhaust, radiation, &c.	43·7	...	45·9	46·9
Total heat ... ..	100·0	...	100·0	100·0
Fuel, Ethyl Alcohol 95 per cent Compression ratio 7 : 1				
R.P.M. ... ..	975	1300	1500	...
Piston speed, ft. per min. ...	1300	1733	2000	...
Heat to I.H.P., per cent ...	37·6	38·1	38·3	...
„ cooling water, per cent	25·4	24·3	23·9	...
„ exhaust, radiation, &c., per cent ... ..	37·0	37·6	37·8	...
Total heat ... ..	100·0	100·0	100·0	...

reduced proportionately as the load was reduced. The circulating water temperature was maintained constant throughout at 140° F.

TABLE II—GROUP B

Fuel, Ethyl Alcohol 95 per cent Compression ratio 5·45 : 1 R.P.M. 1500				
Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	34·8	35·0	35·0	34·8
„ cooling water, per cent	24·1	26·0	29·2	33·0
„ exhaust, radiation, &c., per cent ...	41·1	39·0	35·8	32·2
Total heat ...	100·0	100·0	100·0	100·0
Fuel, Petrol, sample (A) Compression ratio 5·45 : 1 R.P.M. 1500				
Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33·5	34·0	34·1	33·5
„ cooling water, per cent	26·5	28·2	31·8	35·5
„ exhaust, radiation, &c., per cent ...	40·0	37·8	34·1	31·0
Total heat ...	100·0	100·0	100·0	100·0

From these tests it will be observed that :

(1) Provided that, as in the above cases, the time of ignition is advanced as the weight of charge is reduced, and therefore the relative proportion of residual exhaust products is increased, the thermal efficiency remains practically constant over a wide range of load.

(2) That as the load is reduced a greater proportion of the exhaust heat appears in the cooling water.

Lastly, a set of tests were run on hydrogen (Table III.). On this fuel alone it is possible to reduce the power output over a wide range

# DISTRIBUTION OF HEAT IN A FOUR-CYCLE ENGINE 79

by controlling the supply of fuel only, i.e. by qualitative governing, without relying upon stratification. These tests are particularly interesting as they are, it is believed, the only experiments which have yet been carried out in which the power output has been varied solely by varying the mean mixture strength of the working fluid.

TABLE III

Fuel, Hydrogen Gas Compression ratio 5.45 : 1 R.P.M. 1500 Mixture strength at maximum load 10 per cent weak				
Percentage of maximum I.H.P.	100	80	60	40
Heat to I.H.P., per cent ...	33.3	35.6	38.2	40.0
„ cooling water, per cent	23.6	24.9	25.3	28.6
„ exhaust, radiation, &c., per cent ...	43.1	39.5	36.5	31.4
Total heat ...	100.0	100.0	100.0	100.0

In these tests it will be observed that :

(1) The thermal efficiency increases rapidly as the load is reduced, due to the lower mean temperature.

(2) The proportionate heat supply to the cooling water increases slightly as the load is reduced, but at nothing like the rate shown in the previous tables, when governed quantitatively.

It is interesting to compare the results obtained on hydrogen with those found in the previous group. In the latter case the load is varied by varying the weight of working fluid per cycle, the temperature remaining practically constant, while in the tests with hydrogen the weight remained constant and the temperature was varied. As might be expected when the temperature is varied, so the efficiency follows suit, rising as the mean cycle temperature falls. Incidentally, also, the results obtained in Group B serve to illustrate how small a part direct heat loss to the cylinder walls actually plays, for as the weight of charge is reduced at nearly constant temperature, the *relative* heat loss must be increased substantially and yet the thermal efficiency remains unchanged. That quite a substantial change in heat loss has so little effect on the thermal efficiency illustrates how small, in itself, this whole source of loss must be—



that it actually effects no reduction at all in the thermal efficiency is probably due to the fact that as the weight of charge is reduced, so is the proportion of residual exhaust products increased, with the result that the temperature is slightly lower, and this very slight lowering of the flame temperature results in a gain in efficiency sufficient just to balance the increased relative heat loss.

The conclusions to be drawn from the preceding observations and tests are:

(1) That the direct loss of heat to the cylinder walls plays a comparatively insignificant part in the performance of an Internal-Combustion engine, and that even were the whole of this loss completely suppressed, the gain in power output and efficiency would be equivalent only to the conversion of an extra 2·5 to 3 per cent of the heat of the fuel into useful work.

(2) That, of the total heat carried away by the cooling water, only a very small proportion could be converted into useful work, and by far the bulk would appear in the exhaust.

(3) That when running with wide open throttle, the heat flow to the cooling water is very nearly proportional to the speed of the engine, i.e. the proportion remains nearly constant.

(4) When the load is reduced by throttling a much larger proportion of the heat rejected to the exhaust is taken up by the cooling water, before the gases enter the exhaust pipe.

It is frequently stated that the relative inefficiency of a combustion chamber with a large surface volume ratio is due to the increased heat losses. From the above it is clear that this cannot have any great influence. The most probable cause of the comparative inefficiency is, as shown in Chapter IV, that, owing to the lack of turbulence in such types of combustion chamber, an appreciable portion of the charge clings to and is so chilled by the walls that it either escapes combustion altogether or burns so late in the expansion stroke as to be of little value.

**Supercharging.**—Though first developed by Sir Dugald Clerk more than thirty years ago, the use of supercharging has only recently come into prominence as a means of increasing the power output of high-speed engines when working at ground level. That it has not been employed earlier may be explained by the fact that it is only in quite recent years that high-speed engines have been produced whose efficiency and freedom from detonation have rendered them capable of dealing with the intensity of heat flow which supercharging involves.

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The high-speed engine of ten years ago could barely stand the normal atmospheric density and was often either deliberately or inadvertently throttled at the valves, carburettor or both. As the efficiency improved, and therefore the proportion of destructive waste heat diminished, so the breathing capacity of engines was increased, until to-day we have so far reduced the waste heat that we can afford to supplement the normal atmospheric density, in other words, to supercharge. It must not, however, be supposed that because one engine of good modern design will withstand supercharging, another of inferior design will do likewise. To couple a supercharger to an inefficient engine is merely to court disaster.

TABLE IV.—1500 R.P.M.  $R = 4.3$

Intake Press., Atmos. Abs.	Intake Air, Temp. deg. F.	B.H.P.	B.M.E.P., lb./sq. in.	Heat to Cooling Water, H.P.	Heat to Cooling Water, per cent of B.H.P.
1.0	87	14.7	93	13.4	91.5
1.52	90	27.8	175	19.6	70.3
2.02	90	40.0	252	25.1	62.8
2.50	92	51.5	324	30.7	59.6
3.11	94	64.2	404	37.0	57.7
3.49	96	72.8	457	40.6	56.0
3.62	102	75.6	475	41.8	55.3

With a view to determining how the distribution of heat was influenced by supercharging, and also the effect of the latter on detonation, a special engine was designed and constructed at the author's laboratory for supercharging experiments up to very high pressures, namely, up to initial pressures of four atmospheres (absolute). For such work a single sleeve valve was employed, since the use of such a valve eliminated at once both the difficulty of balancing the inlet valves against high intake pressures and also the obvious one of burning out of exhaust valves. The engine in question is of  $4\frac{1}{2}$ -in. bore by  $5\frac{1}{2}$ -in. stroke with a fixed compression ratio of  $4.3:1$ ; it is designed safely to withstand a maximum pressure of 2200 lb. per sq. inch (which pressure has on occasion been exceeded), and to run at a normal speed of 1500–1800 R.P.M.

In view of the very high maximum pressures to be catered for the reciprocating parts are necessarily very heavy, and the mechanical efficiency in consequence is somewhat low, viz. approximately

(E 246)

75 per cent at 1500 R.P.M., and normal intake pressure. As the intake pressure is increased, the absolute friction losses increase but little, with the result that the mechanical efficiency rises rapidly.

Induction air is supplied under any desired pressure from a large piston compressor and a system of intercoolers is arranged so that the temperature of the intake air can be maintained at any de-

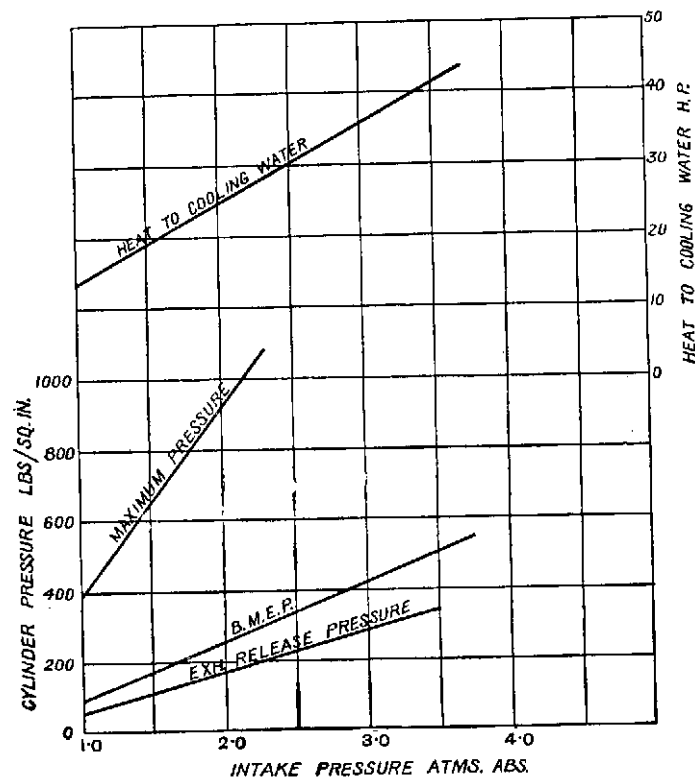


Fig. 28

sired figure between 80° F. and 180° F. Since the air compressor supplying the supercharge is separately driven, and in any case is of large excess capacity, its power absorption has not been taken into account in the following figures, nor has allowance been made for the energy returned to the engine piston by the supercharge air during the normal suction stroke.

Since measurement of the air supply involved very great difficulty the air/fuel ratio could not be determined with sufficient accuracy to justify expressing the heat balance in terms of the total heat of the fuel as in the case of the preceding experiments on the

## DISTRIBUTION OF HEAT IN A FOUR-CYCLE ENGINE 83

variable compression engine. It is therefore expressed in terms of the brake horse-power of the engine. The figures, in consequence, are not directly comparable with those given previously, but they show accurately the effect of supercharge on heat flow.

In fig. 28 is shown the increase in B.M.E.P. (uncorrected for the

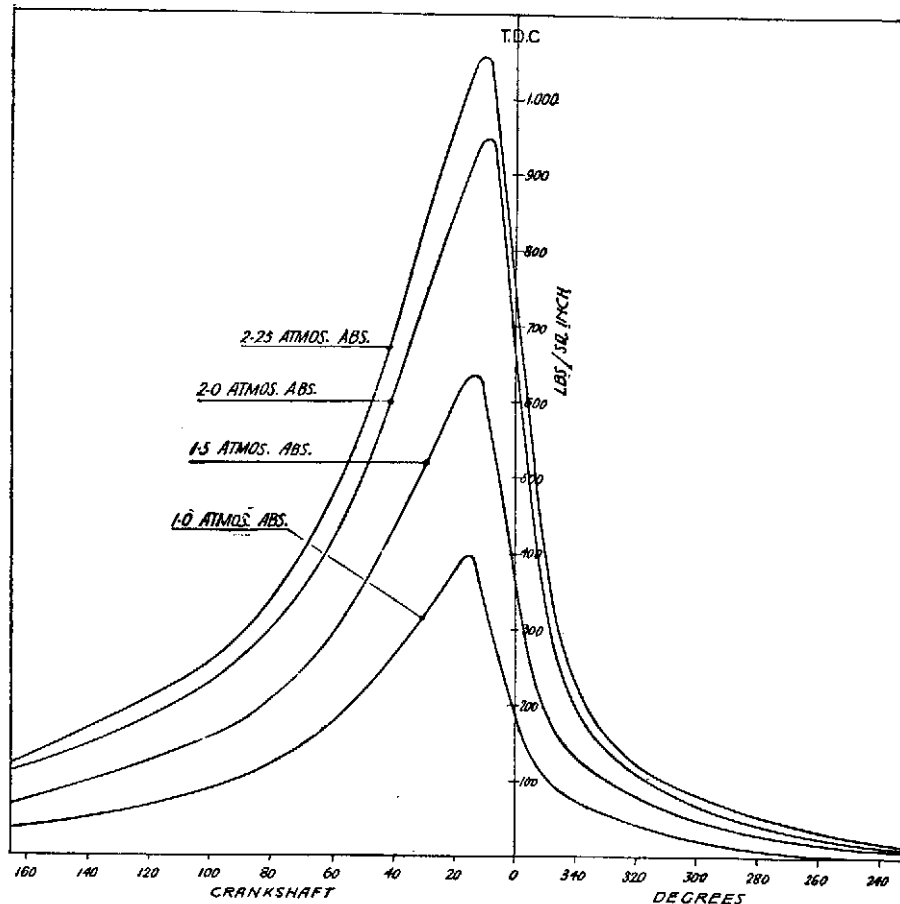


Fig. 29

power absorbed by the supercharger) in heat flow to the jackets, in maximum pressure, in mean effective pressure and in exhaust release pressure, as the pressure of the air is raised from normal atmospheric density to 3.5 atmos., while fig. 29 shows superimposed a series of indicator diagrams at varying initial pressures.

Investigations were also made into the influence of supercharging on the tendency to detonate. Since the engine has a fixed compression ratio it was necessary to prepare a number of different fuel

mixtures with various proportions of benzole added to a standard petrol. Starting with pure petrol the intake pressure was increased very gradually until detonation became apparent; the particular fuel in use at the time was then tested on the variable compression engine and its H.U.C.R. determined. A prepared benzole/petrol mixture was then supplied and the intake pressure further raised until detonation again supervened. This process was repeated step by step with both "cold" air, i.e. 80°–90° F. and hot air 170°–180° F. with repeated check tests of each mixture on the variable compression engine. The results of these tests were as shown in fig. 30 from which it will be seen that the general tendency to detonate increases with increase of intake pressure more or less according to the empirical formula

$$\left\{ \frac{R_1}{R_2} \right\} = \left\{ \frac{P_1}{P_2} \right\}^{0.78} \text{ where } R = \text{Highest useful compression ratio.}$$

$$\text{where } P = \text{Intake pressure.}$$

**Ignition Advance.**—The ignition advance required at various intake pressures, with maximum power mixture strength, is as follows:

Intake Press., Atmos. Abs.	Ignition Advance., degrees E.
1.0	40
1.5	40
2.0	38
2.5	36
3.33	33
3.66	31

Since this engine is not very sensitive to small changes in ignition timing these values are necessarily approximate.

The general gist of the whole of this investigation may be summarized in the graphs shown in fig. 31, which show the increase in all directions with increase of intake pressure over and above normal atmospheric pressure.

**Influence of Cylinder Jacket Temperature.**—It is frequently found that engines give more power and run more efficiently when the jacket water is hot. This is usually, but quite wrongly, attributed to the reduced heat loss to the cylinder walls when the jacket temperature is higher. It is due rather to the fact that the warming of the air and induction system is generally effected by means of the cylinder circulating water, consequently when the cylinder jackets are cold, so also is the induction system, with the result that a considerable proportion of the fuel precipitates, the distribution becomes defective, and both the power and efficiency suffer in conse-

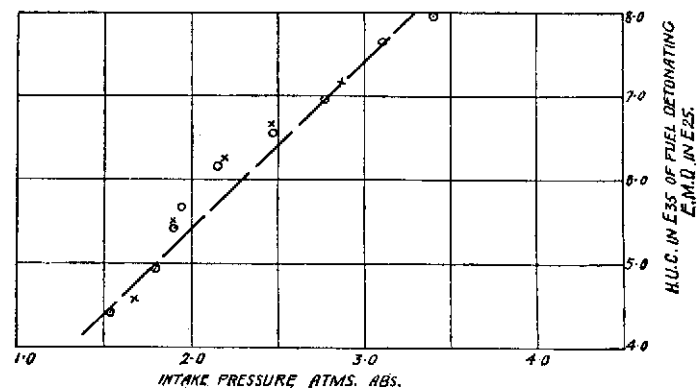


Fig. 30

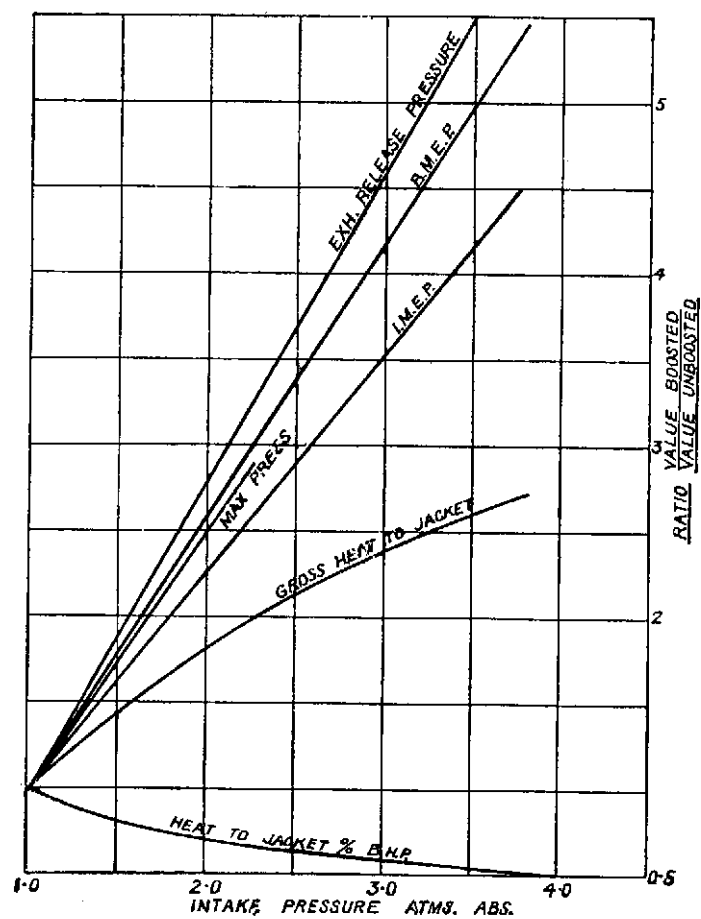


Fig. 31

quence. It is due also to the fact that the piston friction is dependent upon the viscosity of the oil on the cylinder walls and this, in turn, is dependent upon their temperature.

In engines in which the temperature of the induction system is controlled independently of the cylinder temperature, the difference in power, if any, is much less marked, and it then becomes a question of the relation between three factors.

- (1) The variation in heat loss to the jackets with temperature.
- (2) The variation in volumetric efficiency with temperature.
- (3) The variation in piston friction with temperature.

With regard to the heat loss to the jackets. It has already been shown that if the whole of the loss during combustion and expansion were completely suppressed, the indicated horse-power would be increased by less than 10 per cent, taking into account the loss due to dissociation, &c., at the higher temperatures, which would then be reached. Now the mean temperature during combustion and expansion may be taken as about  $3800^{\circ}\text{F.}$ , and that of the inner surface of the cylinder walls with boiling jacket water as, at most, about  $300^{\circ}\text{F.}$ , the temperature difference between the two being about  $3500^{\circ}\text{F.}$  If now the temperature of the jacket water be reduced by  $150^{\circ}\text{F.}$  (that is, from boiling to  $72^{\circ}\text{F.}$ ) the temperature difference between the gases and the cylinder walls will only be increased to  $3650^{\circ}\text{F.}$ , an increase of little more than 4 per cent. Assuming that the heat loss is proportional to the temperature difference (and this is approximately true), then the difference in the indicated efficiency and power due to the greater heat loss with cold jackets will only be 4 per cent of 10 per cent, or about 0.4 per cent. With a very badly designed combustion chamber it might conceivably amount to as much as 1 per cent, but in any case it is but a trifling amount.

With regard to the variation in volumetric efficiency the difference here is much more marked and is in the reverse direction. With hot cylinder walls at  $300^{\circ}\text{F.}$ , the rise in temperature of the working fluid due to contact with the inlet valve and cylinder walls during its entry will amount to about  $80^{\circ}\text{F.}$  With cold water it will be about  $55^{\circ}\text{F.}$ , assuming that the mean surface temperature of the walls is, in each case, about  $90^{\circ}\text{F.}$  higher than that of the jacket water. Experiments quoted elsewhere show that with a cylinder of normal design the change in the temperature rise of the entering gases is about one-sixth of that of the cylinder walls. The mean absolute temperature of the working fluid after its entry to the

cylinder may be taken as about 700° F. absolute. The weight of charge taken into the cylinder and therefore the power output will be proportional to the absolute temperature, so that if this is reduced by 25° F. the weight of charge taken into the cylinder, per cycle, will be increased in the proportion of  $\frac{700}{675}$ , or about 3.75 per cent. From the above considerations it will be seen that while the power and efficiency may be reduced by from 0.4 per cent to 1 per cent, due to increased

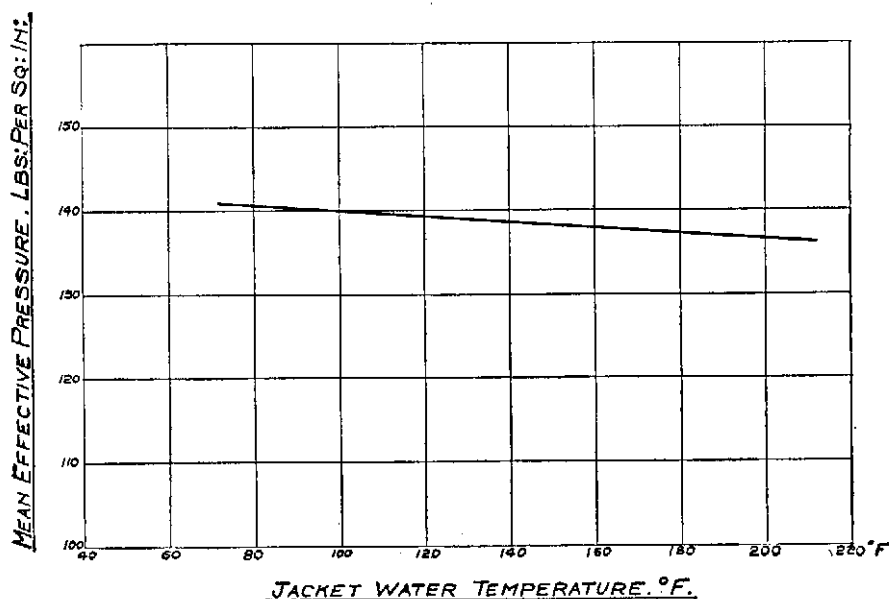


Fig. 32.—Variation in indicated Mean Pressure with Cylinder Temperature

heat losses, the weight of charge taken in will be increased by about 3.75 per cent, the net result being a gain of about 2.75 to 3.25 per cent in the indicated horse-power by reducing the temperature of the cooling water from boiling to about 72° F. or what is usually termed stone cold.

So far as the indicated horse-power is concerned, therefore, any reduction in the temperature of the cooling water involves an appreciable increase in the power developed in the cylinder. Fig. 32 shows the mean results of a large number of actual tests, from which it will be seen that the observed results agree very closely with the above deductions.

The third factor, namely piston friction, plays a very important part. In Chapter IX, dealing with piston design, it will be emphasized that piston friction is dependent in a very large measure on the



viscosity of the lubricating oil, and therefore upon the temperature of the cylinder walls—the colder the walls, the higher the viscosity of the oil clinging to them, and therefore the greater the friction. In the case of pistons with large bearing surfaces and heavy inertia pressures the difference in friction between hot and cold jackets may amount to as much as 8 per cent of the indicated horse-power of the engine, in which case the gain in indicated horse-power from the use of cold circulating water would be completely swamped by the extra friction, and the net result would be a loss of about 5 per cent in brake horse-power. On the other hand, with a very light piston, with the minimum of bearing surface, the difference may be as little as 3 per cent, in which event the brake horse-power of the engine will be independent of the jacket water temperature. Quite recently tests have been carried out on three single-cylinder engines, each of about the same size, in which the carburettor and induction system were heated independently of the circulating water. One of these engines was fitted with a cast-iron trunk piston of orthodox design, one with a very light aluminium slipper-type piston, and one with a cross-head piston, in which the friction of the piston on the cylinder walls is practically negligible. The results obtained are given below:

	B.H.P. at 1200 R.P.M.			
	Engine No.	Jacket Water, 212° F.	Jacket Water, 80° F.	Difference per cent.
(1) Cast-iron trunk piston ...	B. 8	27	25.7	- 5
(2) Aluminium slipper - type piston ...	E. 35	28.3	28.3	Nil
(3) Cross-head piston ...	T.S. 1	29.4	30.0	+ 2

In the case of the engines used for tanks which were fitted with cross-head pistons and independently heated induction systems, it was invariably found that the brake horse-power was from 1.5 per cent to 2 per cent higher with "cold" jacket water, i.e. about 80° F., than with "hot," i.e. about 180°–200° F. The difference in power was quite appreciable in service, and tank drivers frequently expressed surprise that their engines appeared to be more "lively" when the cylinders were cold.

To sum up, apart from any question of carburettor temperature,

the power output of an engine may increase or decrease as the jacket temperature is raised, depending upon the piston friction; if the piston friction is high, the power output will increase; if low, it will not increase. The increase in heat loss over the most extreme conditions is so small as to be negligible. The increase in volumetric efficiency is comparatively large, but is usually not sufficient to balance the increase in piston friction, hence the loss of power so often observed.

In most cases, however, the temperature of the carburettor and induction system is dependent also upon the jacket-water temperature, and when operating with fuels of low volatility the variation of temperature of these parts may play a supremely important part, particularly if the distribution is inherently defective or the mixture is rather weak.

**Gas Velocity and Indicated Mean Pressure.**—From analysis of a very large number of observations and test results, the author has found that the best all-round compromise between such various conflicting conditions as (1) the attainment of the highest possible volumetric efficiency, (2) the attainment of the necessary degree of turbulence, (3) the reduction of the fluid pumping losses to the lowest possible limit, is achieved when the mean gas velocity through the valves is in the neighbourhood of 130 ft. per second. Clearly from the point of view of conditions (1) and (3) it is desirable to keep the velocity as low as possible, while to fulfil condition (2) a high gas velocity is required. The gas velocity, as is customary, is expressed in terms of feet per second through the valve opening on the assumption (*a*) that the valve is wide open throughout the entire stroke; (*b*) that the mean and not the maximum piston speed is taken into account.

There is now abundant evidence that where the valves open directly into the main body of the combustion chamber a somewhat higher gas velocity may be employed without impairing the volumetric efficiency; on the other hand, with such a combustion chamber free from pockets or recesses turbulence will be better maintained and a lower gas velocity will suffice. In other words, when the valves are so placed that they open freely into the combustion space, both the efficiency and the mean pressure are less dependent upon gas velocity. In practice, however, the size of the valves and valve passages is largely controlled by mechanical limitations. Practical experience indicates that, when valves are fitted directly in the cylinder heads, only as much area of opening should

be given as can be provided without allowing the symmetry and compactness of the combustion chamber to suffer. It would seem preferable, therefore, to increase the gas velocity up to as high as 160 ft. per second, rather than to distort the combustion chamber in order to accommodate larger valves.

The valve-opening diagram is clearly a very important factor, and in the absence of precise information it must be assumed that the valve-opening diagram is in each case that best suited to the general design of the engine and its pipe-work.

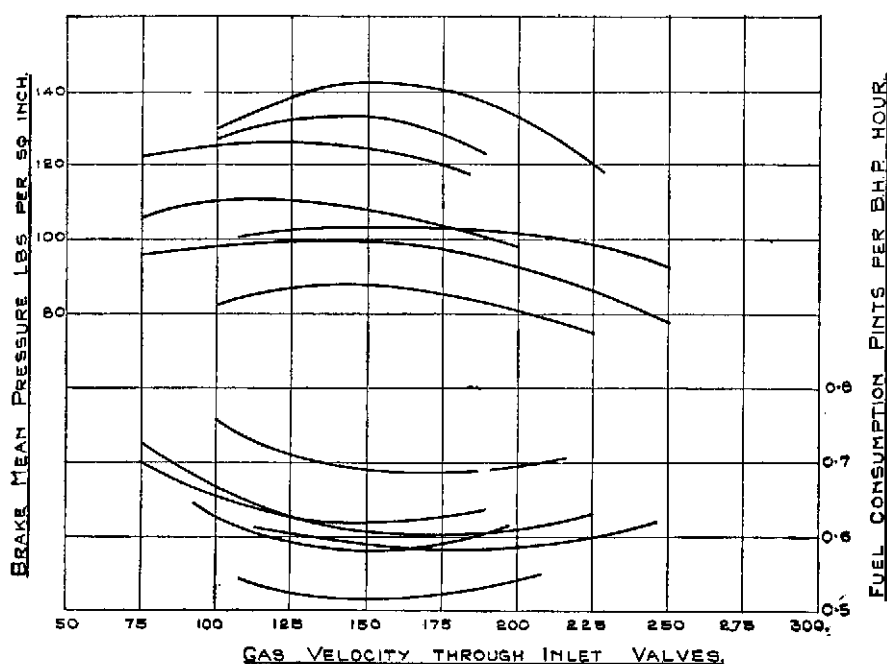


Fig. 33.—A Selection of Curves showing M.E.P. and Fuel Consumption on a Gas-Velocity Basis

The author has now a great deal more information on this subject, derived from a very large number of engines tested in his own laboratory and under his own observation. Further, since all these engines were designed on the same basis having the same valve-opening diagrams, and all running at sufficiently high speeds to eliminate any appreciable variation in the heat losses, the results are directly comparable; and the deductions to be drawn from them are probably quite reliable. The results of some of these tests are shown in fig. 33, from which it will be seen that, while both the mean pressure and fuel consumption differ very considerably, depending both upon the compression ratio and the efficiency of the

combustion chamber, the relation between mean pressure, efficiency, and gas velocity is substantially the same in all cases.

Again, a series of experiments was carried out on the special single-cylinder research engine illustrated in figs. 3 and 4; this engine is fitted with two inlet and three exhaust valves. Provision is made for throwing any of the valves out of operation. The results obtained with various combinations of valves in action are shown in fig. 34.

Many useful deductions can be drawn from the results of this last series, the more so because: (1) The tests were carried out under conditions ensuring an exceptional degree of accuracy; (2) the area of valve opening; and therefore the gas velocity, could be altered without disturbing any other condition.

All experience serves to indicate that the velocity through the exhaust valves may be as much as 50 per cent higher than through the inlet valves without causing any appreciable resistance or other loss. Although the same weight of working fluid has to pass through both the inlet and exhaust valves in much the same time the conditions under which it passes are very different. In the first place, at the time when the exhaust valves open, the pressure in the cylinder is relatively high, generally from 50 to 70 lb. per square inch above atmosphere. The sudden release of the gases at this high pressure results in the setting up in the exhaust pipe of a very high velocity, and the kinetic energy acquired by the gases in this pipe assists greatly in the withdrawal of the remainder from the cylinder. Unless, therefore, the silencer offers any undue resistance the energy expended in driving out the exhaust products is almost negligible, and, within limits, is almost independent of the gas velocity.

While distinct evidence of wire-drawing accompanied by a rapid increase in the fluid pumping losses becomes apparent so soon as the inlet gas velocity exceeds about 150 to 160 ft. per second, there is no evidence of any measurable fluid resistance, at all events until the exhaust gas velocity exceeds about 240 ft. per second.

One reason for this is that owing to their high release pressure the bulk of the exhaust products is discharged while the piston is more or less at rest. It must also be remembered that while a back pressure of, say, 1 lb./in. on the exhaust stroke only decreases the effective mean pressure by that amount, a negative pressure of 1 lb./in. at the end of the suction stroke will reduce the mean pressure by  $\frac{1}{14.7}$  or nearly 7 per cent, a much greater loss.

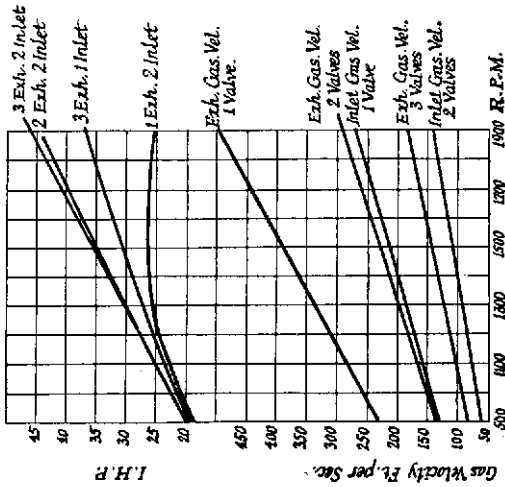


Fig. 34.

## TESTS WITH ONE OR MORE VALVES THROWN OUT OF OPERATION

I.H.P. (Compression ratio 6:1 in all cases.)

## Number of Valves in Operation and Mean Gas Velocity.

R.P.M.	Column No. 1.		Column No. 2.		Column No. 3.		Column No. 4.	
	No. of Valves 2 inl. 3 exh.		No. of Valves 2 inl. 2 exh.		No. of valves 3 inl. 1 exh.		No. of valves 1 inl. 3 exh.	
	Gas velocity, ft. per sec.		Gas velocity, ft. per sec.		Gas velocity, ft. per sec.		Gas velocity, ft. per sec.	
	Inlet.	Exhaust.	Inlet.	Exhaust.	Inlet.	Exhaust.	Inlet.	Exhaust.
900	66.1	86.6	66.1	137.2	66.1	235.0	132.2	86.6
1100	81.0	106.0	81.0	168.0	81.0	287.5	162.0	106.0
1300	95.5	125.0	95.5	198.0	95.5	339.0	191.0	125.0
1500	111.0	144.5	111.0	228.5	111.0	391.0	229.0	144.5
1700	124.8	163.0	124.8	259.0	124.8	443.0	249.6	163.0
1900	139.6	183.0	139.6	290.0	139.6	496.0	279.2	183.0
	I.H.P.		I.H.P.		I.H.P.		I.H.P.	
	19.9		20.0		18.8		19.1	
	24.5		24.55		22.9		23.0	
	29.4		29.4		26.0		27.0	
	34.5		34.4		28.7		30.7	
	39.8		39.3		29.4		33.8	
	45.1		44.3		25.2		36.0	

From the more reliable data now available it is possible to construct a curve giving the relation between inlet gas velocity and volumetric efficiency with a very fair degree of accuracy. As already explained, however, the volumetric efficiency is also influenced to a very large extent by the vaporisation of fuel within the cylinder and upon the degree of pre-heating before entry; it is therefore dependent upon these factors also. The curve shown in fig. 35 gives to a close approximation the relation between the volumetric

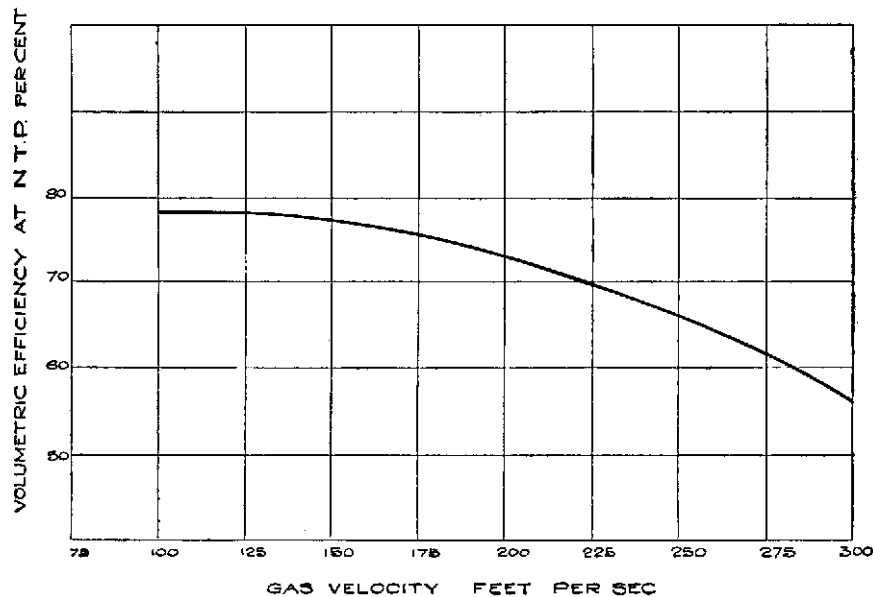


Fig. 35.—Volumetric Efficiency and Gas Velocity

efficiency obtained and inlet gas velocity for an efficient and up-to-date design of petrol engine with normal valve timing; for an engine running on benzol it is about 2.5 per cent too low, and for alcohol about 8 per cent too low, provided, of course, that the amount of pre-heating is the same for each fuel. The curve also assumes that the induction piping and carburettor offers no abnormal resistance, and that not more than four cylinders draw their supply from any one carburettor.

To sum up, it is clear that all the evidence available to-day tends to confirm the assumption that the most satisfactory compromise is attained when the gas velocity through the inlet valves is from 130 to 160 ft. per second, depending upon their position in the combustion chamber; the gas velocity through the exhaust

valves may, however, be as much as 50 per cent greater without any ill effect.

**Influence of Cylinder Size on Power and Efficiency.—**

For equal compression ratios the indicated power and efficiency of an engine is influenced by two factors: (1) Incomplete combustion of the layer of working fluid which clings to the walls of the combustion chamber; (2) direct loss of heat to the walls of the combustion chamber.

Both depend clearly upon the relation between the surface of the combustion chamber and its volume, but the former depends also upon the thickness of the layer adhering to the walls, which in turn is dependent upon the degree of turbulence or scouring effect. Both these sources of loss are reduced to a minimum when the combustion chamber is as compact and symmetrical as possible, for not only is the surface/volume ratio then reduced to a minimum, but the scouring effect of turbulence has the fullest possible play.

Assuming that the combustion chamber is similar in all cases, then it is evident that the relation of surface to volume will be reduced as the size of the cylinder is increased; since, so long as the proportions are similar, one factor varies as the square of the size and the other as the cube. It follows, therefore that under similar conditions the power output per unit volume and efficiency of an engine will increase with increase of size. Exactly how much it will increase must depend upon the proportion which these sources of loss bear to the whole, that is to say, the gain in power and efficiency with larger cylinders must depend largely upon the design of the combustion chamber. When the combustion chamber in all cases is as compact and symmetrical as possible, the gain with increase of size will be at a minimum.

It should be emphasized that the above arguments apply for equal compression ratios. When, however, the effect of cylinder size on the incidence of detonation is taken into account, it will be found that the compression ratio of the smaller cylinder can be increased to an extent which will compensate for the greater heat, loss, &c., with the result that at its appropriate compression ratio the performance—viz. mean pressure and efficiency of any petrol engine—is virtually independent of cylinder size.

## CHAPTER IV

### INFLUENCE OF FORM OF COMBUSTION CHAMBER

Of all the features of design which control both the power output and efficiency of an internal-combustion engine, by far the most important is the form of the combustion chamber. Upon this depends not only the efficiency with which the fuel is burnt, and therefore the power output and efficiency of the engine, but also, to a very large extent, the liability to detonation; and detonation by limiting the compression ratio that may be used sets an additional limit on the efficiency.

The term "Efficiency", never very closely defined, has, of late years, become much abused. It is quite customary nowadays to describe an engine as "efficient" merely because it is capable, mechanically, of running at a very high speed, though this is clearly a misuse of the term.

While the term "efficiency" is often abused, its true importance is still more often overlooked. It is argued frequently that since fuel is cheap and air still cheaper "efficiency" is of but little practical importance. From the point of view of fuel cost, this is often true, but it cannot too strongly be emphasized that besides determining both the power output and fuel economy the efficiency of an engine is also a measure of its reliability and maintenance of tune, for it must be remembered that the reliability of an engine depends, not so much, it is true, on the proportion of the total heat converted into useful work, but rather upon the proportion of the total heat which is not so converted, and which is left over to make trouble. Fully 90 per cent of the ills from which internal-combustion engines suffer are due, directly or indirectly, to excessive heat flow; directly in the form of cracked cylinders and valve seats, seized or burnt pistons and burnt out valves; in the form of carbonization, or gummed up piston rings, &c., and indirectly in the form of partially carbonized or contaminated lubricating oil, resulting in choked oilways and consequent bearing failures. Now each and all of these



sources of trouble, whether they result in complete breakdown or in loss of tune, are attributable to excessive heat flow, and it must constantly be borne in mind that the whole of this destruction is wrought by the flow of waste heat, and that one is really more interested in reducing the amount of heat left behind than anything else. The only possible way in which the waste or residual heat can be reduced is by converting a greater proportion of it into useful work.

Let us consider two engines of the same given cylinder capacity; each, at the same speed, will consume the same weight of air. Whether that air be saturated or supersaturated with fuel, the same amount of heat will be liberated, and we will suppose that it is the equivalent of 100 H.P. in each case. If now one engine has a thermal efficiency of 30 per cent and the other of 20 per cent, it follows that not only will the former engine give 30 H.P. as against 20 H.P. in the other case, but the amount of heat left over to make trouble will be only 70 per cent in the former case and 80 per cent in the latter, so that we can actually get 50 per cent more power from the same engine, and that with a lower heat flow. Now suppose that we run both engines at the same power output, say 20 H.P. To get this power output in the case of the more efficient engine we shall need to liberate only the equivalent of 66·6 H.P. worth of heat, 20 of which will be available as power and 46·6 will be left over. In the case of the less efficient engine we have seen that to get 20 H.P. we have to liberate 100 H.P. worth of heat, and have the equivalent of 80 H.P. left over to make trouble. It will be seen, therefore, that by increasing the efficiency from 20 to 30 per cent, the waste heat is reduced to very little more than one-half, and the more efficient engine will run probably ten times as long at the same power output without trouble or loss of tune. It is on the score, therefore, of reliability and of sustained and consistent performance that the necessity for efficiency is most insistent.

In the design of the combustion chamber the most important considerations are:

- (1) The maintenance of the turbulence set up by the gases during their entry;
- (2) The position of the ignition plug;
- (3) The avoidance of any pockets where the gases may become stagnant;
- (4) The provision of a free and unobstructed entry for the gases after passing through the inlet valve.

**Turbulence.**—The maintenance of turbulence until the time of ignition is probably the most important consideration of all, for upon turbulence depends the rate at which combustion takes place. If the combustible mixture were completely stagnant at the time of ignition, the initial flame brought into life by the passage of the spark across the points of the ignition plug would spread so slowly that, even in a comparatively slow-speed engine, barely half the working fluid would be burnt before the exhaust valve opens. No matter what fuel be used (except possibly hydrogen), we have to rely almost entirely on turbulence or mechanical disturbance to distribute the wan and timid flame rapidly throughout the combustible mixture, and this becomes the more important as the density of the charge is reduced by throttling; for as the density is reduced so is the proportion of residual exhaust products increased, and these, being diluents, tend to lower the flame temperature and so greatly to discourage and retard the process of inflammation. We rely, therefore, entirely upon mechanical disturbance, or turbulence, to speed up the process of combustion and to spread inflammation rapidly throughout the whole bulk of the working fluid.

Apart, however, from the question of accelerating combustion, turbulence plays another and highly important part. In any internal-combustion engine cylinder there is always a layer of working fluid adhering more or less to the cold cylinder walls. This, by its close proximity to cool surfaces, can get rid of its heat so rapidly that it escapes complete combustion. It is upon turbulence that we have to rely, to scour away this layer and to distribute it throughout the combustion chamber. There is little doubt but that the effective thickness of this layer, and its influence upon the power output and efficiency of an engine, depend largely upon the degree of turbulence within the combustion chamber.

Turbulence in itself, so far as the author has been able to ascertain, has no very important influence upon the tendency of the combustible mixture to detonate. If the mixture is completely stagnant and ignited from any one point, the flame will extend from that point very slowly at first, but at a steadily increasing velocity until ultimately it reaches a speed at which the unburnt gas, compressed by the rapidly approaching flame, receives heat from compression, and also by radiation and conduction at a rate in excess of that at which it can get rid of it to the cool cylinder walls, with the result that its temperature is raised above its self-ignition temperature. It then ignites spontaneously and almost instantaneously throughout its

whole bulk, thus setting up an explosion or detonation wave. If instead of being stagnant the combustible mixture is in a state of violent disturbance at the time when ignition is started, the whole process is speeded up enormously; but it is the *whole* process that is so speeded up and not any one part of it, with the result that the phenomena are precisely the same, though on a different scale so far as speed is concerned.

It is generally supposed that a combustion chamber having a large surface in relation to its volume owes its low efficiency entirely to direct heat loss to the cylinder walls, but in Chapter III. we have seen that direct heat loss plays a relatively small part in the performance of an engine, and it is far more probable that this form of combustion chamber owes its low efficiency rather to the fact that not only is the area of the layer of gas which clings to the walls and so escapes combustion considerably greater, but, in addition, its thickness also is greater, due to the fact that in such a chamber there is more resistance to the motion of the gases, and the scouring effect of turbulence is therefore less. Although it is not possible to state definitely that this is the case, yet there is a great deal of presumptive evidence in favour of such an assumption; not the least of which being that in any given design of combustion chamber an increase in turbulence, e.g. by increasing the velocity through the inlet valve, will often produce an increase in efficiency without altering in any respect the flow of heat to the cooling water. This, in itself, would not be conclusive evidence, but coupled with a great deal of other indirect evidence it carries a good deal of weight.

**Position of the Ignition Plug.**—The position of the ignition plug in the combustion chamber is very important, for upon it depends:

- (1) The tendency to detonate;
- (2) The time taken to complete combustion.

The influence of sparking plug position upon detonation has already been discussed in Chapter II and need not be reiterated here.

Apart from the question of detonation, the distance from the point of ignition to the farthest point in the combustion chamber controls, for an equal degree of turbulence, the time taken for inflammation to spread throughout the working fluid. This is very important, more particularly so on light loads, for, other things being equal, the rate of burning depends upon the flame temperature, and this, in turn, depends upon the proportion of residual exhaust products,

As the load is reduced by throttling, so is the proportion of inert gases increased, and, in consequence, the flame temperature is reduced and with it the rate of burning. It may be argued that, within limits, the rate of burning matters but little, because it is always possible to advance the time of ignition to suit. This, however, does not quite meet the case, even on full load, for, if the period of combustion be prolonged unduly, the direct heat loss to the cylinder walls at this vital period may begin to assume serious proportions, while when the load varies rapidly, as it does in all forms of motor traction, it becomes almost impossible so to adjust the time of ignition as to suit all conditions.

If the plug is placed centrally in the combustion chamber, a small ignition advance will serve for all conditions of speed or load, and the engine will run at all times at a high efficiency with fixed ignition timing. If, however, the plug is placed in such a position as to leave a long travel for the flame, not only will a considerable ignition advance be required at full load, but the engine will, at all times, be much more sensitive to the time of ignition, and will not run efficiently at light loads unless the ignition is still further advanced. It should be noted that both the time taken to complete combustion and the tendency to detonate depend, for equal degrees of turbulence, upon the absolute, not the relative, distance from the sparking plug to the farthest point in the combustion chamber. Thus, both the time of combustion and the tendency to detonate will be nearly the same in a cylinder of 8-inch diameter with the plug in the centre as in one of 4-inch diameter with the plug at one side, or for similar plug positions both the tendency to detonate and the time of combustion will be greater with the larger cylinder. This factor would militate against the large cylinder were it not for the fact that in the larger cylinder turbulence is the better maintained, the gases are in more rapid motion, and combustion therefore spreads relatively faster. Hence the efficiency does not suffer, though the tendency to detonate necessarily remains greater in the larger cylinder.

In the design of the combustion chamber the three points to be aimed at are:

(1) To provide a chamber into which the valves open directly, so that after entry the gases need turn no corners and so lose their initial velocity;

(2) To arrange that the sparking plug be placed centrally in the combustion chamber;

(3) That there should be no valve or other pockets where the gases may become stagnant.

At first sight it might appear that the ideal combustion chamber would be a true sphere, but although this would be excellent from the point of view both of surface/volume ratio and of turbulence, it would be ideal only if the point of ignition were at the centre, which is obviously impossible. Since the point of ignition must of necessity be situated somewhere on or very near the surface, it follows that the distance which the flame will have to travel will be equal to the full diameter of the sphere, and this will give rise to detonation and necessitate the use of a lower compression ratio,

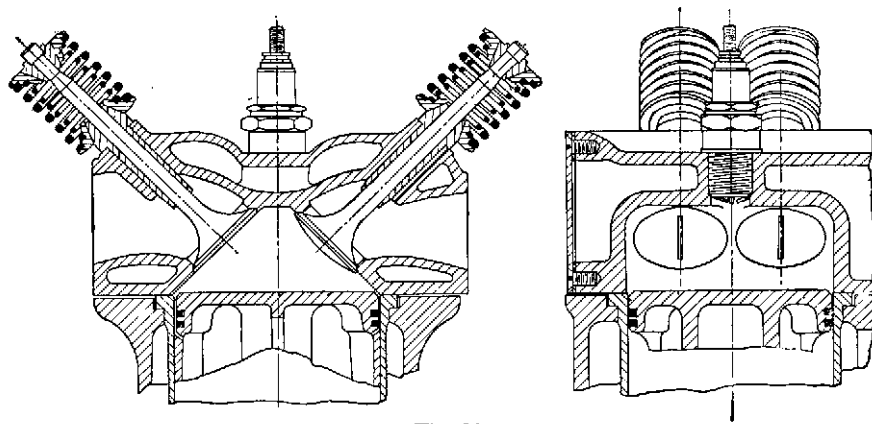


Fig. 36

with consequent loss of power and efficiency. Probably the nearest practical approaches to the ideal combustion chamber is that shown in fig. 36.

This form, while almost ideal on thermo-dynamic grounds, is not very convenient from a mechanical point of view, because it necessitates the use of overhead valves operated from two camshafts, for the central position of the sparking plug precludes the use of a central overhead camshaft. It is, however, particularly applicable to sleeve-valve engines, and the fact that it is inherent to this type of engine is, in the author's opinion, one of the strongest arguments in favour of the sleeve valve. For engines required to develop a high power output and efficiency, similar forms of combustion chamber but with the sparking plug placed at the side, are very commonly employed, and are, indeed, practically universal for all aircraft engines. The effect, however, of placing the plug at the side is to increase both the degree of ignition advance and the

tendency to detonate, and so necessitate the use of a somewhat lower compression ratio and a consequent reduction both in power and efficiency. By using two sparking plugs, however, placed at opposite sides of the combustion chamber, this disadvantage can largely be overcome, for the distance which the flame has to travel from the point of ignition until combustion is completed is then almost halved. In the case of aero-engines where dual ignition is insisted upon, on the score of reliability, this form has become almost a standard one.

The form of combustion shown in fig. 37, in which two horizontal valves are placed in a small cylindrical combustion chamber, is one

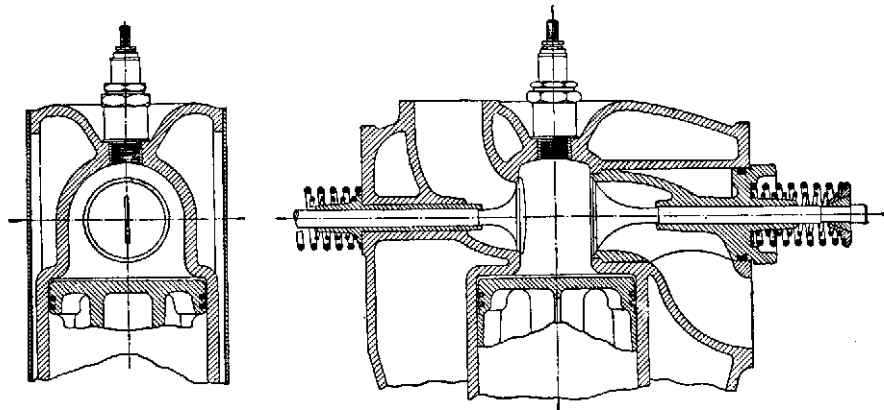


Fig. 37

of the best possible from the point of view both of detonation and combustion efficiency so long as the clearance between the piston and the cylinder head is reduced to the very limit necessitated by mechanical considerations, and so long also as the engine runs at a comparatively low piston speed and therefore does not require large valves. It is therefore a particularly efficient form for short-stroke engines, but though excellent for power output and efficiency it is unfortunately very inconvenient from a mechanical point of view when applied to a vertical engine on account of the valve operation and pipe work.

The form shown in fig. 38, in which the inlet valve is placed vertically over the exhaust in a side-pocket, and in which both the piston and cylinder crown are concave, is surprisingly efficient in maintaining turbulence despite the fact that the valves are pocketed, but, as may be expected with so long a travel for the flame, it is very bad from the point of view of detonation and therefore can only

be used with a low compression ratio. With this form of combustion chamber and with the sparking plug fitted in the valve pocket, the tendency to detonate is considerable, while with the plug fitted on the side, remote from the valve pocket, it is one of the very worst possible forms, from the point of view of detonation, as indeed might be expected. This form, therefore, necessitates the use of a low compression ratio, but gives a very good power output and efficiency at that compression.

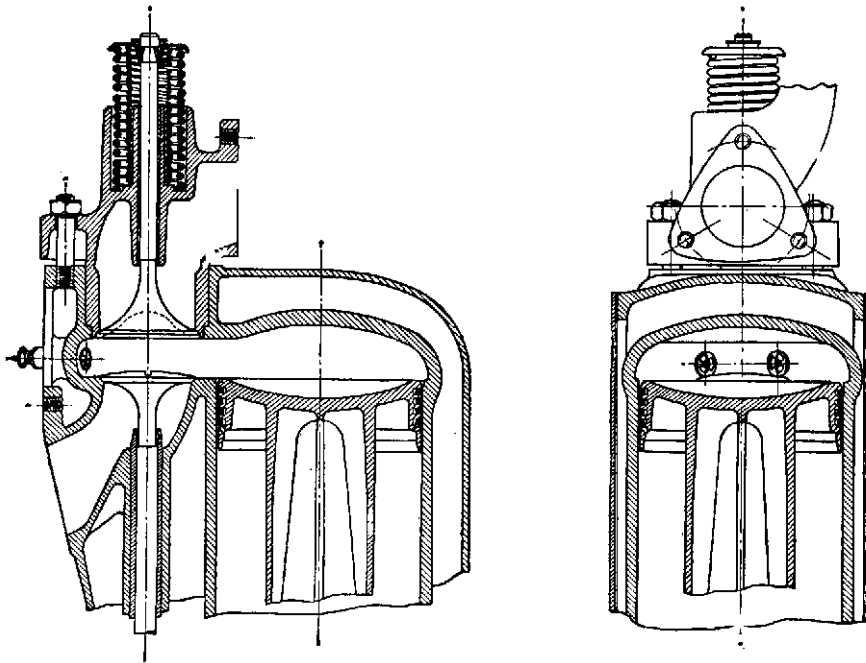


Fig. 38

Some ten years ago, and as the result of a good deal of previous experimental work carried out before and during the War, the author patented a form of cylinder head for side-valve engines; an example of which is shown in fig. 39; this has come to be known as the "turbulent head", and is now widely used. The primary objects aimed at in this design of head were:

(1) To create additional turbulence during the compression stroke in order to:

(a) Increase the rapidity of burning and so obtain both a greater effective expansion ratio and, at the same time, render the engine much less susceptible to ignition timing.

(b) Scour away as far as possible the layer of gas which normally clings to the cool surfaces of the combustion chamber and is

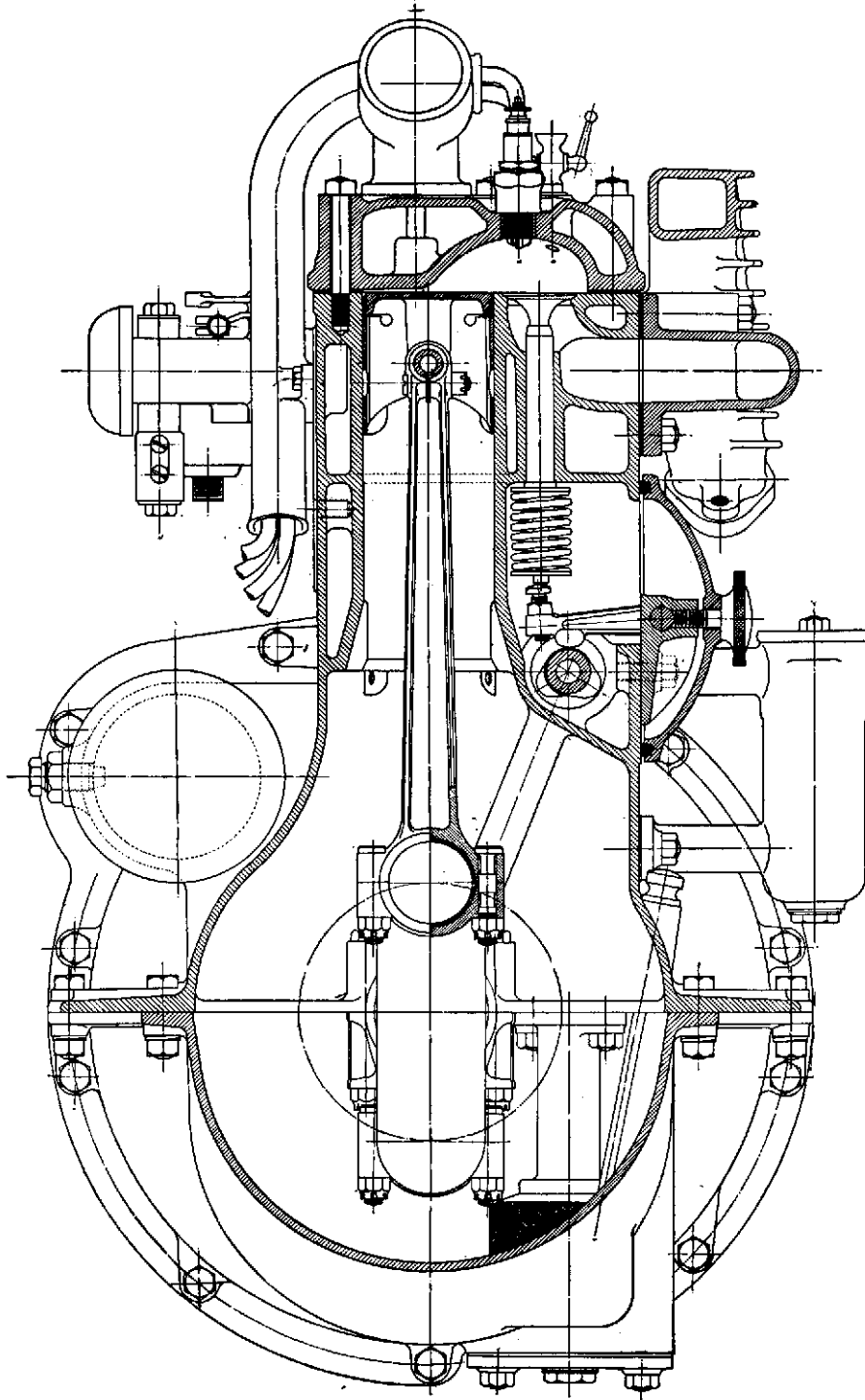


Fig. 39



therefore chilled to such an extent as to escape complete combustion either entirely or until so late in the expansion stroke as to be of little value.

(c) Reduce the tendency to detonate by keeping the unburnt gas in rapid motion, thus enabling it the more readily to get rid of the heat of compression by the oncoming flame front and, at the same time, to break up the flame front.

(2) To reduce the length of flame travel from the sparking plug

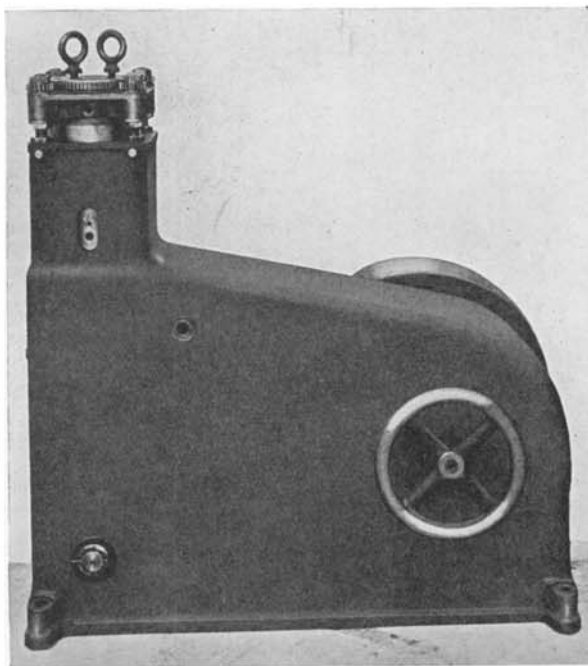


Fig. 40

to the farthest point in the combustion chamber by rendering inoperative, so far as detonation is concerned, the part of the combustion chamber over the farthest side of the piston. This was accomplished by bringing the piston into such close contact with the head that the gas between these two relatively cool surfaces was so chilled as to avoid detonation.

(3) To keep the flame travel as short as possible by placing the sparking plug in a central position.

(4) To reduce, to the minimum, the surface/volume ratio and therefore the heat loss during combustion, though this latter is relatively small.

The above were the intentions of the design at the time it was

first produced; by such means it was found possible to obtain from a side-valve engine a power output and economy at least as good as those previously obtainable from an overhead valve engine of

equal dimensions, with the added advantage of a reduction both in the tendency to detonate and in the "sensitiveness" to ignition timing.

During the past few years the author and his staff have been carrying out a series of investigations in relation both to the effects of turbulence and detonation. For this purpose, the following specially designed apparatus has been employed:

(1) A special single-stroke variable compression engine in which the piston makes one rapid compression stroke and is immediately caught and locked on top dead centre. The charge in the combustion chamber can be fired either

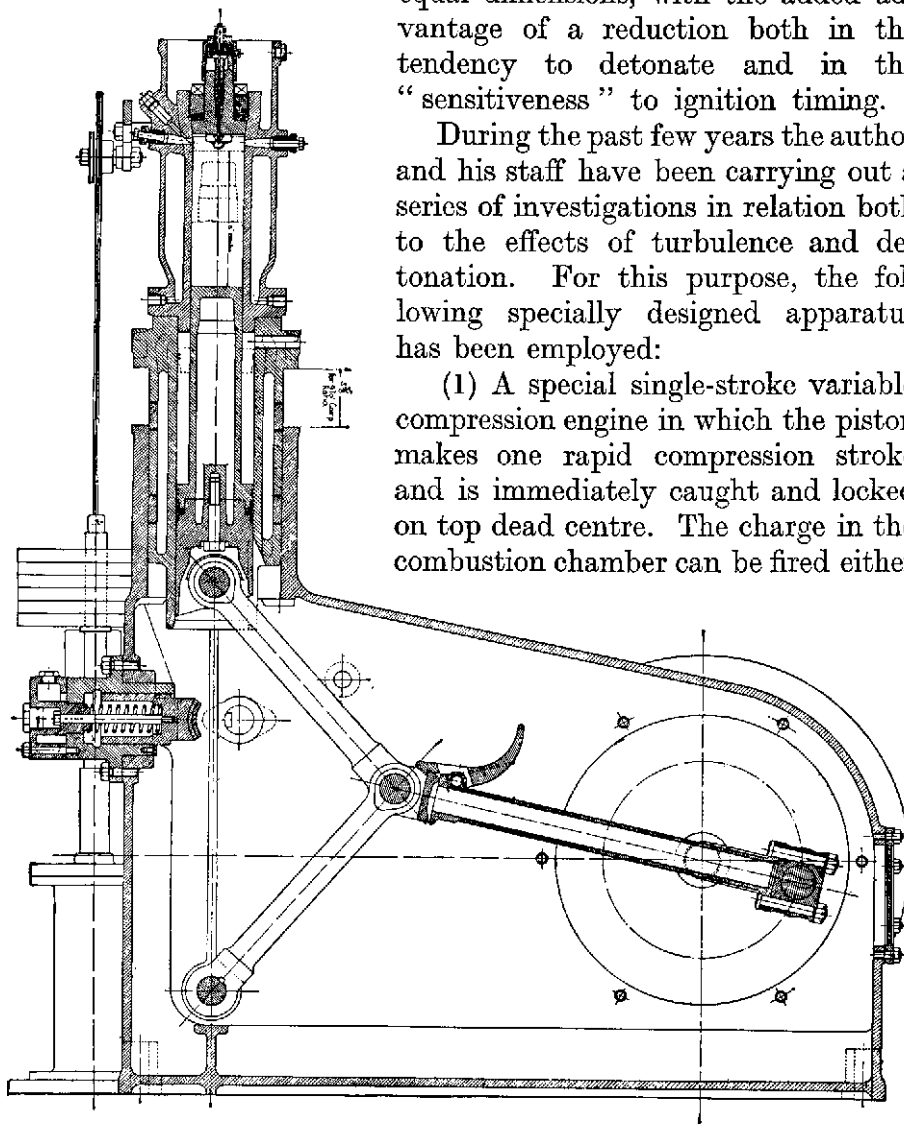


Fig. 41

by a sparking plug in the ordinary way, or by compression ignition alone. A small fan is fitted inside the combustion chamber to give any desired degree of turbulence. See figs. 40 and 41.

(2) A special side-valve research engine (figs. 42 and 43) equipped

with very delicate means for measuring the pressure during combustion at each degree of crank-angle, and with a series of quartz windows controlled by a stroboscope for observing the rate of spread of flame outwards from the sparking plug to the farthest point in the combustion chamber. The engine is equipped with readily detachable heads and over eighty different forms and shapes

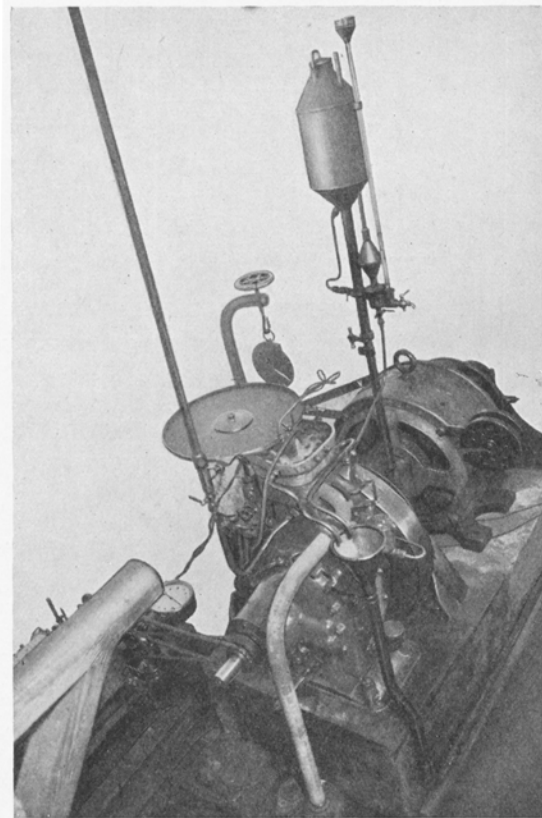


Fig. 42

have been tested and examined as to power output and efficiency, rate of pressure rise, sparking plug position, speed of flame travel, tendency to detonate and all such relevant factors.

(3) A variable compression engine in which also various combinations and positions of both valves and sparking plugs can be tested.

(4) A sleeve-valve engine in which the turbulence can be controlled both in direction and extent by means of guide vanes immediately outside the inlet ports, and recorded by means of

a form of anemometer fitted within the combustion chamber.

(5) A considerable number of research engines built originally for other investigations, but applicable for the purpose of such experiments. In all cases a Dobbie McInnes "Farnboro" electric indicator was employed to determine and record the pressure changes in the cylinder and more particularly the rate of pressure rise during the progress of combustion.

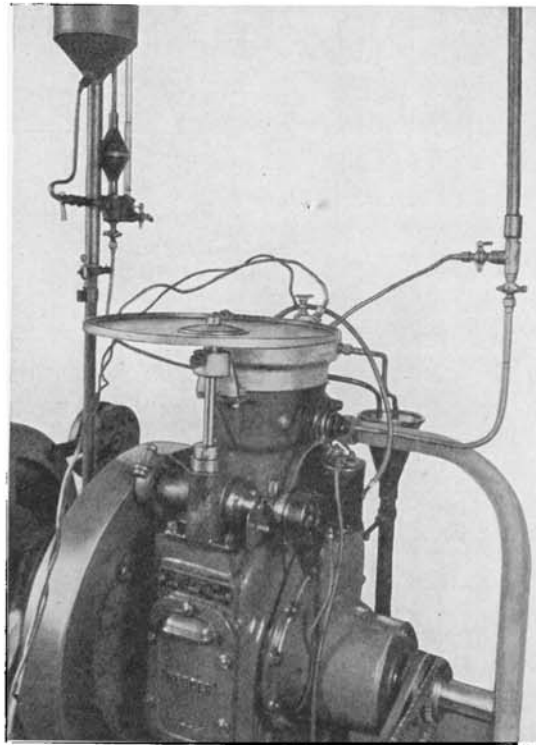


Fig. 43

Such an investigation naturally takes many years and is excessively tedious, for the most meticulous care must obviously be taken to ensure that all conditions are strictly comparable, especially such factors as mixture strength, air, water, and oil temperatures, &c. Moreover, it is obvious that the experimental apparatus must be so designed and constructed as to ensure the highest possible consistency from a mechanical point of view.

The general gist of the experiments carried out has, on the whole, confirmed the conclusions previously expressed, but it is now possible to separate out and to evaluate the various individual factors.

Briefly, the more recent experiments have shown:

(1) That the rate of burning depends primarily upon the degree of turbulence, and may be expressed in terms of increase of pressure per degree of crank-angle.

(2) That, to a secondary extent, the rate of burning depends upon the compression ratio; the higher the compression the more rapid the burning and therefore the less need for turbulence. The diagrams, fig. 45, taken consecutively from the variable compression engine, described in Chapter I, at ratios of 4, 5 and 6.0 : 1, but with exactly the same degree of turbulence in each case, show this very clearly.

(3) That, at any compression ratio, the best power output and efficiency are obtained on full load when the rate of pressure rise is

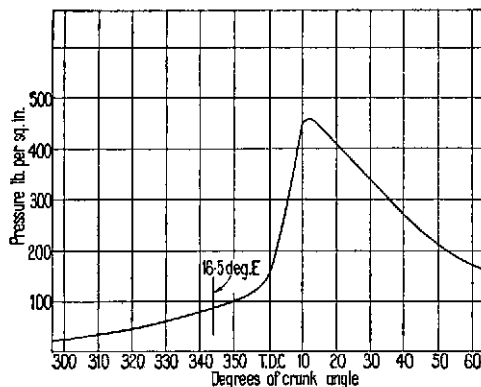


Fig. 44.—Max. press., 466 lb. per sq. in. Pressure rise, 34.4 lb. per sq. in. per deg. Ignition advance 16.5 deg. E.

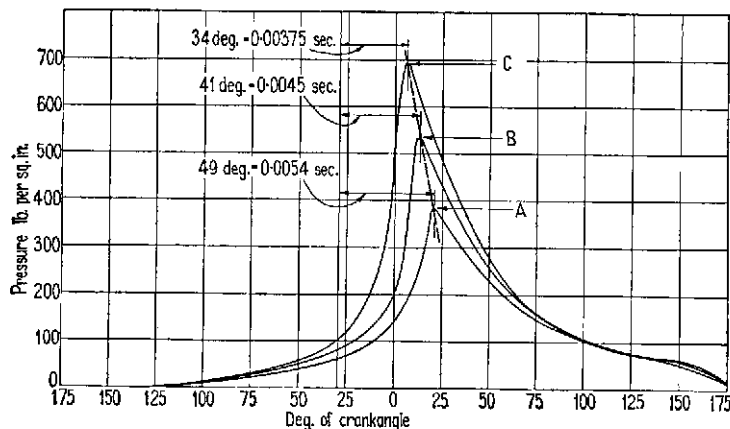


Fig. 45—A, R = 4.0 : 1. 1-M.E.P. = 120 lb. per sq. in.  
B, R = 5.0 : 1. " " 136 " " "  
C, R = 6.0 : 1. " " 146.5 " " "

approximately 30 lb. per square inch per degree of crank-angle, but that at light loads a somewhat greater degree of turbulence is desirable. Fig. 46 shows the observed relationship between power output and rate of pressure rise. It will be seen that over the range

25 lb. to 35 lb. per square inch per degree the curve is fairly flat; below 25 lb. the loss by delayed and incomplete combustion becomes serious; in practice, very serious, for delayed combustion

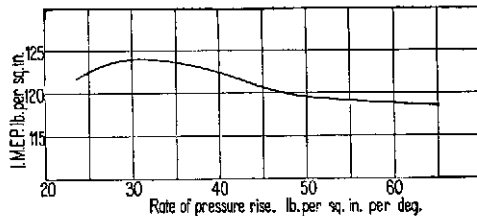


Fig. 46.

means heavy carbonization, a tendency to burn out or pit exhaust valves, a "sensitivity" to ignition timing and a tendency to fire back into the carburettor if the mixture is weakened to the economical limit. Above about 35 lb. per square inch

the heat loss by intense convection exceeds the gain due to the more rapid burning and the performance once more begins to fall. Also, at these high rates of pressure rise the very abrupt application of pressure intensifies crankshaft spring, piston slap, &c., and renders the engine running very harsh, noisy, and rough.

(4) That in the so-called "turbulent head", turbulence is set up to an almost equal degree by:

(a) The initial velocity through the inlet valve.

(b) The velocity through the restricted throat during compression.

It is intensified also by the final ejection, just prior to ignition, of the gases entrapped between the piston and cylinder head.

During the series of experiments referred to, each of the above factors was varied individually step by step, and its influence evaluated.

(5) That though each of the factors (a) and (b) influences the degree of turbulence to an almost like extent, it is usually undesirable, because of secondary considerations, unduly to intensify either of them at the expense of the other; that is to say, it does not pay, on the grounds of volumetric efficiency, to provide a very small inlet valve and a large throat, thus relying for turbulence mainly on intake velocity, nor, on the other hand, does it pay to provide a very large valve and rely for turbulence on a very restricted throat; for either method alone or when exaggerated unduly tends to restrict the breathing capacity of the engine.

(6) That turbulence from each of the three sources increases proportionately with increase of speed so that the relationship of rate of pressure rise to crank-angle remains substantially constant throughout the whole speed range of the engine; thus a combina-

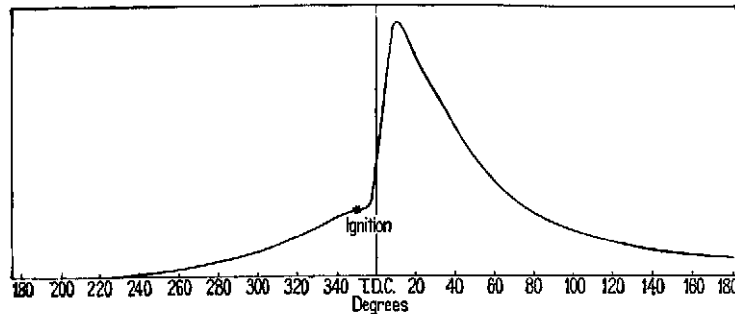


Fig. 47.—Pressure rise 32.5 lb. per deg. Ignition 10 deg. E. Max. press. 438 lb. Speed 800. Full load. O.V.S. Detonation. Max. power mixture strength. I.M.E.P. 122 lb.

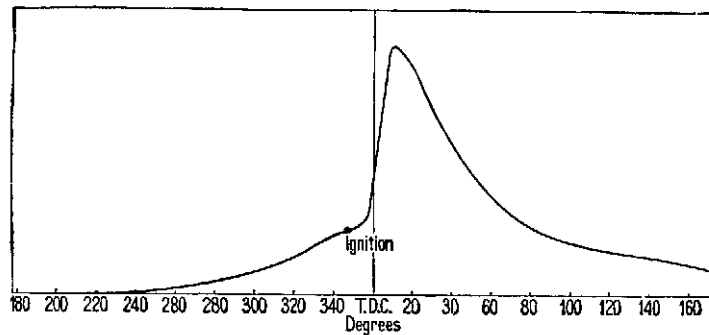


Fig. 48.—Pressure rise 32.5 lb. per deg. Ignition 13 deg. E. Max. press. 446 lb. Speed 1200. Full load. No detonation. Max. power mixture strength. I.M.E.P. 127 lb.

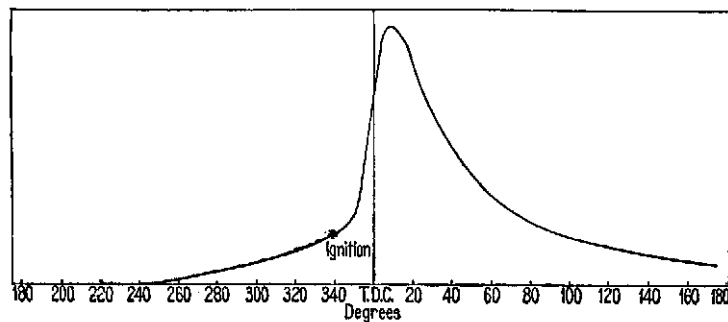


Fig. 49.—Pressure rise 31.8 lb. per deg. Ignition 21 deg. E. Max. press. 442 lb. Speed 1800. Full load. No detonation. Max. power mixture strength.

tion of valve size and combustion chamber proportions designed to give a rate of pressure rise of 30 lb. per square inch per degree at full throttle will give this rate at any engine speed. A typical

example of this is shown by the series of indicator diagrams taken from a commercial vehicle engine equipped with such a head. (Figs. 47, 48 and 49.)

(7) That in all cases the best possible position for the sparking plug is nearly in the centre of the "effective" combustion chamber, not quite in the centre, but with a slight bias towards the exhaust valve head, since this neighbourhood is necessarily the hottest zone and should therefore have the shortest flame travel. Only a slight bias is desirable however. If pushed too far over towards the exhaust valve the length of flame travel in the other direction then becomes too great.

Experiments on a turbulent head to investigate the effects of various positions of the sparking plug (see fig. 50) gave the following results, shown in Table I.

TABLE I

Position of Sparking Plug.	Full Throttle Performance corrected to C.R. 5:1.		Ignition Advance, degrees Early.	H. U. C. Ratio on Shell No. 1 Petrol.
	B.M.E.P., lb./sq. in.	Fuel Consumption, Pt/B.H.P./hour.		
1	111.9	0.552	17.6	5.87:1
2	109.7	0.577	19.8	5.65:1
3	110.5	0.577	23.0	5.3 :1
4	109.5	0.579	22.0	5.45:1

It will be noticed that position No. 1 is that giving the best results both in power and efficiency, together with the least tendency to detonate. Although in position No. 2 the sparking plug is well over the exhaust valve, a hot region, the tendency to detonate is

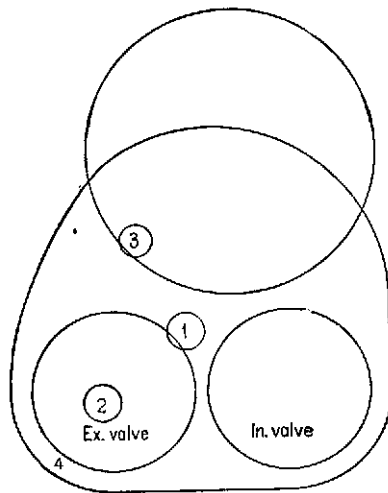


Fig. 50—Sparking Plug Positions  
No. 1, Normal position in Ricardo Turbulent Head. No. 2, Over Exhaust Valve.



greater than for the more central position No. 1. The reason for this lies in the greater length of flame travel in position No. 2. In Position 4 the effect is still more marked.

(8) That so far as detonation is concerned, the clearance between the piston and cylinder head in the "inoperative" portion is the main determining factor, and is critical. Here detonation is liable to take place, as shown by observation through the quartz windows, so soon as a certain minimum clearance is exceeded. Fig. 51 shows the observed relationship between clearance space and the tendency to detonate, from which it will be seen that a reduction in clearance from 0.15 in. to 0.10 in. alone permits of the compression ratio being raised by no less than 0.85 ratio, viz. from 5.75 to 6.6.

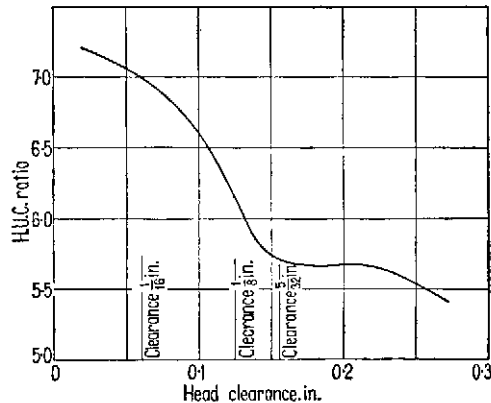


Fig. 51.—Curve showing tendency to detonate plotted against head clearance. Spark in normal position for Ricardo head.

(9) That the effect of turbulence alone upon the tendency to detonate and as distinct from much more important factors such as length of flame travel is appreciable, but not outstanding. It is extremely difficult in any practicable engine to vary the degree of turbulence without varying also other relevant factors which are liable to confuse the issue, so far as detonation is concerned. For the purpose of this part of the investigation the single-stroke engine, with and without the fan running, proved the most instructive. In this apparatus the compression could be varied within wide limits, while the fan speed, and therefore the turbulence, was under complete control. It was found that as the fan speed was increased, so the compression ratio could be raised progressively, when the same fuel was used, at the same mixture strength and the same temperature.

Again, a series of experiments on the research engine No. 2 on which only the area of the throat or passage connecting the combustion chamber proper with the cylinder was varied, showed that, as this area was reduced, and the turbulence and rate of pressure rise increased thereby, so the ratio of compression within the detonation limit could be raised.

## INFLUENCE OF FORM OF COMBUSTION CHAMBER 113

Thus a series of tests on a large number of heads all fitted to the same cylinder, and with the same minimum clearance above the piston, but differing only in the throat area, yielded the results shown in Table II.

TABLE II

Degree of Turbulence (expressed in terms of rate of pressure rise, lb. per square inch per degree of crank angle).	Highest Useful Com- pression Ratio of Shell No. 1 Petrol.
25	5.35:1
30	5.55:1
35	5.80:1
40	6.10:1
45	6.55:1
50	6.85:1
55	7.05:1
60	7.15:1

The above tests may be considered as strictly comparable in every respect except that as the throat area was reduced, so perforce was the maximum length of flame travel, though only to a very small extent. It will be seen that as the turbulence is intensified so is the permissible compression ratio increased, but that over the range practicable with any normal design of engine, namely from 25 to 35 lb. per square inch per degree, the effect of turbulence upon detonation is not of a very large order.

In a similar manner the influence upon detonation of the depth of the layer of gas contained in the clearance between the piston and cylinder head was determined by varying this alone. A series of experiments which could be carried out without the disturbance of any other relevant factor except that decrease of clearance, by increasing the rapidity of ejection of the entrapped gases, tended somewhat to increase the turbulence. The result of these observations taken with five different clearances and with a fixed throat area giving rather high turbulence are given in Table III.

It will be noted that as between heads B and C a difference in clearance of from 0.085 to 0.147 accounts for over one whole ratio of compression, though the observed difference in rate of pressure rise as between these two tests was less than 3 lb. per square inch per degree. The very large difference is here to be accounted for by the fact that a clearance of 0.147 was sufficient to allow detonation to occur in the shallow pocket so formed. In other words the farther portion of the combustion chamber over the piston has, at this

larger clearance, become "operative" so far as detonation is concerned. This was confirmed by observation through the stroboscope and quartz windows when the flame could be seen to pass across the clearance space at full velocity, whereas with the smaller clearance space it slowed down very appreciably after entry. Thus, as between heads B and C the effective length of flame travel is increased by  $2\frac{1}{2}$  in., i.e. the length of the clearance space lying between the piston and cylinder head.

The effect on turbulence of varying the size of the inlet valve was almost exactly similar to that of varying the restricted throat, that is to say, when the area of the inlet was doubled, the rate of pressure rise was approximately halved and vice versa, thus showing

TABLE III

Head.	Clearance between Piston and Head. Inches.	Highest Useful Compression Ratio of Shell No. 1 Petrol.
A	0.022	6.9 : 1
B	0.085	6.8 : 1
C	0.147	5.67 : 1
D	0.210	5.48 : 1
E	0.272	5.37 : 1

that turbulence can be promoted to an almost like extent by restriction either at the inlet valve throat or in the communicating passage.

A series of tests was carried out in order to determine the influence of shape generally. Keeping the clearance above the piston and the throat area constant, various contours ranging from a plain rectangular section to rounded sides with a "streamline" slope over the piston were also tried several years ago. The effect on performance was negligible, even between the two extremes; the difference in power output, fuel consumption, detonation and all other relevant factors was observed to be less than 1 per cent.

**Limitations of the Turbulent Head.**—The chief limitations of the ordinary turbulent head are:

(1) That with high degrees of turbulence, although the maximum pressure is not increased to any appreciable extent, the very abrupt application and rise of pressure following ignition causes "spring" of the moving parts, especially the crankshaft, and so gives rise

to a high periodicity vibration accompanied by an indeterminate drumming noise, the combination of the two being usually described as "roughness". In practice, this roughness determines and limits the degree of turbulence which may be employed. Conversely the degree of turbulence which may be used without roughness is a measure of the rigidity and stiffness of any engine, more particularly as regards its crankshaft. A really well-designed engine with an adequately supported crankshaft will run quite smoothly with a rate of pressure rise up to 30 lb. per square inch per degree, but only an exceptionally "stiff" engine will stand 35 lb. per square inch per degree without roughness.

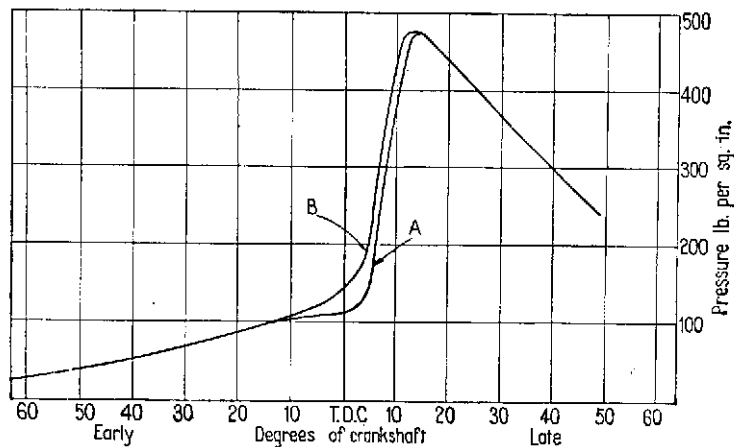


Fig. 52

An examination of the problem of roughness has, therefore, played a very prominent part in the general investigation.

It has been found that roughness is due, not to the absolute rate of pressure rise but rather to the very abrupt application of pressure, and that, if the pressure is applied at first gradually, a very rapid rate of pressure rise may be employed later without any trace of roughness, much in the same manner as a cam-operated valve, even with a very high rate of acceleration, can be made quite silent in operation by the provision of a gradual approach curve to take up all clearance and backlash. In order to provide a gradual initial pressure rise and thus gently to take up all spring and backlash, means were sought whereby a small proportion of the total charge could be isolated, and burned in a more or less stagnant condition, while the main bulk was in a state of normal turbulence. By igniting first this stagnant portion, a slow but gradually increasing rate

of pressure rise could be attained initially, followed by extremely rapid burning so soon as the flame reached the main portion of the charge. By isolating and stagnating 15 per cent of the total charge, it was found possible to burn 85 per cent of the charge at a rate of pressure rise even as high as 50 lb. per square inch per degree without any appreciable roughness and certainly with markedly less roughness than in an ordinary turbulent head at a rate of 30 lb. per square inch per degree. The two superimposed indicator diagrams in fig. 52 illustrate the different conditions; in one case the whole charge was burnt at the rate of about 50 lb. per square inch per degree; in the other, about 15 per cent was burnt slowly and the remainder at the rate of about 50 lb. per square inch per degree. In the former case, the running was intolerably rough; in the latter, quite reasonably smooth.

The cylinder head shown in fig. 53, patented recently by the author, and known as the "Shock-Absorber" type shows the practical application of this principle. In this type of head, a shallow flat-roofed pocket is provided with the sparking plug fitted centrally therein.

The pocket opens out into the main combustion chamber through a wide but shallow mouth extending about half-way round the circumference, and having a very abrupt rectangular opening. This pocket may be placed anywhere in the cylinder head, but it is convenient to place it over either the inlet or exhaust valve. The capacity of the pocket may be varied at will, depending upon the amount of gradual "take up" required. In practice, the best compromise is reached when the capacity of the pocket ranges between 10 and 25 per cent of the total capacity of the combustion chamber. The gas driven into this pocket during the compression stroke is broken up by the abrupt and sharp-edged

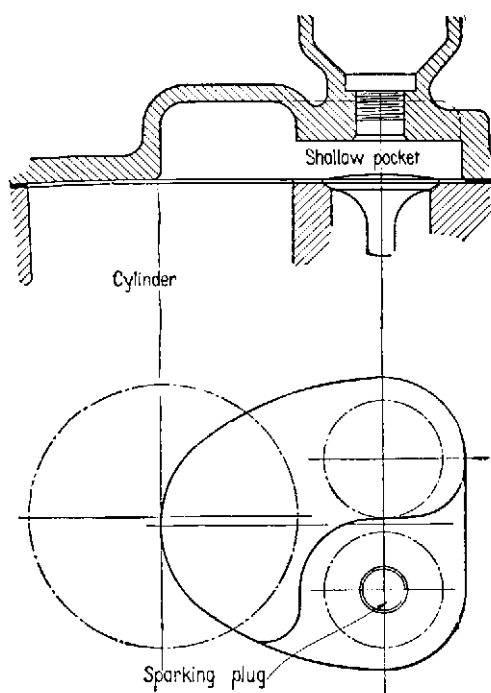


Fig. 53.—"Shock Absorber" head

entry into local eddies which die down very soon in so restricted a space.

On ignition, the gas in the pocket burns slowly, but at a steadily increasing velocity, until the flame reaches the open mouth when, owing to the abrupt change of section, the flame front is broken up and disseminated rapidly throughout the main body of the combustion space; stroboscopic and pressure rise observations on one experimental head, in which the capacity of the pocket was approximately 15 per cent of the total volume of the combustion chamber, showed that the time from the actual passage of the spark until the flame reached the open mouth amounted to  $30^\circ$  when the

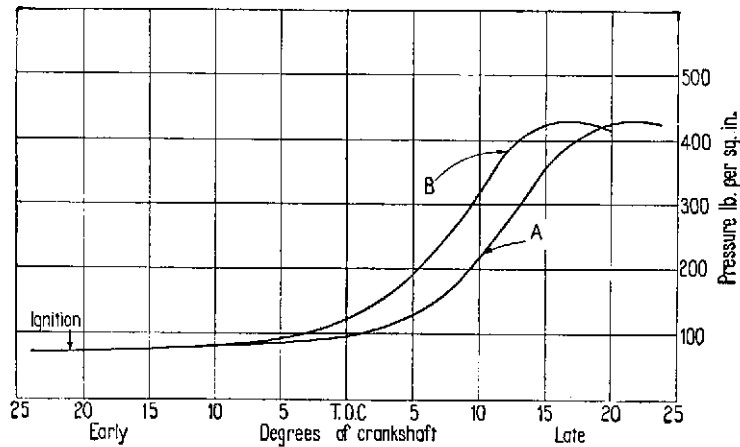


Fig. 54

shallow pocket was placed over the inlet valve, while the completion of combustion (up to the point of maximum pressure) of the main body of the charge amounted only to  $12^\circ$ . In other words, the burning of 15 per cent of the total charge occupied  $30^\circ$  of the crankshaft, while that of the remaining 85 per cent occupied only  $12^\circ$ . When the pocket was placed over the exhaust valve, the rate of burning therein was faster by about 4 crankshaft degrees, due to the higher temperature both of the gas and of its surroundings when in this position. See fig. 54.

From the point of view of detonation, although the sparking plug is no longer centrally situated, the "Shock-Absorber" type of head is very slightly better than the ordinary turbulent type, because, so far as detonation is concerned, the effective length of flame travel is little more than that from the open mouth of the pocket to the farthest point in the combustion chamber.

It is found with the "Shock Absorber" type head that the disturbance caused by the burning of the pilot charge and the emergence of the flame on a wide front, tend to create additional turbulence in the main body of the combustion chamber, so much so that when, to a given turbulent head, having a normal rate of pressure rise of 35 lb. per degree, a shallow pocket was applied, the rise of pressure rose to over 45 lb. per degree, though none of the controlling factors were altered. This means, in effect, that for a "Shock-Absorber" type head, the throat connecting the combustion chamber with the cylinder may be enlarged considerably and still allow of the same degree of effective turbulence and the same rate of pressure rise, as with the ordinary turbulent head, in fact, the restriction, as such, becomes less of a necessity.

In either the side valve or in the conventional overhead valve type of engine, the valves are placed in a single row, along the engine, consequently the total available valve area, or in other words, the breathing capacity, is limited by the length of the engine or rather that of the cylinder block.

In practice, the requirements of the crankshaft and its bearings dictate the minimum cylinder centres and therefore the length of the cylinder block. For any given size of cylinder the total available valve area obtainable when all the valves are in a single row and minimum cylinder centres are adhered to, is such as to limit the peak of the power curve to a piston speed of about 3000 ft. per minute, and this limitation applies alike to either side valve or overhead valve engines so long as the valves are in a single row. In order, therefore, to obtain the peak of the power curve at piston speeds well in excess of 3000 ft. per minute, it becomes necessary to arrange the valves in two rows. In racing-car engines it has now become the invariable practice to employ two rows of inclined overhead valves, each row operated by a separate overhead camshaft, thus allowing of the use of a centrally placed sparking plug between the two camshafts. This provides probably the most efficient form of head possible with any arrangement of poppet valves, but is open to the objection that it is complicated, costly, and inaccessible; for this reason it is seldom used except for purely racing purposes. The object of the "High Power" head is to reproduce as nearly as possible the above conditions, without resort to an overhead camshaft, and in a form which is relatively inexpensive and accessible.

The high-speed capacity and the high efficiency of the twin overhead camshaft engine are due to:

(1) The ample breathing capacity afforded by having the inlet and exhaust valves in separate rows, thus doubling the length available for valve accommodation.

(2) The central position of the sparking plug.

(3) The very short length of flame travel due to the central sparking plug and the absence of any corners or pockets where the gases may be entrapped and detonated, thus allowing of the use of a high ratio of compression.

(4) The low weight of the moving masses between the cams and the actual valve heads, thus allowing of high rates of acceleration and therefore of a very efficient valve opening diagram.

In the "High Power" type of head, these conditions are reproduced as far as possible in the following manner:

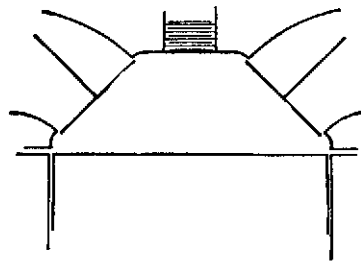


Fig. 55

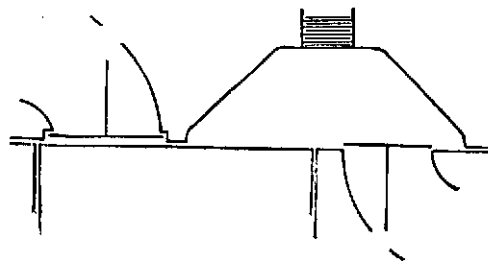


Fig. 56

(1) A double row of valves is obtained by placing the inlet valve overhead and operating it through a push rod and rocker; the exhaust valve, however, is retained in its usual position, as in a side-valve engine. In this there is, of course, nothing in the least novel.

(2) The sparking plug is placed in the centre, or almost in the centre, of the effective combustion space.

(3) The overhead inlet valve is situated in the flat "inoperative" portion of the combustion chamber, the latter is offset as in the ordinary turbulent head and embraces only the exhaust valve and a small proportion of the piston area; thus, the chamber, in plan, is similar to but considerably more compact than in the case of the ordinary turbulent head, since it embraces one valve only, and the length of flame travel is no greater than in the twin overhead camshaft type of engine. The whole combustion chamber, in fact, is very nearly similar to that of the racing engine, but is offset sideways in relation to the piston, as shown in figs. 55 and 56. The conditions as regards turbulence are practically the same as in the



ordinary turbulent head, and, of course, a "Shock-Absorber" pocket can be employed if desired. The location of the inlet valve entirely outside the operative portion of the combustion chamber is, of course, both the novel and the essential feature of this type of head.

(4) Since the inlet valve opens directly into the cylinder barrel and beyond the combustion chamber, there is no resistance in series with it due to the restricted throat, and conversely there is no need to limit the restriction, i.e. the degree of turbulence, to avoid throttling. Again, since we do not depend for turbulence on the entering velocity through the inlet valve, this valve can be made of almost unlimited size; moreover, since it is outside the combustion chamber, its size does not influence either the area of the chamber or the length of flame travel.

(5) In the case of the exhaust valve the mass interposed between the cam and the head is little or no greater than in the overhead camshaft type. In the case of the inlet valve it is, of course, considerably greater, but the inlet valve, unlike the exhaust, can be masked to any desired extent, and an efficient opening diagram can be obtained thereby with a relatively low rate of acceleration and therefore allow of a very high engine speed. Again, the inlet valve is not opened against a high back pressure so that the operating mechanism can be made considerably lighter than would be the case if the exhaust valve were placed overhead. Also, since the overhead valve is the cool one, no large clearance is required to allow for expansion.

Samples of this head were tested on the No. 2 research engine; these, at the same compression ratio and the same speed, gave approximately 7 per cent more power and 5 per cent lower fuel consumption than the best of the ordinary turbulent heads. Using a throat area which would give a rate of pressure rise of 40 lb. per square inch per degree in the ordinary turbulent head, the rate in the "High Power" head was only 29 lb., showing that the removal of the inlet valve to beyond the combustion chamber had reduced the turbulence. With this type of head, brake mean pressures as high as 122 lb. per square inch have been maintained up to well over 4000 R.P.M. on ordinary commercial petrol.

The high-power type of head has been adopted and is in use in several very high-speed engines on the Continent, where it has given results almost identical with those obtained from the conventional twin overhead camshaft type. Owing to its high inherent efficiency

and to the mechanical design, which allows of very ample water circulation round the exhaust valve seating and valve stem, it is particularly well adapted to supercharging; all the more so, since the short flame travel allows of the use of a fairly high ratio of compression even when supercharged.

#### **Determination of the Combustion Chamber Efficiency.**

—It has been shown that the efficiency of the combustion chamber is the most important of all the factors which control both the power output and the thermal efficiency of an engine. Unfortunately, however, it is, in most cases, very difficult to determine even the relative efficiency of two different designs of combustion chamber, in the case of multiple-cylinder engines, because irregularities in distribution play so important a part as to render a determination based on the known consumption of fuel per horsepower hour of very little value. It is very commonly found that of two engines the measured fuel consumption per horse-power hour is the same, despite the fact that one may have a considerably more efficient combustion chamber, and therefore develop more power than the other. At first, it would appear that the design of both engines is equally efficient, whereas in fact the former is much the more efficient of the two, but the practical value of its high inherent efficiency has been annulled by greater irregularities in distribution. Were it practicable to change the whole distribution system from one engine to the other a very marked change in efficiency would at once be observed; but this in practice is seldom, if ever, possible, because in almost all existing designs of multi-cylinder engines a considerable proportion of the induction passage is embodied in the cylinder casting.

The most practical method which the author has yet found for determining the true efficiency of a multi-cylindere engine independently of all irregularities in distribution, defective carburation, &c., depends upon the fact that in the case of any hydro-carbon fuel or alcohol the heat energy liberated by the combustion of any given weight of air is almost exactly the same, however much excess of fuel may be present, provided always that there is an excess of fuel, or, in other words, so long as the mixture is over-rich the thermal efficiency, based on that portion which is burnt, will be the same over a very wide range of mixture strength on the rich side, with the result that over the range from 5 per cent rich to 35 per cent rich the heat liberated by unit weight of air is constant to within extraordinarily narrow limits.

If, now, instead of measuring the weight of fuel consumed, we measure the air consumption and calculate the efficiency obtained in terms of lb. of air consumed per horse-power hour, we shall arrive at a means of determining the true thermal efficiency of an internal-combustion engine irrespective of any losses due to irregularities in distribution, defective carburation, precipitation of liquid fuel, etc. If, by air measurement, we find that the thermal efficiency of an engine is low, then we know that the combustion chamber design is at fault, and that no amount of juggling with the carburettor or distribution system will avail.

On the other hand, as so often happens, an engine apparently well designed and with what would appear to be an efficient form of combustion chamber shows on test a very poor fuel economy; in such a case, we can, by ascertaining the air consumption, determine at once whether the fault lies in the engine design or in that of its distribution system. Given reliable means for measuring accurately the air consumption, it is necessary only to set the carburettor to give a rich mixture, in practice to set it to the mixture giving maximum power (which on petrol is about 15 to 20 per cent rich) and take readings of air consumption; all that is needed being to ensure that the mixture strength is such that the weakest cylinder in the group is receiving a mixture not less than about 5 per cent rich. Since there is an extremely wide range available beyond this, i.e. up to about 40 per cent rich (after which combustion becomes delayed again as on the weak side), this presents no difficulty whatever.

The absolute value found for the thermal efficiency by this method is, in all cases, somewhat higher than that obtained from the fuel-consumption readings when working with the most economical mixture strength, and this is so even in the case of single-cylinder engines, where distribution losses can be almost entirely eliminated, the difference being due to :

(1) The larger increase in specific volume when an over-rich mixture is used.

(2) The slight loss due to condensation of liquid fuel on the cylinder walls which escapes combustion and ultimately finds its way past the piston into the crankcase.

(3) The small loss due to precipitation of liquid fuel on the walls of the induction pipe; this fuel enters the cylinder in gulps of liquid which are not completely vaporized, at any period during the cycle.

(4) The small loss of liquid or vapour due to "blow-back," or

reverse flow in the induction pipe due to the sudden closing of the inlet valve.

Once the air consumption is known, the thermal efficiency of any engine can be determined from the formula :

$$E = \left( \frac{\text{I.H.P.}}{\text{lb. of air per hour}} \right) \times C,$$

where C is a constant representing the amount of heat liberated by the combination of 1 lb. of air ; for all petrols it may be taken as 196.

The following table gives the effective calorific value, the mixture strength for complete combustion in terms of air/fuel ratio by weight, the amount of heat liberated by the combination of 1 lb. of air, and the value of C in the formula above, for a representative selection of fuels, from which it will be seen that over the whole range of available fuels the heat liberated by the combination of 1 lb. of air is substantially the same.

The figures given in the fourth column are calculated as for a

TABLE IV

Fuel.	Effective Lower Calorific Value B.Th.U.s per lb.	Air Fuel Ratio.	Heat liberated by 1 lb. of Air B.Th.U.s.	Value of Constant.
Petrol samples—				
(1) ... ..	19,200	15.05	1275	197.0
(2) ... ..	19,020	14.7	1295	195.0
(3) ... ..	19,120	14.8	1293	195.5
(4) ... ..	18,900	14.6	1295	195.0
(5) ... ..	19,090	14.9	1282	197.0
(6) ... ..	19,250	15.0	1285	196.5
(7) ... ..	18,920	14.7	1288	196.0
Kerosene ... ..	19,100	15.0	1275	197.0
Hexane ... ..	19,390	15.2	1275	197.0
Heptane ... ..	19,420	15.1	1285	196.0
Benzene ... ..	17,460	13.2	1320	192.5
Toluene ... ..	17,660	13.4	1315	193.0
Cyclohexane ... ..	18,940	14.7	1290	196.0
Heptylene... ..	19,320	14.7	1320	192.5
Ether ... ..	16,830	13.0	1295	195.0
Ethyl Alcohol—				
99 per cent ... ..	11,950	8.95	1333	190.0
95 „ ... ..	11,125	8.4	1330	190.5

mixture giving complete combustion, as also the value of the constant  $C$ ; but over a very wide range of mixture on the rich side, the variation is, in all cases, very small indeed, and appears to be almost exactly the same in the case of all volatile liquid fuels, so that the possible error due to variations in mixture strength is extremely small.

The following tables, Nos. V to VII, give the results of typical

TABLE V  
Petrol R=5:1, Sample No. 4  
Date of test, 5/7/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pres- sure, lb. per sq. in.	Indicated Horse- Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct ... ..	196.0	132.0	32.0	6.13	32.1
Plus 5 % excess fuel	196.5	135.0	32.7	6.0	32.8
" 10 "	197.0	136.5	33.1	5.95	33.1
" 15 "	197.5	137.5	33.3	5.93	33.2
" 20 "	198.1	138.0	33.4	5.94	33.15
" 25 "	198.8	138.0	33.4	5.96	33.05
" 30 "	199.5	137.5	33.3	5.98	32.85
" 35 "	200.0	136.5	33.1	6.04	32.7
Maximum thermal efficiency calculated from fuel consumption 32.1 per cent, with mixture 16 per cent weak.					

TABLE VI  
Ethyl Alcohol 99 per cent R=5:1  
Date of test, 27/8/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pres- sure, lb. per sq. in.	Indicated Horse- Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct ... ..	199.0	141.0	34.2	5.82	32.9
Plus 5 % excess fuel	199.5	143.0	34.6	5.77	33.3
" 10 "	200.0	144.5	35.0	5.72	33.65
" 15 "	201.0	145.5	35.25	5.70	33.7
" 20 "	202.0	146.5	35.5	5.69	33.75
" 25 "	203.0	147.0	35.6	5.70	33.7
" 30 "	204.0	147.3	35.7	5.72	33.65
" 35 "	205.0	147.6	35.8	5.73	33.6
Maximum thermal efficiency calculated from fuel consumption 33.0 per cent, with mixture 15 per cent weak.					

TABLE VII  
Ethyl Alcohol 95 per cent R=5:1  
Date of test, 16/8/21

Mixture Strength.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency.
					Per cent.
Correct ... ..	203.0	145.0	35.1	5.79	33.1
Plus 5 % excess fuel	203.5	146.7	35.5	5.72	33.6
" 10 "	204.0	148.0	35.8	5.71	33.65
" 15 "	205.0	149.0	36.0	5.70	33.7
" 20 "	206.0	149.5	36.15	5.70	33.7
" 25 "	207.0	150.0	36.25	5.70	33.7
" 30 "	208.0	150.3	36.3	5.71	33.65
" 35 "	209.0	150.5	36.35	5.74	33.4
Maximum thermal efficiency calculated from fuel consumption 32.9 per cent, with mixture 15 per cent weak.					

tests on petrol and alcohol (the latter both nearly pure and 95 per cent) over a wide range of mixture strength from correct to 35 per cent over-rich, taken on the variable compression engine described in Chapter II, while at the foot of each table the maximum thermal efficiency as deduced from the fuel consumption at the most economical mixture strength is given for purposes of comparison.

In Table VIII are given the results of air measurement tests at varying compression ratio, ranging from 4.0:1 to 7.0:1, the fuel used in this case being benzol. Here again the agreement between the calculated thermal efficiency from the air consumption with a rich mixture, and from the fuel consumption with a weak mixture, is on the whole very consistent.

Similar tests carried out on a six-cylinder aero-engine with a compression ratio of 4.7:1 showed an air consumption of 6.4 lb. of air per I.H.P. hour over a range of mixture strength from 10 per cent to 25 per cent over-rich, using standard aviation petrol. The air efficiency in this case works out at 31 per cent, a figure very considerably greater than that obtained from the fuel consumption, which was only 28.2 per cent at the most economical mixture strength. The discrepancy in this case represents the loss due to irregularities in distribution, etc. A comparison of these results with those given previously for the variable compression research engine is very interesting, all the more so because both engines have

TABLE VIII  
 Varying Compression Ratio  
 Fuel, Benzol, about 20 per cent rich  
 Date of test, 16/10/21

Compression Ratio.	Lb. of Air per Hour.	Indicated Mean Pressure, lb. per sq. in.	Indicated Horse-Power.	Lb. of Air per I.H.P. Hour.	Indicated Thermal Efficiency as found by Air Measurement.	Indicated Thermal Efficiency as found by Fuel Measurement 15 per cent weak.
4 ... ..	203	125.0	30.3	6.70	Per cent. 28.8	Per cent. 27.7
5 ... ..	195	136.5	33.1	5.90	32.8	32.0
6 ... ..	189	145.0	35.2	5.37	35.9	35.0
7 ... ..	185	152.0	36.8	5.03	38.3	37.3

nearly identical forms of combustion chamber, very nearly the same cylinder capacity, and, in both cases, the charge is ignited by sparking plugs at opposite sides of the combustion chamber. That the aero-engine does not show, by the air-consumption test, so high an efficiency as the research engine at the same compression ratio, namely, only 31 per cent as against 31.7 per cent, is to be explained probably by :

(1) Although two plugs were used they were sparked by two separate magnetos, and therefore not so accurately synchronized.

(2) The aero-engine had a relatively shorter stroke, so that the combustion chamber was flatter, and therefore somewhat less efficient.

(3) The mechanical condition of the aero-engine was probably not so good, i.e. there was probably more leakage loss.

The difference in fuel consumption was, however, much more marked, for the maximum thermal efficiency was 30.9 per cent reckoned on the fuel consumption at a compression ratio of 4.7 : 1 in the case of the single-cylinder research engine and only 28.2 per cent in the six-cylinder aero-engine. After allowing for the different combustion-chamber efficiencies of the two engines, the loss by irregularities in distribution, loss of unburnt fuel, etc., of the aero-engine is about 10 per cent, while in the single-cylinder it is only about 2.5 per cent. The indicated mean pressure was found to be exactly the same in both engines, namely, 133 lb. per square inch ; but measurements of volumetric efficiency showed that, while the research

engine had a volumetric efficiency at this compression ratio of 76.2 per cent, that of the aero-engine was 78 per cent, a difference which very approximately compensates for the lower combustion chamber efficiency of the latter engine. Again, tests carried out on a four-cylinder commercial vehicle engine with a compression ratio of 4.24 : 1 showed an air consumption of 7.75 lb. per I.H.P. hour = 25.6 per cent thermal efficiency as against  $6.58 = 30.4$  per cent in the variable compression engine at the same ratio, from which it may be deduced that the efficiency of the combustion chamber was only 85 per cent that of the research engine at the same comparison ratio. In this particular engine the design of the cylinder head was very defective, the valves being placed in deep-set shallow valve pockets with the sparking plugs directly over the inlet valves. The fuel efficiency at the most economical mixture strength was, however, about 23.8 per cent, showing that the efficiency of distribution, if such a term may be used, was, in the case of this four-cylinder engine, as high as 93 per cent. In other words, this engine made up, to some extent, so far as fuel efficiency was concerned, for bad cylinder design by having a quite unusually efficient distribution system, but the bad cylinder design showed itself in the low power output obtainable, the indicated mean effective pressure being only 93 lb. per square inch as against 129 lb. per square inch in the variable compression engine under exactly similar conditions. Had the volumetric efficiency been the same in both cases, the mean effective pressure would have been  $\frac{85}{100} \times 129$ , or about 110 lb. per square inch; that it was, in fact, only 93 lb. was due again to defective cylinder design, whereby the free entry of the gases after leaving the inlet valve was obstructed by the surrounding walls of the shallow valve pockets. Air measurements showed also that, while the volumetric efficiency of the research engine at this compression ratio, and at the same temperature, was 77 per cent, that of the four-cylinder engine was only 66 per cent, so that the maximum M.E.P. should have been  $\frac{66}{77} \times 110$ , or 94 lb. per square inch, a figure which agrees very closely with the 93 lb. actually measured during the tests. This latter is rather a striking example of an inherently defective engine, which showed a comparatively good economy because its induction system was unusually efficient.

In the absence of any means for measuring the air consumption, a fair estimate of the efficiency of different forms of combustion



chamber can be gained by comparing the maximum mean effective pressure, but this again assumes that the volumetric efficiency is the same in both instances.

Such an assumption is, of course, not always justifiable, but it is at least fair to assume that the variations in volumetric efficiency as between two somewhat similar types of engine will be very much less, and will have a much smaller influence on the determination of the combustion-chamber efficiency, than the variations in mixture strength, as between individual cylinders, despite the rather exceptional example quoted above. Where means are available for air measurement, the efficiency of any form of combustion chamber can very readily be determined from the measured air consumption.

If an engine consumes its air efficiently, then it is an efficient engine, and to render it economical in fuel is a question solely of carburation and distribution. If its air consumption is heavy, then no amount of finessing with carburettor adjustments or distribution design will render it efficient.

## CHAPTER V

### LUBRICATION AND BEARING WEAR

Three factors have to be considered in the design of any bearing : first, the ability to carry the necessary duty in the space available with a reasonable margin of safety against breakdown ; next, the rate of wear of the bearing surfaces ; and, finally, the energy lost in friction.

Piston friction will be dealt with in a later chapter, so that the bearings only will be considered here.

These are generally of the "plain" or sliding friction type. Ball and roller bearings, being of a fundamentally different nature, are not included.

Where two surfaces, apparently in contact, are moving relatively one to the other, there are three possible cases to be considered.

In the first case, that of "dry" friction, the surfaces are in actual contact without any lubricant. In this case the friction is very great, and only very low loads and speeds can be imposed without seizure. This case never occurs when a bearing is functioning properly, so need not be dealt with further.

The second condition, that of "greasy" friction, occurs when the surfaces, though virtually in contact, are actually lubricated with some substance which discourages their mutual adhesion. The lubricant in such cases appears to function by exerting some kind of chemical action on the metallic surfaces.

Lastly, there is "viscous" friction, in which the surfaces are completely separated by a film of lubricant : this is clearly by far the most desirable state of affairs, and, fortunately, it is one which can easily be attained in a well-designed and adequately lubricated bearing, and may, in fact, be regarded as the normal condition.

In a journal bearing the necessary oil film between, the loaded surface is maintained by a wedging action, due to the fact that the shaft sets itself eccentrically in its journal. This is made clear in fig. 57, where it can be seen that the oil in the wide space A is

dragged by the rotation of the shaft into the narrow space B, thus forcing the surfaces apart. Under these circumstances the frictional loss and the thickness of the oil film for any given conditions can be

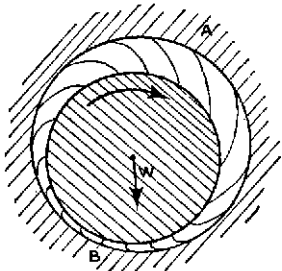


Fig. 57

evaluated theoretically, and it has been found that the results so obtained hold good in practice. These two factors—film thickness and friction—depend, for any given bearing, solely on the load, the speed, and the viscosity of the lubricant.

The influence of these three factors is roughly as follows: Increase of load, *alteris aequis*, increases the friction, though not nearly in direct proportion, and decreases the film thickness. Increase of speed increases both the friction and the film thickness, as does also an increase in the viscosity of the lubricant. However, increase of either load or speed, by increasing the rate of energy loss, heats up the oil, and thus decreases its viscosity.

This fact, in the case of an increase of load, to some extent nullifies the increased friction; it still further decreases the film thickness, while with increased speed the theoretically increased film thickness is actually reversed. Thus both increased load and speed tend to reduce the film thickness.

It is clear that, as no surfaces are perfectly smooth, there is a limiting thickness of oil film at which the high spots of the two surfaces begin to bridge across the oil film.

At these points the oil film is no longer of appreciable dimensions, so that the laws of "greasy friction" begin to apply. The frictional force, under these circumstances, is, at such points, considerably higher than with viscous friction, and obeys totally different laws. The chief factor appears to be a property, probably of a chemical nature, called "oiliness," which tends to reduce friction between two surfaces in contact. It is quite distinct from viscosity, as quite thin oils, such as sperm or rape, can have it in a far greater degree than more viscous substances, such as treacle, from which it appears to be entirely absent. Unfortunately the data on this subject is very limited and largely contradictory, but it is fairly well established that oils of animal or vegetable origin, such as sperm, rape, and castor oils, are considerably "oilier" than the hydrocarbon mineral oils, while viscous substances of a "sticky" nature are entirely devoid of the property.

It is clear that if the lubricant has but little "oiliness," the friction at the "high spots" will be exceedingly high, and that the heat so generated will still further reduce the thickness of the oil film, thereby making matters worse, and so starting a "vicious circle" ending in seizure, unless the bearing lining has so low a melting-point that it can fuse locally without boiling off the lubricant as in the case of white-metal.

With an oily lubricant, on the other hand, the increase in friction at the "high spots" will be much less, and the risk of seizure correspondingly deferred.

The influence of "oiliness" is thus only of importance where, owing to excessive loading, unsuitably disposed surfaces, or inadequate oil supply, a complete film of oil cannot be maintained. It has apparently no influence on the friction under normal conditions, though it may enable one to use, in any given case, a thinner oil, with correspondingly reduced friction, by relying, as a reserve in case of abnormal conditions, upon oiliness rather than upon excess viscosity.

To return to the consideration of normal conditions, the maintenance of an adequate oil film necessitates efficient arrangements both for the removal of the heat generated and for the continuous replenishment of the oil in the bearing.

The heat is got rid of, to some extent, by the oil which forms the actual oil film, but mostly either by conduction from the bearing surfaces, or by excess oil which runs through or over the bearing without actually forming the load-carrying film. The advantages of forced lubrication are mainly due to the cooling effect of the excess lubricant.

The supply of oil to the bearing is greatly facilitated by the "pumping" action caused by alternating loads, it being found that any given bearing will carry an alternating load considerably in excess of the maximum steady unidirectional load. Again, a narrow bearing loses a far greater proportion of its oil through side leakage than does a wide one, so is correspondingly less efficient when subject to "natural" lubrication, while with forced lubrication a narrow bearing will carry a heavier continuous load per square inch of projected area because of the greater quantity of oil which can be circulated through it and therefore the better cooling. It is also necessary that the surfaces should be of such a shape that the wedging action, which maintains the oil film at the highly loaded parts of the bearing, can take place, for the pressure which must exist in order

to carry the load is far in excess of any that a forced lubrication system can supply.

Fortunately, journal bearings are naturally suitable, as explained earlier, but with thrust collars special measures have to be taken, as in the Michel bearing. However, in practically all cases, in so far as high-speed internal-combustion engines are concerned, such thrust loads as are involved are most conveniently dealt with by the adoption of ball thrust races.

When conditions are too severe for even a molecular oil film to survive, the surfaces come into actual metallic contact with one another—that is to say, when the oil film becomes so thin that the two surfaces come within the zone of molecular attraction, an exceedingly high temperature is at once set up, resulting in local fusion of the surfaces. In some cases such local fusion may, by removing a high place in the surface of the bearing, relieve the loading at this particular point and so permit of the re-formation of the oil film. Unless this occurs, and the oil film re-forms immediately, the surface fusion will spread until the bearing either seizes solid or the bearing metal melts and runs out. When both bearing surfaces are hard and of anything approaching the same melting point, the surface fusion is generally accompanied by partial welding, and the two surfaces become inextricably locked. When the material forming one surface is relatively soft and has a low melting point, as in the case of white metal, the bearing merely melts, either locally or completely; in the former case, as already pointed out, local melting may be due to the existence of a high spot, and the fusion of this high spot may at once relieve the cause of trouble and permit of the restoration of the oil film in time to prevent any further spread of surface fusion: this is possible because the melting point of white metal is well below the boiling point of the lubricant. Local fusion is a very common occurrence in white metal lined bearings, especially when new and not too well fitted, and is often quite harmless.

The intensity of heat flow when an oil film breaks down locally is very surprising; it is quite common to find two case-hardened steel surfaces fused together locally and the temper of the surfaces undisturbed within less than  $\frac{1}{2}$  in. from the point of fusion. This corresponds to a temperature difference of something like 2500° F. in  $\frac{1}{2}$  in.

The case of bearings submitted to very heavy loads and low rubbing speeds is somewhat different. Here the time element

enters prominently into the question. Since at low speeds the wedging action, tending to maintain the oil film, is very slow, the effect of the pumping action, due to change of load, is of correspondingly increased importance. There is abundant evidence that under severe loads and favourable conditions even bronze bearings may be crushed without breaking down the oil film. So long as facilities are available for the replenishment of the oil, and so long as the rubbing velocity is such that the product of loading and rubbing velocity does not exceed a certain figure—that is to say, so long as the heat generated does not exceed the rate at which it can be dissipated by replenishment with cool oil or by conduction—a journal bearing will not fail from pressure. Failures of heavily loaded, slow-moving journal bearings are almost invariably due either to an interruption in the oil supply or more frequently to bending or distortion of one of the members, causing excessive local pressure and heat flow. The only bearings in a high-speed internal-combustion engine submitted to very heavy loading with low rubbing velocity are the gudgeon-pin bearings. Experience has shown that when the gudgeon-pin is supported in such a manner that it does not distort appreciably through bending, maximum pressures up to 6000 lb. per square inch may quite safely be carried, without forced lubrication and without perceptible wear.

**Wear of Bearings and Shafts.**—In view of the fact that most bearings are completely oil-borne, it is perhaps a little surprising at first sight that wear should take place at all, since the two surfaces never actually come into contact. The explanation appears to be that all wear is due to the abrasive action of small particles of grit carried by the oil. These particles, which are, for the most part, so small that they cannot be removed by filtration, are carried by the oil into the bearing and there embed themselves in the softer of the two surfaces; thus in a white metal lined bearing the particles of grit invariably embed themselves in the soft white metal. Once partially embedded they proceed to lap the shaft. It is clear that only those particles of grit which project far enough from the softer material to span the oil film and so actually touch the other member can cause wear. Other things being equal, therefore, the rate of wear depends upon the thickness of the oil film, which, in turn, depends upon the pressure and temperature—the cooler the lubricant, or the lighter the pressure, the greater is the thickness of the oil film and, therefore, the greater the distance across which the particles of grit must reach before they can come in contact with the harder member.

Again, the rate of wear depends upon the hardness of the surfaces of the materials. It is common knowledge that when a soft-steel shaft runs in white-metal bearings it is the shaft and not the bearing material which wears; this is perfectly natural, since the particles of grit will always tend to embed themselves in the softer surface of the two and so proceed to cut or lap the other and harder surface. In order to reduce wear it is desirable that the difference in hardness between the two surfaces should be as large as possible: thus in the case of a steel shaft running in white-metal bearings, the softer the white metal the more readily will the particles of grit embed themselves entirely out of harm's way, while the harder the shaft the less readily will it submit to the cutting action of such particles of grit as are not completely embedded.

All available evidence indicates that the once popular idea that a hard white metal should be used, and that its surface should be rendered even harder by hammering or driving a taper mandrel through it, is quite erroneous—the surface of the white metal should be as soft as is consistent with the necessary resistance to crushing. In the case of two very hard surfaces, such as case-hardened steel against cast-iron or hardened steel, very little wear occurs, the probable explanation being that the hardness of both surfaces exceeds that of the particles of grit, so that the latter are merely ground up between the two surfaces and do not get any opportunity of embedding themselves in either or of cutting them.

It has been observed that such bearing surfaces, even when freely exposed to road dust, etc., do not wear readily, but that when carborundum is introduced along with the lubricant very rapid wear takes place.

It is when two surfaces, both relatively soft and of somewhat similar hardness, are employed that the most rapid wear takes place. A soft-steel shaft running in bronze bearings wears very rapidly indeed, unless the load factor is so low as to permit of a very thick oil film being maintained.

One very striking example of excessive wear between two surfaces of nearly similar hardness is to be found in the case of copper aluminium alloys and phosphor-bronze. An alloy consisting of 88 per cent aluminium and 12 per cent copper affords an excellent bearing material for hardened-steel shafts. It is light, which is often important, is an excellent conductor of heat, and is readily cast and machined. When, however, a phosphor-bronze shaft is run in bearings of this aluminium alloy, the shaft, which is slightly

the harder of the two, wears away with almost incredible rapidity. Similarly, experiments which the author has carried out with bronze piston rings in an aluminium cylinder and with an aluminium piston in a bronze-lined cylinder resulted, in the former case in the rings wearing down to half their original thickness in eight hours, and in the case of the bronze-lined cylinder the liner wore about 0.010 in. oval in a run of twelve hours. In neither case did the aluminium show any appreciable wear.

Again, a soft-steel shaft, running in a copper-aluminium bearing, wears away very rapidly.

Probably the worst possible results are to be found when two similar and relatively soft materials are used for the two members of a bearing, for then not only is the difference in surface hardness reduced to zero, but the opportunities for welding together in the event of a failure of the oil film and consequent local fusion are at a maximum.

In the case of cast-iron or hardened steel, both surfaces are so hard as to be very little affected by grit, but, in the event of a breakdown of the oil film, the two are very liable to become welded.

Experience with aluminium pistons has shown :

- (1) That when these are fitted in soft-steel cylinders the cylinder bore wears very rapidly ;
- (2) When fitted in hard-steel cylinders—0.4 carbon—the wear is very slight ;
- (3) When fitted in cast-iron cylinders finished by grinding, wear of the cylinder bore takes place if the grinding material has not been thoroughly removed. Such wear does not take place when the cylinders are reamed or when lapped after grinding.

**Maximum Pressures on Bearings.**—So far as high-speed bearings are concerned—that is to say, when the rubbing velocity exceeds about 8 ft. per second—the load factor only, that is, the product of load and speed, need be taken into account ; the maximum pressure, so long as it is not high enough to distort or crush the bearing material, is of little moment, since it is not applied for a long enough period to have any influence on the conditions of lubrication.

**Limiting Load Factor.**—The highest load factor which can safely be carried by a bearing depends upon :

- (1) The system of lubrication, whether forced, natural, or fed with a measured quantity of fresh oil.
- (2) The viscosity of the oil.



(3) The facilities available for conducting away the heat generated in the bearing.

Under the conditions ruling in high-speed engines in which the circulation of oil is limited to that with which the piston scraper rings can cope, with forced lubrication, and good facilities for dispersing the heat, load factors as high as 30,000 lb. ft. per second can be safely carried in a journal bearing with alternating load. With such a load factor the rate of wear is of course considerable, but there are many examples of crankshaft centre bearings in aero-engines with load factors as high and even higher than this, but about 12,000 to 15,000 lb. ft. per second appears to be the upper limit of sustained load for a splash- or trough-lubricated bearing.

Where very high rubbing velocities are involved, much higher load factors can be carried when floating bushes are employed. Such bushes, floating freely between the two members, rotate at an intermediate speed, so that the rubbing velocity between either face is halved; also they permit of a much greater circulation of cooling oil through them. Under such conditions the load factor may be increased by 50 per cent without imperilling the bearing.

**Maximum Load.**—Where the rubbing velocity is low the only limit to the maximum load is set by the rigidity of the members. There is no danger of the oil film being broken down by pressure alone, provided there is scope for natural replenishment. When serious distortion takes place, the load factor may be increased locally—that is to say, the pressure may all be concentrated on one point in the bearing; and since the rubbing velocity is the same at all points, it follows that the product of pressure and rubbing speed may be excessive at one point, causing rapid local heat flow and ultimate breakdown of the oil film.

**Load Factor and Wear.**—Other things being equal, the rate of wear may be taken as being almost directly proportional to the load factor. It depends, however, very largely on the facilities for cooling; thus, in the case of crankshaft bearings, for equal load factors with forced lubrication the rate of wear on the main journal bearings is more rapid than on the crankpin bearings, because the crankpin bearings are always better served and therefore cooled with oil, since it is flung out to them by centrifugal force.

**Oscillating Bearings.**—When the motion is oscillating only, the wear on the members is no longer uniform; this objection can, however, usually be overcome by allowing the harder member to float freely—thus, in the case of a gudgeon pin, if this is fixed, either

to the piston or the connecting-rod, local wear will take place, but if allowed to float freely in bearings, both in the connecting-rod and piston, local wear on the pin can be avoided; further, a much heavier load can be carried, because the rubbing velocity between any of the members is halved.

From the above considerations, it is clear that, other things being equal:

(1) The friction of a bearing, when freely lubricated, is nearly proportional to the load factor on the bearing, and depends, though to a lesser extent, upon the nature of the surfaces—the smoother the surface the lower the friction.

(2) The rate of wear is also proportional to the load factor.

(3) When oil of higher viscosity is used, the friction is greatly increased at first; but on account of the greater amount of energy dissipated in shearing the oil film, the heat flow is greater, the temperature is therefore higher, with the result that, after running some time, the reduction in viscosity due to the higher temperature nearly compensates for the higher initial viscosity, and so the conditions as regards friction and the thickness of the oil film ultimately become nearly similar. They do not become quite similar, because owing to the higher temperature of the bearing the rate of dissipation by radiation and conduction is greater, consequently the bearing never reaches so high a temperature relative to the viscosity of the oil as when a thinner oil is used.

**Types of Oils.**—The oils used in internal-combustion engines fall into two main divisions—mineral oils and those of animal or vegetable origin. “Compounded” oils, and mixtures of the two types, are also used.

*Mineral oils*, which are composed of various hydrocarbons, mostly of the paraffin series, are by far the most frequently used. This is partly because of their lower cost, and also on account of their chemical stability, which renders them less prone either to carbonization or to oxidation or gumming. On the other hand, they do not appear to possess the property of “oiliness” to the same extent as vegetable or animal oils. Except in cases where bearings are very severely loaded and near their limit, mineral oils are probably the most suitable, and so far as the author has been able to determine by experiment there is no advantage to be gained from the use of vegetable or compounded oils in any well designed and properly lubricated engine.

For ball or roller bearings a pure mineral oil would appear pre-

ferable, since such an oil is less liable to form corrosive acids in service.

*Animal oils*, such as whale and lard oils, and vegetable oils, such as rape or castor oils, are largely composed of the esters of fatty acids. Their chief virtue lies in their high "oiliness," which is of use in cases where the oil supply is necessarily limited, as in crank-case compression two-stroke engines, or where severe local over-loads, due to distortion, etc., are probable. Their defects lie in their comparative instability, which renders them liable to become gummy and acid by exposure to the air, and also causes them to carbonize more rapidly than mineral oils. They are also expensive, and, the supply being necessarily limited, would become more so if their employment became general. Their use, therefore, should be, and generally is, limited to exceptionally high-duty engines, and a few other special cases.

### Carburation

The function of the carburettor is not, as is so often supposed, to gasify the fuel, but rather to provide constant proportions of finely divided liquid fuel and air under all conditions of speed or load. The gasification or vaporization of the liquid fuel takes place in part in the induction system, and in part in the cylinder of the engine.

The requirements of a good carburettor are that it shall—

- (1) Provide a constant predetermined ratio of fuel and air at all speeds and at all loads, under constant conditions.
- (2) Pulverize the fuel as finely as possible under all conditions.
- (3) That when the throttle is opened suddenly it shall provide, momentarily, an over-rich mixture, for reasons which will be explained later.
- (4) Provide an over-rich mixture for starting or running idle at very slow speeds.
- (5) Be provided with automatic or at least readily controllable means of enriching the mixture throughout the whole or at least the lower part of the range, until the carburettor and induction system are fully warmed up.
- (6) Be simple to adjust in the first instance and unlikely to get out of adjustment in use.

Probably no carburettor yet made conforms to all these conditions, though they are not impossible of compliance.

It is worth while to examine each of these conditions separately and to see what their compliance involves.

The first consideration, namely, that of providing a uniform mixture strength under all conditions, or of "metering," as it is generally termed, is the basic problem in carburettor design. The simplest possible expression of a carburettor is a jet to which liquid fuel is supplied at a constant level, such jet being situated in the centre of a venturi nozzle, or choke-tube, through which the whole

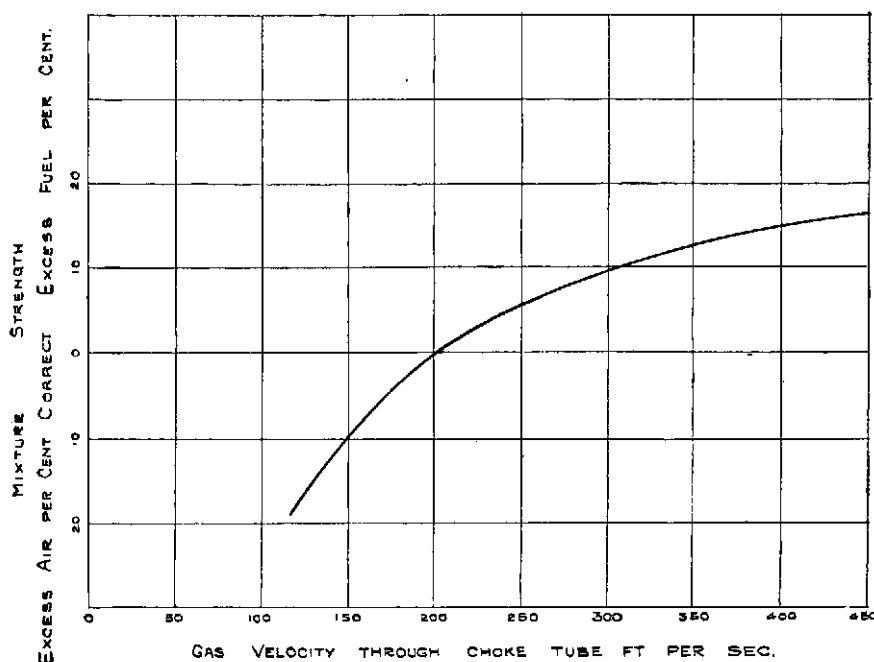


Fig. 58

of the air passes on its way to the engine. The depression in the choke-tube is therefore at all times a function of the I.H.P. of the engine, and this depression is relied upon to draw petrol from the jet. Unfortunately, the laws governing the flow of liquid from a jet, and of air through a venturi throat, are not the same, for the one medium is a liquid and the other is a gas. As the speed at which the air flows through the choke-tube increases, so the flow of fuel also increases, but at a considerably greater rate, with the result that, if the proportionate sizes of jet and choke-tube diameters are chosen to give a "correct" mixture at any one speed, the mixture will be too weak at a lower speed and too rich at a higher speed, as shown approximately in fig. 58.

To this simple form of carburettor some means of compensation must therefore be provided, and for most purposes it must be automatic. There are numerous methods of compensating, but they may be divided broadly into two main groups:

(1) In which means are provided for supplying, automatically, additional air as the power output increases.

(2) In which means are provided for supplying automatically additional fuel as the power output decreases.

Intermediate between these groups are methods providing means for checking the flow of fuel through the jet by obstructing it by means of a reversed air-flow, etc.

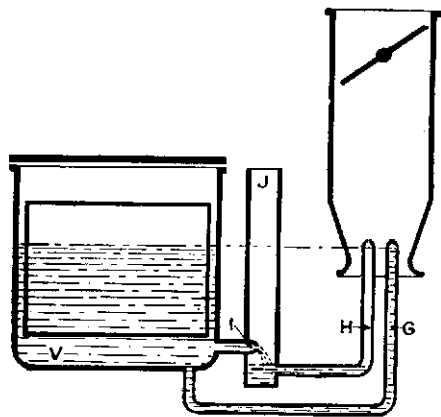


Fig. 59.—Diagrammatic Arrangement, Zenith Carburettor

The first group includes all such devices as automatic extra air valves operated by suction. Broadly speaking, these are not very satisfactory, because they involve the addition of a constantly moving part, which cannot readily be lubricated, and unless the movement of this part is controlled by an efficient dash-pot serious wear is liable to occur; on the other hand, if it is controlled by a dash-pot, then its movement will be somewhat sluggish, though this is not

necessarily a disadvantage in view of the third requirement stated previously. In any case, however, it is always desirable to avoid the use of an additional moving part if possible.

The second group includes those carburettors in which compensation is effected by means of an additional jet fed by gravity from the float chamber and open to atmosphere; the flow of such a jet is unaffected by the depression in the choke-tube. Carburettors belonging to this group can be adjusted to give fairly accurate metering under all conditions of speed or load; and since they contain no moving parts to wear or possibly to stick, they are, in the author's opinion, to be preferred. The basic principle of this type of carburettor is illustrated in fig. 59, while fig. 60 shows approximately the rate of flow from either jet. It will be seen that, as the power output and therefore the depression in the choke-tube increases, the flow of liquid from the main jet increases rapidly, while

that from the gravity-fed jet, which has a constant head of only about one and a half or two inches, remains substantially constant. Its proportional flow therefore decreases. It is obviously possible by a suitable selection of jet sizes to keep the fuel air ratio very nearly constant over a wide range of speed and load.

In addition to these general groups there are large numbers of purely mechanical devices whereby either the fuel supply, the air supply, or in some cases both, are varied mechanically by the move-

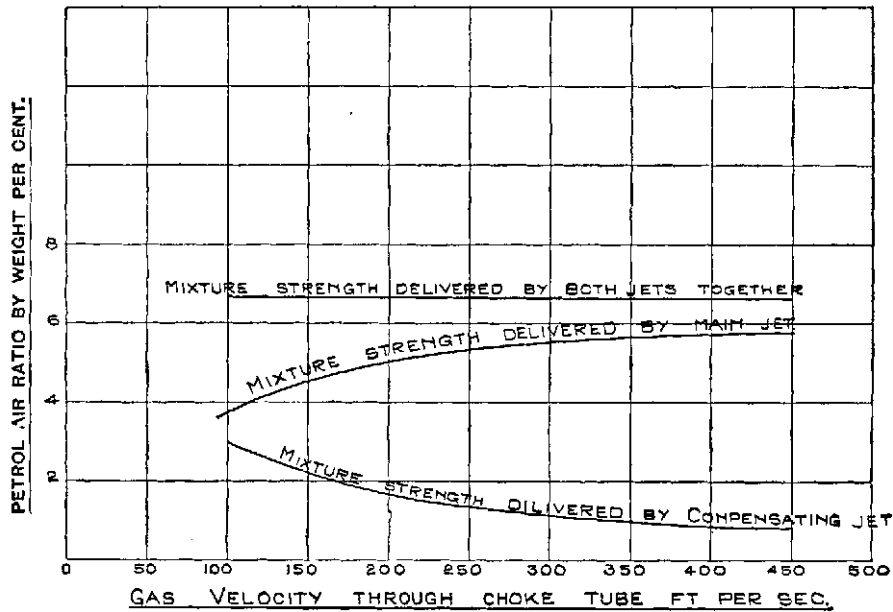


Fig. 60.—Mixture Strength delivered from Main and Compensating Jets, Zenith Type Carburettor.

ment of the throttle level. Such devices cannot possibly afford true compensation for all conditions of speed or load, since no change can be effected without movement of the throttle. For certain purposes, however, such, for example, as marine work in which the torque and speed vary in a fixed relation, mechanically compensated carburettors are probably quite satisfactory. The advantages of this form are that a mechanically compensated carburettor can be made very cheaply; it has only one jet to look after and no adjustment which can be deranged. It will fulfil the requirements of a marine engine, but certainly will not give accurate metering when applied to engines in which the speed or torque may vary without movement of the throttle.

The second condition, namely, thorough pulverization, is not at all easy to comply with. It is, however, an exceedingly important factor, because it is most desirable from every point of view to keep the suction temperature as low as possible. Whether the fuel enters the cylinder in a liquid or a gaseous state, it will, so long as it is finely divided, be completely evaporated after its entry to the cylinder, on coming in contact with the hot residual products therein.

In a previous chapter it has been shown that it is always very desirable to keep the suction temperature as low as possible, and to this end it is often preferable to allow the fuel to enter in a finely divided but still liquid state, and so to make use of the latent heat of evaporation of the liquid to lower the temperature in the cylinder. This, however, is possible only when the liquid is very finely pulverized, and when the induction system is so arranged that reasonable uniformity of distribution can be obtained without too much differentiation due to the unequal inertia of air and liquid particles; if delivered in coarse drops, these will coalesce and precipitate on the walls of the induction system, finally entering the cylinder in gulps of liquid, which will never evaporate. These pass through the cylinder unburnt, while a proportion of the liquid fuel will find its way past the piston and into the crankcase, where it will contaminate the lubricant.

In most carburettors the velocity of the air past the jet is relied upon to pulverize the fuel, and, for this purpose, every effort is made to obtain a high velocity at the least possible sacrifice in power output.

Unfortunately, however, pulverization becomes of most importance when the engine is running at low speeds or reduced loads, i.e. when the quantity of air passing, and therefore its velocity, are at a minimum. With a venturi orifice of the best possible design the volumetric efficiency, and therefore the power output of the engine, will be penalized severely if the air velocity exceeds 400 ft. per second, and the author has found that to obtain a good compromise between maximum power output on the one hand and good economy on reduced loads on the other, the mean velocity through the choke-tube should not exceed 400 ft. per second when four cylinders are drawing from a single carburettor, 330 ft. per second for three cylinders, and about 250 ft. per second for single cylinders, the lower velocity in the latter cases being permissible because:

(a) With less than four cylinders the suction is intermittent and the maximum velocity therefore considerably greater.

(b) The fewer the number of cylinders drawing from any one carburettor the shorter the total length of induction pipe.

To obtain better pulverization one or other of two methods may be employed:

(1) A very small choke may be used and the bulk of the air admitted elsewhere, the bulk supply of air being cut off as the load

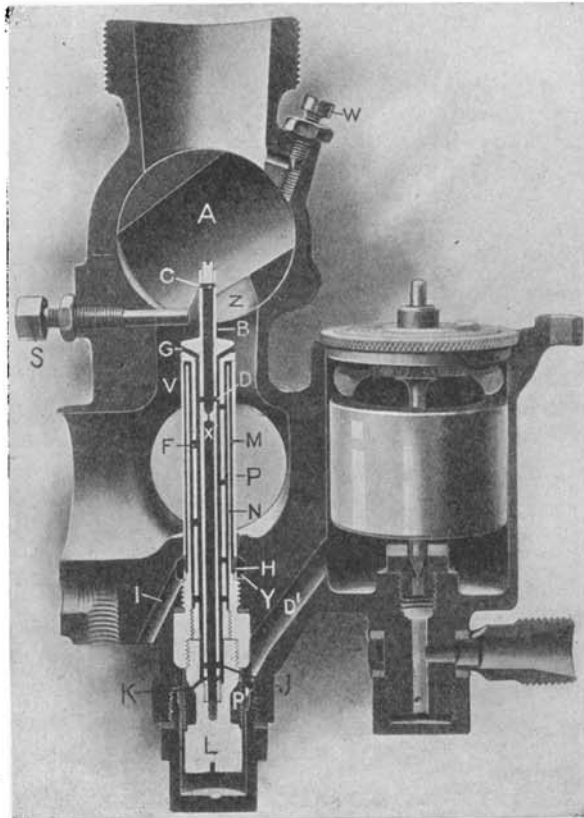


Fig. 61.—Section, Claudel Carburettor

is reduced—this entails a combination of mechanical and fluid compensation.

(2) What is termed a shrouded or diffuser jet may be used in which air is drawn through the liquid to form an emulsion, which is then delivered from the jet, as employed in the Claudel carburettor, figs. 61 and 62.

The former has the advantage that it becomes possible at reduced loads to maintain the velocity not only past the jet, but also throughout a considerable proportion of the induction system. It carries



with it, however, the disadvantage that the carburettor becomes somewhat complicated.

The diffuser or shrouded jet gives good pulverization at the jet itself, but, owing to the low velocity in the whole of the induction system at light loads or low speeds, particularly the latter, the finely divided particles are given too much opportunity to coalesce. It carries with it, however, the additional advantage that the flow of air through the fuel in the jet tends to effect a certain, though limited, measure of compensation.

For the carburation of all engines liable to sudden demands, such

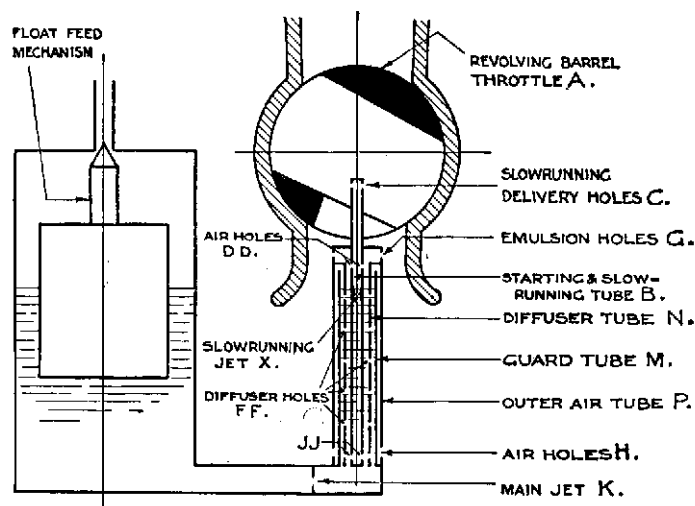


Fig. 62.—Diagrammatic Arrangement, Claudel Carburettor.

as all road vehicle engines and all engines under the control of a sensitive governor, it is most important that, on the sudden opening of the throttle, the carburettor shall deliver momentarily an over-rich mixture. The reason for this is as follows:

When an engine is running light or at a very much reduced load the pressure in the induction system may be only about 5 lb. per square inch absolute. At this pressure and even at quite low temperatures almost the whole of the fuel flowing through the induction system will be evaporated and the walls of the induction passages will be dry. If now the throttle be opened suddenly the pressure in the system will at once rise from, say, 5 lb. per square inch to nearly 15 lb. per square inch absolute, while the temperature conditions will remain unaltered. Now, although the fuel may evaporate completely when under a pressure of only 5 lb. per square inch,

when the pressure is raised by nearly three times, this will no longer be the case, unless the induction system be very hot. The immediate result is that the first portion of the fuel admitted after the throttle is opened deposits at once on the walls of the induction system, and, unless the mixture supplied by the carburettor is very rich, that which reaches the cylinders is far too weak to burn; this state of affairs will continue until sufficient fuel has been supplied thoroughly to wet the walls of the induction system, for when working under full throttle conditions the whole of the walls are normally lined with a layer of liquid, the thickness of which depends both on the temperature of the walls themselves and upon the velocity of flow through them. The practical effect of this is that, when the throttle is opened suddenly after idling, the engine will splutter and backfire or even stop firing altogether for a few revolutions; if now the throttle be closed again, the half-formed wet layer will immediately re-evaporate and the engine will run steadily again; by repeatedly opening and closing the throttle the necessary wet layer can be built up gradually. To obviate this difficulty one or other of three expedients must be adopted:

(1) The carburettor must be set to deliver, at all times, an over-rich mixture.

(2) The walls of the induction system must be maintained at so high a temperature that little if any liquid fuel can lie upon them, even at atmospheric density.

(3) The carburettor must be provided with some means whereby a little liquid fuel is stored up when running idle, and delivered to the induction system immediately the throttle is opened.

The first of these may be dismissed at once as altogether too extravagant. The second method can, at best, be but a partial remedy only, though a certain amount of pre-heating is essential in the case of present-day petrols, whose mean volatility is low and most of which have a final boiling point of over 400° F. To raise the induction system, however, to such a temperature as will prevent entirely any condensation even of the highest boiling fractions is practically out of the question, and would, in any event, so reduce the power output, increase the tendency to detonate, and, by raising the whole cycle temperature, so lower the efficiency of the engine, as to be quite outside the range of practical politics.

The third method, namely, the momentary supply of an over-rich mixture, meets the case satisfactorily; it costs nothing in power output, and permits both of working normally with the most

economical mixture strength, and of reducing the heat input to the induction system.

In the case of carburettors using a gravity-fed compensating jet, this condition can be met by providing a well having a capacity sufficient to supply a 100 per cent excess of fuel for, say, 3 or 4 cycles, and fed from the compensating jet. When running on full throttle this well is normally dry, but when idling the well fills up to nearly the level in the float chamber. So soon as the throttle

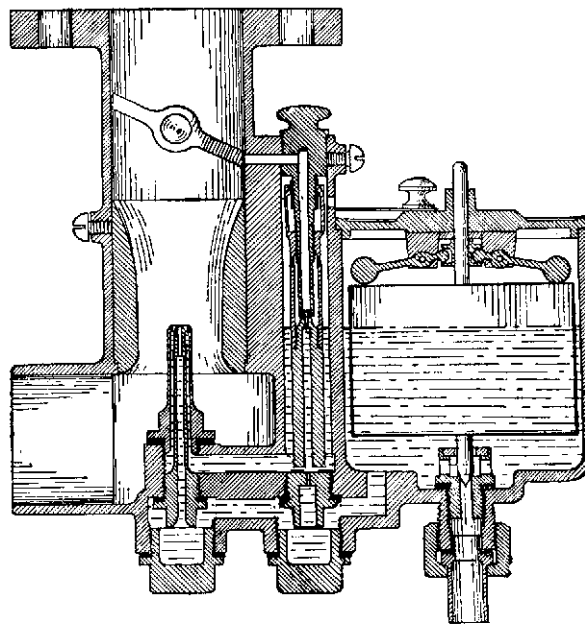


Fig. 63.—Sectional Arrangement of Zenith Carburettor, showing details of Pilot Jet

is opened the sudden depression caused by the inflow of air to the induction system draws the whole of the contents of the well into the induction system and thus provides, momentarily, an over-rich mixture.

When a shrouded or diffuser jet is used the same effect can be brought about by providing in the annular passage around the jet sufficient capacity to meet this instantaneous demand.

Both the Zenith and the Claudel carburettors cater for this condition, the former by means of a well fed by gravity, as shown in fig. 63, and the latter by the use of a diffuser jet with large capacity. Neither, however, provides, in the author's opinion, sufficient capacity to meet the case of a four-cylinder engine with a relatively cool

induction system, and both could certainly be improved by the addition of a small plunger pump actuated mechanically by the throttle lever.

The fourth condition, namely, that it shall provide an over-rich mixture for starting, is met, in nearly all cases, by the provision of a separate, or pilot jet, and some provision is usually made whereby this jet drops out of action so soon as the engine has attained sufficient speed to bring the main jets into operation. In those carburettors using a gravity-fed compensating jet, the pilot is fed from the compensating jet and automatically drops out of action so soon as the level in the well falls beyond a certain point.

To start from cold, i.e. 60° F., on ordinary commercial petrol, it is necessary to provide a minimum mixture strength from three to four times over-rich, depending upon the vapour tension of the fuel and upon the actual temperature of the mechanism. For the purpose of starting merely, there is no objection to using an excessively over-rich mixture, provided that its supply is discontinued so soon as the engine is under way. When the pilot jet draws from a well fed by the compensating jet and is but slightly submerged, as in the Zenith or Claudel carburettor, it will deliver an excessively rich mixture only when the level is at a maximum, and this again can occur only when the engine is at rest, for the very minimum running speed will suffice so to lower the level in the pilot jet chamber as to cut this jet either partially or completely out of action. When, however, the pilot jet is fed direct from the float chamber, it is necessary, in order to obtain a sufficiently rich mixture, either to flood the carburettor and so both raise artificially the level in the pilot jet and at the same time expose for evaporation a considerable surface of liquid petrol in and around the air intake to the carburettor, or partially to close the air intake to the carburettor and thereby subject the main jet to excessive suction.

The fifth condition, namely, adjustment of mixture strength to compensate for changes in temperature, is met only in those carburettors which are provided either with variable jets or with a control on the air supply. It is an important condition, but one which is generally unprovided for; indeed, it is not at all an easy one to meet adequately. Although an engine should require the same mixture proportions, once the induction system has been wetted, whether it is hot or cold, yet it is none the less desirable to reduce slightly the size of the jet as the carburettor warms up, because the viscosity of petrol and indeed of most volatile liquid fuels varies considerably

with temperature, and, with a given size of jet and a given depression, the quantity of fuel passed will increase as the temperature of the liquid rises, due to the reduced viscosity. The influence of the variation in the viscosity of the liquid can be reduced to the minimum by employing always very short jets or at least a very short length of orifice of small diameter, but it cannot be eliminated entirely by such means.

In all carburettors used for aircraft, and which are therefore required to operate under wide variations both of density and temperature, it is absolutely essential to provide some means of varying bodily the mixture strength. This is effected by means of what are termed altitude controls. Altitude controls take several different forms, but the most usual is that in which the float chamber is hermetically sealed and the air space in it connected to the carburettor at two points, one below the choke-tube and the other between the choke-tube and the throttle. Between these two points there is a considerable difference in pressure, due to the resistance offered by the choke-tube. A control cock is fitted in the passage connecting above the choke-tube, and this passage is made very much larger, the other being in the nature of a small permanent leak only. When the control cock is closed the pressure in the float chamber is equal to that of the outside air, since it is balanced by the permanent leak, but on opening the control cock the pressure in the float chamber falls to something between that ruling in the passage above the choke-tube and that of the outside air, depending upon the respective areas of the connecting passage-ways. As the pressure in the float chamber is reduced, so the level in the jet falls, and less liquid is delivered from it. By adjustment, therefore, of the control cock it is possible to lower the level of liquid in the jet to any desired extent, and so to vary the mixture strength. This method and the several variations of it, serve admirably for aircraft when the engine is driving a propeller and the torque-speed characteristic is therefore definite throughout the range; but it is in the writer's opinion somewhat doubtful whether it forms a suitable means of control for such purposes as motor vehicles, owing to its influence either on the gravity-fed compensating jet or upon the diffuser when such jets are used. For carburettors with mechanical compensation it would no doubt work admirably.

The sixth condition, namely, that it shall be simple to adjust in the first instance, and unlikely to get out of adjustment in use, is a very important one indeed, because in practice it is often very

difficult to determine whether the metering is correct or not—too weak a mixture, either throughout or at any point in the range, shows itself at once by backfiring into the induction system, but it is not always so easy to make sure that the mixture is not too rich. Carburettors in which a gravity-fed compensating jet is used are always rather difficult to adjust accurately in the first place unless the engine is run under conditions wherein its fuel consumption, etc., can readily be determined at all loads and speeds, as on the test bench. On the other hand, such carburettors, once adjusted, will remain in adjustment for all time.

Carburettors with moving parts controlling the compensation are, on the whole, much easier to adjust in the first instance, but are liable, owing to wear, leakage, etc., to lose their adjustment. Finally, there are, on the market, several carburettors which, though they do not under any circumstances give correct metering, are none the less so easy to adjust to give a fair average proportionality, that in unskilled hands they are often found to yield results as good as or better than those of more scientific design.

As stated previously, there is, so far as the author is aware, no single carburettor which conforms to all the conditions he has laid down; but despite this, a good modern carburettor when properly adjusted is a remarkably efficient piece of apparatus. Though often much abused, the fault more often lies with the distribution system than with the carburettor itself.

It must be remembered always that the carburettor and the distribution system are closely interdependent. When the distribution system, either by its large exposed surface, the low velocity maintained in it, or its low temperature, is such as to encourage precipitation, then it is desirable to use a carburettor which will give very thorough pulverization, even at the expense of some loss by wire-drawing, and which will supply a large excess of fuel when the throttle is opened suddenly. On the other hand, with a different distribution system quite other characteristics may be desirable. It is quite wrong to suppose that any standard carburettor may be tacked on to any existing distribution system without regard to the particular characteristics of either.

**Ignition.**—By a process of elimination ignition systems for high-speed internal-combustion engines have been reduced to two main types:

- (1) The high-tension magneto.
- (2) The high-tension coil and battery system.

The former was in almost universal use until a few years ago; the latter is a reversion to an earlier type, and has come into prominence again because nearly all vehicles and aircraft are now equipped with electric-lighting sets, so that an ample supply of low-tension current is always available.

*Intensity of Spark.*—Although when working on full throttle with a “correct” or slightly over-rich mixture the intensity of the spark is of very little importance, yet on reduced loads or weak mixtures the intensity plays a very prominent part. It has been found experimentally that when the nature, conditions, or consistency of the fuel/air mixture are such as to yield very rapid burning, then a very feeble spark will suffice. So far as maximum power output alone is concerned, the intensity of the spark does not play any serious part; it has, however, a considerable influence upon the maximum efficiency attainable, since it governs, to some extent, both the range of burning on the weak side and the rate of burning when throttled.

In the ordinary high-tension magneto, the intensity of the spark is at a maximum only at the point of maximum flux, and is therefore reduced when the time of ignition is either advanced or retarded beyond this point. In the coil and battery system, on the other hand, the intensity remains the same irrespective of the time of ignition. This is a substantial argument in favour of the latter system.

In the past, the coil and battery system was superseded by the high-tension magneto on the score of reliability for two reasons:

(1) Because there was no charging dynamo available, with the result that the battery was not kept fully charged and was frequently allowed to run down, with consequent complete failure of the ignition system.

(2) The early low-tension contact-breakers were generally ill designed and badly made, and gave continual trouble.

When the high-tension magneto first appeared on the market it had two substantial advantages over its rival system: namely, a constant supply of low-tension current, and a really well-designed and well-made low-tension contact-breaker.

These advantages are not so prominent to-day, for nearly every high-speed engine is equipped with a charging dynamo for lighting and starting.

It is not proposed to discuss the details of either system, for these are now well known and can be found in any text-book on the subject.

*Sparkign Plugs.*—Probably no part of an internal-combustion engine is more complained of and abused than the sparking plug, though the complaints levelled against it are often unjust. It is generally complained either that sparking plugs oil up and so become inoperative, or that they give rise to pre-ignition; but the fault quite as often lies in the choice of an unsuitable plug for the engine or in defective design, with the result that too much oil passes into the combustion chamber.

In any type of sparking plug there is a limited range of temperature between which the points will be sufficiently hot to burn off any oil which may be deposited upon them, and sufficiently cool to avoid pre-ignition.

In those engines in which, owing to defective piston design, the quantity of oil passing the piston is excessive, it is necessary, in order to burn off the oil, to employ a type of plug with thin points which will keep hot when running on reduced loads. In this connection it should be remembered that, in a throttle-controlled engine, the actual temperature of the working fluid is nearly the same at any throttle opening, and that it is the total quantity of heat and not the temperature which varies. If the plug points are thin and their facilities for getting rid of the heat are poor, they will attain a temperature corresponding to the mean temperature of the cycle irrespective of the quantity of heat liberated. If, on the other hand, they are provided with good facilities for getting rid of the heat imparted to them, then their temperature will be governed rather by the quantity of heat, but will at all times be somewhat lower. In other words, in plugs with long thin points the points will always be hot but will remain at much the same temperature at any load; while in plugs with short, thick points and good facilities for getting rid of their heat, the points will keep cooler at all loads, but their temperature will vary over a wider range as the load is varied. From the point of view of maintaining an equable temperature at all loads, comparatively long, thin points are preferable, and so long as the compression is low and there is no detonation to increase the temperature and rate of heat flow they will not give rise to pre-ignition. But when the compression ratio is high and the proportion of diluent is therefore small, pre-ignition will occur the more readily; moreover, under these circumstances detonation is the more liable to occur, and this, as has been shown previously, will give rise to overheating of the plug points. For such engines, therefore, it is necessary to use a type of plug whose points will keep as cool as possible. When,



by careful design, the flow of lubricating oil to the combustion chamber is reduced to the lowest limit, it becomes possible to use a "cool" plug without trouble from oiling up, and it is then preferable to do so, for the thick points naturally last longer, and there is less risk of overheating from momentary detonation or other causes.

For low-compression engines or for engines whose duty is comparatively light a "hot" plug may be used with advantage, more especially if they have any tendency to pass an excess of oil; while for high-compression engines a "cool" plug must be used, and the tendency to oil up must be overcome by adequate piston design. No single plug can at present be made to suit a high-compression engine which passes excess oil into the combustion chamber, but the remedy lies in the design of the engine, and it is not fair to abuse the sparking plug because it is being called upon to meet conditions outside its legitimate range.

## CHAPTER VI

### MECHANICAL DESIGN

In the design of an internal-combustion engine, as in all creative work of this nature, the æsthetic side must not be overlooked. In the first place, beauty of form and of proportions is in itself an admirable guide to mechanical correctness; for mankind has come to regard as beautiful that which is mechanically correct, whether it be in nature, in architecture, or in engineering.

In general, beauty and efficiency—in the widest sense of the term—are synonymous, and the appeal of any design to the æsthetic sense is often as reliable a guide as is a mathematical analysis of its mechanical features. Again, the æsthetic side makes a powerful though an unconscious appeal to the user, whose artistic sense, mute and inarticulate though it may be, will always be roused.

The designer's first aim should be to ensure that the products of his work will receive the care and even affection which he hopes will be bestowed upon them by their users. To this end he should make an appeal to them through their artistic sense rather than to fads or fashion, for the former is innate in all mankind, while the latter may vary widely.

There is a prevalent but quite erroneous belief that the reliability and even the efficiency of an engine are, to a large extent, a function of the actual number of parts it contains. Speaking generally, there can be no greater fallacy. While it is obvious that the number of parts should be kept down to the minimum compatible with efficiency and mechanical correctness, this can very easily be overdone. Fewness of parts too often denotes excess of compromise. All design must necessarily be based on compromise, and it is upon the soundness of judgement by which the compromise is arrived at that the success of an engine ultimately depends. In an internal-combustion engine many of the parts are subjected to complicated stresses, both heat stresses and pressure stresses; and when, by multiplying

the number of parts, such stresses can be reduced or dealt with separately, this should be done unhesitatingly. No member should be subjected to compound stresses if by the provision of additional members these can be split up; for example, when a member is subjected to combined torsion and bending it is preferable, where possible, to replace it by two separate members, one designed to deal with the bending alone and free from torsion, and another subject to torsion only and free from bending. To do this may involve the introduction of perhaps six or eight times as many parts in this particular piece of mechanism, yet the safety and reliability will be many times greater. Again, in the design of an engine the problem often arises of carrying a shaft in two bearings whose perfect alignment with each other cannot be ensured in machining or is liable to disturbance in use owing to distortion, &c. In such cases the choice lies between fitting a fairly flexible shaft which will accommodate itself by flexure to slight errors in alignment, or the provision of double universal joints between the two bearings. In the former case the shaft is liable to fail ultimately from fatigue through constant flexure, however slight, while the bearings are liable to give trouble, and in any case the friction will be much greater. In the latter case safety and the minimum of friction are ensured, but at the cost of several extra parts. A choice of this nature confronts every designer almost daily, and he has to decide whether he will risk the simple expedient or resort to the more complicated one. He is generally too liable to adopt the former course on the ground of manufacturing cost, but in such case he has no right to boast that he is using fewer parts—he is doing so only because he cannot afford to use more, or because he has neither the knowledge nor the experience to appreciate the risk he is incurring. Again, in the larger sizes of engine it is often desirable to duplicate the exhaust valves, even though this may involve the duplication of the whole valve gear also. By doing so, smaller valves can be employed, and both the temperature of the valve heads and the stresses in the valve gear will be reduced, the valves will remain in good condition for a far longer time, while the margin of safety in the valve gear will be greatly increased. The net result of duplicating the exhaust valves will be that the engine will retain its efficiency for a much longer period, and will at all times be more reliable. It is better far to use 500 parts if need be, to comply with the laws of sound mechanics, than to defy them with a single part. The craving to make one single member

perform several distinct functions is often very difficult to resist, but it should be firmly controlled.

It is possible to produce an internal-combustion engine composed of only seven parts (exclusive of studs and nuts). If fewness of parts were a criterion, such an engine should sweep the board. In practice this type of engine has earned almost universal condemnation because of its unreliability and the ingenuity with which it devises different ways of going wrong. At the other end of the scale the aero-engine which, during the war, showed itself capable of running for, by far, the longest period, without overhaul, was the Rolls-Royce "Eagle," an engine which contains at least 50 per cent more parts than any other engine in the service.

That increased number of parts necessarily involves increased care and maintenance on the part of the user is a sheer fallacy. Fewness of parts saves manufacturing costs to some small extent, but it certainly confers no benefit whatever upon the user. Even to the manufacturer it does not always effect a saving, for the amount of fitting work required is often inversely proportional to the number of parts, and fitting is nowadays the most costly of all classes of work. Designers will do well to realize that "simplicity" as ordinarily understood is by no means always a virtue; in nine cases out of ten it is a positive vice.

The foregoing remarks must not be read as implying a disregard of manufacturing cost. On the contrary, manufacturing cost is generally much the most important consideration once the needs of all-round efficiency have been catered for, and by efficiency in this sense is meant not merely thermal efficiency but reliability and durability.

In days gone by, material and skilled fitting were both comparatively cheap, while tooling was a costly item. To-day, however, owing to the vast improvements in machine-tool design, tooling and grinding have become relatively cheap, while the cost of material has risen enormously and skilled fitting has become almost unobtainable. The designer therefore must accommodate himself to these altered conditions and economize material and hand-fitting wherever possible. In this connection he will often find that it is an actual economy to employ a greater number of parts if the total weight of material used is no greater and some hand-fitting is saved thereby. With well-thought-out design and accurate machine work it should be possible to eliminate hand-fitting almost entirely. Much of the hand-fitting which is done to-day is unnecessary and even

undesirable; the scraping-in of bearings, for example, is a custom which dates from the time when accurate machine work could not be relied upon and when designers did not realize the value of self-alignment. Now that crankshafts can be finished by grinding to within extremely close limits, and all their bearing housings can be machined at one operation, scraping is no longer necessary; indeed, no amount of hand-scraping will give so accurate or uniform a bearing as that afforded by the machining. Again, from an economical as well as from a mechanical point of view it is essential that all shaft-bearing housings shall either be machined at one operation or their alignment be ensured by spigoting. If, for any reason, neither of these is possible, then it is not only better but often even cheaper to provide universal joints and so be independent of alignment, rather than to rely upon a doubtful alignment secured by costly hand-fitting. Not only is hand-fitting expensive and unreliable, but it is also the most effective barrier against interchangeability. The old belief that a "hand-made" piece of mechanism is preferable to a "machine-made" dies hard, but the sooner it is buried the better; to-day it is an anachronism.

**Design and Material.**—It is commonly supposed that, to be successful, a high-speed internal-combustion engine must of necessity be made from very carefully selected and highly specialized materials. While, of course, it is obvious that the higher the quality of the material the better, yet, with appropriate design, the ordinary materials of commerce will be found to give perfectly satisfactory results, and are, in fact, much more widely used than is generally supposed. It is only when the weight is very closely limited, as in the case of an aero-engine, or when an exceptionally high output is desired, that fancy materials become necessary.

The ordinary multi-cylinder high-speed engine necessarily involves the use of some very complicated castings, and the choice of material for these is, in practice, governed almost solely by foundry considerations. The material of the cylinder block and crankcase, for example, must be such that it will flow freely in the mould and yield a homogeneous casting, a consideration which generally dominates all others.

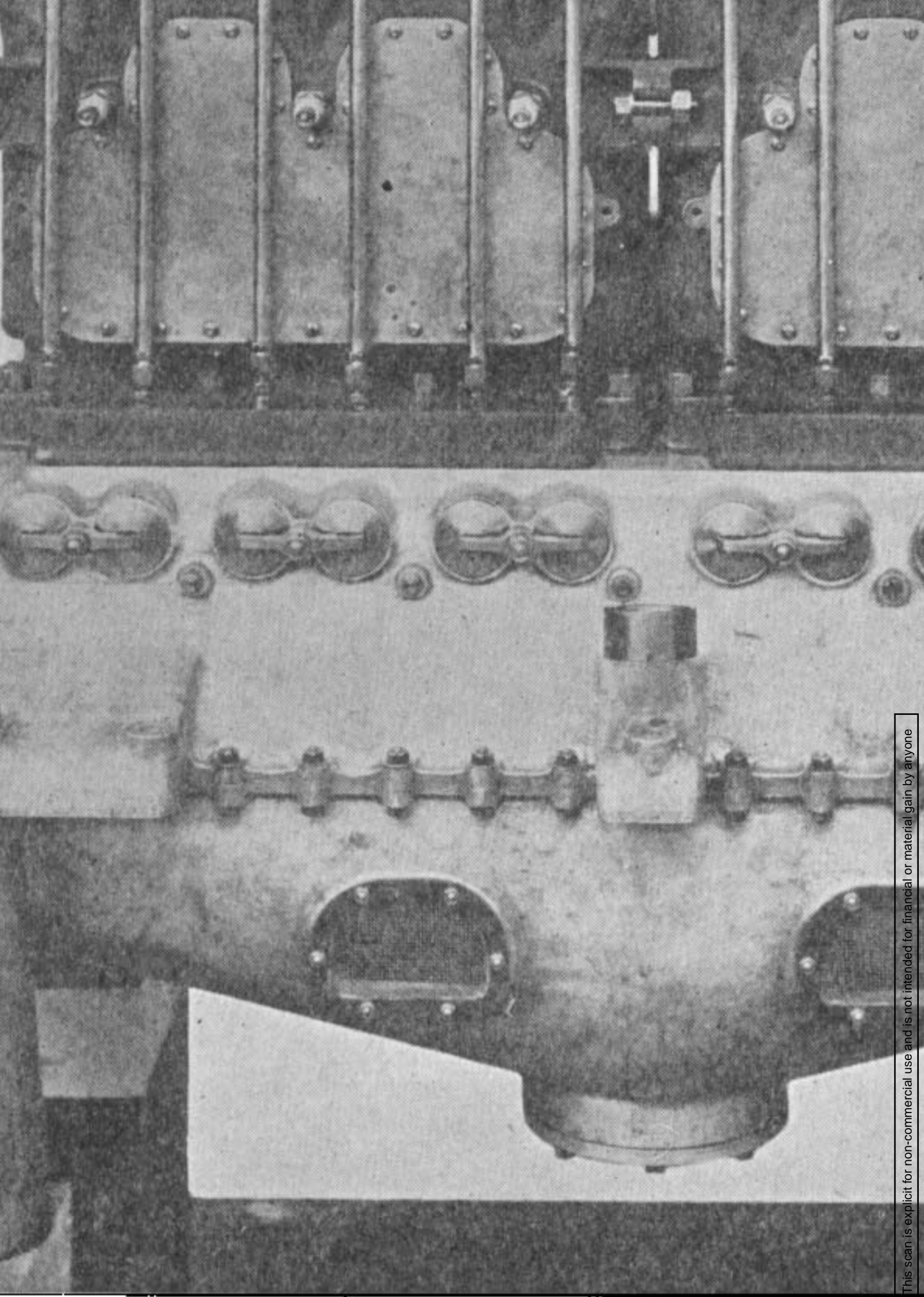
In general, surface hardness is of far more importance than tensile strength; for the necessity for extreme rigidity, which is the first principle in high-speed engine design, compels the use of such heavy scantlings in any case, that tensile strength plays a very secondary part. From the point of view of rigidity all steels are

substantially equal, for the modulus of elasticity is practically the same for all.

**Crankcase.**—For the crankcase, either cast iron or aluminium may be used, depending upon whether weight is an important factor or not. The chief consideration in the design of a crankcase for a multi-cylinder engine is to provide sufficient depth to resist the bending moment due to the opposed couples in a four- or six-cylinder engine. This bending moment is not severe in so far as it affects the structural strength of the crankcase, but it is often quite sufficient to cause very tiresome vibration at certain critical speeds, and in order to prevent this it is necessary to employ a design of crankcase so strong to resist both bending or torsion that, from a structural point of view, it has an enormous factor of safety. Since rigidity is a function of design rather than of material, it follows that, within limits, any material which will cast well and possess reasonable physical qualities will serve the purpose.

To ensure rigidity against bending and torsion it is very desirable, if not essential, that the lower half of the crankcase shall form a structural part of the whole, and particularly is this the case when separate cylinders are used. When, as is customary, the crankcase is divided on the centre line of the crankshaft, the two halves should be bolted together with a large number of closely pitched bolts, to ensure a good friction joint between the two, and the lower half should in itself have ample rigidity considered as a girder. The custom of using merely a light oil tray attached to the upper half by a few loose-fitting bolts cannot be too strongly condemned. Although the lower half of the crankcase may not and usually does not play any part in the mounting of the crankshaft, and apparently is subjected to no direct stresses, yet it has a very important function to fulfil in completing the structure of the crankcase as a whole, and assisting in and ensuring its absolute rigidity. This consideration becomes particularly important in the case of six-cylinder engines, in which crankcase vibration is, or used to be, a very common source of annoyance. Fig. 64 is an illustration of a light 150 h.p. engine, which may be taken as a good example in this respect.

Another consideration which must always be borne in mind is the question of noise due to the vibration of flat panels, &c. As far as possible all flat surfaces should be avoided, or, if this is not convenient, they should be broken up by internal ribbing. There is some evidence to show that for similar designs and scantlings



(b) The cost of machining aluminium is considerably less than that of cast iron; and since in a crankcase there are usually a very considerable number of machining operations, this often outweighs the higher initial cost of the material.

The difficulty in securing studs in aluminium is a real one, but it is usually possible so to design a crankcase as to eliminate or almost eliminate the use of studs. By extending the main crankshaft bearing bolts through to the top of the crankcase and utilizing them to hold down the cylinder block, two groups of studs can be eliminated and a very sound mechanical job results. Where studs must be used they should be screwed in from 2 to 3 diameters, and are then satisfactory, if properly fitted in the first place.

**Cylinder Block.**—Where a number of cylinders are cast together in one block, cast iron is almost invariably used, though recently aluminium blocks fitted with steel or cast-iron liners have been adopted for aircraft engines and for a few motor-car engines also. So far as the plain cast-iron block is concerned there is little to be said except that two considerations should be aimed at in the choice of a material; that it shall cast readily and be free from blow-holes or porous places, and that it shall be as hard as possible compatible with the first consideration. There appears to be no merit in the use of the close-grained iron so often called for in specifications, for an open grain may be just as good and is sometimes better. In this connection it may be mentioned that cylinders finished by grinding seldom wear so well as those which have been reamed; this appears to be due to the retention of small particles of the grinding material in the surface of the bore. These soon embed themselves in the piston, which then proceeds to lap the cylinder walls, more particularly when the piston is of aluminium alloy. The habit of grinding cylinder bores came into force at a time when it was found difficult to obtain a satisfactory finish by tooling alone; and it was soon found to be both a cheaper and a more satisfactory means of obtaining at once a reasonably accurate bore and a good finish, and so found wide favour in the eyes of manufacturers. With modern improvements in machining methods it is now possible to obtain excellent results by reaming and honing, and this process would appear to be preferable. No doubt, with care and a suitable choice of grinding wheels, embedding of the grinding material in the cylinder bore can be avoided, but in many instances this is not the case to-day.

In recent years it has become rather the fashion to make the whole cylinder-head detachable. This practice has many important



advantages. In the first place, it simplifies the cylinder block casting very considerably and enables the cylinder to be bored straight through—a very substantial advantage. In the second, it permits of the valves being placed closer together, and so in the case of side-valve engines, reduces the area of surface in the valve pockets and permits of the cylinder centres being reduced, both very valuable considerations. Thirdly, it eliminates the use of valve plugs, which are always an objectionable feature, since they are uncooled and are liable to leakage.

The principal objection to detachable heads is that they necessitate the making of a gas- and water-joint over a large area and the breaking of this joint in order to get at the valves or clean out the combustion chambers. By the use of suitable copper asbestos gaskets, by providing plenty of holding-down studs suitably spaced, and an ample depth of head, these objections may be overcome. Where trouble with cylinder-head joints has arisen, it can generally be traced either to lack of rigidity in the head or to insufficient or badly placed holding-down studs. Given careful design and ample rigidity, the use of detachable cylinder-heads has much to recommend it.

It is now customary to embody the induction manifold in the cylinder block casting. This certainly makes for neatness, and, by reducing the number of joints in the system, lessens the tendency to leakage; but when these points are conceded—and admittedly they are important points—there is nothing else left to recommend the practice, which has three serious defects:

(1) The design of the induction system, and therefore the efficiency of distribution, must be subordinated to foundry requirements. It is seldom, if ever, possible to design an efficient distribution system which can be embodied in the cylinder block casting.

(2) With modern petrol having a final boiling point of about 400° F., jacketing of the induction system with warm water is of very little use; it merely serves to supply heat to the working fluid before its entry to the cylinder, without ensuring evaporation of the heavier fractions. In previous chapters it has been shown that the less heat supplied to the gases before their entry to the cylinder the better. To prevent precipitation some heat must be supplied but to be of real use it must be supplied at a high temperature. It is far better, therefore, to supply locally a small quantity of high-temperature heat just at those points where, owing to changes in velocity or direction, precipitation of liquid fuel is most liable to

occur, rather than to subject the whole system to a continuous supply of low-temperature heat, which serves merely to reduce the power output without evaporating any but the lighter fractions of the fuel.

(3) When the induction system is cast in the cylinder block the whole of its internal surface is rough; this greatly increases the tendency of the liquid particles of fuel to precipitate, and it increases also the frictional losses.

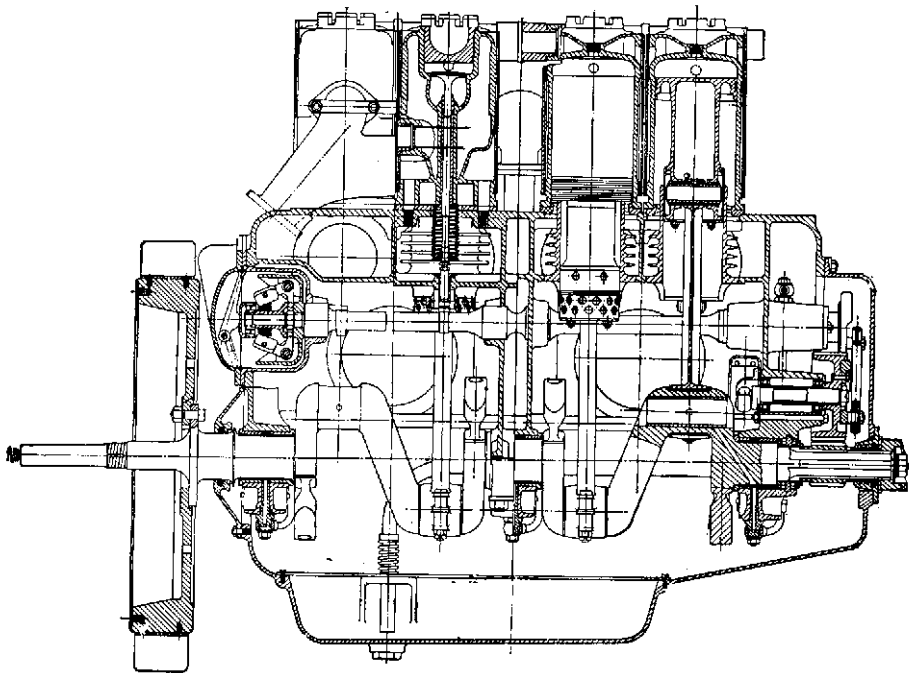


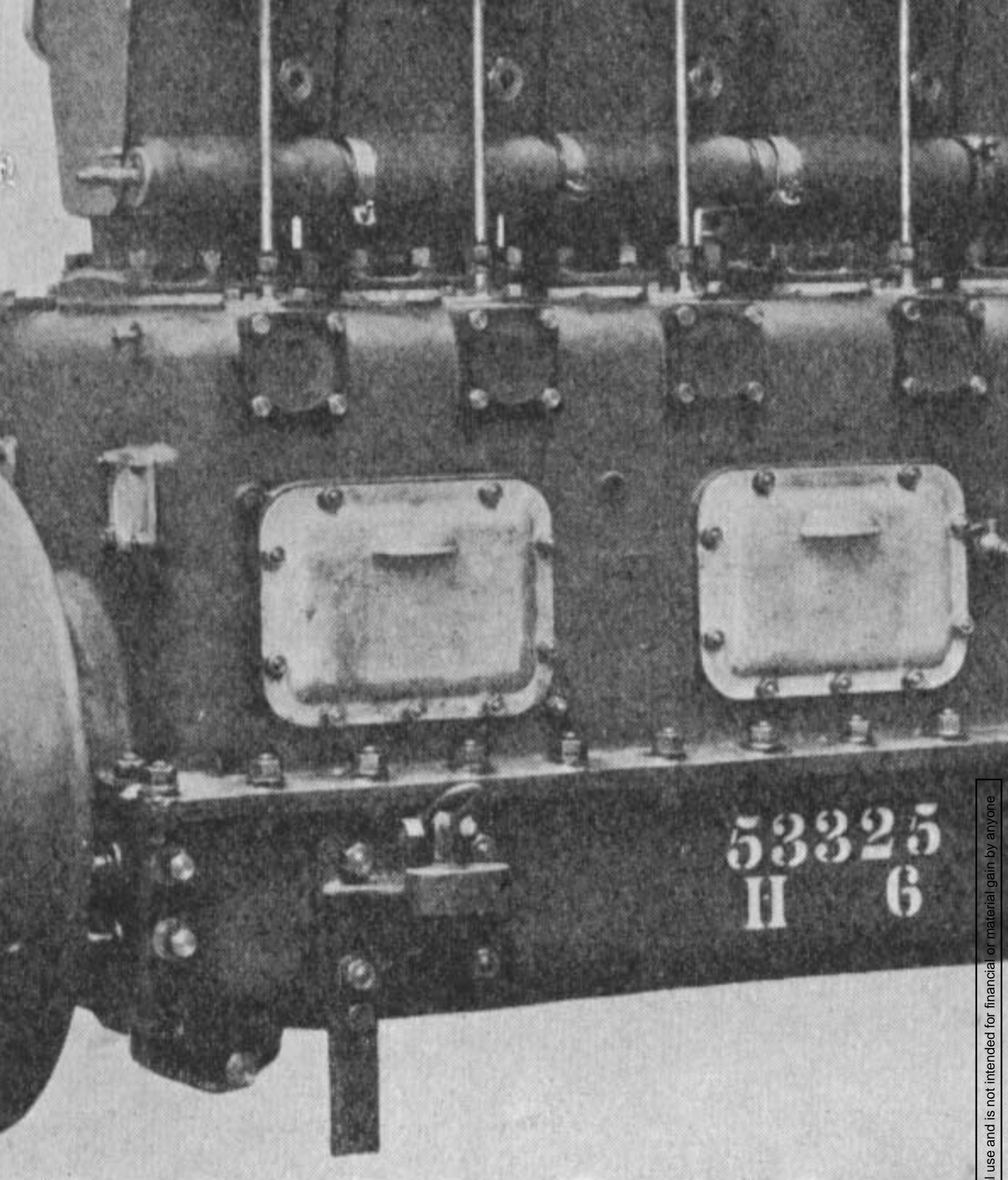
Fig. 65.—Section of Four-cylinder Engine, showing Open-sided Cylinders

Apart from the orthodox multi-cylinder monobloc type, there are many other forms of cast-iron cylinder. Fig. 65 shows a fairly common form for separate cylinders in which the sides of the water-jacket are cast open and subsequently closed with thin sheet-steel covers. This has the advantage that the casting is very simple, can be inspected internally, and the core sand thoroughly removed. Also, it permits of a number of cylinders being packed together more closely than would otherwise be possible with separate cylinders, thus both reducing the length of the crankshaft and increasing the rigidity of the crankcase, as shown in fig. 66.

Fig. 67 shows a design for large cylinders, in this instance of 100

(H 246)

12



**B.H.P. Tank Engine, showing close packing of components, and the ease of inserting in it loose steel lining.**

(4) The water connections can be secured to the main casting, and not to light sheet-metal jackets.

Yet another form of cylinder construction which is sometimes favoured consists in casting the whole cylinder block in aluminium alloy with cast-in valve seats, and either screwed or pressed-in steel liners. This construction is open to the objection that it is

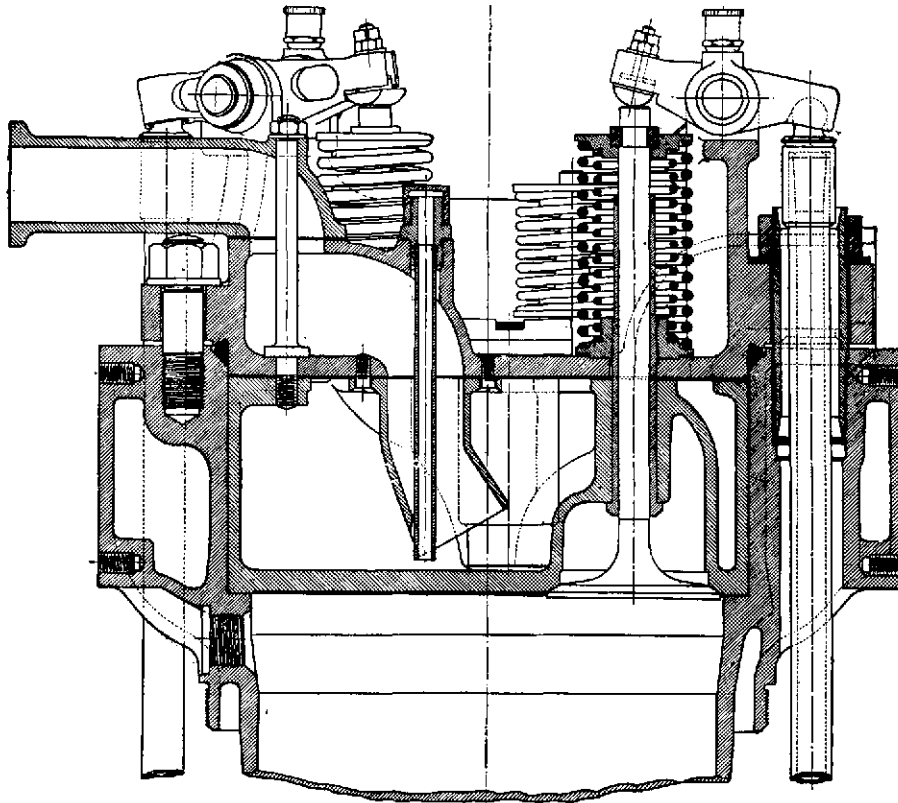


Fig. 67.—Section of Cylinder and Head, 100 B.H.P. Engine.  $8\frac{1}{2}$ -inch bore,  $9\frac{1}{4}$ -inch stroke. Speed, 1150 R.P.M.

always difficult to ensure good thermal contact between the liner and aluminium walls, on account of the very great difference in the coefficient of expansion of the two materials.

A form of cylinder construction which has found great favour for aircraft engines is that in which both the cylinders themselves and the water-jackets are built up out of steel by welding. In such a construction it is customary to forge the cylinder and cylinder head as a plain thimble, to screw and weld the valve ports and valve guides into this, and finally to weld over all a light steel water-jacket

pressed from thin sheet and made up in two halves. This form of construction is very light and sound, and thanks to the absence of castings, and therefore to the small proportion of scrap, it is not nearly so expensive as would appear at first sight. The chief objections to it are:

- (1) It is applicable only to separate cylinders, or at least it

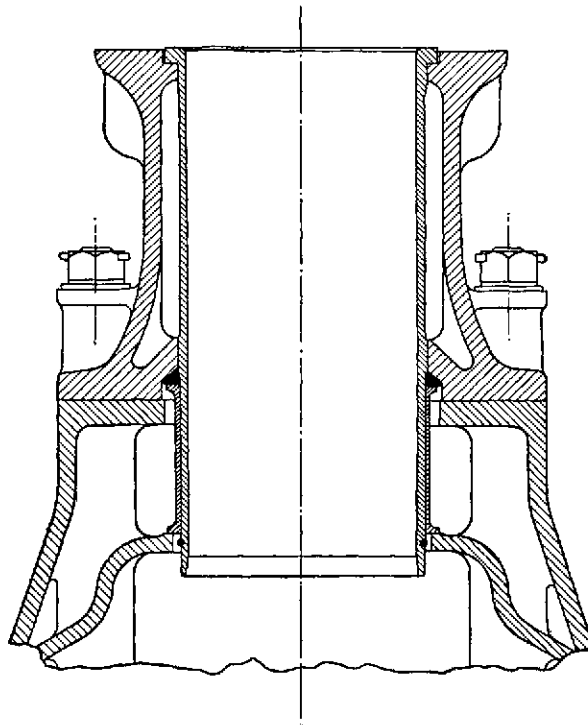


Fig. 68.—Section of Cylinder showing loose hardened Steel Liner sealed by Rubber Ring

becomes very complicated when the water-jacket embraces more than one cylinder.

- (2) The water connections are attached only to light sheet-metal jackets, which are liable to crack from vibration.

- (3) Freedom of design is rather limited by constructional difficulties.

As regards material for built-up steel cylinders, it would appear that a high-carbon steel about 0.45 to 0.55 per cent carbon gives good results as regards wear.

For plain cast-iron cylinders a mixture should be selected which will flow freely and yield a homogeneous casting with as hard a surface as possible and regardless of "grain."

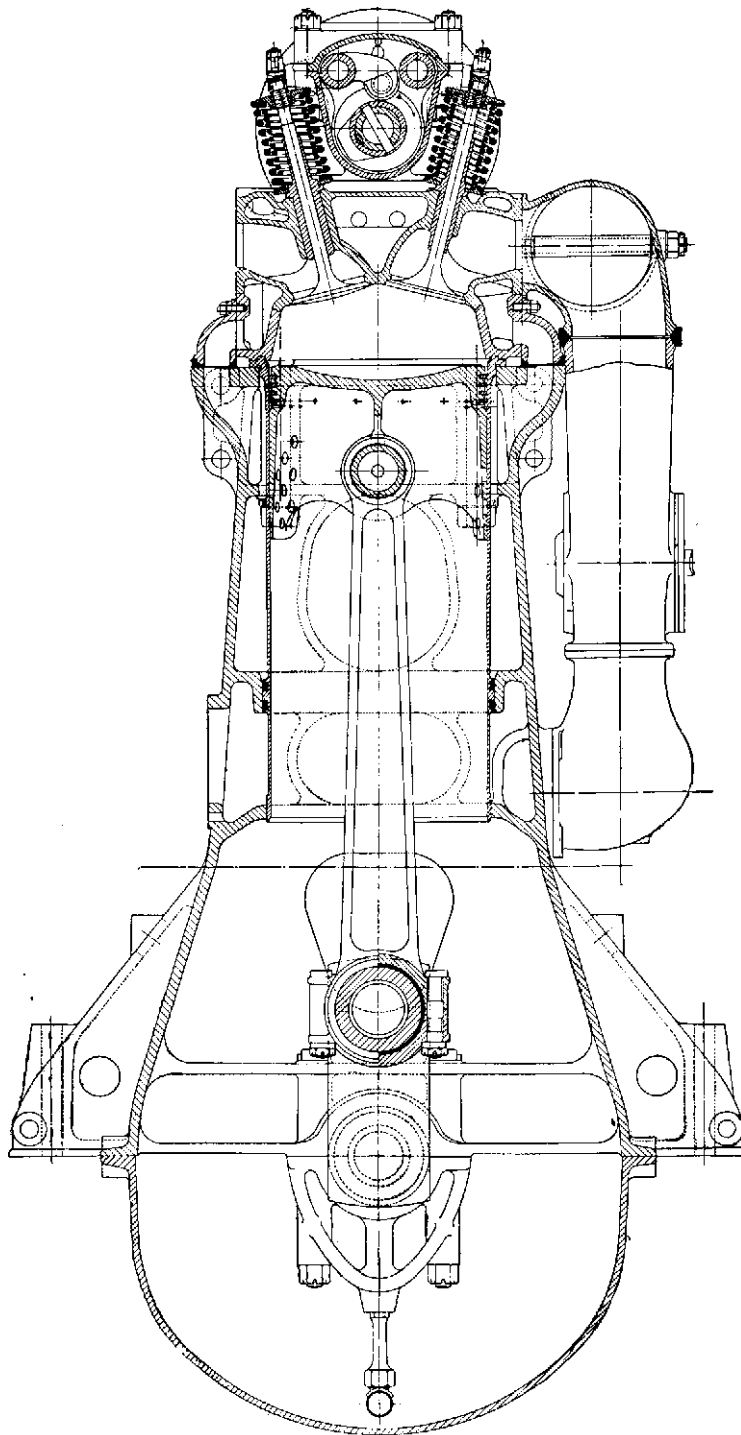


Fig. 69.—Section showing cylinder construction of large Experimental Aero-engine developing 130 B.H.P. per Cylinder. Bore 8 inch, stroke 11 inch, speed 1400 R.P.M.

**Crankshafts.**—There is always a considerable difference of opinion as to the most suitable material for crankshafts. Except in the case of aircraft engines, tensile strength and yield point are factors which need hardly be considered, for by the time the requirements of torsional rigidity, freedom from whipping, stiffness and rigidity of crankpins and journals, and the provision of adequate bearing surface in a limited length, have all been met, it will be found that the scantlings of the shaft are such as to put all risk of failure from any source but fatigue, quite out of the range of probability. The two essential requirements are resistance to fatigue and surface hardness. In the author's opinion it is almost a waste of time to calculate out the stresses in a crankshaft, for, if the design is adequate from the points of view enumerated above, it will invariably be found that the calculable stress in the material is quite absurdly low. It must be remembered that the modulus of elasticity for all steels is substantially the same, and since it is rigidity which controls the design of a crankshaft, all steels are nearly on a par in this respect. In the author's opinion a straight carbon steel containing about 0.45 per cent carbon fulfils all the requisite conditions quite satisfactorily for most commercial purposes except for aircraft engines. These latter are excepted because—

(1) Having no flywheel, aircraft engines are less susceptible to torsional oscillations in the crankshaft, and the latter can therefore be made considerably lighter.

(2) In order to save weight in the shaft and in the design generally, a bearing is always fitted between each crankthrow, hence the tendency to whip is greatly reduced.

(3) Owing to their relatively short working life, the area of bearing surface is generally cut down in order to reduce weight.

Apart from the question of cost of material, the advantages of using a straight carbon steel for ordinary engines are:

(1) That every engineering establishment is familiar with the methods of forging, heat treating, and machining it.

(2) That it is very uniform in quality, and much less sensitive to errors in heat treatment.

In other words, it is decidedly more dependable than high tensile alloy steels, all of which require careful heat treatment, and if wrongly treated, are liable to be dangerous.

Fracture of a carbon steel crankshaft of reasonable design is a very rare occurrence indeed, and is generally due either to want

of adequate fillets at the crankpins and journals, or to periodic torsional vibrations; either of these causes will result in eventual fracture from fatigue, no matter what the material may be. The former may be avoided by providing an ample radius, and the latter by avoiding periodic vibration, either by fitting a vibration damper or by altering the scantling of the shaft so as to raise or lower the periodic speed out of the normal running range. In the case of four-cylinder engines with reasonably light reciprocating parts, it is generally quite easy so to design the crankshaft that there shall be no severe torsional vibration at any speed of which the engine is capable. In the case of six-cylinder engines, however, it is by no means so easy to do this, and it then becomes desirable to employ vibration dampers or other means to check torsional vibration.

The most important consideration to-day is to reduce the rate of wear. This, as shown previously when dealing with lubrication, is a function of the surface hardness, both of the shaft itself and of the bearing material in which it runs. The harder the shaft and the softer the bearing material the better, provided that the latter will not crush. Wear appears to be due almost entirely to particles of grit embedding in the soft bearing material and so lapping the shaft. The rate at which they will lap away the shaft depends upon (1) the surface hardness; (2) the load, which governs the thickness of the oil film and therefore the distance which the particles have to bridge before coming into contact with the shaft.

In general, the load on the crankshaft bearings is so severe that it is not always possible to use a very soft bearing material, but it should be borne in mind that, other things being equal, the softer the bearing material the less will be the wear.

**Balance Weights.**—In four- or six-cylinder engines, the centre crankshaft journal is subjected to very severe loading on account of the cumulative centrifugal and inertia pressure from the two cranks on either side of it, since these are always in the same plane; for this reason, either much greater surface must be given to this bearing, or the crankshaft must be fitted with balance weights to counteract the centrifugal pressure from the two centre cranks. Unfortunately, if the two centre cranks are fitted with balance weights, so also must the others. The provision of balance weights, however, while it relieves the load on the journal bearings, and more particularly on the centre bearing, and also reduces the tendency to vibration of the crankcase, is very objectionable because, by adding



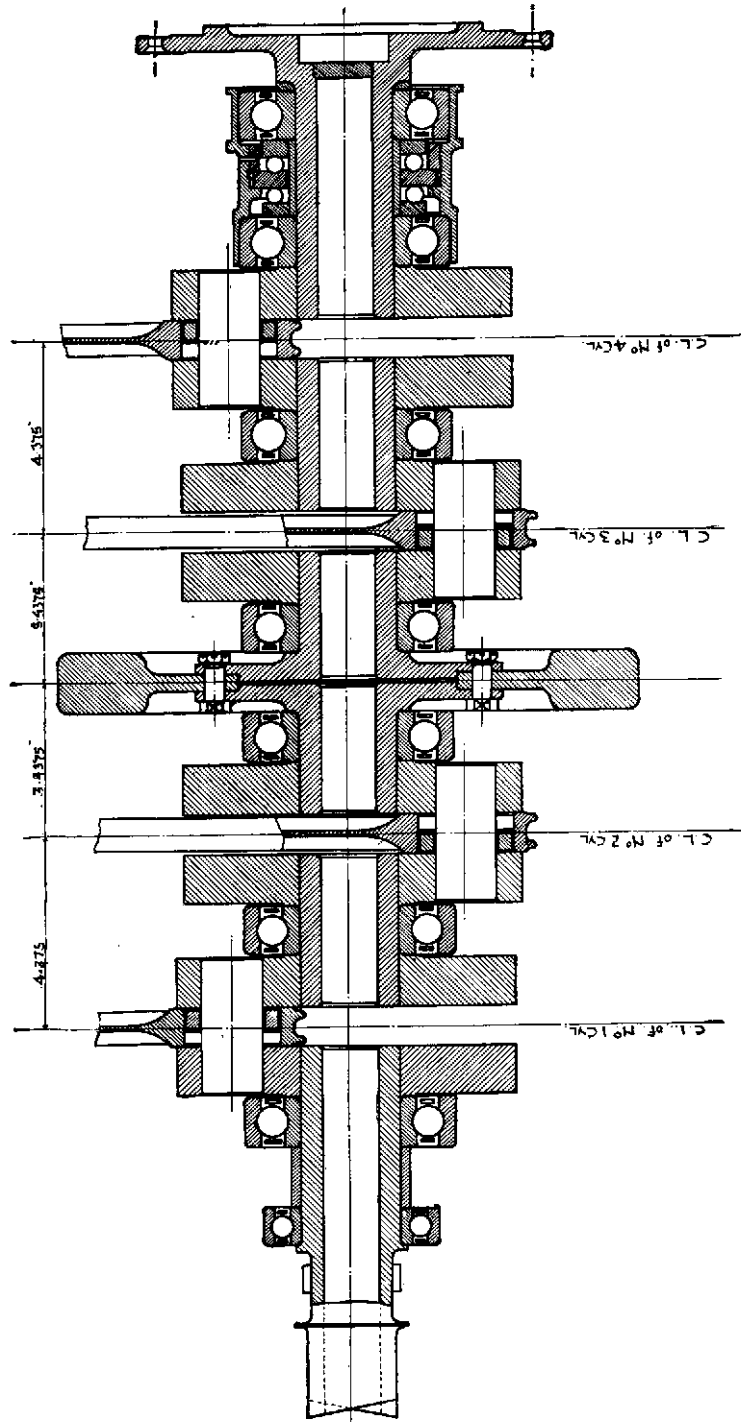


Fig. 70.—Crankshaft Assembly as used in 3 litre 1922 Vauxhall Racing Engines

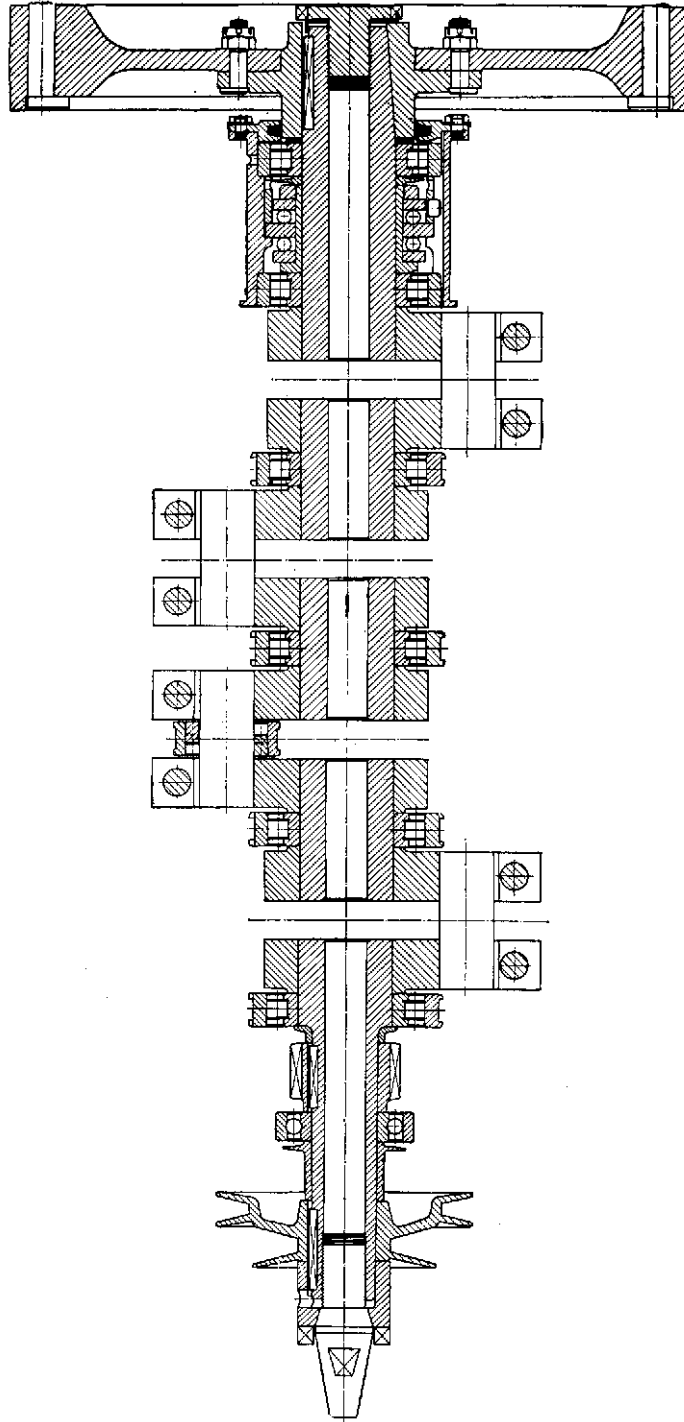


Fig. 71.—Roller-bearing built-up Crankshaft for heavy Commercial Vehicle Engine

masses with considerable inertia to the various throws of the crank, it increases greatly the tendency to torsional vibration of the crankshaft, and at the same time tends to lower its critical speed. This may be, and often is, a serious objection to the use of balance weights, for torsional vibration of the crankshaft is much more serious in its effects and more difficult to deal with than that of the crankcase. The use of balance weights, therefore, is by no means always to be recommended; they may be of advantage in reducing wear, or they may set up severe and dangerous periodic vibration, depending upon the actual circumstances. In cases where, owing to restrictions upon length or for other reasons, the area of bearing surface of the centre bearing is limited, it sometimes becomes necessary to provide balance-weights, and in such cases their ill effects can be counteracted by the use of the well known Lanchester torsional vibration damper.

There can be little doubt but that the ideal crankshaft should have case-hardened journals and pins. It is very difficult satisfactorily to case-harden a one-piece multiple-throw crankshaft on account of its tendency to buckle when quenched, except in the case of very short cranks such as are used in engines with two opposed cylinders. It is, however, in the author's opinion quite open to question whether the advantages of case-hardened bearing surfaces are not such as to justify the use of built-up cranks, with the webs shrunk on to the journals and the crankpins pressed or clamped into place. Fig. 70 shows an actual example of a crankshaft constructed in this manner for an engine of 125 B.H.P., running at a normal speed of 4000–4500 R.P.M., which has been found to give very satisfactory results. Fig. 71 shows an alternative design with the crankpins clamped into place, and roller bearings used throughout. This design is intended for an engine of more normal speed and performance. The bearings on such a shaft should prove well-nigh everlasting, but in the event of wear or any accident such as the fracture of a roller, the whole crankpin, complete with its roller bearing, could be replaced quite easily. Apart from their virtual immunity from wear, ball or roller bearings have the advantage that they are not dependent upon continuous lubrication to maintain the oil film and are therefore more reliable; also, although the friction loss with plain bearings amounts, in any average design, to less than 2 per cent of the maximum power of the engine at high speeds, and at normal working temperatures, this loss is almost independent of load, and is liable actually to increase

as the speed is reduced. While, therefore, the total frictional losses of the plain bearings on a crankshaft are relatively small at high speeds and full load, at reduced speeds and loads they begin to play an important part. In the case of motor vehicle engines which, for the bulk of their existence, operate under very light loads, and whose average load factor is only from 30 to 40 per cent, the use of a ball- or roller-bearing crankshaft would result in a very appreciable gain in fuel economy, probably about 6 to 8 per cent.

**Wear of Crankshafts.**—In general terms, it may be said that while the greatest wear in crankshafts generally occurs in the main journal bearings, actual failure of the bearing material is more frequent on the crankpins, even when both journal and crankpin bearings are subjected to the same load factor. The causes of this state of affairs are not far to seek. The greater liability to failure of the crankpin or connecting-rod big-end bearings as compared with the main journal bearings when both are subjected to the same mean loading per square inch is due to :

(1) The smaller heat capacity which the bearing has to draw upon.

(2) Inadequate support of the bearing in the connecting-rod, the big-end eye of which is generally lacking in rigidity.

In the event, therefore, of a temporary stoppage or slowing down of the oil circulation, the crankpin bearing will, in a given time, attain a much higher temperature than the main journal bearings, and, if the stoppage be prolonged, it will reach the critical temperature at which breakdown occurs (*viz.*, about 360° F. for ordinary white metal), long before the main journal bearings, which have behind them the heat capacity of the whole of the crankcase.

With regard to wear, in most lubrication systems the oil is delivered first to the main journal bearings, and passes thence to the crankpin bearings, with the result that the journal bearings receive and retain most of the grit in the lubrication system, and so the more readily lap the shaft.

In general, the rate of wear appears to be almost directly proportional to the load factor on the bearing, and the surface hardness of the shaft, but is almost always most rapid at the bearing which receives its lubrication first and so retains most of the grit.

**Connecting-rods.**—The chief considerations in the design of a connecting-rod are :

(1) That it shall be stiff enough to resist not only bending and crushing, but also vibration.

(2) That it shall be as light as possible.

(3) That the big-end eye of the rod shall be sufficiently rigid to ensure adequate support to the bearing.

Adequately to fulfil conditions Nos. 1 and 3, means in effect that the scantlings of the rod must be such that it has an ample margin of safety so far as ultimate strength is concerned. When I. section rods are used, and this is probably the most desirable form both from the point of view of manufacture and of the disposition of the material, the section must have sufficient width to resist vibration in the plane of the crankshaft, as well as sufficient depth to resist bending—this is too often forgotten and many mysterious troubles and noises are probably due to sideways vibration of the connecting-rod.

It is clearly most important to keep the weight of the connecting-rod as low as possible consistent with fulfilling the other requirements, and in this connection it should be emphasized that it is the weight of the rod as a whole, and not that of the reciprocating end only, which has to be considered. In many cases it is quite as important to keep down the weight of the rotating as it is the reciprocating end, for while the reciprocating mass of the rod affects the balance of the engine, its rotating weight is of more importance in so far as it influences the average pressure on the crankpin and crank journal bearings due to centrifugal loading. In a six-cylinder engine in which the reciprocating parts are balanced inherently, rotating weight plays a more important part than reciprocating; on the other hand, in a four-cylinder engine in which the secondary disturbing forces due to the inertia of the reciprocating mass are cumulative, it is the reciprocating weight of the connecting-rod which must be considered first.

It is in the design of the eye of the big-end that particular care is required.

The first consideration is to obtain a uniform support for the bearing, and, to this end, not only must the eye of the rod be made as rigid as possible in itself, but also the load transmitted down the shank must be distributed over it as uniformly as possible. It is generally useless to provide a wide bearing because of the practical impossibility either of obtaining the necessary rigidity or of distributing the load over it. It is, in the author's opinion, very doubtful whether any useful bearing surface can be obtained when the width of the connecting-rod big-end bearing exceeds three-quarters of the diameter of the crankpin; in any case it is very desirable to splay

out the outer webs of the shank at their junction with the big-end eye, in order to distribute the loads transmitted down them.

Another frequent source of weakness in connecting-rod design is the lack of provision of sufficient abutment for the two halves of the connecting-rod bearing. The cap of the rod should be considered as though it were an arch loaded at its centre, but in tension not in compression. Viewed in this light it is obvious that unless the arch has a wide abutment it will tend to close in and so nip the crankpin at the sides. In high-speed engines it is common practice,

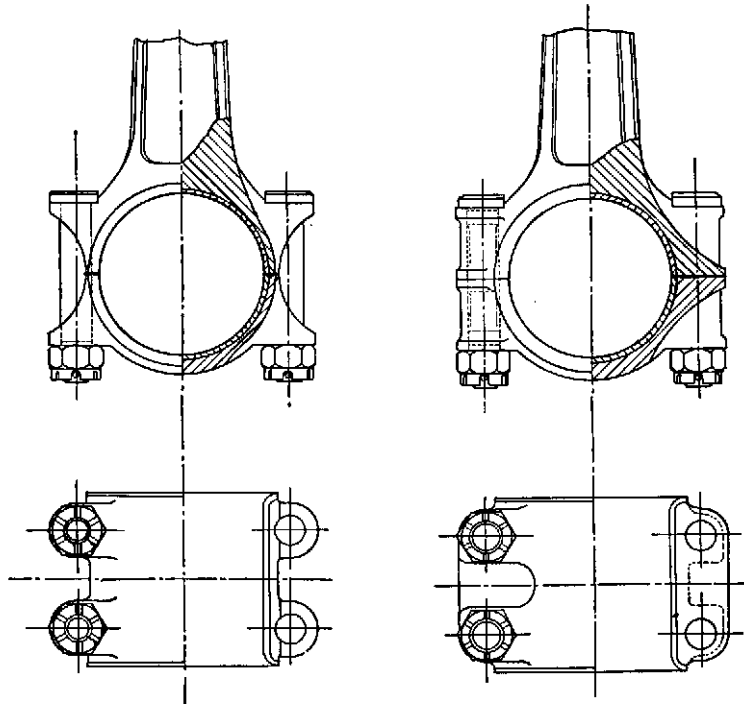


Fig. 72

Fig. 73

in order to save weight, to cut down the width of the abutment to the lowest possible limit, and many failures of big-end bearings are directly attributable to this cause. Fig. 72 shows a design of connecting big-end used in some aero-engines which gave rise to constant failures of crankpin bearings. Fig. 73 shows the same rod after the design had been corrected, when no further trouble arose.

The problem of fitting the bearing itself in the connecting-rod big-end has always been a difficult one. There can be little doubt but that the most reliable method is to run the white metal direct

into the eye of the rod, for by this means alone can good thermal contact be obtained. In some cases, particularly when aluminium alloy connecting-rods of high conductivity are used, good thermal contact is no longer desirable, for the flow of heat from the piston down the shank of the rod may itself be such as to raise the temperature of the bearing. In such cases we must rely entirely on oil-cooling, and good thermal conductivity becomes a thing to avoid. There are, however, several practical objections to this method:

(1) In the event of the lining being run out, it is not easily replaced.

(2) If the lining runs out, the steel surface of the rod will come in contact with the crankpin and may do considerable damage.

In spite of these objections, however, direct white metalling of the rod itself is probably the most satisfactory method for high-duty engines. The alternatives are the use of die cast white-metal bearings or white-metal-lined bronze shells. The former appear to be suitable only for light duty, because of the difficulty in securing uniform support and close contact with the metal of the rod, also they are open to the objection that if the white metal runs out the steel of the rod will come in contact with the crankpin; in this respect the position is considerably worse than when the metal is run direct into the rod, because of the much greater clearance left. For medium-duty engines the use of bronze shells lined with a very thin lining of white metal appears to offer the most satisfactory compromise, for, in the event of failure, the shells can readily be replaced and there is much less danger of the crankpin suffering damage. In any event, the white-metal lining should always be kept very thin, from 0.015 in. to 0.30 in. appears to be ample.

As regards the small-end bearing there is little to be said except that to ensure rigidity it should be short and of large diameter. If lined with chilled phosphor-bronze this bearing should never give the slightest trouble, nor show measureable wear, provided that the gudgeon pin is stiff enough to resist deflection (which is seldom the case), and is adequately supported from the piston. Troubles with the small-end bearings of connecting-rods are almost invariably due to deflection of the gudgeon pin, which causes heavy local pressure; so long as this pin is stiff enough to afford uniform bearing, a very small area of surface will suffice. As in the case of the big-end bearing, it is quite useless to provide a width greatly in excess of the gudgeon-pin diameter, both on account of the impossibility of

providing adequate support to the bearing and also on account of the deflection of the gudgeon pin itself.

So far as material for connecting-rods is concerned, the problem is somewhat the same as in the case of the crankshaft, but there is one important difference, namely, that the material of the rod is not called upon to function as a bearing surface, so that surface hardness is not required. Since to fulfil the requirements as regards rigidity necessitates automatically the provision of an ample margin of safety, there is seldom any occasion to call for high tensile alloy steels. Plain carbon steels or a mild nickel steel, which can readily be forged or stamped, will be found suitable in all but very extreme cases.

**Gudgeon Pin.**—Lack of stiffness and inadequate support of the gudgeon pin in the piston are common failings in many high-speed engines. Excessive wear and occasional seizure of the gudgeon-pin bearings are still epidemic in some engines, and although the cause is generally attributed to inadequate bearing surface or scanty lubrication, careful examination will almost invariably reveal that the real trouble is deflection of the pin causing excessive local pressures. Although the pressure on a gudgeon-pin bearing is very high, the rubbing velocity is low, and the average load factor is certainly by no means a high one. Given that the pressure is distributed uniformly over the surface of the bearing, the rate of wear and the risk of seizure in this bearing should be insignificant. In the normal design of trunk piston, the gudgeon pin is carried in bosses which are attached only to the walls of the piston skirt, with the result that the fluid pressure is conveyed to the pin at its two extreme ends, for although the bosses themselves may be rigid enough, their attachment to the piston is by no means so. Calculation will show that, in most instances, the deflection of the gudgeon pin under the maximum fluid pressure is altogether excessive and quite sufficient to concentrate the loading on the two extreme ends of the bearing. It is clearly of the utmost importance either that the gudgeon pin shall receive its load from the piston crown at points as near its centre as the connecting-rod bearing will allow, or its diameter shall be such that, when loaded at the two extreme ends, there shall be no appreciable deflection. To conform with the latter condition is nearly impossible, since in many instances it would involve the use of a gudgeon pin of such large diameter and weight as to be prohibitive. When, however, the gudgeon pin is loaded at two points, about half the diameter of the piston apart,



there is no difficulty in obtaining the necessary rigidity with a reasonable diameter. As a general rule, for engines of normal compression, the diameter of the gudgeon pin should be one quarter of that of the piston, and its true points of support one half the diameter of the piston apart. With such proportions and with a full floating gudgeon pin the working life of this bearing will be almost indefinite, even with scanty lubrication.

Unless the pin be of abnormally large diameter, it is quite useless to provide a wide bearing at the connecting-rod small end, since its provision necessitates spreading the points of support and so increases the deflection and renders the extra bearing surface valueless. In

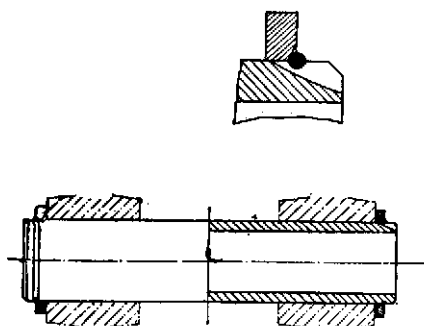


Fig. 74.—Method of locating a full floating Gudgeon Pin

the author's opinion it is very doubtful whether any use can be made of a bearing width exceeding 0.35 of the piston diameter. Given sufficient rigidity, the wear on a gudgeon-pin bearing is extremely small, but, owing to the small angle through which the connecting rod oscillates, it is also extremely local and tends to wear the pin oval. This tendency can be overcome by the use of a floating gudgeon pin, that is to say, by permitting the

pin to turn freely both in the connecting-rod bearing and in the piston bosses, so that it will rotate slowly and wear uniformly all round its circumference. The use of a floating gudgeon pin removes also the difficulty of locating it endwise in the piston, which is a serious trouble, and becomes acute in the case of aluminium pistons, in which, owing to their large coefficient of expansion, the gudgeon pin can be tight only when the piston is cold. Fig. 74 shows the mounting and means of location which the author has found the most satisfactory after much experience. In this design the load is transmitted to the gudgeon pin by two heavy webs extending straight down from the crown of the piston as near the centre as the connecting-rod bearing will permit. The gudgeon pin floats freely in the piston bosses, and is located endwise by means of light washers secured by wire circlips sprung into grooves near the end of the pin.

For the material of the gudgeon pin one consideration, namely, surface hardness, should override all others, for if the pin is stiff

enough not to deflect appreciably under load, it will be strong enough to resist fracture. From the point of view of surface hardness the best possible material is case-hardened mild steel, and a steel should be chosen which will give a glass-hard surface. In very exceptional cases air-hardened steel may be used, but this is seldom necessary.

**Valves.**—In the design of the valves, it is necessary always to remember that the objects in view are:

- (1) To provide the freest possible entry and exit for the gases.
- (2) To keep them as small as possible, consistent with the first condition.

In order to comply with these conditions it is essential to make the orifice coefficient of the valve and its surroundings as high as possible, in order that it shall pass the maximum weight of gas with the minimum of pressure difference. To this end care should be taken—

- (1) That the flow of the gases on either side of the valve port is as free as possible—so far as possible there should be no abrupt bends or changes of section on either side of the valve head.

- (2) The lift should always be equal to at least one quarter and preferably even to 30 per cent of the port diameter.

The use of unduly large valves, and more particularly of large valves with a reduced lift, should always be avoided for the following reasons:—

- (1) Except in very slow-speed engines the provision of relatively large valves must almost invariably be detrimental to the compactness of the combustion chamber.

- (2) For a given frictional resistance, and therefore for a given volumetric efficiency and fluid pumping loss, a much higher entering gas velocity may be employed when the valve is small, has a high lift, and is well stream-lined, so that the degree of turbulence and therefore the power output and efficiency are greater.

- (3) Since both the inlet and exhaust valves get rid of the bulk of their heat through their seatings, it follows that the larger the valve the higher will be its temperature. It is very important to keep the temperature both of the exhaust and inlet valves as low as possible. The former because their durability is a function of their working temperature, and the latter because the entering gases are only too ready to take up heat from the inlet valves and so penalize the volumetric efficiency. Heat picked up from the inlet valves may be regarded as purely detrimental, for it is received too late in the induction process to be of any use in assisting uniformity of

distribution. Its addition from this source results merely in reducing the density of the charge and raising the whole temperature of the cycle, both of which are highly undesirable from every point of view.

(4) Large diameter valves and, in particular, exhaust valves, result in unnecessarily heavy stresses on the valve-operating gear, since on full load the exhaust valves are opened against a pressure ranging from 50 to 80 lb. per square inch. That the valves should always be as light as possible consistent with mechanical strength and heat dissipation is of course obvious, but there is a tendency in many designs of high-speed engine to carry weight cutting in the valves altogether too far, with the result that stretching, distortion,

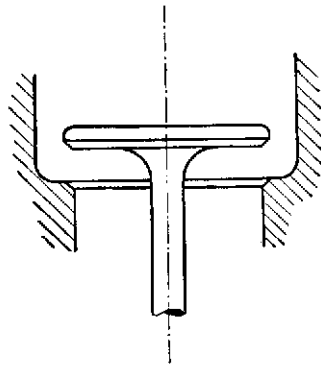


Fig. 75

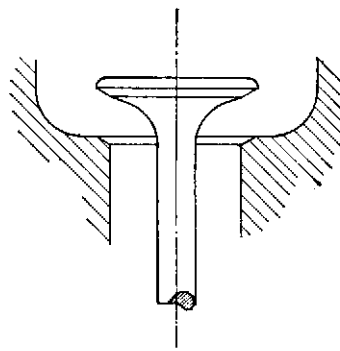


Fig. 76

and overheating are liable to occur. When the whole reciprocating mass of the valve and all its auxiliary gear down to the cam are taken into account, it will be found that the weight of the valve head alone forms but a very small proportion of the whole, and it is generally bad policy to stint metal in the valve head and stem for the sake of the relatively small saving in weight effected thereby.

Since a valve gets rid of its heat largely through the seating, it follows that the seating should be made fairly wide in order to provide a sufficient area of contact when the valve is at rest. Very narrow valve seats save some trouble, both in the first instance and subsequently when grinding in, but there is little doubt that the use of wide seats tends to a higher and better sustained efficiency.

Figs. 75 and 76 show to scale two actual designs of valve and the surrounding walls both being in the full open position. That shown in fig. 76 is much the more efficient type of the two, and owing to its better stream lining and more efficient orifice coefficient it may be made considerably smaller and, though the head is much

more massive, it is for the same weight of gas delivered considerably the lighter valve of the two, and will have much the longer life.

**Number of Valves.**—In order to keep the temperature of the valve heads as low as possible and so to increase both the durability of the exhaust valves and the volumetric efficiency, it is, in large engines, always preferable to duplicate the inlet and exhaust valves rather than to resort to the use of very large

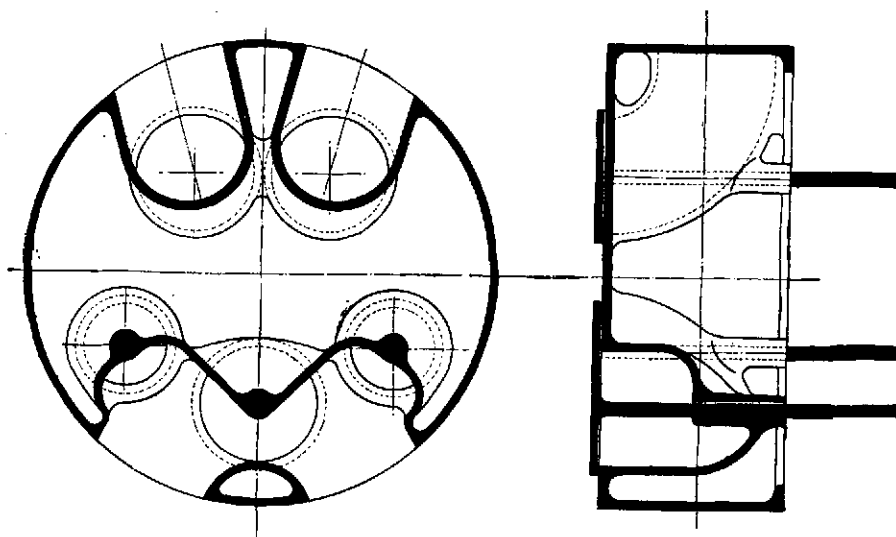


Fig. 77

sizes. In the author's opinion, as a rough general rule for more or less normal engines using volatile liquid fuels, no single valve, either inlet or exhaust, should be asked to deal with more than 200 lb. of air per hour, this giving in an engine of average efficiency from 30 to 35 I.H.P. This rule applies nearly irrespective of the speed at which the engine runs. It must of course be considered merely as a first approximation, for there are many other factors which enter into the consideration and are often of such magnitude as to overrule it. First of these is the form of the combustion chamber. When, as is usual in the larger sizes of engine, the valves are placed vertically in the cylinder head, duplication of the inlet and exhaust valves introduces no difficulty and in no way penalizes the efficiency of the combustion chamber. In some cases it is preferable to employ two inlet and three exhaust valves, an arrangement which lends itself very conveniently to an efficient design of combustion head. Such an example is given in fig. 77,

which shows the cylinder head and valves of an engine developing 80 B.H.P. per cylinder. In this instance two small exhaust valves and a large one are used, the former being opened some 30° earlier, in order to act as pilot valves to get rid of the bulk of the high-pressure high temperature exhaust products before the large valve opens, and so to relieve this valve and its operating mechanism.

## CHAPTER VII

### MECHANICAL DETAILS

**Ball and Roller Bearings.**—The use of ball and roller bearings in internal-combustion engines is becoming more and more extended. The great advantage of such bearings lies in—

- (1) Their low coefficient of friction.
- (2) Their independence as regards lubrication.
- (3) Their freedom, under favourable conditions, from wear.

The disadvantages attaching to them are :—

- (1) Their high first cost.
- (2) Their tendency, under certain circumstances, to set up a disagreeable growling noise.

In general it may be laid down that ball or roller bearings should be used—

- (1) In all places where the provision of adequate lubrication is difficult.

- (2) In places where it is difficult or inconvenient to provide a sufficient surface hardness for ordinary plain bearings.

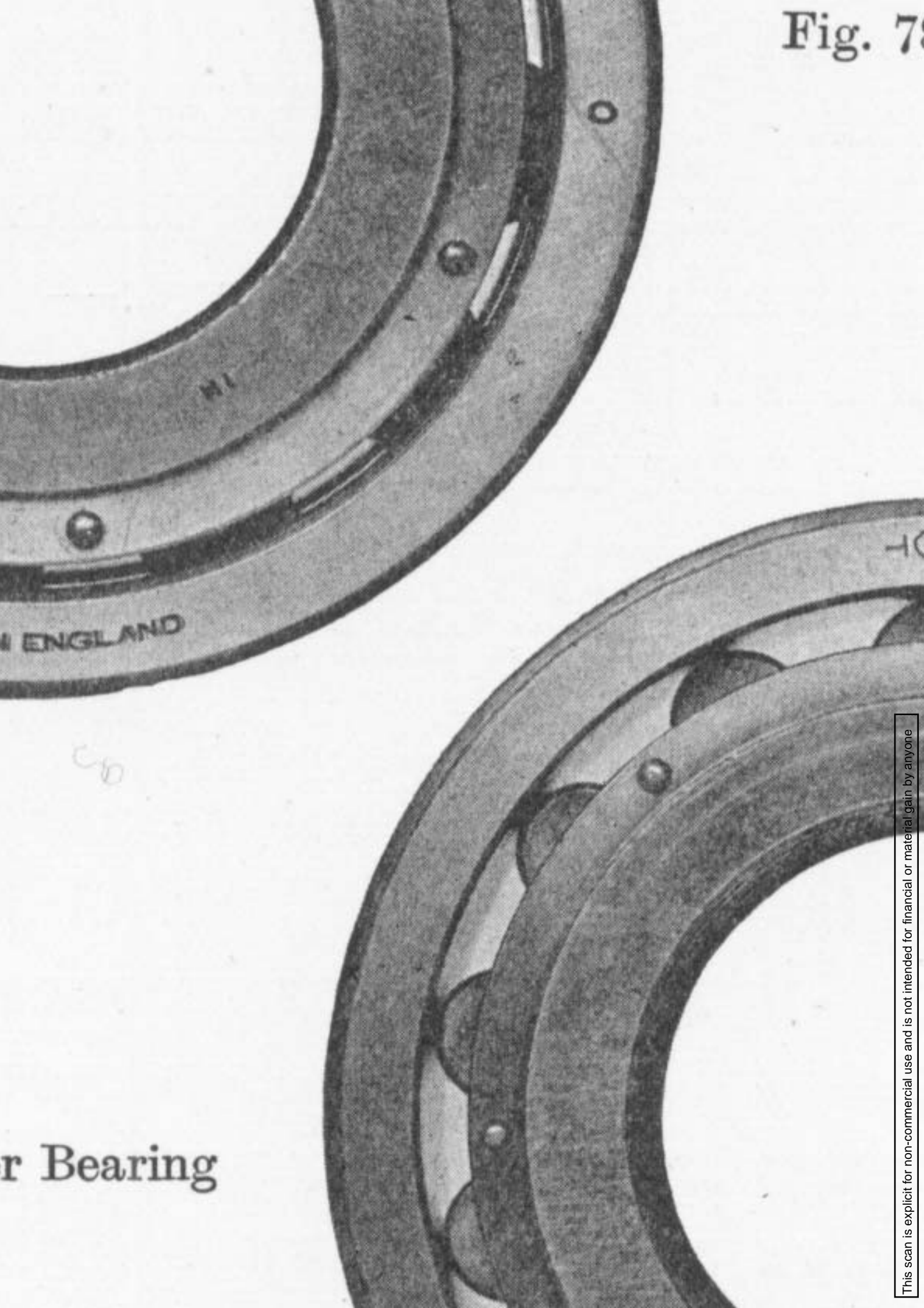
Ball or roller bearings appear to be particularly unsuitable and to give rise to noise when applied to any shaft liable to flexure, as, for example, when fitted to the main journal bearings of a very light crankshaft.

Unlike plain bearings their durability and safety are nearly independent of speed, but are dependent rather upon the *maximum* load to be carried. On the score of reliability, therefore, they show to great advantage in situations where the mean load is heavy and the rubbing velocity very high, i.e. where the “load factor,” as opposed to the maximum load, is very high.

Fig. 78 shows a typical example of ball journal bearing, and fig. 79 of a similar roller bearing.

In the author's opinion a suitable situation for ball, or preferably, roller bearings, is in the connecting-rod and main crankshaft bearings because, in the first place, these are very heavily loaded, and, at the

Fig. 78



r Bearing

such bearings cannot be split and must be threaded over; this entails the use of a light and lanky crankshaft, the one thing which should most sedulously be avoided. With such a crankshaft, considerable flexure is bound to take place, and this by tilting the inner race and so partially jamming the balls or rollers gives rise both to undue wear and to the characteristic growling noise so often associated with ball bearing crankshafts. It would appear desirable, therefore, if ball or roller bearings are to be used, to build up the crankshaft, using very heavy and massive crankwebs shrunk or pressed on to the journals and so reduce as far as possible any tendency to flexure. In the case of single-cylinder engines with inside fly-wheels and built-up cranks ball or roller bearings may be used with particular advantage, for in this case the main crankshaft bearings are shielded from the maximum shock pressures by the inertia of the fly-wheels, while the use of combined fly-wheels and crankwebs permits of the fitting of a readily detachable crank-pin which can be case-hardened and ground, and being detachable, allows of the connecting-rod eye being unsplit. Fig. 80 shows an actual example of such an arrangement as applied to the "Triumph" motor-cycle engine, which has been found particularly satisfactory from every point of view.

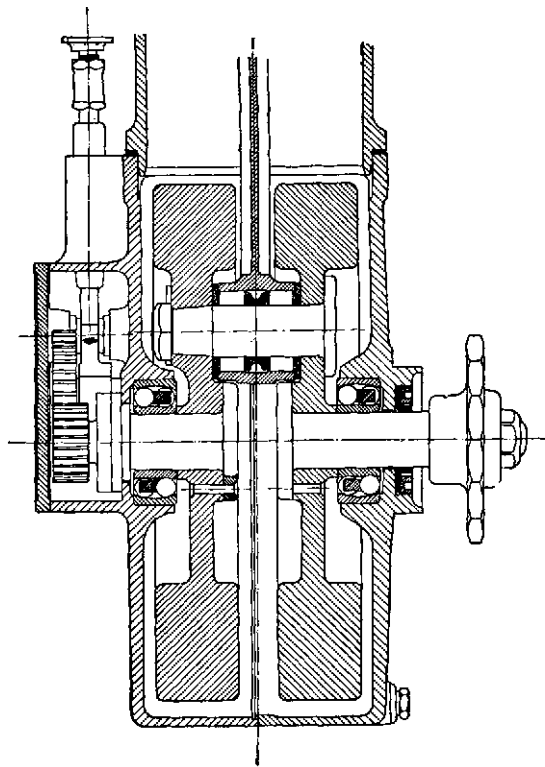


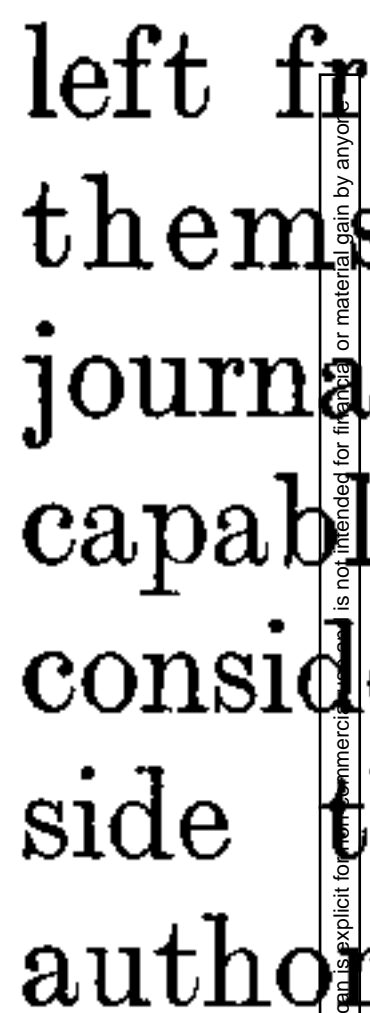
Fig. 80.—Section of "Triumph" Crank-chamber, showing Ball or Roller Bearings

Where the loads to be dealt with are heavy, as in the case of crankshaft bearings, it would appear preferable to employ roller bearings despite the objection that such bearings provide no end-wise location for the outer races.





course essential that a single location by the outer race, a



recommended, such as the Skefko bearing shown in figs. 81 and 82.

Such bearings, however, are liable to give trouble in erection owing to the outer housing slewing round radially when being threaded into position. The author has seen several instances in which it has been well-nigh impossible to erect certain parts of an engine on this account.

**Auxiliary Drives.**—In the design of any high-speed engine one of the most difficult problems, if not the most difficult, to deal with is that of driving all the auxiliary gear. This gear consists usually of the camshaft, magneto, oil-pump, water-pump, and in many cases also a fan and a dynamo. It is no easy matter to dispose all these auxiliaries in convenient and accessible positions and to drive them, each at their respective speeds, without noise or undue mechanical complication. From the point of view of silence there can be little doubt that the use of the so-called silent chain is the best means, but it has its limitations. In the first place, it is absolutely essential to provide means for adjusting the tension of the chains if anything approaching a long working life is to be attained. Secondly, it is necessary to provide an arc of contact of the chain with its sprocket wheel of not less than  $110^{\circ}$  to  $120^{\circ}$ . This latter consideration precludes the use of a single chain embracing a number of sprocket wheels. At the very outside, a single chain can embrace only three wheels, and even this applies only when the wheels are disposed approximately at the apices of an equilateral triangle—a disposition which is not by any means always convenient. It is therefore generally necessary to provide two chains, an arrangement which on account of their width is very bulky and cumbersome and also very expensive, for chains and sprocket wheels are at the best of times very costly items.

Fig. 83 shows an arrangement wherein a single chain is employed to drive the camshaft and magneto; in this instance the oil pump is driven directly from the camshaft, and there is no positive drive to the water-pump, fan, or dynamo, though the former, if it were required, could be driven from the shaft driving the magneto. In this arrangement provision is made for adjusting the chain by mounting the magneto shaft and wheel in an eccentric housing: the cylindrical portion of this housing is extended through the main wall of the crankcase for a length of about 2 inches, and the magneto bracket and magneto are clipped on to this projection, so that the alignment of the magneto is entirely unaffected by any rotation of the eccentric housing.

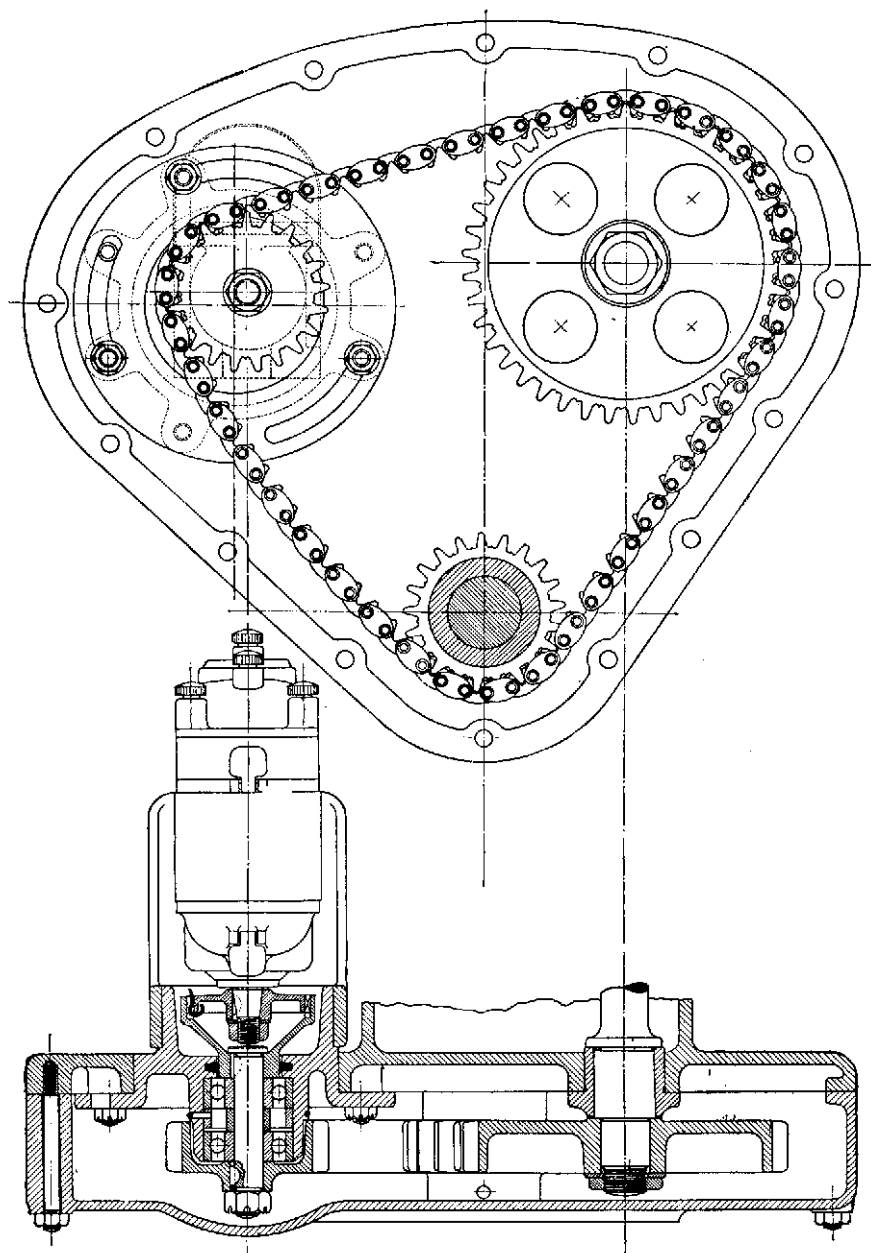
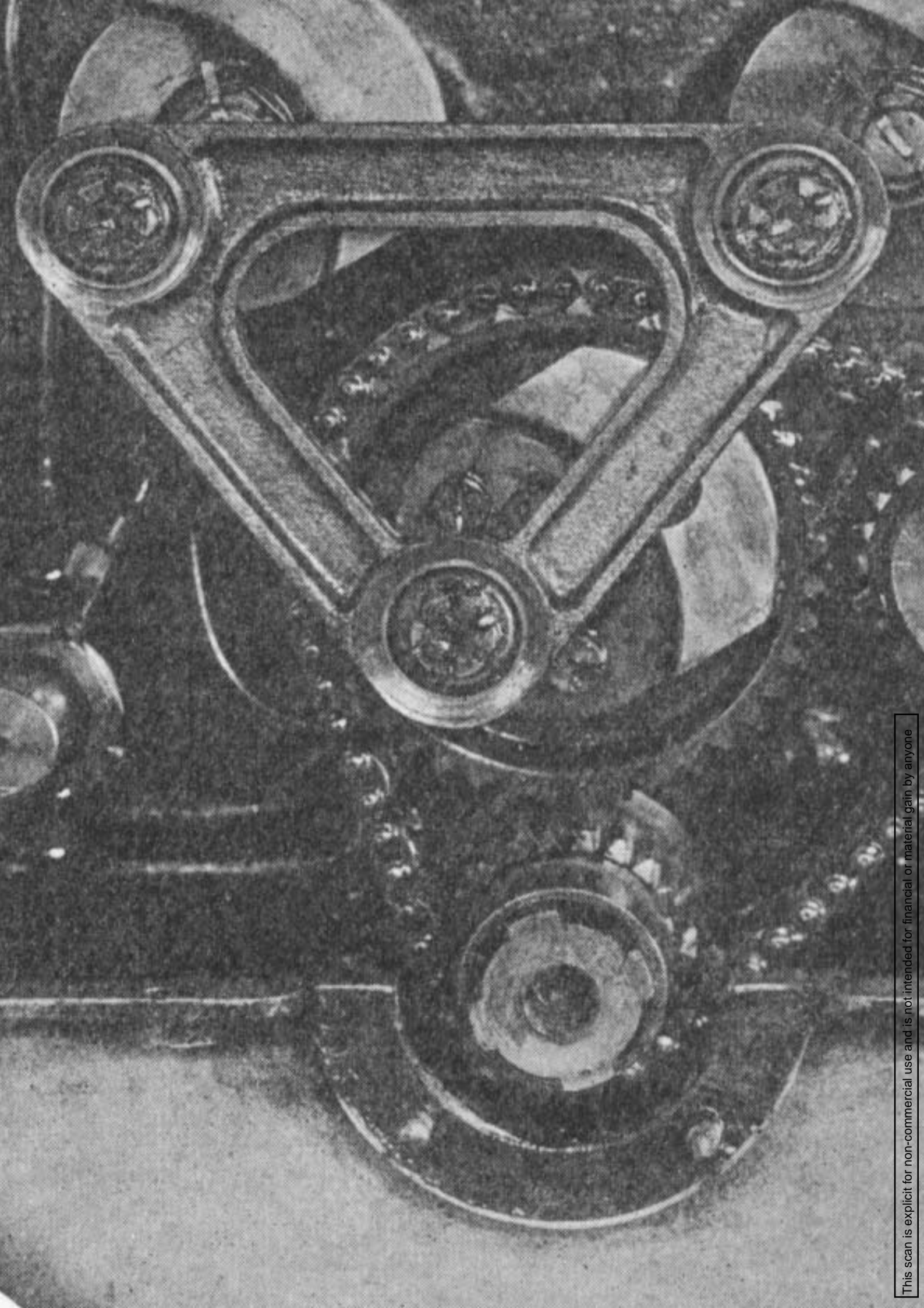


Fig. 83.—Single Chain triangulated Drive with eccentric Adjustment

Fig. 84 shows an arrangement for driving two camshafts, the magneto, water and oil pump. In this arrangement one chain encircles the crankshaft sprocket, a half-speed idle wheel placed immediately above, and a jockey wheel at the side, forming a nearly



From the point of view of silence alone, it would be far better, where spur gears are used, to fit these at the fly-wheel end of the crankshaft where the angular velocity is virtually constant. In practice, however, there are generally serious objections to this, both on the score of accessibility and because the shaft is usually provided with a large flange for carrying the fly-wheel, formed integrally with the crankshaft, so that it is not possible to thread the crankshaft wheel into place. Such a position would, however, be vastly preferable on the score of silence, particularly in the case of six-cylinder engines, for the one serious objection to the use of spur gearing lies in its noise, and this is very largely due to variations in the angular velocity of the crankshaft at the end remote from the fly-wheel.

When spur gearing is used, it is of the utmost importance to provide that the wheels shall be correctly meshed. To ensure this it is preferable always to provide some means of initial adjustment, for it is well-nigh impossible to ensure sufficiently correct machining of the centres. When using spur gearing for the auxiliary drives, the author prefers always to mount one or more of the idle wheels in any train in a separate spider carrying the bearings for the wheel and bolted up against the face of the crankcase. This allows of some initial adjustment, for the spider can be attached by bolts or studs having clearance holes, and adjusted until the meshing is correct, when it may be finally and definitely located with a dowel pin.

Fig. 85 shows an arrangement in which two oil pumps and a camshaft are driven from a train of three spur wheels with the intermediate or idle wheel mounted on a spider as explained.

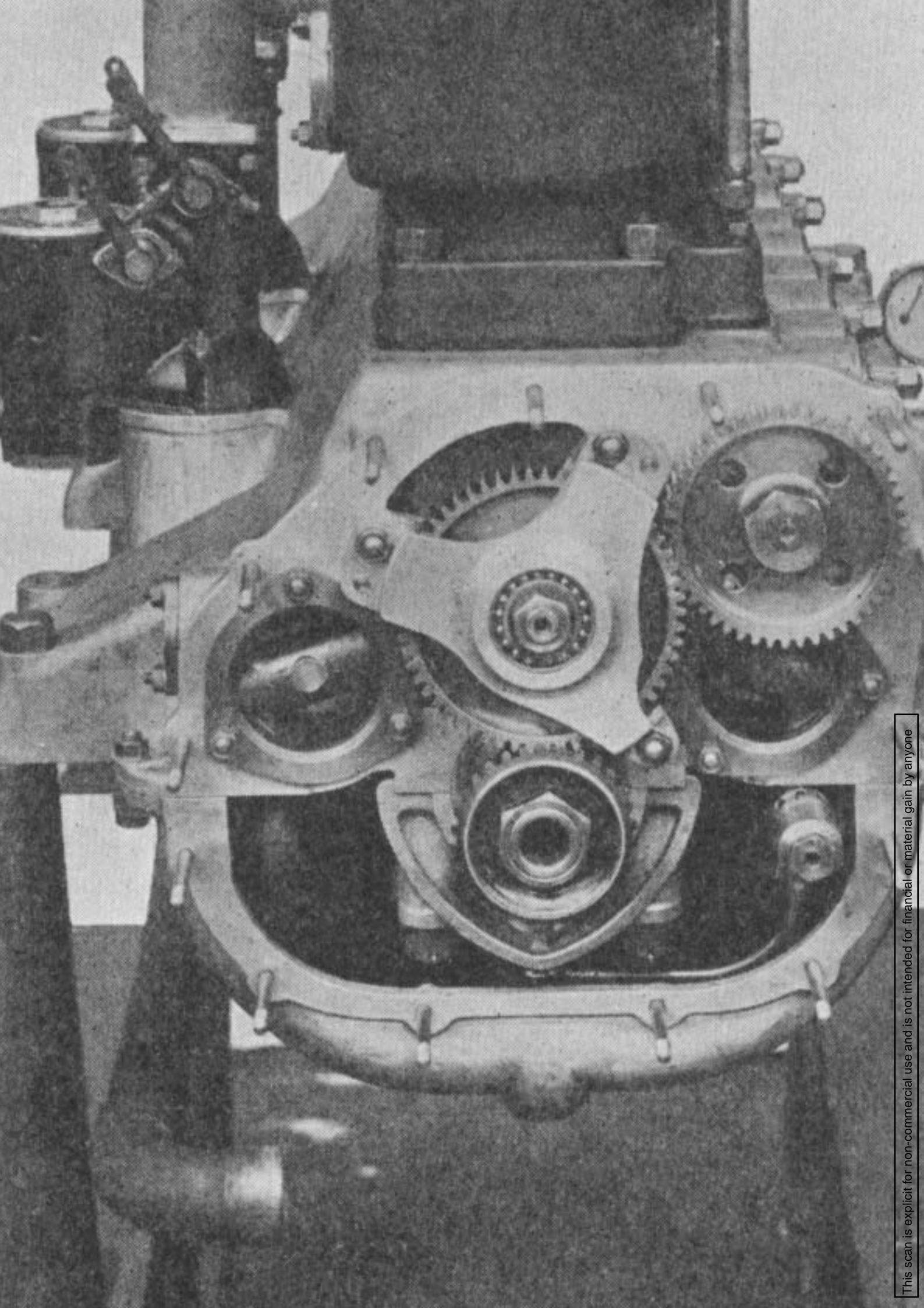
Where the number of auxiliaries is very large, it is often convenient to employ a cross-shaft driven by means of skew or spiral gearing; this arrangement is neat and compact and is much in favour for the drive of the magneto and water pump, both of which can then be disposed in a very accessible and convenient position. Such a drive is satisfactory and silent provided—

- (1) That adequate provision is made for dealing with the end thrust involved and for taking up any longitudinal backlash.

- (2) That ample lubrication is provided; for such gears, being of the rubbing rather than of the rolling variety, are naturally more dependent upon continuous lubrication.

There are, of course, endless possible combinations and permutations of auxiliary drive, but, generally speaking, the all-spur drive





of the speed, so that while at 1000 R.P.M. it may absorb only  $\frac{1}{4}$  H.P., at 2000 it will absorb about 2 H.P., and at 3000 nearly 7 H.P. The belt drive provided is usually nothing like adequate to transmit 7 H.P. at 3000 R.P.M., nor would it be reasonable to expend anything approaching this horse-power in cooling the engine. The practical result is that at any speed in excess of, say, about 1500 R.P.M. the belt slips. This results in rapid wear both of the belt itself and its pulleys. In many cases, if not in most, the normal running speed of the engine is such that the fan belt is constantly slipping, which is highly undesirable, and gives rise to most of the familiar troubles with this part of the mechanism. In general it is far better so to gear the fan that it runs at a considerably lower speed than that of the engine, and so to proportion the belt drive that it will slip only when the engine is running at momentarily excessive speeds.

There are some arguments in favour of driving the dynamo by belt, for the armature of a lighting dynamo has a considerable mass and will therefore tend to run at a constant angular velocity; if it is driven positively from a member which has a varying angular velocity there is bound to be acute disagreement. For this reason the use of any form of toothed gearing for driving the dynamo is undesirable, unless the gearing is situated at the flywheel end of the crankshaft. Chain-driving may be satisfactory, because a chain, thanks to its backlash, its elasticity, and its own weight, is capable of dealing with moderate variations in angular velocity, though a belt is the best of all in this respect. Most of the troubles with belt drives for dynamos are due to inadequate size and, in some cases, to the pernicious use of a three-cornered belt drive for the dynamo and fan, a practice which cannot be condemned too strongly.

**Lubrication Systems.**—Broadly speaking, the various systems of lubrication may be divided into two classes; namely, those which supply the needs of lubrication alone and those which make use of the oil both as a lubricant and as a cooling agent. To the former belong all systems of trough or splash lubrication and those in which a small measured quantity of oil is fed to each bearing, while to the latter belong all those systems in which oil is fed under pressure directly into the bearing itself. Both systems have their advantages and disadvantages, and the choice must depend upon a consideration of all the circumstances. As a broad generalization it may be said that in cases where the load factor on the bearings is high, pressure lubrication is essential for the sake of the cooling effect obtained,

while in cases where the load factor is comparatively low, splash lubrication or measured feed may be preferable, on account of the lower rate of wear, since less abrasive grit is imported into the bearing. It must be remembered that so long as oil and means of ingress are available, sufficient will always enter the bearing to maintain the oil film, whether it be applied under pressure or not, so that from the point of view of lubrication alone, as apart from cooling, the pressure system scores little or no advantage over the splash.

**Pressure Lubrication.**—The primary object of using pressure lubrication is to maintain a continuous circulation of a large quantity of cool oil through the bearings, in order to carry away the heat generated by friction. Bearing this in mind the objects to be aimed at are:—

- (1) To circulate as much oil through the bearings as possible.
- (2) To keep the oil as cool as possible.

The amount of oil that can be circulated through the bearings of an engine depends upon the pressure at which it is supplied, the clearance in the bearings, and the viscosity of the oil. It should always be remembered that the pressure in itself means nothing; it is only as a measure of the rate at which the oil is being circulated that it has any significance at all. In any pressure system, with a pump of given capacity and normal characteristics as to volume and pressure, the tighter the bearings or the higher the viscosity of the oil the greater pressure will be required to force a given quantity of oil through the bearing in a given time. If now, owing to wear in the bearings or to the use of an oil of lower viscosity, the same quantity can pass through the bearings more freely, the pressure will fall, but this does not mean that either the lubrication or the cooling effect has been reduced.

On starting up, with the oil and the bearings cold, a very high pressure will naturally be required to circulate the oil through the engine, but so soon as the engine warms up the pressure will fall rapidly, due to the reduced viscosity. Under these circumstances, however, the bearings are being equally well lubricated and in fact better cooled than when the oil pressure is high, since the drop in pressure is due merely to the increased flow through the bearings. This point has been emphasized, because there is a very prevalent belief that, with pressure systems, a low oil pressure indicates defective lubrication, and it is not at all uncommon for operators to use an oil of high viscosity in order to maintain a high pressure in the system, though by doing so they are really both reducing the



flow and increasing the coefficient of friction and therefore the heat generated at the bearings; in other words, defeating their own object.

In order to ensure a free circulation of oil it is desirable, with pressure feed lubrication, to keep the bearings reasonably slack. From the author's experience it would seem that a minimum clearance of about 0.0015 in. should always be permitted in all crankshaft and connecting-rod bearings when forced lubrication is employed. The practice of putting bearings up tight and allowing them to run in cannot be recommended, since it results merely in both checking the oil circulation and causing undue wear on the crankshaft itself.

Where pressure lubrication is employed and the oil is led into or near the centre of the bearing it is preferable not to use any oil grooves in the bearings, since these merely permit of the escape of oil without compelling it to circulate over the whole face of the bearing and so pick up heat from all parts; it is, however, sometimes desirable to provide flats on the crank to help distribute the flow from the oil hole. Both on the score of reducing friction surface and eliminating the danger of nipping, it is well to relieve away at the sides of the journal bearings, though such relief should not be carried to the extreme ends and so provide a free escape for the oil.

With pressure feed lubrication it is generally desirable to use an oil of low viscosity, since this will permit of a greater quantity being circulated through the bearings in a given time and, by reducing the friction, will reduce the heat generated. So far as the author can discover, the only advantages of an oil of high viscosity are: (1) That the oil film is thicker, the surfaces are therefore kept farther apart, and there is less chance of minute particles of grit bridging the oil film and wearing the metal surfaces; (2) it appears to be less liable to work its way past the piston rings into the combustion chamber, though this latter is very doubtful; and (3) the leakage loss is reduced. From every other point of view all the advantages would appear to lie with the use of a thin oil.

**Rate of Circulation.**—For engines of high duty the capacity of the pump should be such that, at normal speed, about  $\frac{1}{2}$  gallon of oil is delivered per hour for every brake horse-power of the engine. This of course is a purely empirical rate based on practical experience. In effect it means that even with very slack bearings and with an oil of low viscosity it is about the maximum that can be circulated under a reasonable pressure.

More recent experience with improved oil scraper rings on the

pistons and with improved design of oil outlets on the crankpins, has shown that as much as  $1\frac{1}{2}$  gallon per hour per brake horse-power can be circulated through the bearings, and a considerably higher load factor can in consequence be carried. For extreme cases such as racing aero-engines still further oil-cooling can be obtained by maintaining a brisk circulation of cool oil through the crankshaft itself from end to end in addition to that supplied to the bearings.

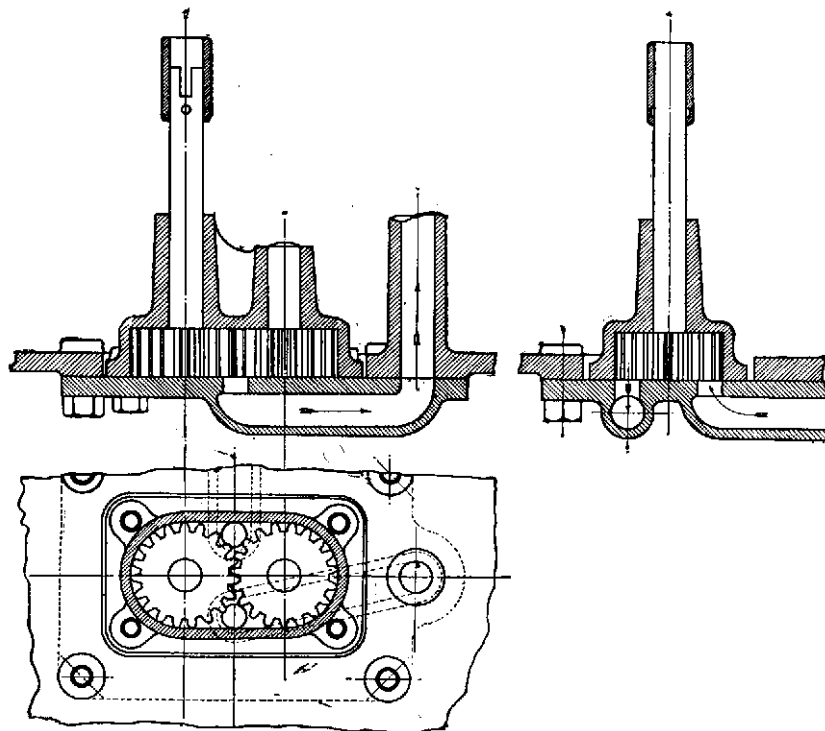


Fig. 86

**Type of Pump.**—There are three types of pump in general use: the gear pump, the ordinary plunger pump, and the valveless oscillating plunger pump. Typical examples of each of these are shown in figs. 86, 87, 88.

The former has the advantage that its motion is purely rotary, and that it is cheap and easy to accommodate, but it has four serious disadvantages:—

- (1) Its volumetric efficiency falls very rapidly with wear.
- (2) When slightly worn it has little or no suction lift, and unless actually submerged it is liable to let go of the oil.

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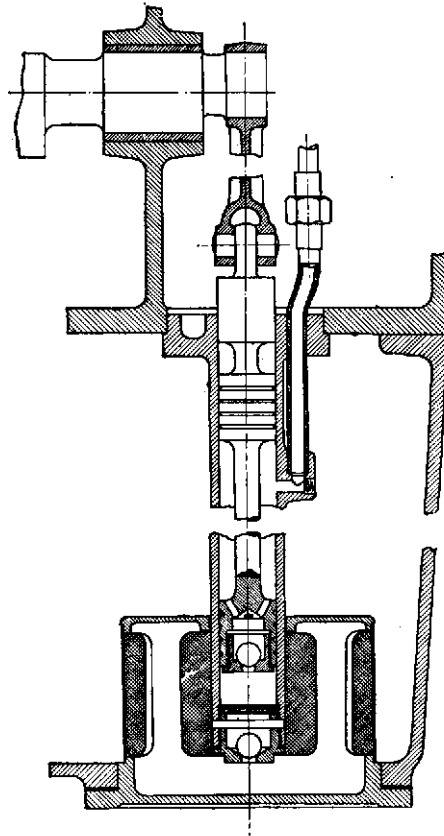


Fig. 87.—Reciprocating Oil Pump

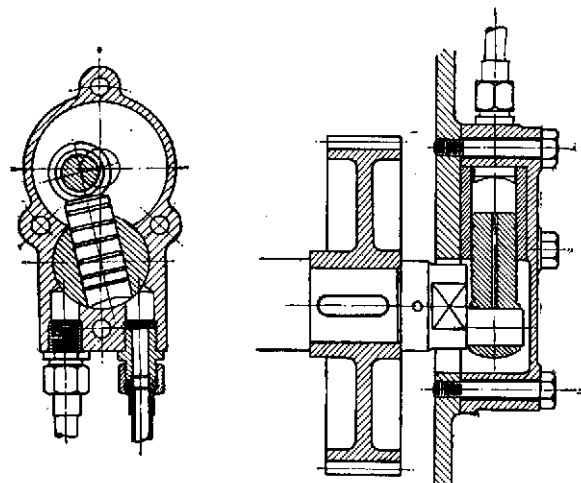


Fig. 88.—Valveless oscillating Plunger Pump

(3) It is very easily jammed by the entry of particles of foreign matter.

(4) It delivers a very rapidly pulsating flow as the teeth engage, and this is liable to set up high periodicity vibrations in the oil piping and so to cause fatigue and fracture of these pipes.

The ordinary plunger pump has the advantage that its volumetric efficiency varies little with wear, and also that it has a high suction lift; but, on the other hand, the suction valve is liable to stick, either open or closed, and so put the pump out of operation; also it limits the speed at which the pump will operate (fig. 87).

The valveless plunger pump as shown in fig. 88 is, in the author's opinion, the most satisfactory of the three types, for it has no valves to stick or to limit the speed of operation; also it is capable of dealing with dirt, particles of felt, and the other foreign bodies with which lubricating oil is often freely supplied. When run at high speeds it is necessary to provide an air vessel on the suction side in order to maintain a continuous flow in the suction pipe.

**Oil Relief Valves.**—In order to avoid the setting up of dangerous pressures at starting, or when using thick oil or tight bearings, it is necessary in forced lubrication systems to provide a pressure relief valve, and this should be set to blow off at about 25 lb. per square inch. When gear pumps are used and the flow of oil is practically continuous, almost any type of spring-loaded relief valve will serve, but when single-acting plunger pumps are used and the flow is pulsating, the design of the relief valve requires careful consideration. A form of relief valve which the author has found to give very satisfactory results is shown in fig. 89. It consists of a spring-loaded plunger sliding in a cylindrical casing and arranged to uncover relief ports when at the end of its travel. The diameter of the plunger should be from 50 per cent to 100 per cent greater than that of the pump, and its stroke, before uncovering the relief ports, should be nearly equal to that of the pump plunger, so that it has a swept volume equal to two or three times that of the pump. It should be loaded by a long flat-rated spring and will then act as a kind of mechanical air vessel, steadying the pressure at all times, and relieving it when it exceeds any predetermined limit. With such a relief valve the pressure supply from a single-acting plunger pump is kept almost free from fluctuation and will appear practically steady on the pressure gauge.

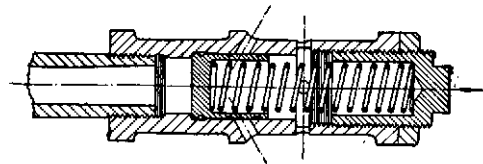


Fig. 89.—Plunger Type Oil Relief Valve

Such a relief valve should be fitted always as remote from the pump as possible in order to ensure that the oil has had access to all the bearings before reaching the relief; at the same time it is desirable that the pressure gauge connection be taken from as near the relief as possible, both in order to record a steady pressure and also to ensure that it reads the minimum pressure in the system.

**Oil Filters.**—The primary objection to pressure feed lubrication is that the circulation of a large quantity of oil through the bearings necessarily involves the circulation also of a large quantity of grit, and therefore tends to more rapid wear. The size of grit which causes ordinary wear, as apart from visible scoring, is such as no ordinary filter can hope to cope with, for it must be remembered that the thickness of the oil film may be of the order of one-tenth of a thousandth of an inch, and that it only requires a particle of abrasive slightly in excess of this dimension to span the oil film

and abrade the metal surface. It is obvious that no gauze or other such filter can possibly restrain the passage of particles of this size, and that the most that can be hoped for is that it will stop the larger particles, mostly of soft material, such as threads of felt from the filtration of the oil, or lumps of carbon from the pistons. The filter therefore may keep out foreign bodies which are otherwise liable to choke the oil system generally, but it cannot hope to influence the rate of wear. From the point of view of reducing wear it would appear that the best remedy is to provide ample settling space in the oil sump where the fine particles of grit can sink to the bottom undisturbed. On the whole, therefore, it would appear desirable to provide a coarse mesh filter, capable of arresting large particles which might choke up the system, and which will not readily itself become choked, and to rely upon an ample settling space for the precipitation of the fine abrasive matter.

Whether such a filter should be fitted on the suction or delivery side is still rather an open question. If on the suction side it is liable, in the event of neglect, to stop the flow of oil to the pump; if on the delivery side it is, when choked, liable to burst and deliver the whole of its collection into the bearings at one gulp. On the whole, the best compromise appears to be to fit a very coarse mesh filter capable of eliminating nuts, split pins, etc., on the suction side, and a somewhat finer mesh but substantial filter on the pressure side.

Experiments have recently been carried out with centrifugal filters or separators, and there seems a prospect that these may prove capable of eliminating some at least of the finer particles of abrasive matter and so reduce the rate of wear on the bearings. So far, however, very little is known as to their behaviour.

The system of lubrication in which a small measured quantity of oil is fed to each bearing has a good deal to recommend it for engines in which the bearings are not heavily loaded and which do not therefore require oil cooling.

The quantity of oil so fed should be the minimum required to maintain and replenish the oil film and provide a reasonable margin of safety.

The advantages of this system are :

- (1) That only clean oil enters the bearings, and that only in relatively small quantities, hence the amount of grit or abrasive matter imported into the bearing is reduced to the minimum, with the result that wear also is reduced to the lowest possible limit.
- (2) The amount of oil splashing about inside the crankcase is

reduced to the minimum, so that the cylinder walls cannot be over-lubricated; this is an advantage in one respect, but on the other hand it renders independent lubrication of the cylinders necessary and therefore introduces an extra complication.

The principal objection to this system, and it is a very serious objection, is that the life of each and every bearing so fed is dependent entirely on the continuous operation of the pump or other feeder, and it provides no reserve to tide over a temporary stoppage.

The use of sight-feed drip lubricators and ring oilers may be cited as the simplest expression of this form of lubrication, while, in its more developed form, it is customary to employ either a battery of small slow-moving pumps, each feeding one individual bearing, &c., or a single pump and distributing valve, the principal objection to the latter system being the relatively long interval which must elapse between replenishments.

The system of lubrication as shown in fig. 90 in which oil is fed continuously into troughs into which the connecting-rods just dip, is, in its effects, intermediate between the two foregoing. As regards wear, it has the advantage that comparatively little oil is actually forced into the bearings, and therefore the quantity of abrasive introduced is relatively small. It does not, of course, provide for any cooling by oil, since the quantity actually passing through or over the bearings is comparatively very small, but it does provide a good deal of reserve capacity in the event of any accidental stoppage of the pump. One of the difficulties, however, with this system is that it is often troublesome to provide, at all times, sufficient oil for all the crankshaft bearings without over-lubricating the cylinder walls. This difficulty can be met to a large extent by lowering the troughs, and so reducing the depth of immersion when the throttle is partially closed and the load factor on the crankshaft bearings is reduced. This can be accomplished by hinging one end of the troughs and connecting the other through suitable linkage gear with the throttle lever of the engine, as is done in the case of the Daimler sleeve valve engines.

When trough lubrication is used, it is sometimes possible to dispense with an oil pump and to use either the flywheel of the engine or a disc mounted on the crankshaft to throw oil up into a gallery from which it may be led down to the troughs by means of suitable ducts. This arrangement has, however, only a limited application, for it is not always possible to provide a disc of diameter sufficient to reach the oil, the highest level of which must be well

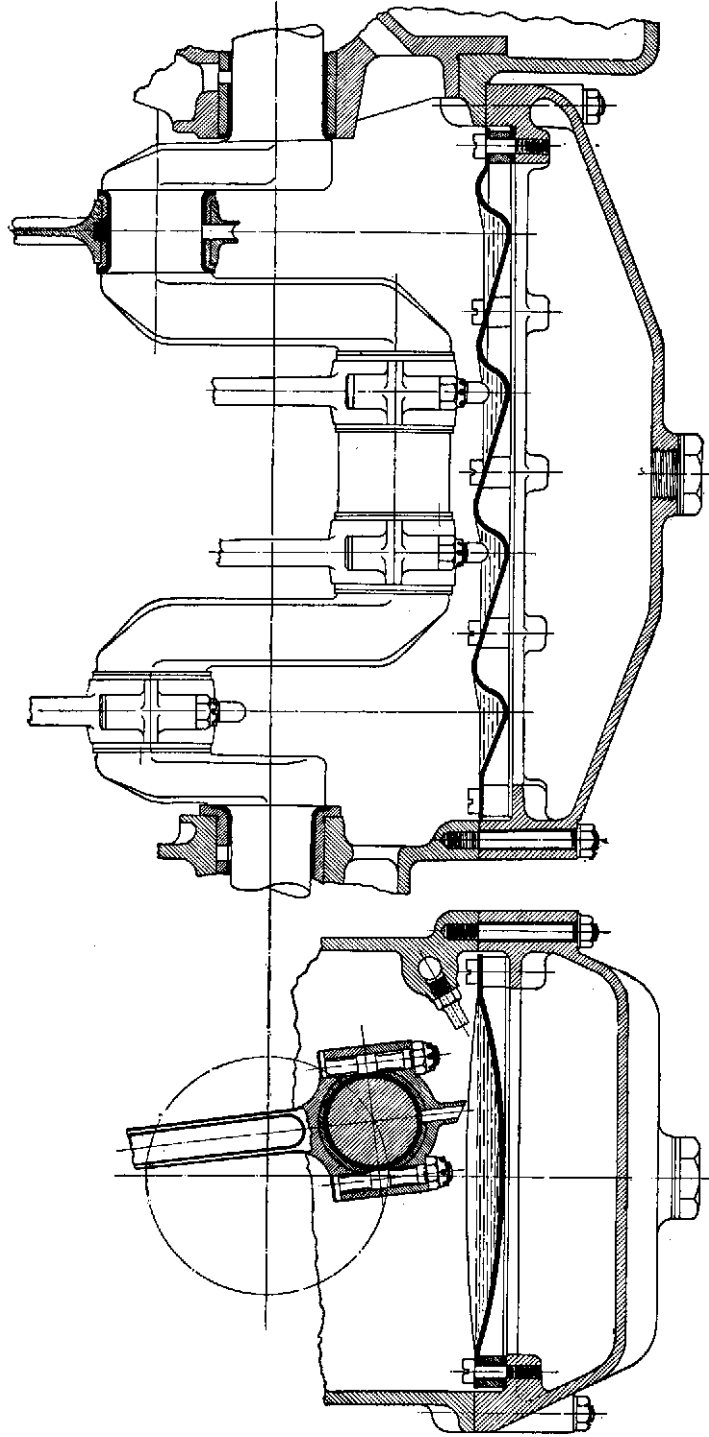


Fig. 90.—Typical Example of Constant Level Splash System

below that of the troughs. Unless there is room to fit a very large diameter disc, the permissible range of oil level becomes dangerously narrow.

To sum up : There can be no doubt but that wherever a high duty is required, involving either heavy pressures or high rubbing velocities, or both, forced lubrication alone can be relied upon, since this system provides the oil cooling so essential where high load factors have to be dealt with. The one and only objection to forced lubrication is the increased rate of wear due to the large quantity of extremely fine abrasive material introduced into the bearing. This objection can best be met by :

(1) The provision of ample settling capacity in the base chamber of the engine.

(2) The employment of materials for the bearings giving the maximum possible difference in surface hardness.

(3) The provision of centrifugal filters or separators, though not much is known as yet as to the efficacy of these devices ; ordinary filtering is, however, of little use from the point of view of wear.

Measured feed lubrication is no doubt very satisfactory for lightly loaded bearings and gives the minimum of wear, but it involves the use of many oil pumps and provides no reserve capacity ; as such it is rather dangerous, since the stoppage of any one pump in the group may prove disastrous. Trough lubrication is, on the whole, a good compromise for engines in which the duty is comparatively light. It tends to show a lower rate of wear than the forced system, though higher than the measured oil system. It has the advantage of providing a very fair reserve capacity, and is on the whole a fairly reliable and satisfactory system once its limitations are realized.



## CHAPTER VIII

### VALVES AND VALVE GEAR

The timing and operation of the valves are problems which require very careful consideration, for they are factors which have a powerful influence on the performance of the engine, and as such deserve the most careful consideration. Since the timing and opening periods must be decided upon before the cam gear can be designed, it will be well to consider this side of the problem first.

The features to be aimed at as regards the inlet valves are :

- (1) To induce the maximum possible weight of charge into the cylinder at full loads.
- (2) To expend the least possible energy in the process at all loads.
- (3) To produce the maximum of turbulence during the period of entry.

As regards the exhaust valves, the problem is merely that of getting rid of the exhaust with the least possible back pressure and the least distress to the valve gear. So far as the exhaust valve timing is concerned, there is very little to be said except that owing to the high terminal pressure at the time when the exhaust valve is first opened, the velocity past this valve is very high indeed and the heat flow at this period very intense. The high pressure of release, however, usually provides sufficient kinetic energy to counteract the friction and inertia in the exhaust pipe, so that a high mean velocity, both through the valve opening itself and through the ports, &c., is permissible without introducing any appreciable back-pressure, provided, of course, that there is no great resistance imposed on the flow of the gases at the outer end of the exhaust pipe. On the ground of heat dissipation it is desirable, and on the grounds of back-pressure it is permissible, to use small exhaust valves and to work with a high velocity through the valve opening and ports. In practice it is perfectly satisfactory to work with a velocity through the exhaust ports 50 per cent greater than that

through the inlet, provided of course that the latter is not already excessive. In a previous chapter it has been shown that, for all-round performance, the best mean velocity through the inlet valves is in the region of 150 ft. per second. The mean velocity through the exhaust valves may, therefore, be in the region of 220 feet per second.

In the case of the exhaust valves, it is particularly desirable, from every point of view, to use small valves with a high lift, because, in the first place, an exhaust valve can only get rid of the bulk of its heat through the valve seating, for the heat carried away along the stem forms only a very small proportion of the whole. It follows therefore that the smaller the diameter of the valve and the greater its lift, the better chance there is of keeping it reasonably cool. In this connection it is necessary to emphasize that, although the maximum area of opening is attained in the case of a flat-seated valve when the lift of the valve is equal to approximately one-quarter of the port diameter, it does not in the least follow that this should represent the maximum lift, because, in the first place, the valve is fully open only for a small proportion of its total opening period, and in the second, the orifice coefficient increases rapidly as the lift is increased; or, in other words, for a given pressure difference a greater weight of gas will pass through a given area of opening when that area is provided by a small valve with a high lift rather than by a large valve with a low lift. From these arguments it will be seen that, wherever possible, the total lift of an exhaust valve should always be at least equal to, and preferably greater than, one-quarter of the diameter of the port.

Again, the exhaust valve, unlike the inlet valve, has to be lifted from its seat against a pressure which may amount to anything up to 80 lb. per square inch, and for this reason also it is obvious that the diameter should be kept as small as possible in order to reduce both noise and wear and tear.

From every point of view, therefore, it is highly desirable to use the smallest possible diameter of exhaust valve and a high lift.

As regards the timing of the exhaust valve, two factors must be taken into account. It must be opened sufficiently early to permit of the exhaust pressure falling almost to atmospheric before the return stroke of the piston commences, and it must be held open sufficiently late to permit of the residual exhaust gas escaping right up to the very end of the stroke. It is impossible to give hard and fast figures for the most suitable setting for an exhaust valve, because

this must necessarily depend upon so many variable factors, such as the mean gas velocity through the valve port and the rate of acceleration of the valve. For a mean gas velocity, however, of about 200 feet per second through the valve port and a normal rate of acceleration, the best setting, in the author's experience, is that the exhaust valve should already have travelled through 50 per cent of its total lift when the piston is at the bottom dead centre and should be still 5 per cent open when the piston is at the top centre. The actual point at which the valve leaves and returns to its seating is practically meaningless as a guide to the valve setting.

With regard to the inlet valve setting, we have to take into account a number of factors which need not concern us in the case of the exhaust valve. Also we have to be much more careful, because not only does the volumetric efficiency, and therefore the power output, of the engine depend very largely upon the inlet valve setting, but also the negative work during the suction stroke may be unnecessarily high. We have to consider how many cylinders are drawing from any one source of supply and also how far our efforts to obtain maximum power output should be subordinated to the attainment of economy on reduced loads. It will be best to consider first the conditions as they apply to a single-cylinder engine for full power, and, later, to note what modifications are necessary to meet other conditions. As in the case of the exhaust valves, we ought to use a relatively small valve with a high lift, though for different reasons. In this case we want to obtain the maximum possible turbulence at the minimum expenditure of energy, therefore we require the highest possible orifice coefficient. For full-load running, in particular, we want to get the highest possible charging efficiency, and to achieve this we want to obtain a high velocity in the valve passage during the earlier part of the suction stroke, and to make use of the kinetic energy we have acquired during this period thoroughly to fill up the cylinder towards the end of the piston's stroke. To this end we require to open the valve rather gradually at first in relation to the piston's movement, to keep it as wide open as possible towards the end of the stroke, and to shut it as quickly as possible after the end. This does not necessitate the use of an unsymmetrical cam, as may at first sight appear, but it depends rather upon the angular setting of the cam in relation to the crankshaft. To illustrate the point :

Fig. 91 shows an ordinary symmetrical constant acceleration valve

lift diagram plotted against crankshaft degrees where  $0^\circ$  represents the inner centre and  $180^\circ$  the outward centre of the piston.

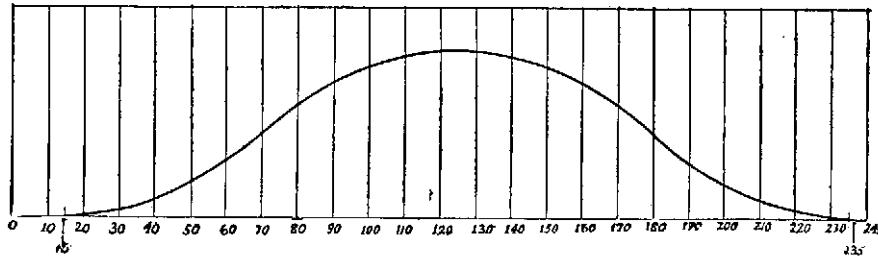


Fig. 91.—Valve Diagram, Constant Acceleration throughout

Fig. 92 shows the same diagram re-plotted in terms of piston displacement, while the dotted line represents the relative velocity

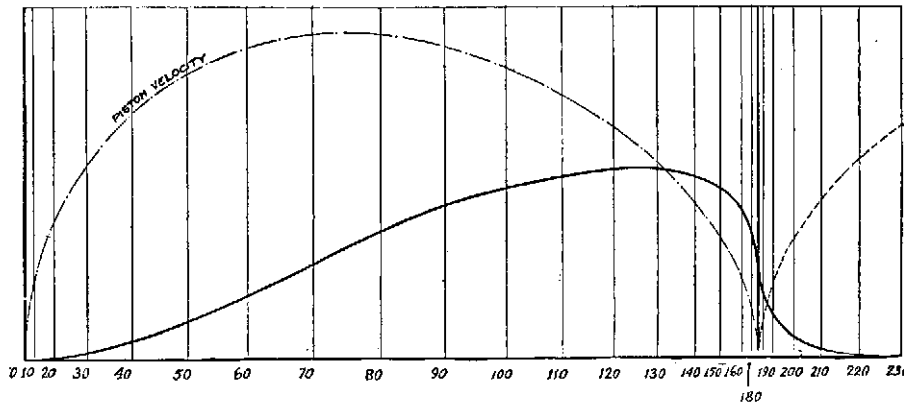


Fig. 92.—Valve Opening Diagram in relation to Piston Displacement

of the piston throughout its stroke (assuming in both cases that the ratio of connecting-rod length to crankthrow is 3.6 : 1).

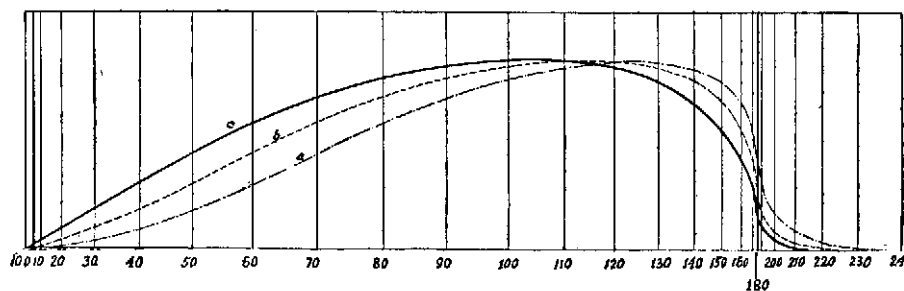


Fig. 93.—Effect of displacing Valve Opening Diagram in relation to Crankshaft

In fig. 93, *a*, *b*, *c* show the effect of displacing the angle of the camshaft by ten crankshaft degrees in each case, and illustrates

very clearly how the general form of any given valve opening diagram varies with the angular relation to the crankshaft.

Fig. 94 shows the change of velocity through the valve throughout the stroke in the case of diagram fig. 92, and assuming, for simplicity, that the pressure is constant or that the fluid is a liquid instead of a gas. From this diagram it will be seen that between the middle and the end of the stroke the velocity falls from 300 feet per second to nil, while the kinetic energy, due to this change in velocity, is made use of to pile up a static pressure in the cylinder. With careful design and with a mean gas velocity of 130 feet per second, it should be possible to pile up sufficient kinetic energy, during the earlier portion of the stroke, to overcome the frictional

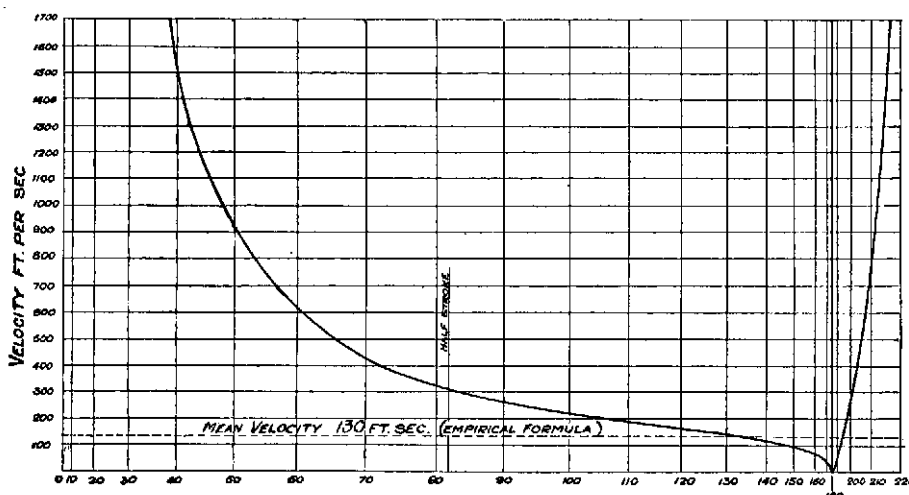


Fig. 94.—Equivalent Gas Velocity through Inlet Valve in relation to Piston Displacement

resistance of the valves, &c., and to charge the cylinder up to full atmospheric pressure by the end of the stroke; it then remains to close the valve as rapidly as possible after the bottom dead centre to avoid loss by expulsion during the early part of the compression stroke. This, then, is the setting for maximum power output, but it is not the best for fuel economy, for two reasons:

(1) Work is done by the piston in accelerating the air column during the period when it is travelling at a high velocity, and this is not returned until the piston reaches, or almost reaches, the bottom centre, consequently the pumping losses are relatively high, though, from the point of view of maximum power output, as apart from fuel efficiency, this is more than compensated for by the increased weight of the charge retained in the cylinder.

(2) When the inlet valve closes and the high velocity flow of gas towards the cylinder is stopped more or less abruptly, a reaction takes place, with the result that the air flows back through the carburettor and some fuel is liable to be blown out and wasted.

If, now, it is desirable to sacrifice maximum power for the sake of better fuel economy, it will be preferable to extend the period of opening of the inlet valve by about  $20^\circ$ , as shown in figs. 95 and 96, to allow it to open considerably earlier and to close a trifle later, the former in order to reduce the pumping losses and the latter to give more time to fill up the cylinder at the lower velocity. In this case,

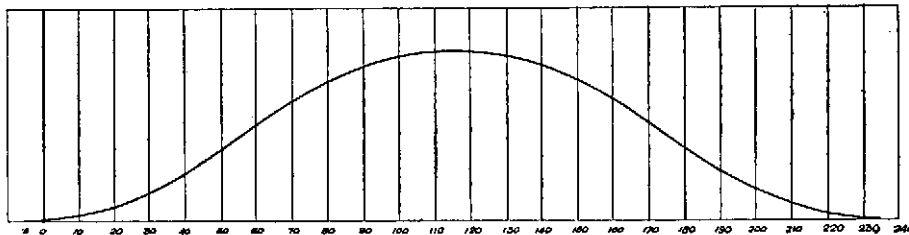


Fig. 95.—Valve Opening Diagram with Period extended  $20^\circ$  of Crank Angle

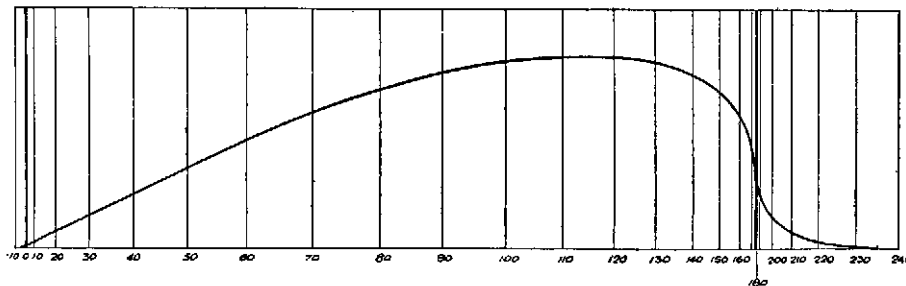


Fig. 96.—Valve Opening Diagram in relation to Piston Displacement

although the opening period of the valve is  $20^\circ$  longer and the actual effective opening area is considerably greater, yet the maximum power output will be slightly less.

The above considerations hold good only when the carburettor is placed, as it always should be in all single-cylinder engines, reasonably close to the inlet valve port. When any considerable length of induction pipe is interposed between the carburettor and the valve port, pressure oscillations of considerable magnitude will be set up, and these will tend to surcharge the cylinder at certain speeds and to starve it at others, while at all times they will tend to increase the blow back through the carburettor.

When working on reduced loads, by throttling, we are no longer

concerned with trying to fill up the cylinder, and our aim then becomes to maintain turbulence as far as possible and to reduce the fluid pumping losses. So far as the former is concerned, we can only rely on using the smallest possible valves, upon keeping the orifice

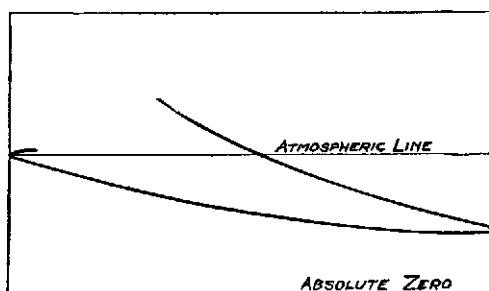


Fig. 97.—Indicator Diagram, Suction Stroke

coefficient as high as we can, and upon ensuring that the gases have as unobstructed an entry to the cylinder as possible after passing the valve. Another point of importance is the position of the throttle. If this is close up to the valve port, so that there is little capacity between the throttle and the

valve, then it is clear that during the idle strokes, this capacity will fill up to atmospheric pressure, or very nearly so, in which case the inlet should open early in the stroke and the suction diagram will be as shown in fig. 97; if, on the other hand, there is a considerable capacity between the throttle and the inlet port, then, at the commencement of the outward stroke, the pressure in the cylinder will be approximately atmospheric while that in the port will be considerably below atmosphere, with the result that so soon as the inlet valve opens the pressure in the cylinder will be reduced by gas flowing out through the inlet valve, and there will be some unnecessary negative work on the piston, as shown in the diagram, fig. 98. In such a case it will

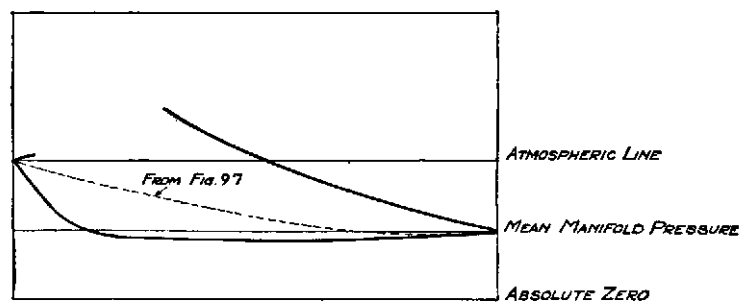


Fig. 98.—Indicator Diagram, Suction Stroke

be preferable not to open the inlet valve until the gases in the cylinder have been expanded down to a pressure corresponding to that in the port; this will give a diagram such as shown in fig. 99, which should be compared with the previous figure. It is not of course practicable

to obtain a valve timing which will be ideal for all conditions of load, and the best that can be done is to arrange the timing to suit the load at which the engine will be running for the majority of its existence. Generally speaking, for single-cylinder engines it would appear to be best to use a rather late opening inlet valve and to keep the throttle as close as possible to the valve; this will give the best results in normal working. Apart from the question of negative work, it is always desirable to reduce the capacity between the throttle and the inlet valve as far as possible, in order to subject the carburettor at reduced loads to a pulsating suction, and thus obtain better pulverization of the fuel, rather than to a continuous suck at a low velocity. Consideration will show that if the capacity between the throttle and the inlet valve were infinite the velocity through the carburettor would be continuous throughout the cycle;

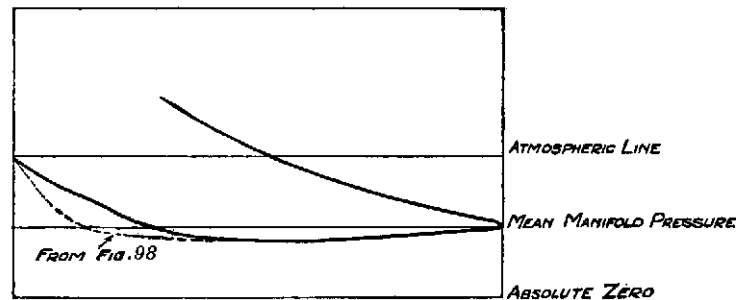


Fig. 99.—Indicator Diagram, Suction Stroke

while, on the other hand, if there were no capacity, then the mean velocity through the carburettor during the suction stroke would be just four times as great and the pulverization of the fuel correspondingly better.

The case of the single-cylinder engine is relatively simple as compared with that when several cylinders draw from one source of supply. Also it is difficult to treat the problem of valve timing and distribution separately, for they are so closely interdependent; together they form an intensely complex problem, and, probably, the least understood of all problems connected with the modern internal-combustion engine.

We will, however, assume for the time being that we are dealing with a homogeneous mixture of gas and air flowing in the induction pipe, and endeavour to see how best to deal with certain of the more common cases.

CASE 1. Two cylinders with cranks at  $360^\circ$  and even firing.



2. Two cylinders with cranks at  $180^\circ$  and consecutive firing.
3. Four cylinders with cranks at  $180^\circ$ .
4. Six cylinders with cranks at  $120^\circ$  fed by two carburettors, each distributing to a group of three cylinders.
5. Six cylinders with cranks at  $120^\circ$  fed by one carburettor.

So far as the exhaust valve is concerned the conditions are substantially the same in all cases, and it is the inlet valve timing alone which need be considered.

CASE 1. This is simply two single-cylinder engines operating alternately ; there is no overlapping, and the problem is the same as that of the single-cylinder engine.

CASE 2. This is always a very difficult one to deal with. Probably the only satisfactory solution is to provide two separate carburettors and two exhaust pipes and so treat as two separate single-cylinder engines.

When only one carburettor is fitted from which the two cylinders draw consecutively, the best method is probably to employ very late opening inlet valves in order to avoid overlap. It is obvious that if the first piston sets up a high velocity in the induction system and relies upon the kinetic energy so acquired to fill up the cylinder at the end of the stroke, then it is fatal to allow the second inlet valve to open until the first cylinder is completely filled, for the energy acquired will be expended simply in forcing gas into the second cylinder at the commencement of its suction stroke when it is not required, while the first cylinder will be starved. So long as there is any appreciable overlapping in the period of opening of the inlet valves the first cylinder will always be starved while the second will be surcharged. On the other hand, to avoid overlapping and yet keep the inlet valves open long enough to allow the cylinders to fill up is difficult and necessitates a very late opening indeed. Also the short period introduces difficulty in the case of high-speed engines in regard to operation. Again, such very late opening as is necessary to obtain equality between the cylinders will render the suction of the engine very noisy when running on full load, for this is always an objectionable feature of very late opening inlet valves. It would appear that on the whole the best method of dealing with this very unsatisfactory form of engine is, in the case of comparatively slow-speed engines, to open the inlet valves very late ; and in the case of high-speed engines, to employ two altogether separate carburettors and throttles. In either case it is most desirable to fit two separate exhaust pipes, for overlap between the exhaust valves cannot possibly

be avoided, and unless two pipes are fitted, cylinder No. 1 will discharge high pressure and highly heated exhaust products into No. 2 just before the completion of the exhaust stroke, thus filling the clearance space of this cylinder with exhaust gas under pressure, at the one point in the cycle at which the presence of highly heated gas is most undesirable. In some instances a fairly uniform power output may be obtained from such an engine because the starving of No. 1 is balanced by the drowning out with exhaust products of No. 2, with the result that on full throttle both cylinders give a much reduced but more or less equal performance, though in such cases the evils no longer balance one another on reduced loads, when the presence of an excess of exhaust products is more than usually objectionable.

CASE 3. Four cylinders drawing from a single carburettor.

Except that the flow in the branch pipe from the carburettor is relatively constant, this case is almost as difficult to deal with as the last. So far as valve timing is concerned, either all overlap of the inlet valves must be avoided, with the resultant difficulties of an unduly short opening period and noise when running on full throttle, or a certain amount of irregularity, coupled with a reduction in the maximum power output due to robbery of one cylinder by another, must be tolerated. In a four-cylinder engine, however, the branch pipes from the throttle to the several cylinders are generally of considerable length, and in some cases use can be made of the kinetic energy in these branches to fill any one cylinder despite attempted robbery by another, more especially so in the case of very high-speed engines. So far as throttled conditions are concerned, this case may be regarded as one in which the capacity between the throttle and the inlet valves is infinite, and, therefore, in which a late opening inlet valve is definitely desirable. On the whole, it would appear that, for this type of engine, it is best to use very late opening and relatively early closing valves in the case of moderate-speed engines and those which operate normally at a comparatively low load factor, because this will tend to give more economical running at reduced loads; and to use comparatively early opening and late closing valves for engines which are normally run at very high speeds or at a high load factor despite the fact that for maximum power at comparatively low speeds this setting will be inferior to the late opening and earlier closing.

Probably the best results from every point of view can be obtained when two independent induction systems are used, one feeding the inside, and the other the outside pair of cylinders. This arrange-

ment eliminates all question of overlap, and has been applied by the author in the case of several four-cylinder engines designed to give a very high power output and economy.

The once common practice of permitting overlap between the inlet and exhaust valves of the same cylinder is certainly not to be recommended. The arguments for providing overlap in this manner are (1) to make use of the kinetic energy of the gases in the exhaust pipe to scavenge the cylinder and so obtain a greater weight of "live" gas in the cylinder when running at full power; and

(2) To lengthen the period of valve opening, with a view to reducing the stresses on the valve mechanism at very high speeds.

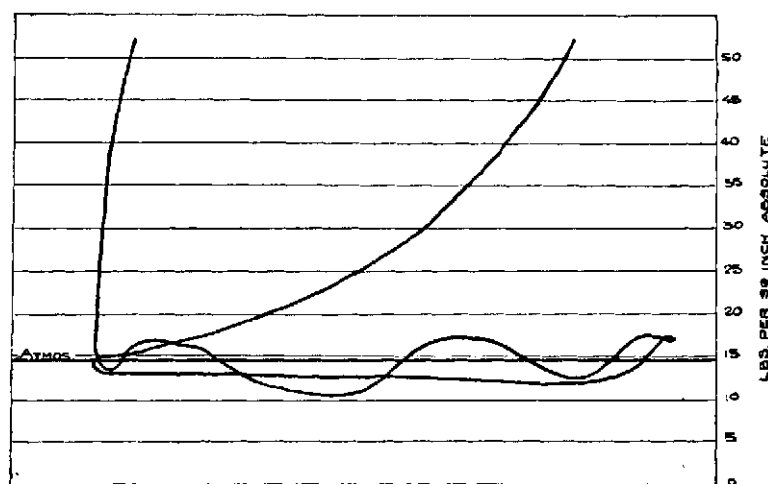


Fig. 100.—Indicator Diagram at 1500 R.P.M. shows Pressure Variations during Exhaust Stroke

The first is wrong in inception and bad in practice, for, in the first place, the gases flow out through the exhaust pipe in a series of pulsations, the pressure ranging usually from about 3 lb. above to 3 lb. below atmosphere, depending upon the length of the pipe, as shown in the indicator diagram, fig. 100, which is taken from one cylinder of an engine running at 1500 R.P.M. It is just as likely that the pressure at the exhaust valve, at the moment when the inlet valve opens, will be above atmospheric as it is that it will be below, in which case a reverse process will occur and exhaust gases will be driven back into the induction pipe. Hence, any advantage which may be gained at one particular speed of rotation will be counteracted by a corresponding but larger loss at other speeds. Again, the valves are seldom far enough apart in the cylinder to permit of any useful scavenging effect being obtained, fresh gas

merely being drawn out of the inlet port and into the exhaust, and so lost. On reduced loads the practice of overlapping is particularly bad, for it must be remembered that at this period the pressure of the residual exhaust products in the exhaust system is round about atmospheric, while the pressure in the induction system may be half an atmosphere or less, with the result that exhaust products are simply sucked back from the exhaust system into the induction system, and that under just those conditions when the presence of exhaust diluent is most detrimental to efficiency. The second argument need not apply if the valve mechanism be properly designed, as will be shown later. In any event, care should always be taken to prevent adjacent cylinders from exhausting into each other. It is usual nowadays to combine together the exhaust ports of the two central cylinders and to connect the exhaust manifold to the two outside and the central pair; this arrangement is fairly satisfactory, but it is better still to use either three or four quite separate exhaust pipes between the valve ports and the manifold, though in practice this is sometimes inconvenient.

CASE 4. Six cylinders drawing from two carburettors, each feeding one group of three.

This case is simple, there is no overlap of the inlet valve periods, and in so far as valve timing is concerned each group may be treated as three single-cylinder engines.

CASE 5. Six cylinders fed from a single carburettor.

In this case (A) the capacity between the throttle and the inlet valve may be regarded as infinite; (B) overlapping of the opening periods of the several inlet valves cannot, under any circumstances, be avoided; (C) unless long separate branch pipes be provided to each cylinder—which is almost impracticable on the grounds of distribution, confusion of pipe-work, &c.—little or no use can be made of the kinetic energy of the gases in the induction pipe owing both to the excessive overlapping and to the constant reversals of direction of flow in the induction manifold. It will therefore be impossible fully to charge any of the cylinders, or, indeed, to attain a final suction pressure in any of them appreciably in excess of the mean pressure in the induction manifold. Since the capacity between the throttle and the inlet valves is so large, it will pay—particularly on light loads—to employ rather late opening inlet valves in order to reduce to the minimum the negative work in the cycle. In such a case the late opening of the valves will cause very little noise at full load because the suction through the carburettor

is fairly continuous in any event. Although with six cylinders drawing from one source of supply it is practically impossible to obtain so high a power output as when supplied in groups of three, yet certain advantages may be taken as compensation. In the first place, with such an arrangement, there is much less objection to blow back through the inlet, since any discharge from one cylinder during the early period of the compression stroke will be taken up by another cylinder of the group and will not result in a reversal of flow through the carburettor. It will be worth while, therefore, to allow the inlet valves to close much later than usual, and also to allow them to close much more gradually, so that at low speeds an appreciable proportion of the gas drawn into the cylinders will be discharged back into the induction manifold, while at very high speeds this will not occur, since the suction pressure will be lower and, at the same time, the wire-drawing of the nearly closed valve will be more effective. The practical result of permitting blow back during the early period of the compression stroke will be to reduce the effective compression ratio, leaving the expansion ratio unaffected. The efficiency, therefore, will not be reduced, but the compression pressure at full load will fall automatically as the speed is reduced.

Now, since the tendency to detonate depends in practice very largely upon both the compression pressure and the time element, it thus becomes possible to effect a balance between compression pressure and time. At low speeds, when the time is relatively considerable, the compression pressure will be lower and the tendency to detonate will be equal or less (according to the actual proportions chosen) instead of much greater, as the speed is reduced on full throttle.

It may be argued that, with late closing inlet valves, the useful torque at low speeds will be considerably reduced, but this is true only when comparisons are made with an engine of very low compression and, therefore, low efficiency.

In the case of a high compression engine with early or normal closing inlet valves, full use cannot usually be made of the greater weight of charge retained in the cylinder on account of the excessive detonation set up; it then becomes necessary either to retard the time of ignition, to set the carburettor to give an over-rich mixture at low speeds, or partially to close the throttle. In any case the full torque at low speeds is lost. In effect, the use of late closing inlet valves as applied to such engines is that a considerably higher ratio of compression can be used without trouble from detonation and a

substantially higher economy obtained thereby. The actual available torque at low speeds is little, if any, reduced, while the torque at high speeds is increased, owing to the longer period of charging.

The cases considered cover practically all the range; where greater numbers of cylinders than six are employed they are always divided into groups, which come under one or other of the categories we have considered.

To sum up, so far as the exhaust valve is concerned, the problem of its design and operation is the same for all engines irrespective of grouping or numbers of cylinders. It should be as small in diameter as possible; the lift should in no case be less than one quarter of the port diameter, and preferably it should be as much as 30 per cent of the port diameter. In all cases it should be lifted and closed as rapidly as possible, while, as regards timing, it is a good rule that it should be at about half lift at the outward centre of the piston and 5 per cent open at the end of the exhaust stroke.

As to the inlet valve, this, too, should be kept small, with a lift not less than one quarter of the port area, in order to obtain the maximum of turbulence. Its time of opening and closing must depend, to some extent, upon the number and grouping of the cylinders, but, except in the case of six cylinders drawing from one carburettor, it should always be closed as rapidly as possible, the primary aim being to keep it nearly wide open at the end of the suction stroke and to close it as soon as possible after. In the case of six cylinders drawing from one carburettor, the inlet valve may be opened and closed much more leisurely.

**Cam Design and Valve Operation.**—In designing the cams for operating either the inlet or exhaust valves, the primary considerations are both to open and to close the valves as rapidly as possible with the minimum of stress or noise, and at the same time to arrive at a form of cam which can readily and easily be produced. It is only too often that a cam is designed to give a specific opening diagram, and to provide, say, a constant rate of change of acceleration throughout the whole opening period, which may be ideal on the drawing-board but almost impossible to reproduce with accuracy. It must be remembered that a cam profile cannot usually, if ever, be “generated” in the grinding machine, but must be reduced from a master cam, and that the former must be hand-made—the accuracy of its profile can be ensured only when the contour is made up of simple arcs of circles and tangents.

Again, it is always very desirable to avoid any concave surfaces,

since these limit the radius of the grinding wheel which can be used to produce them, thus imposing a very tiresome limit on the manufacture. By a suitable combination of cam and follower the necessity for concave surfaces can always be avoided.

In the operation of a spring-controlled valve by a cam, the first movement of the cam imparts a positive acceleration to the valve until nearly half lift, when the acceleration changes and becomes negative—the valve is then under the control of the spring, whose tension must be sufficient to overcome the inertia due to acceleration. From about half lift to full lift, and from full lift to half closed, the valve is entirely under the control of the spring. For the first half of the lift and the latter half of the closing period the spring is inoperative and the movement of the valve is controlled directly by the cam. The spring, therefore, does not come into effective operation until the valve is nearly half open, and ceases to operate when the valve is about half closed. The rate of acceleration permissible while the valve is under spring control is governed by the pressure and rating of the spring, but during the first and last portions of the valve's movement the rate of acceleration is governed solely by the permissible pressure against the flank of the cam. To make the best use of the spring material the rate of acceleration during the spring-controlled period should be such as to correspond as nearly as possible with the rating of the spring—that is to say, the rate of acceleration should increase steadily as the spring is further compressed. During the first and latter portions of the valve's movement the rate of acceleration may generally be much greater, but should be kept more or less constant.

These considerations indicate that constant acceleration throughout the whole period is by no means ideal, or even desirable. The acceleration during the period of cam control may usually be very high and more or less constant, while the acceleration during spring control should be as low as possible in order to use light springs, and should vary uniformly with the rating of the spring. In any case, of course, the acceleration must be limited to that at which the spring pressure will always overcome the inertia of the valve, and that by a margin sufficient to cover any friction of the valve in its guide.

Fig. 101 shows a convenient form of sheet for setting out cam designs. On this sheet are shown :

1. The acceleration, both positive and negative, during the opening period.
2. The corresponding velocity curve.

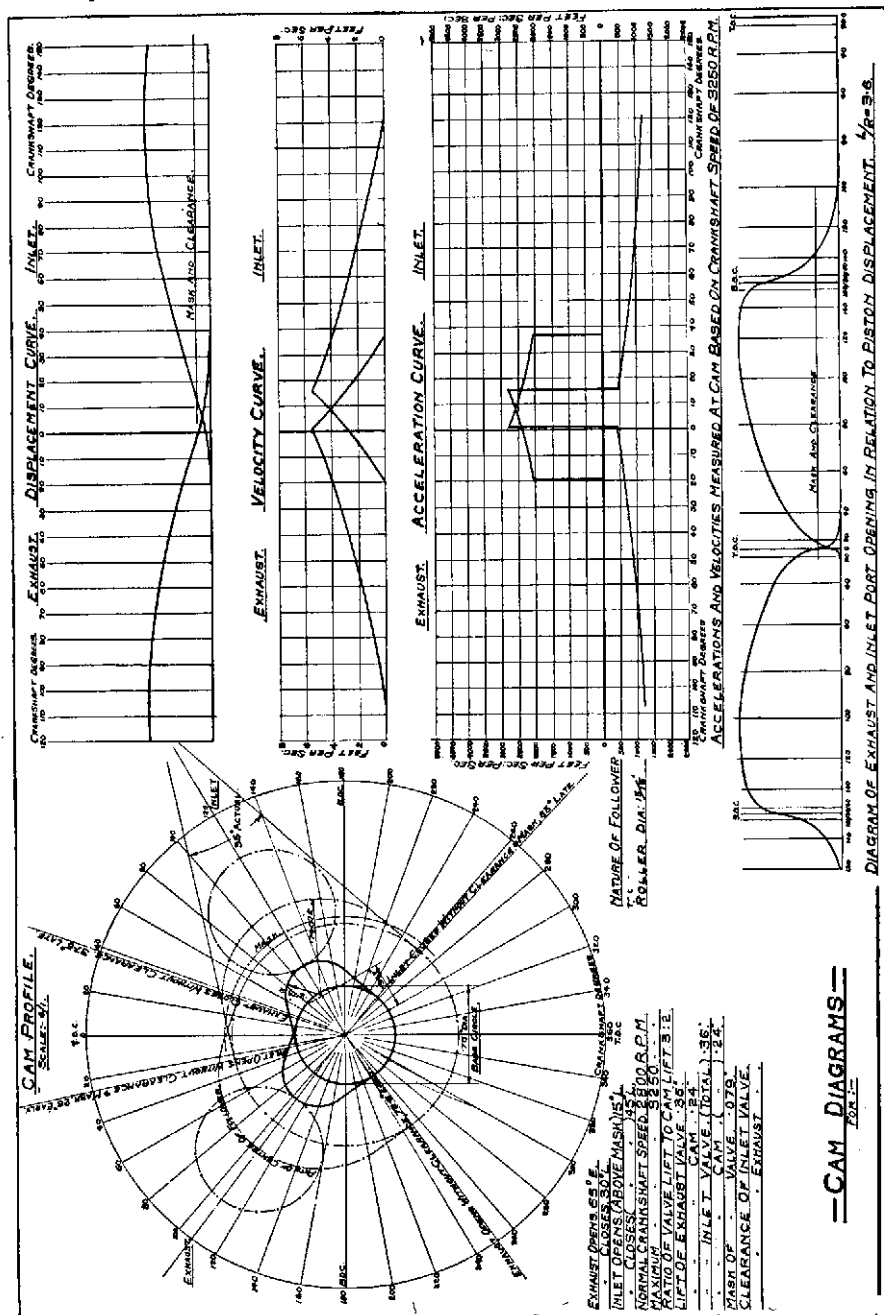


Fig. 101.—Cam Diagrams

3. The corresponding valve movement on a time basis.
4. The valve movement in relation to piston displacement, i.e. the valve opening diagram.
5. The evolution of the contour of the cam.



**Permissible Acceleration.**—This must be considered from two aspects—the highest permissible acceleration while under spring control and the highest permissible acceleration under direct cam control. Both, of course, depend largely upon the total reciprocating weight of the valve and its gear, which must include half the weight of the spring.

With regard to the acceleration under spring control—this is determined by the weight of active spring material and the permissible stress in the material. In the author's experience, so long as the stress in the material is kept down to from 40,000 to 50,000 lb. per square inch, ordinary spring steel coil springs will last almost indefinitely, even in the highest speed engines. When the valve is small, i.e. less than 2.0 inches diameter, and is operated more or less directly from the cam, and when the weight of intervening gear, such as rockers, tappets, push-rods, etc., is comparatively small, a maximum acceleration of about 2500 ft. per second per second, corresponding to a spring pressure of 78 times the reciprocating weight when the spring is fully compressed, is usually permissible, though except in excessively high-speed engines it is seldom necessary to employ so high a rate of acceleration. The acceleration at the point when the spring first takes up the load must, of course, be lower, in proportion to the rating of the spring. In the case of moderate-speed engines there is no need to employ anything like such a high rate of acceleration when under spring control, and for engines of about 15 to 20 B.H.P. per cylinder running at maximum speeds not exceeding 2000 R.P.M. an acceleration under spring control of 800 to 900 ft. per second per second will permit of as favourable a valve opening as can be desired. At the other end of the scale, a limit is fixed for minimum spring tension; this must always be sufficient, in the case of the exhaust valves at all events, to resist the vacuum formed in the cylinder when running throttled. In practice it is found that in order to prevent the exhaust valves from being sucked open, particularly when the engine is in a state of vibration, a spring tension of at least 11 lb. per square inch is required, reckoned on the area of the head of the valve. In the case, therefore, of a valve the area of whose head is, say, 3 square inches, a minimum spring tension of 33 lb. will be required when the valve is on its seating. With a spring of normal rating, the tension, when the valve is fully lifted, will be at least 50 lb. Now the reciprocating weight of such a valve and its attendant gear will probably be in the neighbourhood of 1.5 lb., and the maximum permissible rate of acceleration in such a

case will therefore be  $\frac{50 \times g}{1.5} = 1080$  ft. per second per second. This, however, makes no allowance for friction in the guide, but even after making a generous allowance for this factor it will be seen that the case for all moderate-speed engines, when the valves are directly operated and the intervening gear is not heavy, is easily met by the provision of a spring of only just sufficient strength to prevent the exhaust valve from being sucked open when running idle.

The highest permissible rate of acceleration while the valve is under direct cam control depends upon the type of follower used.

This may be either a roller, a curved slipper, or a plain flat-footed or "mushroom"; examples of each of which are shown in figs. 102,

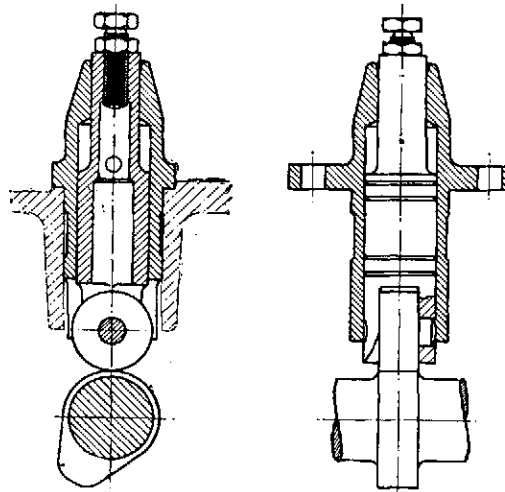


Fig. 102.—Roller Ended Follower

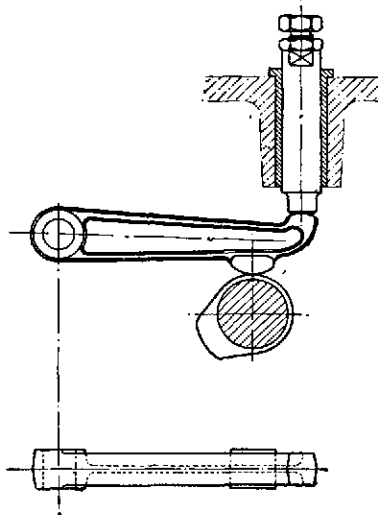


Fig. 103.—Slipper Type Follower

103 and 104. At first sight, it might appear that the roller is the most satisfactory form, but on investigation it will be found that this is far from being the case, and for three reasons:

(1) The whole of the load is taken on the roller pin whose projected area is necessarily very small, this pin cannot conveniently be pressure lubricated, and its facilities for obtaining replenishment of oil are very poor, hence it is easily overloaded.

(2) Owing to the changes in surface velocity as the cam revolves and to the inertia of the roller itself, it follows that the latter cannot truly roll, but must skid, and that just at

the period when the pressure on it is at a maximum.

(3) The use of a roller greatly increases the weight of the tappet

gear. For moderate-speed engines, when the loads are comparatively light, the use of a roller is permissible, but it should never be used in very high-speed engines for the reasons stated above.

The second type, namely, the curved slipper, is better than the

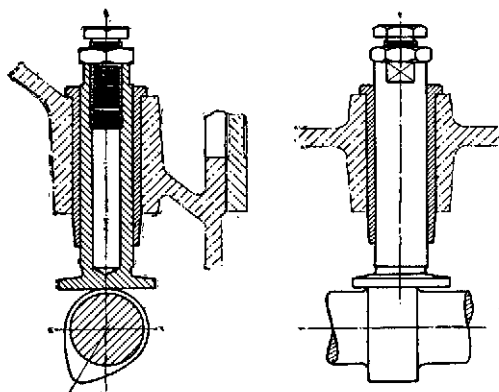


Fig. 104.—Mushroom Type Follower

roller, in so far that it involves no bearing which may become overloaded and break down, but it has the disadvantage that it presents only a very small area of rubbing surface against the cam, and so is liable to wear. Both the roller and the slipper "skid," but the latter skids much more rapidly and presents only one face, while the former

skids slowly and presents a continual change of face. Against this, however, must be set the fact that both the radius and the width of the slipper can be much greater than that of any roller. On the whole, the roller has the advantage on the score of wear so long as the pressure is light and the pin bearing is not overloaded, while the slipper scores when the rate of acceleration, and therefore the pressure, is high, for though it may wear considerably and therefore require renewal, it will not break down altogether.

The third type, namely, the flat-footed or "mushroom" tappet, is, in the author's opinion, the most satisfactory of the three, but it, too, has certain limitations, for it necessitates the use of a cam with a larger base circle, which cannot always be provided. Followers of this type should always be offset sideways, so that the sliding of the cam tends to rotate them. Under these conditions practically the whole surface of the flat foot is made use of, and the wear is less than with either of the other two types, while there is no pin bearing to be overloaded. It has all the advantages of the roller type in that it constantly presents a new surface in contact with the cam, and all the advantages of the slipper type, in that it has no bearing to fail, while the conditions as regards lubrication are ideal. The one practical objection is that it is usually impossible to employ a low rate of positive acceleration when, if ever, this is desired; hence it is difficult to obtain such quiet running as with the other types, though with careful design the difference is very slight. With flat-footed tappets

it is perfectly safe to employ a very high rate of acceleration, for the pressure occurs only when the flank and not the tip of the cam is in contact with the tappet. It is perfectly safe with the latter type, with cams, say,  $\frac{5}{8}$  in. wide and a tappet diameter of, say,  $1\frac{1}{4}$  in., to apply an average pressure of 250 lb. during the period of cam control. In the case of a valve and its gear weighing 1.5 lb. with a spring tension at rest of, say, 40 lb., this will correspond to a rate of acceleration of about 4500 ft. per second per second. With the roller type it is very doubtful whether it would be safe to exceed an acceleration of about 2000 ft. per second per second. Since, from the point of view of wear, it is only the average pressure which need be taken into account, it follows that there is no particular advantage to be gained by keeping the acceleration constant during the period of cam control, and that there is no objection to following the line of least resistance and making the flank of the cam either a tangent or a simple circular arc.

It is desirable to keep the base circle of any cam as small as practicable in order to reduce the rubbing velocity between the cam and its follower, and this applies whether the follower be a roller or a slipper.

It must be remembered always that the "effective" radius of any cam is the actual radius from the centre of the camshaft to that of the roller or to the centre of radius of the slipper. Hence it matters not, so far as the valve motion is concerned, whether the cam be large and the roller small, or *vice versa*, except when a flat-footed follower is used. In this latter case it is necessary to use a cam of comparatively large diameter, but when flat-footed followers are used the wearing surfaces are so large and the facilities for lubrication so good that a high rubbing velocity is much less objectionable.

The author is greatly indebted to one of his assistants, Mr. R. J. Cousins, for the construction and development of the following method for arriving readily at the most suitable cam contour, to comply with any given set of conditions.

When setting out the design of any cam the first question is that of deciding whether the cam profile, as dictated by the valve opening requirements, is permissible mechanically, rather than constructing a cam to conform to some ideal figures for positive and negative acceleration. This being the case, an analysis was made of the general form of cam in which the flanks and nose are composed of circular arcs or straight lines, and a series of graphs prepared, giving

practically on sight the actual acceleration at any point for all reasonable proportions of cam.

Fig. 105 shows the acceleration on tangent flanks (dotted curve) and round noses with circular followers, also harmonic cams with flat or mushroom followers.

Fig. 106 gives the figures for hollow flanks with circular followers.

Fig. 107 deals with round (convex) flanks with circular followers.

Fig. 108 shows various forms of cams, internal and external, and indicates the graph to be used in each case.

The formula is the same in all cases, viz.:

$$\text{Acceleration in ft. per sec. per sec.} = \frac{R \times C \times N^2}{100,000},$$

where  $R$  = radius in inches (see fig. 108),

$C$  = a coefficient from the corresponding graph, depending upon the form of cam and the angle from the base circle (for flanks) or apex (for noses),

$N$  = revs. per minute or crankshaft (assuming camshaft runs at half-engine speed).

A preliminary lay-out of the cam is first made, taking the known factors (which usually include the approximate base circle diameter, lift, period of opening, and room available for roller or follower).

The positive acceleration at the beginning and end of the flank and the negative acceleration (on the springs) at beginning of nose and apex are then read off, and the form of the curve visualized from the suitable graph, when it will be obvious at once if the cam is out of court mechanically, in which case suitable compromises must be made.

Assuming that a tangent cam has been constructed in the first instance, a certain amount of adjustment of the positive acceleration may be made by altering the distance from camshaft centre to the centre of the roller or slipper, the acceleration varying directly in proportion, but if this would make the roller too large on the one hand or make the curvature of the roller or slipper too sharp on the other, the flank may be made curved, concave, or convex as may be necessary, to increase or decrease the positive acceleration.

This will tend to have the opposite effect on the negative acceleration on the nose of the cam, for it will give either more or less time in which to bring the valve parts to rest from maximum velocity.

In cases where the figures are too high all round, it becomes imperative to increase the period to the longest possible and to decrease

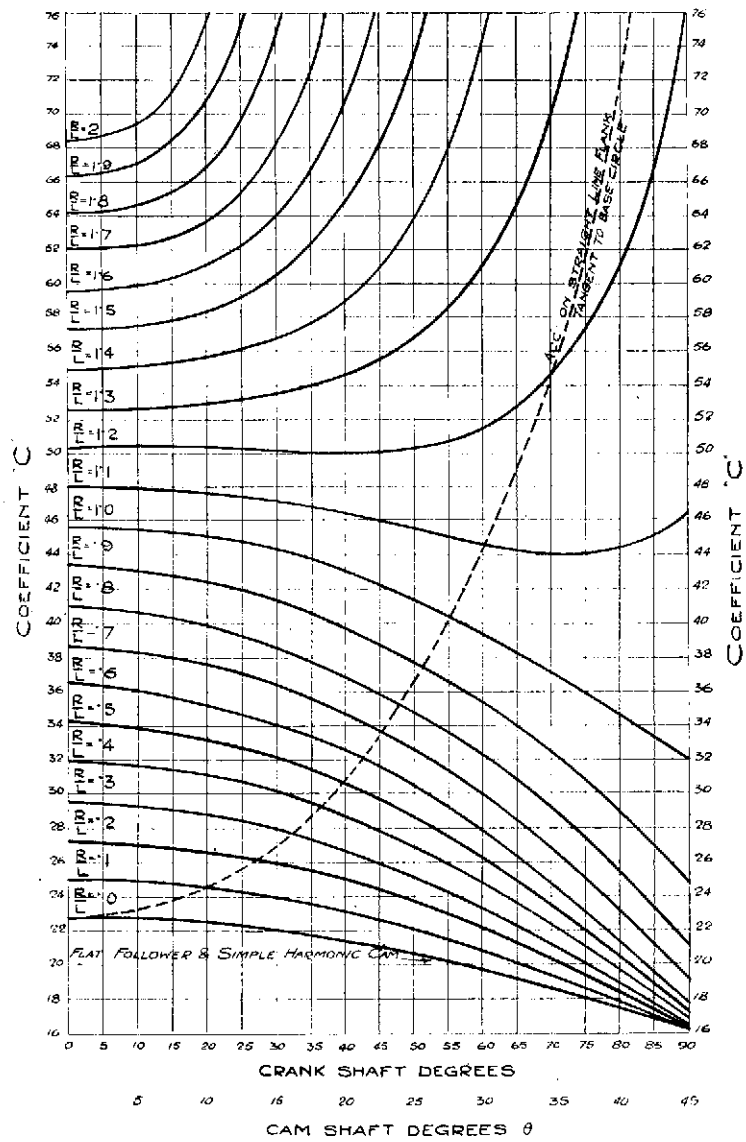
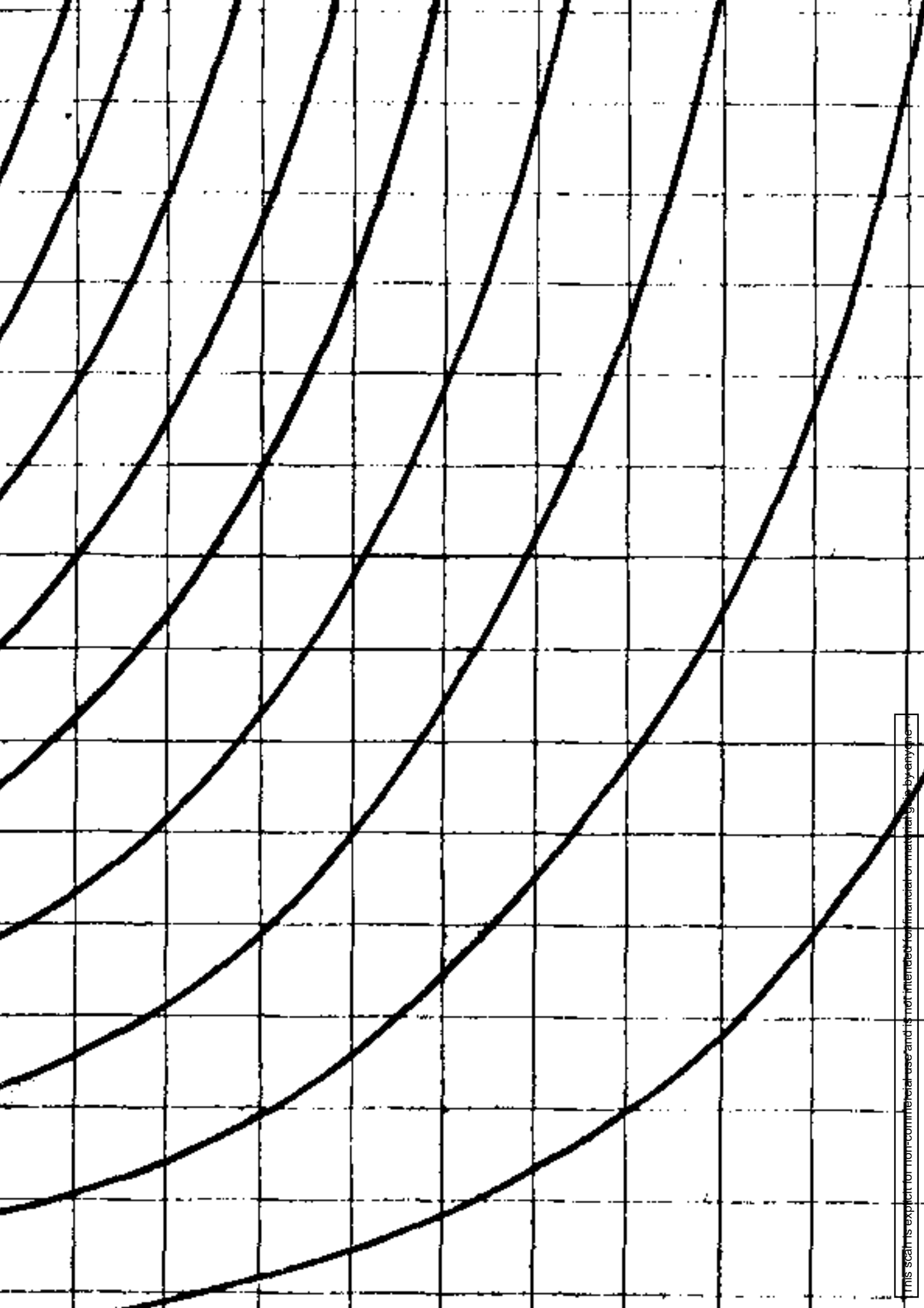


Fig. 105.



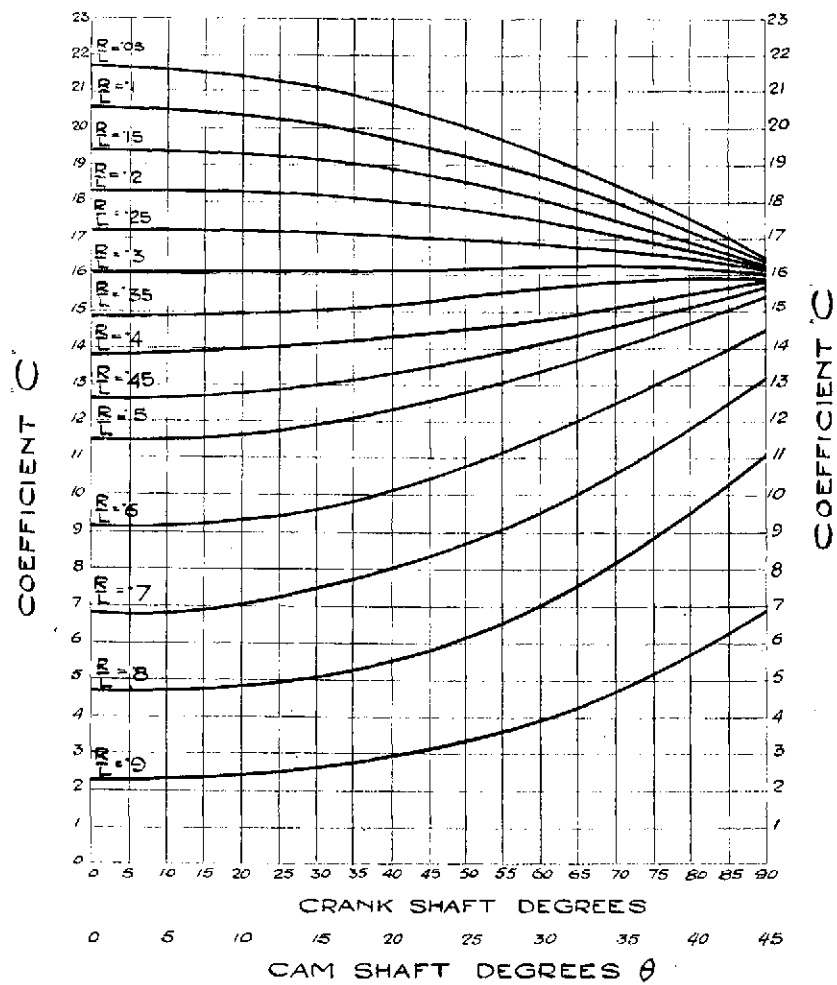


Fig. 107

Facing p. 220.



the lift to the minimum, at the same time providing ample surface on the cam and follower to take the heavy loading.

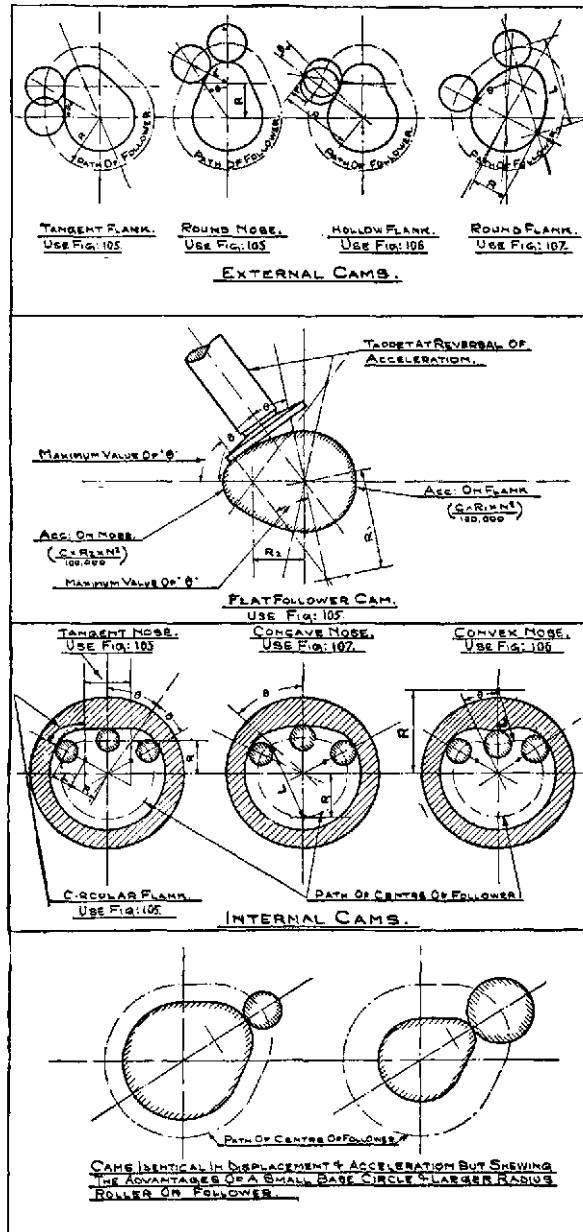


Fig. 108

The negative acceleration on the nose is directly proportional to the distance from camshaft centre to the centre of curvature, it

varies also (but not directly) as the proportion is altered between that distance and the radius of curvature measured to the centre of roller or slipper.

These factors provide a ready means of adjusting the negative acceleration, but it is limited inasmuch as a long radius of curvature cannot be used if the cam period is short or the flank convex.

On the other hand, the radius may be shortened down to a figure very little more than the radius of the follower if the engine speeds and lift are low. This may make the nose of the cam concentric with the shaft for a short distance, giving an improved opening diagram.

It should always be remembered that the nose of the cam does not sustain any pressure at maximum speed because the pressure exerted by the springs should then balance the inertia of the valve parts, while, at low speeds, the pressure on this part of the cam must always be somewhat less than the spring pressure. On the other hand, the flank of the cam has to take the pressure of the spring, the gas pressure on the valve (in the case of the exhaust), and also the force necessary to accelerate the valve parts.

It follows that sharp curvature on the nose of the cam will not lead to undue wear or surface cracks, but that the curvature of the cam flank and follower should be as large as possible.

This consideration points to the use of as small a base circle and as large a roller or slipper as possible for any virtual cam, i.e. for any particular line of motion plotted at the centre of the follower.

As the flank will usually be a straight line or a long radius curve, it is affected little if at all by any reduction in the diameter of the base circle, the nose, as explained, is loaded less and less as the speed increases and need not therefore be considered ( $\frac{1}{16}$  in. radius will stand quite well), while the roller or slipper is greatly improved by the increase in its radius of curvature, and the rubbing velocity is also reduced.

In all cases of cams and followers formed of circular arcs the velocity and acceleration of the centre of the follower are the same as those of a piston having a crank and connecting-rod of lengths  $R$  and  $L$  respectively (see fig. 109). Whenever possible the proportions of the nose of the cam should be such that  $\frac{R}{L}$  is less than unity.

This ensures that the negative acceleration will be at a maximum at the apex and fall off towards the points of junction with the flanks. Since all ordinary forms of spring give an increasing pressure towards the top of the lift, a cam nose so proportioned permits

of the use of a spring which approximately balances the inertia at all points.

If, on the other hand,  $\frac{R}{L}$  is greater than unity the acceleration increases towards the flanks, and as the spring must be at least equal to the inertia at any point, it follows that it is too strong at the apex and throws an unnecessary stress on the valve gear.

**Flat or "Mushroom" Type Followers.**—The cams for use with these are most conveniently constructed of circular arcs, one of small radius for the nose and two others of much larger radius placed symmetrically on either side and tangent to the nose and base circles.

The arcs of the flanks and nose being continued round to form complete circles will be seen to form cranks or eccentrics with which the flat-ended tappet engages in turn. The motion of the tappet is therefore composed of portions of simple harmonic motions of varying amplitude, and the radial velocities and accelerations about the flank and nose are proportional to the distance from the camshaft centre to the centre of curvature in each case.

This is a useful feature of this form of cam for it enables one to determine a relationship between positive and negative accelerations in the first instance and plot the cam accordingly.

A convenient method of construction is appended (fig. 110).

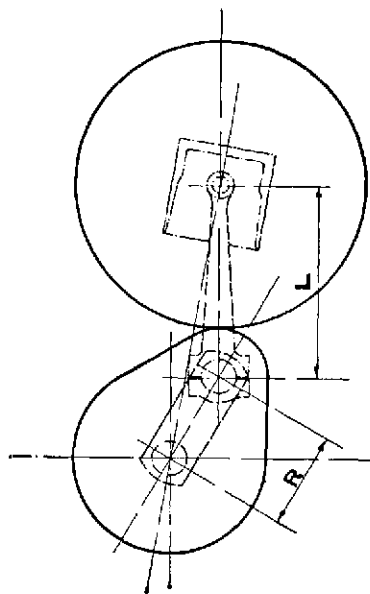


Fig. 109

### Construction of Harmonic Cams

(see fig. 110)

Cam profile formed of circular arcs.

Follower made with a flat face to engage with cam and moving in a straight line normal to that face.

Draw a horizontal line AO of unit length, say 1 in.

Draw BO and CO forming angle AOB, AOC, so that each equals half the valve period plus clearance. (Clearance may be assumed

to be  $12^\circ$  to  $16^\circ$  total (crankshaft degrees), i.e.  $3^\circ$  to  $4^\circ$  a side actually on the cam.)

On BO produced mark off OD, making  $OD : OA :: \text{acceleration on flank of cam} : \text{acceleration on nose (springs)}$ . (This proportion may be determined at first or modified after a preliminary layout. Average cases range from 2 : 1 to 3 : 1.)

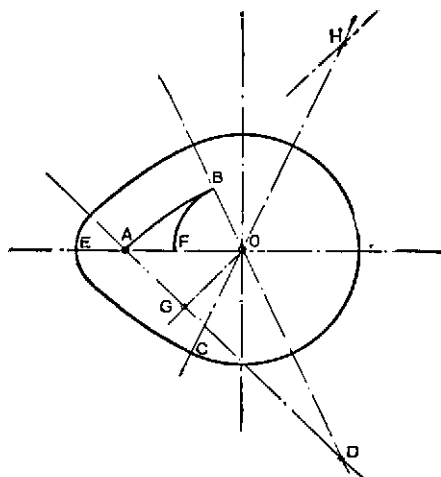


Fig. 110

With D as centre, DA as radius, describe arc AB.

With O as centre, OB as radius, describe arc BF.

Then as  $AF : \text{the required lift} :: OB : \text{required true base circle radius}$ . (Note.—The true base circle radius is smaller than the cam profile by an amount equal to the radius of the nose.)

Construct a similar figure with true base circle and lift as required. Then take a suitable radius (say  $\frac{1}{16}$  in. or more) and describe an arc with centre A forming the nose of the cam, and complete the profile by arcs from centres D, O, and H.

The size of the tappet head is found as follows:—

Join AD and draw OG perpendicular to AD.

Then OG is the maximum eccentricity of the point of contact, and occurs when the tappet is on the junction of the flank and nose curves (i.e. the point of reversal of acceleration from positive to negative).

The radius of the tappet head must be slightly greater than OG, so that the cam may not overrun the edge.

The changes may be rung very easily on the relationship of flank and nose because with this type of cam and follower very heavy positive pressures may safely be used without producing localized wear on the tappet. It will be found that high positive accelerations and long opening periods produce small base circle cams.

Every effort should be made to keep the base circle small, and to this end the radius of the nose may be reduced to  $\frac{1}{16}$  in. if necessary with safety. Another convenient property of these simple harmonic cams is that the negative acceleration is greatest at the apex and falls off in a regular manner towards the point of junction with the

flank, so that if the spring pressure at the apex be made to balance the inertia at that point and the total deflection from free length to maximum compression be equal to the distance from camshaft centre to centre of nose, the spring pressure will exactly balance the inertia at all other points. This renders the determination of the most suitable spring a very simple matter.

**Internal Cams.**—These had a considerable vogue at one time on small single-cylinder engines. When composed of circular arcs and straight lines they follow the same laws as external cams of the same  $\frac{R}{L}$  proportions, but it should be noted that the actual cam is larger than the virtual cam or line of motion of the roller centre, whereas the ordinary external cam is smaller, so that for any particular case the internal cam has a much higher rubbing velocity.

Moreover, since the cam must embrace the roller or slipper the latter is necessarily very much limited in size and the rubbing speed on the pin considerable when a roller is used.

By the use of very good material and workmanship, and particularly by their success in producing excellent surfaces on the pins and rollers, some manufacturers have succeeded in obtaining very satisfactory results, but the type is certainly not to be recommended from either the theoretical or the manufacturing point of view, the internal grinding alone being sufficient to give the decision in favour of external cams.

The graphs given in figs. 105, 106, and 107, and the key diagrams in fig. 108, are worked out for all reasonable proportions, but the formulæ are given below so that extreme cases may be dealt with. True radial movement of the tappet is assumed in all cases. Where the tappet takes the form of a lever, the fulcrum must be so placed that the arc followed by the roller centre approximates to a radial line, otherwise serious distortion of the valve opening diagram will occur, and stronger springs will be necessary, owing to the fact that the acceleration is increased on one side of the nose and decreased on the other, as compared with the value for a radially moving follower.

The acceleration in all cases corresponds to the second differentiation of the radial displacement in respect to time and for the regular forms here dealt with is as follows :

Straight line tangent to base circle

$$\text{Acc.} = W^2 R \frac{(1 + 2 \tan^2 \theta)}{\cos \theta} \dots \dots \dots (1)$$

(F 246)

16

Round nose, and round or hollow flank

$$\text{Acc.} = W^2 R \left( \cos \theta - \frac{n^2 \cos 2\theta + \sin^4 \theta}{(n^2 - \sin^2 \theta)^{3/2}} \right) \quad \dots \quad (2)$$

Simple harmonic cam with flat follower

$$\text{Acc.} = W^2 R \cos \theta, \quad \dots \quad \dots \quad \dots \quad (3)$$

where  $R$  = radius in ft. from shaft centre to roller centre when on base circle in the case of (1) and the radius from shaft-centre to centre of curvature in case of (2) and (3),

$W$  = angular velocity in radians per sec.,

$\theta$  = angle moved through from point of contact with base circle for flank (1), (2), and (3) and, in the case of the nose, the angle from the apex =  $180^\circ - \theta$  (2) and (3),

$L$  = radius of curvature,

$n = \frac{L}{R}$  (see accompanying figures).

**Masked Valves.**—It will be evident that as the period required for the inlet valve opening is shorter than that for the exhaust, the accelerations will be greater, increasing inversely as the square of the time. It will also be noted from any ordinary valve diagram that as the velocity of the valve is zero at beginning and end of its period the value of the opening is very small for the first and last 20 or 30 degrees. By recessing the valve seat in such a way that the outer diameter of the valve acts as a piston valve, it is possible to start the motion earlier and finish it later. While keeping the time of opening and closing as before (because the valve head must clear the recess before any appreciable quantity of gas can pass) this greatly reduces the acceleration and usually permits of the use of the same cam for inlet and exhaust. It has also a considerable effect on the valve opening diagram since the end of the diagram, instead of being attenuated, retains a considerable value and finishes abruptly.

From the point of view of volumetric efficiency this is the most useful feature of the recessed or "masked" valve, as it is usually called, because it gives a large opening at bottom dead centre and closes the valve before the piston has risen sufficiently to pump the charge back through the valve. In normal cases a mask depth of about  $\frac{1}{8}$ th to  $\frac{1}{4}$ th of the lift is suitable.

**Valve Springs.**—These must be considered from four points :

- (1) The force at various points in the travel.
  - (2) The maximum stress in the wire.
  - (3) The stress range from max. to min.
  - (4) Periodic vibrations in the spring itself ("surging").
- (1) From the displacement and acceleration diagrams a force/lift graph is drawn as follows:

A number of points are taken on the displacement diagram from apex to point of reversal (where the nose joins the flank) and are

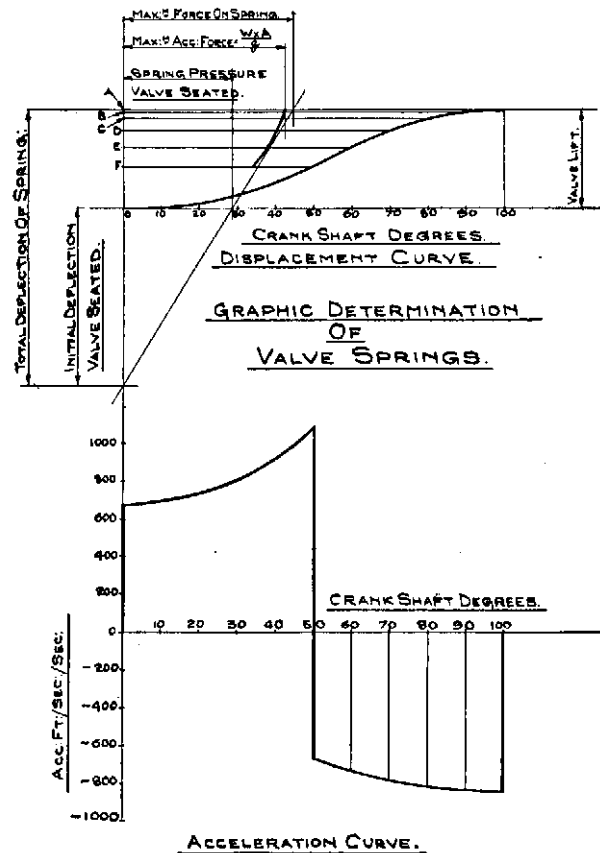


Fig. 111.

projected down to the acceleration diagram and also horizontally to a vertical line.

On the horizontal lines are marked off the distances A, B, C, D, &c., equal to the corresponding vertical ordinates on the acceleration diagram. These points are joined by a curve which shows the force necessary to balance the negative acceleration in terms of the lift (see fig. 111).

Practically all valve springs follow a straight line law, and therefore the straight line which most nearly corresponds to the force/lift graph represents the most suitable spring.

If  $A$  is the acceleration at any point in ft. per second per second the force should be  $F = \frac{A \times W}{32.2}$  plus a small allowance for friction.  $W$  is the weight in lb. of the valve, tappet, etc., including half the weight of the spring. (This latter figure may be assumed and a subsequent correction made if necessary.) The minimum force must be sufficient to hold the exhaust valve shut when the throttle is almost closed and to avoid unnecessary loading on the valve gear, the maximum should be just high enough to allow a safe margin beyond the highest engine speed.

(2) The maximum stress in the wire must be well below the yield point, otherwise permanent set will take place and the free length will become less, thus reducing the force required to compress the spring to any given point.

It is not advisable to exceed 30 tons per square inch, and it is always preferable to keep within 25 tons per square inch maximum.

(3) The stress range, i.e. the difference between the initial stress (valve closed) and the maximum stress (valve fully open), should be kept down in order to avoid fatigue and rapid deterioration of the metal of the spring. The range should not exceed 12 tons per square inch according to the quality of the steel and the life expected.

(4) If the mass of the spring itself be too great in relation to its stiffness (which is proportionate to its rate, i.e. the force in lbs. required to compress it 1 in. axially) the natural period of oscillation of the spring becomes large and may even approach the period of the valve motion. Serious vibrations may then be set up, and the spring is liable to fatigue and may allow the valve gear to jump the cam.

The number of free vibrations per minute of the centre of the spring when the ends are held is :

$$N = 590 \sqrt{\frac{R}{W}},$$

where  $N$  = number of vibrations per minute,

$R$  = rate of spring, i.e. lbs., required to compress it 1 in. axially,

and  $W$  = the weight of the spring in lbs.

If it is found that  $N$  is equal to the revolutions per min. of the camshaft or is a simple multiple of same (say 2, 3, or 4 times), it is



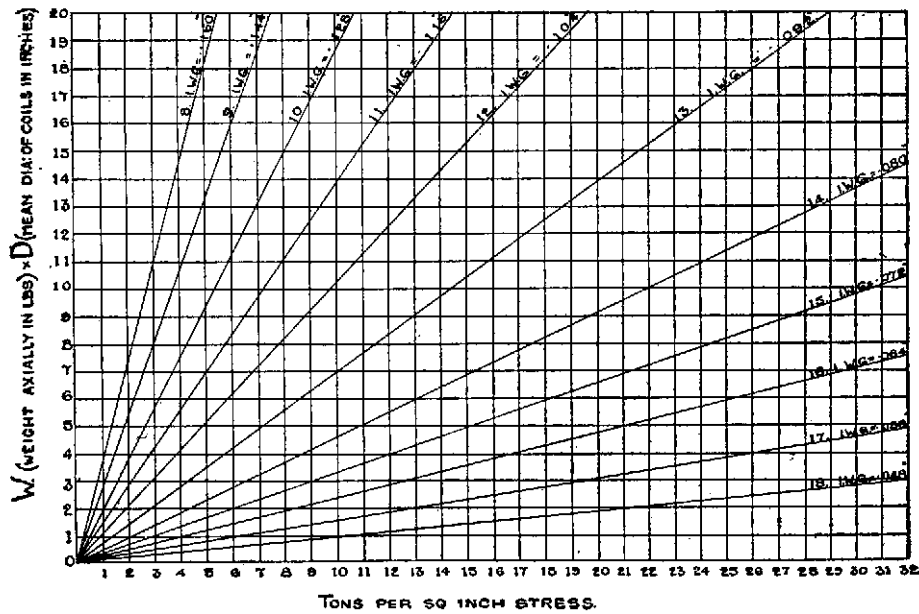


Fig. 112.—Spring Graph—Stresses

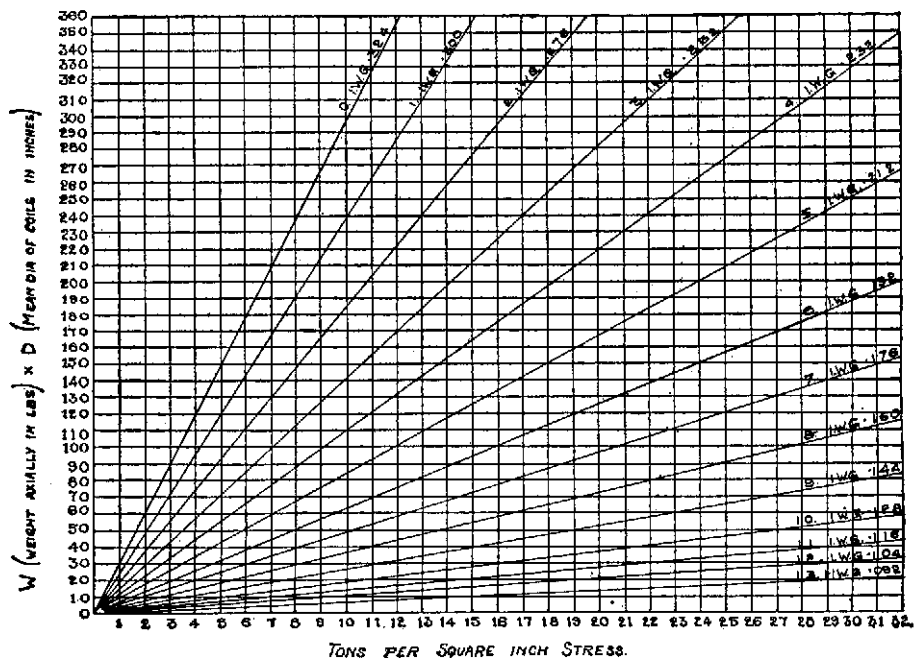


Fig. 113.—Spring Graph—Stresses

practically certain that the spring will shudder and cease to function properly.

A vibrating spring appears blurred when the engine is running, whereas in a spring which is functioning correctly, the central coils can be seen clearly owing to the fact that they are stationary while the valve is shut (say two-thirds of the total time) and the eye retains the impression.

The accompanying graphs will assist in the selection of the gauge of wire (see stress graph fig. 112 for light gauges and fig. 113 for heavier gauges) when the force and approximate diameter of spring are

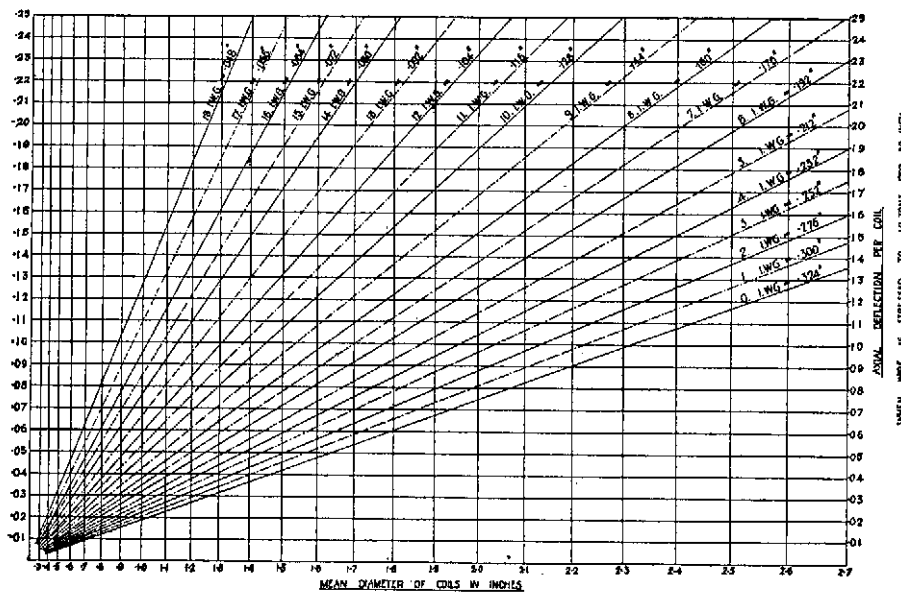


Fig. 114.—Spring Graph—Deflection

known. The deflection per coil ( $d$ ) may be read off from the deflection graph (fig. 114). The total deflection ( $D$ ) being already fixed (see figure for force/lift graph) the number of *effective* coils is

$\frac{D}{d}$  and the total number  $\frac{D}{d} + 2$ .

$$\text{The maximum stress} = \frac{8FD}{\pi d^3} \text{ lb. sq. inch,}$$

where  $F$  = maximum force in lb.,

$D$  = mean dia. of coils in inches,

$d$  = dia. of wire in inches;

$$\text{also the maximum deflection} = \frac{8FND^3}{Cd^4},$$

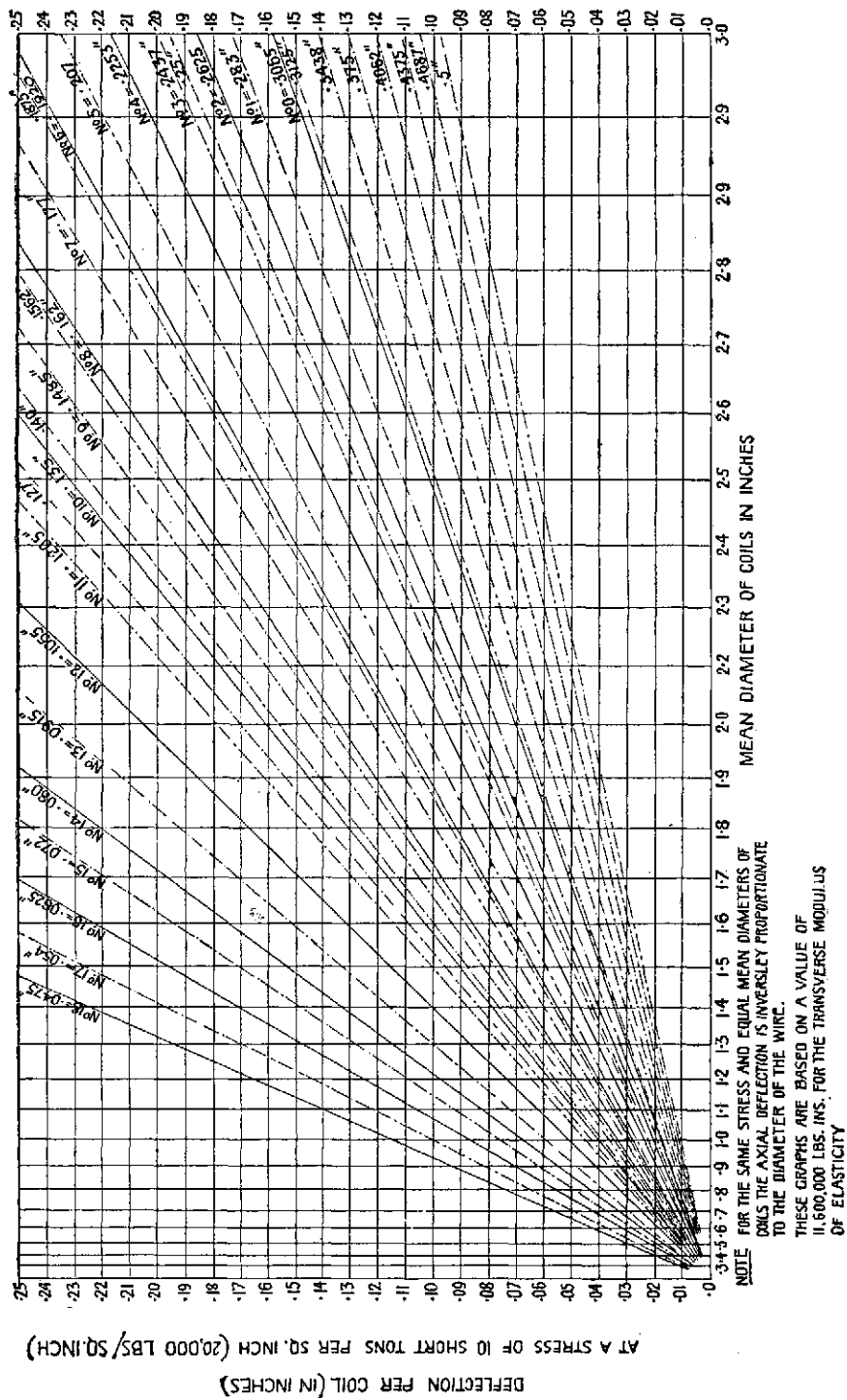
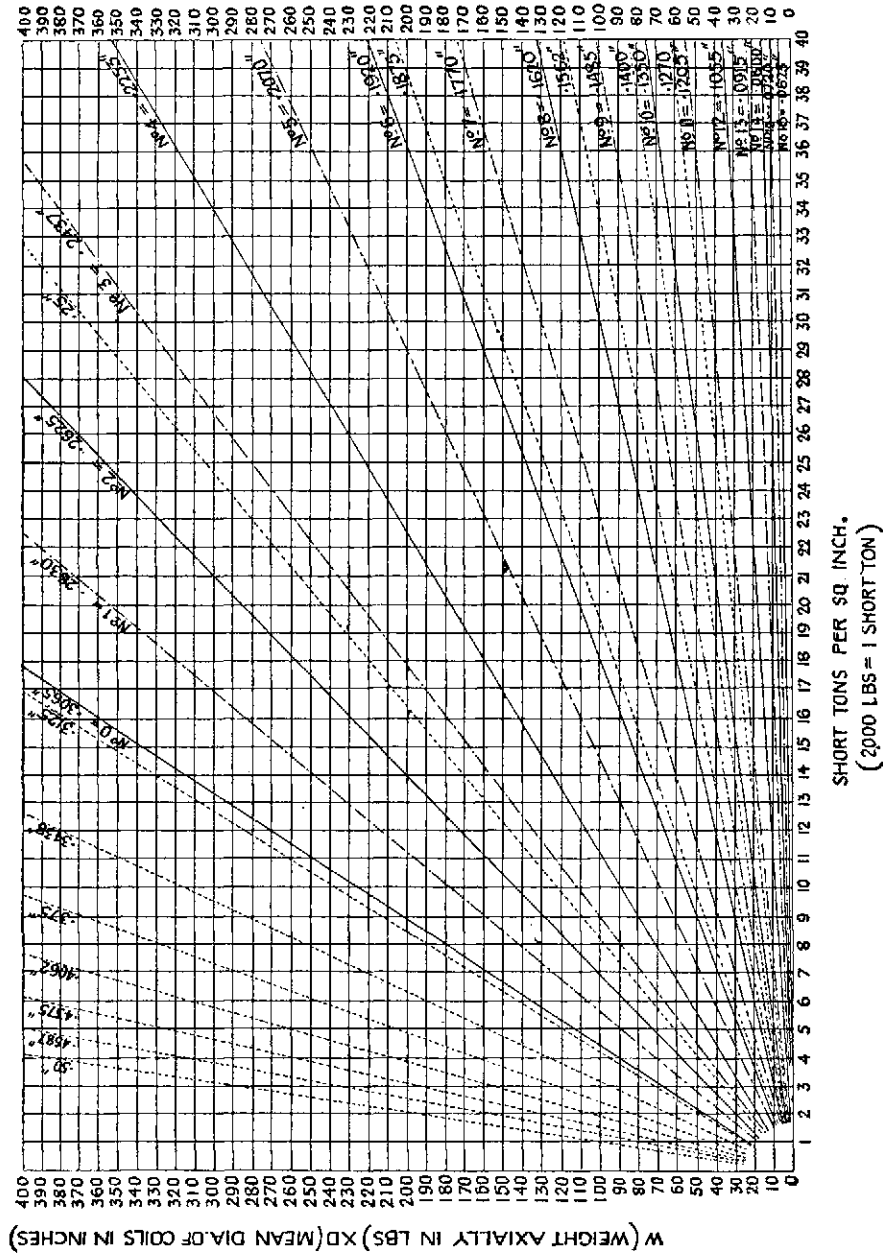


Fig. 115.—Spring Graph—Deflection  
For American (Washburn & Moen) Wire Gauge



where  $N$  = effective number of coils (total minus 2),

and  $C$  = the transverse modulus of elasticity = 11,600,000.

The spring thus arrived at should then be considered under heading (4) before being passed as suitable.

It will be apparent that (3) and (4) are in opposition, for if the stress range be kept *too* low, the spring loses stiffness and may develop slow vibrations in time with the valve gear. There is no difficulty in practice in satisfying both points even in very high-speed engines.

If the length of the spring be made such that the coils are almost crowded when the valve is open the stress due to "surging" may be limited.

## CHAPTER IX

### PISTON DESIGN

Of the total mechanical losses in any internal-combustion engine of normal design, piston friction forms by far the largest individual item; and, of all the independent component parts the piston is probably the most difficult of design in that it has to fulfil so many and frequently such contradictory conditions. Broadly speaking, the main objects to aim at in the design of a piston for the lighter high-speed types of internal-combustion engine are:—

- (1) To reduce friction to the lowest possible limit.
- (2) To reduce the weight.
- (3) To dissipate heat to the walls of the cylinder.
- (4) To prevent the passage of oil into the combustion chamber.
- (5) To provide adequate support for the gudgeon pin.

Conditions 1 and 2 are largely interdependent, for the bulk of the average pressure exerted by the piston against the cylinder walls is, in any high-speed engine, due to the resolved component of the inertia forces which, when averaged over the whole cycle, exceeds the fluid pressure; hence if the weight is reduced the average bearing pressure is reduced also, and for the same bearing pressure per square inch, and therefore for the same durability, the area of bearing surface may be reduced nearly in proportion to the weight.

Piston friction is of course dependent also upon the nature and condition of the lubricating oil adhering to the cylinder walls.

The general question of lubrication and friction has been considered in relation to bearings, etc., in Chapter V, and it has been shown that friction is to a large extent proportional to the area of surface, the viscosity of the lubricant, and, to a much less extent, to the load. In the case of the piston, however, the conditions are somewhat different; in the first place, although the rubbing

velocity is higher, the average load is low and is, compared with any of the bearings, small. Under such conditions the area of surface and the viscosity of the lubricant play a very important part.

With regard to the area of surface, it is clear that only the surface at right angles to the line of the crankshaft is operative, the surface at the sides of the piston receiving no thrust at all. It is therefore clearly desirable to remove all inoperative surface in order to reduce, as far as possible, the area of the oil film in shear. In view of the very light loading to which a piston is subjected, a relatively small area of bearing surface suffices, and for a reasonably light piston an actual bearing surface on either side equal to 50 per cent of the area of the crown should be ample, provided it is properly disposed; that is to say, provided it is disposed equally above and below the gudgeon pin and over a subtended angle of from  $90^{\circ}$  to  $110^{\circ}$ . The author has never heard of any case of a piston seizing from overloading of the bearing surfaces. Seventy per cent of piston seizures are due to insufficient allowance for expansion or to distortion, and the remaining 30 per cent to complete failures of the oil supply. As to wear, in the case of cast-iron pistons at any rate, this is mainly due to the piston rings; it is very unusual to find any serious wear on the bearing surfaces of a piston or in the cylinder bore below the travel of the piston rings.

The rate of wear of the cylinder bore is more or less proportional to the gas pressure behind the rings and in consequence is greatest at the position of the top piston ring when the piston is just below top dead centre.

When compared with any other bearing surface in the engine, the friction of the piston is abnormally high; this is undoubtedly due primarily to the fact that the oil is partially carbonized and its viscosity, and therefore its resistance to shear, is greatly increased. It must be remembered that, at every cycle, most of the oil clinging to the walls of the cylinder barrel is exposed to the full flame temperature of the burning gases. It is probable also that the fluid resistance is greater when the direction of motion is constantly reversed than when it is continuous, as in the case of a shaft running in bearings. As an illustration of the effect of the carbonizing of the oil on piston friction, the author has always observed, when testing engines on electric dynamometers, that if the supply of fuel be suddenly cut off after running under load and the engine motored round by means of the dynamometer the friction torque is at first high, but falls rapidly as the carbonized oil on the cylinder walls is replaced with fresh clean

oil from the lubricating system. Fig. 117 shows a typical curve of total friction torque on a time basis carried out under these conditions. The engine in this instance was run under full load at 1200 R.P.M. for a considerable period, until all temperature conditions had become normal; the supplies of circulating water and fuel were then simultaneously cut off and the engine motored round at precisely the same speed—the change over from full load running to motoring being effected without any measurable interval of time and without any appreciable change in speed. In this particular

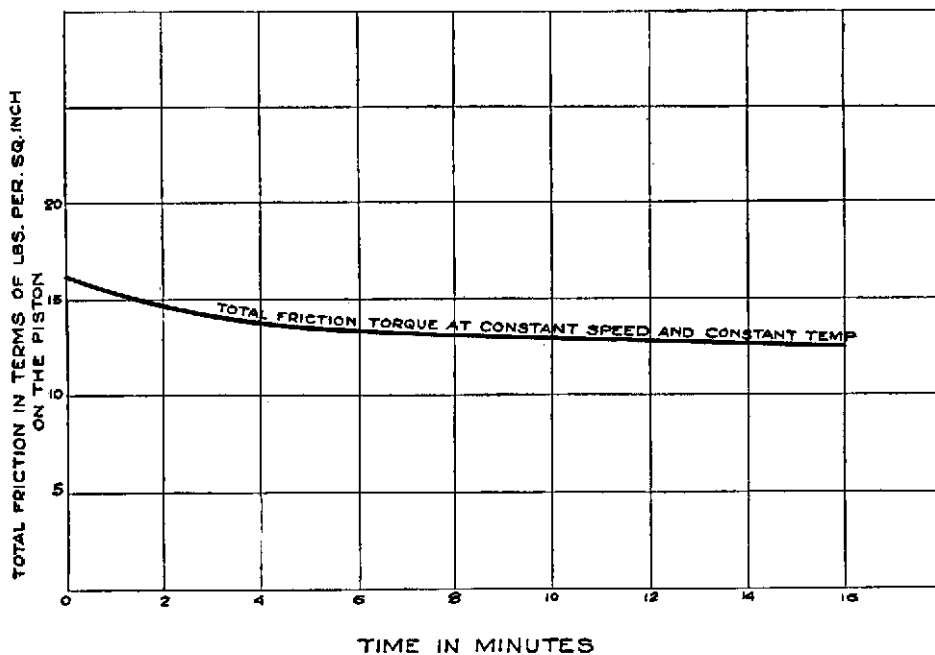


Fig. 117.—Curve showing Drop in Friction Torque as contaminated Oil is replaced by fresh Oil

case the friction losses of the bearings and auxiliaries and the fluid pumping losses had all been ascertained separately, and were found to be equivalent to a mean pressure of 6.5 lb. per square inch at 1200 R.P.M. Deducting these, the piston friction is as shown in the curve, fig. 118, from which it will be observed that it falls from the equivalent of 9.5 lb. per square inch immediately after the fuel is cut off to 6.5 lb. per square inch after ten minutes, by which time it may be presumed that practically the whole of the carbonized oil on the cylinder walls has been replaced by fresh oil.

**Influence of Temperature on Piston Friction.**—As might be expected, the friction of the piston is largely dependent upon the



temperature of the lubricant, and since the temperature of the latter is determined primarily by that of the cylinder walls to which it clings, it follows that the friction is controlled very largely by the temperature of the cooling water. In Chapter III, when dealing with the influence of cylinder temperature upon power and economy, it was shown that the indicated horse-power of an engine decreases with increase of temperature, because the reduction in the weight of charge far more than outweighs the slight gain due to reduced heat losses. In practice the brake-horse-power and economy of an engine generally increase with increase of temperature, because the

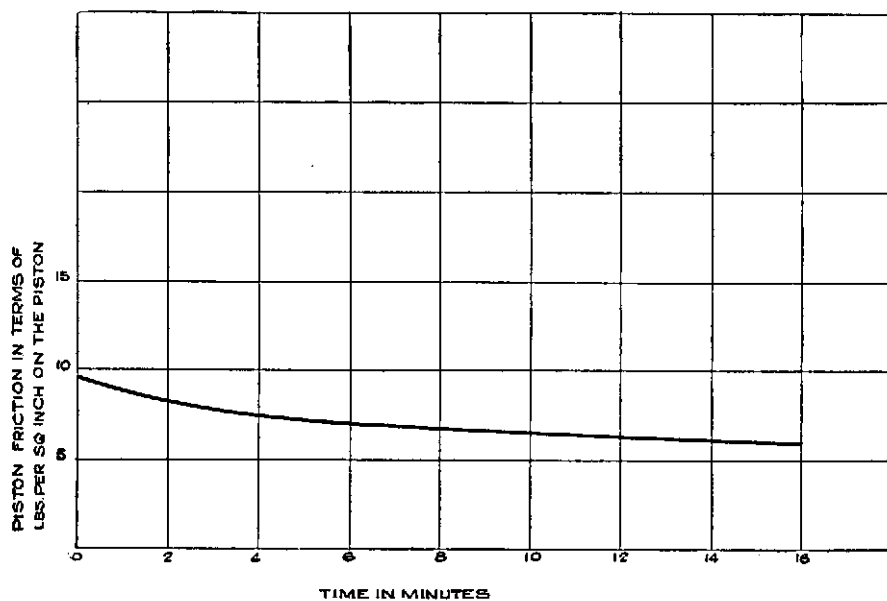


Fig. 118—Curve showing Drop in Piston Friction alone from fig. 117

reduced piston friction more than compensates for the reduction in indicated power. Fig. 119 shows the results of a test on a standard four-cylinder commercial vehicle engine when motored at a speed of 900 R.P.M. In this test, after the engine had been run for some time the water-jackets were thoroughly flushed through with cold water until the cylinders had been cooled down to atmospheric temperature. The supply of circulating water was then cut off and the rise in temperature and the friction torque (expressed in terms of mean pressure on the engine) were recorded at intervals of two minutes.

In the case of this engine, the friction of the bearings and auxiliaries and the fluid pumping losses were determined separately,

and were found to amount to the equivalent of 5.5 lb. per square inch at 120° F. If this figure be deducted throughout, it will be observed that the piston friction falls from 10.5 lb. per square inch with a cylinder temperature of 70° F. to 5.6 lb. per square inch with a temperature of 150° F. Over this range of temperature the decrease in indicated mean pressure, in the case of this particular engine, is about 2 per cent or 2 lb. per square inch, but the drop in piston friction is equivalent to about 5 lb. per square inch, so that the net increase in power at the higher temperature would be about 3 lb.

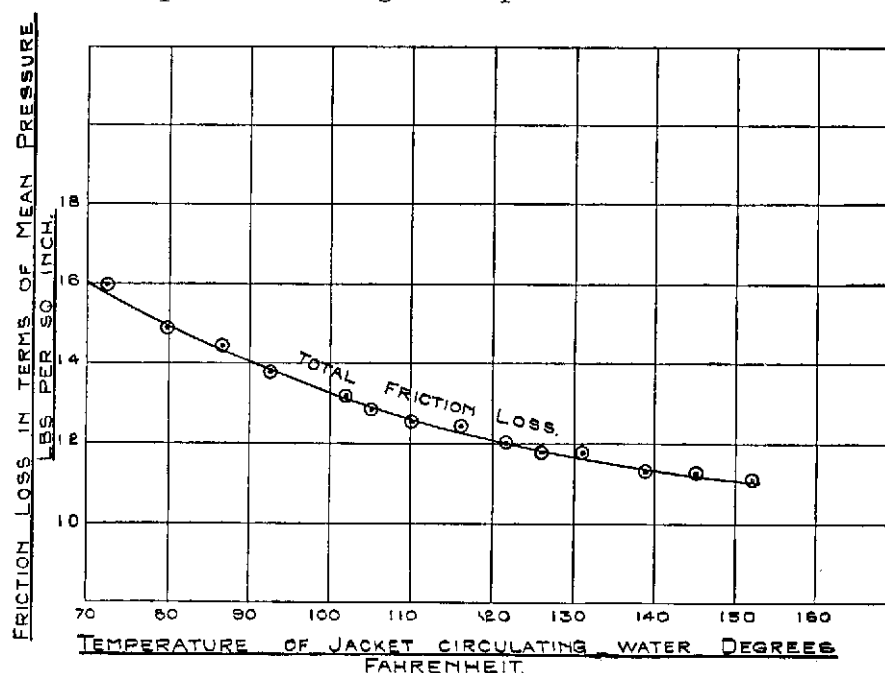


Fig. 119.—Motoring Tests, showing Change of Friction with Temperature at 900 R.P.M.

per square inch or 3 per cent. This agreed fairly closely with actual results obtained when running under power, when the difference was found to be nearly 4 per cent. The conditions are not, however, exactly comparable, because—

(1) When running under power the temperature of the piston and inner surface of the cylinder walls is higher than that of the jacket water; this would tend to reduce the difference in piston friction between hot and cold, because the whole temperature scale is virtually raised.

(2) When motoring, the oil on the cylinder walls was clean, consequently the piston friction was lower; under actual running

conditions the piston friction was no doubt about 30 per cent greater at all temperatures; this would accentuate the difference, and probably more than outweighs (1).

**Dissipation of Heat.**—The heat from the crown of the piston is disposed of—

- (1) Through the rings to the cylinder walls.
- (2) Through the bearing surface to the cylinder walls.
- (3) To the oil and air below the piston.

There is a great deal of evidence in support of the theory that, in comparatively small engines at all events, the bulk of the heat passes to the cylinder walls *via* the piston rings. The author could cite numerous experiments in confirmation of this theory, but it is probably sufficient to state that experiments have shown that, when all transference of heat by way of the bearing surfaces has been cut off, the temperature of the piston crown is found to be very little higher. In any case, it is evident that heat can only be transmitted rapidly to the cylinder walls through that portion of the skirt or bearing surface which is being pressed hard against them by the thrust of the connecting-rod and where the oil film is therefore both thinnest and in most active movement.

The proportion of heat carried away by the circulation of the air and oil below the piston cannot be very large and need not be taken seriously into consideration, except in cases where special arrangements are made to increase these effects.

It is quite clear that the most important consideration is the transmission of heat from the centre of the crown to the circumference; once the heat can be conveyed to the circumference, experience shows that there is no difficulty in getting rid of it. In order to facilitate the transmission from the centre to the circumference, it is obvious that the crown should be made as thick as possible consistent with the weight limitation, and the conductivity of the material should be as high as possible. During the last few years the use of aluminium alloys has come into vogue for pistons; not only is their weight about one-third that of cast iron, but their conductivity is about five times as great. With such alloys, it is found that the rate of heat transmission is so high that, even in the case of cylinders developing over 120 B.H.P., there is no need to make the crown of the piston any thicker than is needed for strength alone. Recently all aero-engines, and many others also, have been fitted with pistons made throughout of aluminium alloys. The objections to an all-aluminium piston are :—

(1) That owing to the very high rate of expansion with temperature a large clearance must be allowed; this causes an audible knock when the thrust is reversed under pressure at the end of the compression stroke.

(2) Aluminium being a relatively soft material permits of particles of grit embedding in it, and is therefore liable to cause wear of the cylinder walls unless the latter have a very hard surface.

(3) Aluminium castings, unless very carefully annealed, are liable both to grow and to distort, so that yet more clearance must be allowed on this account.

None of these objections apply when the crown and ring-carrying portion alone is made of aluminium alloy and the bearing surfaces of cast iron, while advantage can still be taken of the high conductivity and light weight of aluminium.

**Passage of Oil into the Combustion Chamber.**—One common form of trouble with internal-combustion engines, and more particularly with those of the high-speed enclosed type, is the passage of oil into the combustion chamber, where it carbonizes both on the walls of the chamber and on the crown of the piston, and so gives rise to detonation and ultimately to pre-ignition. Passage of oil past the piston rings and so into the combustion chamber is due to—

(1) The oil is forced up against the rings on the downward stroke of the piston because the motion of the piston, combined with its thrust against the cylinder walls, sets up a very considerable hydraulic pressure and the oil is, so to speak, rolled up against the rings.

(2) The motion of the piston rings in their grooves tends to pump the oil into the combustion chamber.

In order, as far as possible, to prevent the passage of oil into the combustion chamber, the following considerations should be taken into account :—

(1) The setting up of a heavy hydraulic pressure can largely be prevented by perforating the bearing surface so that the pressure can relieve itself, and also by freely venting the piston just below the bottom ring.

(2) As the piston travels downwards the rings are all bearing against the top faces of the grooves, and the clearance between the lower sides of the rings and their grooves is filled with oil scraped from the cylinder walls. As the piston rises again, the rings change over and bear against the lower face of the groove; the oil therefore passes round behind the rings to their upper face and, at the top of the

stroke, when the rings again change sides, some of it is squeezed out. It will be seen therefore that each ring functions as a valveless oil pump and tends to deliver oil into the combustion chamber.

In order to reduce this pumping as far as possible—

(1) The rings, with the exception of the scraper ring, should be made as close a fit as possible in their grooves.

(2) Ample venting should be provided below the scraper ring to permit of the free escape of any oil scraped off the cylinder walls.

(3) The tendency of the rings to pump oil can further be checked to a large extent by drilling holes through the ring groove behind the ring, thus permitting of the free escape of any oil as it passes the back of the ring.

This latter expedient should apply only in the case of the lowest ring, since such drilling permits also of the escape of gas. Fig. 120 shows an arrangement of rings, &c., which has been found very effective in preventing the passage of oil into the combustion space.

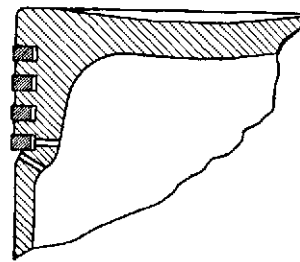


Fig. 120.—Section of Slipper Piston, with Scraper Ring vented by Holes drilled through Groove

There is a widespread belief that the passage of oil into the combustion chamber is dependent primarily upon the pressure or vacuum in the cylinder, and that, when throttled down, the reduced pressure in the cylinder during the suction stroke causes oil to be sucked up past the rings. This belief is founded upon the fact that when an engine is opened out after running throttled up, smoke becomes apparent, indicating an excess of oil in the combustion chamber. Also when an engine has been run with the throttle nearly closed and the cylinders or valve caps are removed, oil in a liquid state is then found in the combustion chamber. In spite of such evidence, however, the theory is quite fallacious, for the actual quantity of oil passing the rings is found to be a function of the speed, and of the speed alone, the pumping pressure set up by the rings being of far too high an order to be influenced appreciably by any relatively slight differences of pressure in the cylinder. When an engine is running at or near its full load, the oil passing into the combustion chamber is burnt along with the fuel; combustion is nearly complete, so that no smoke is visible from the exhaust. When running dead light the flame temperature, owing to the large dilution with exhaust gases and the relatively higher rate of heat

loss, is insufficient to burn the oil, with the result that it accumulates in the combustion chamber until the throttle is opened; the quantity is then so large that there is not sufficient oxygen available for its complete combustion at first, with the result that it is only partially burnt and issues from the exhaust as blue smoke—that is, as partially vaporized but unburnt oil.

In any normal high-speed closed-type engine, about 90 per cent of the lubricating oil consumed is burnt in the cylinder as fuel, a fact which should always be borne in mind when reckoning the efficiency of an engine, for the hourly consumption both of fuel and oil should be taken into account. In most normal engines the proportion of oil consumed is very small in relation to the fuel and does not materially affect the consumption of the latter, but in the case of certain aero-engines, particularly of the rotating cylinder type, the consumption of oil is so high as materially to reduce the fuel consumption, and so give rise to a fictitious fuel economy.

The author has carried out a number of tests in order to ascertain the influence of both pressure and speed on the passage of oil into the combustion chamber, and has tried the effect of motoring an engine and collecting the oil passing the piston under the following conditions :—

- (1) When the pressure on either side of the piston is atmospheric.
- (2) With a continuous vacuum of 20 in. of mercury in the cylinder.
- (3) With a continuous air pressure of 45 lb. per square inch above the piston.

In all three cases the quantity of oil passed per hour was, within the limits of observation, the same—certainly to within 10 per cent. In all cases also the quantity of oil passed by the piston varied almost directly as the speed of rotation.

In fig. 121 is shown a special machine used for testing pistons and rings for—

- (1) Friction.
- (2) Leakage.
- (3) Passage of oil.

It consists of a water-jacketed cylinder mounted on a crankcase in which different types of pistons, rings, &c., can be fitted. The piston is reciprocated by means of a crank and connecting-rod, and, to ensure freedom from vibration, reciprocating balance weights, operated by eccentrics, are provided. The cylinder is heat-insulated from the crankcase and the friction of the piston is measured directly by the temperature rise of the water in the jacket. In order to

ensure uniformity of temperature the water in the jacket is kept circulating by means of a small propeller driven by a belt from the crankshaft. The top end of the cylinder is connected to a large and heavily lagged receiver of sufficient capacity to prevent any

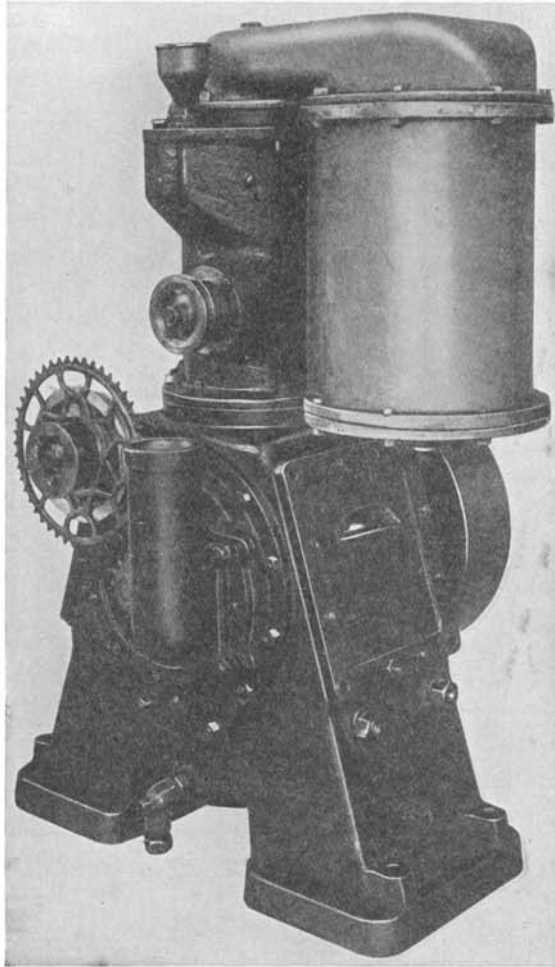


Fig. 121.—Photo of Machine designed for testing Pistons

appreciable variation in pressure, so that the same air is drawn in and out of the cylinder at every stroke, and errors, due to the circulation of cold air inside the cylinder, are eliminated, or nearly so. This receiver is, in turn, connected to an air pump so that the pressure on the piston can be raised or lowered to any desired degree, and the effect of fluid pressure both on piston friction and the passage of oil can be observed.

For lubrication, oil is forced under a pressure of 30 lb. per square inch through the hollow crankshaft, from which it passes out through the connecting-rod big-end bearing and is thrown on to the cylinder walls.

For testing gas tightness and leakage of rings the receiver is removed and a plain cover fitted in its place on the top of the cylinder. This cover is provided with a small and very light automatic inlet valve connected to an air-measuring device. With this cover fitted the piston alternately compresses and expands the air in the cylinder,

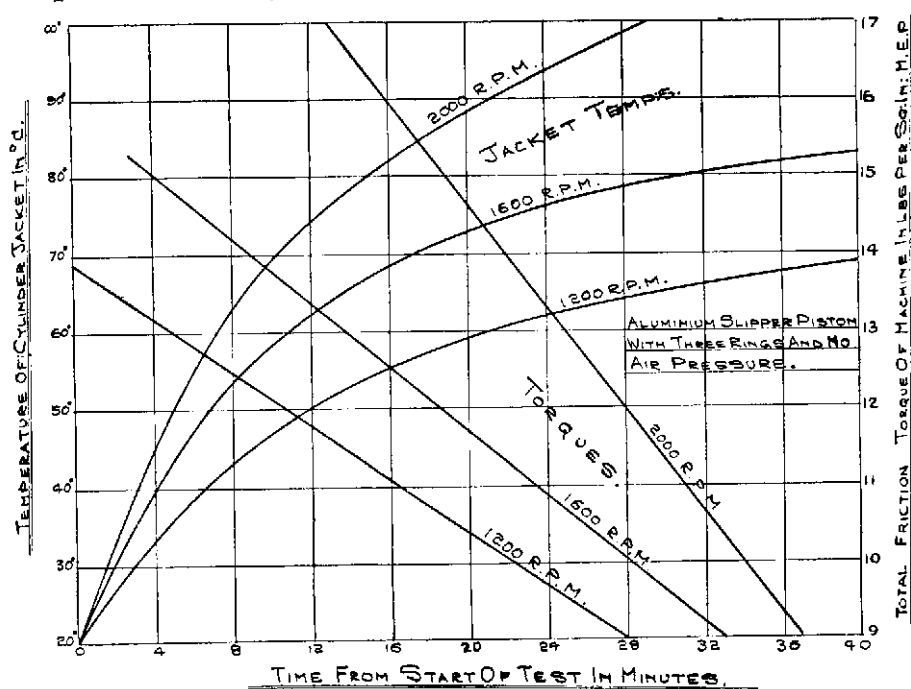


Fig. 122.—Test Readings at three different Speeds

the maximum pressure being about 220 lb. per square inch. Any leakage past the rings is made up by air entering through the inlet valve, and the quantity of air so required to make up for leakage is measured by the displacement of water.

The machine is direct connected to a balanced swinging field electric dynamometer, and can be driven at any speed from 600 to 2500 R.P.M. The total torque required to rotate the machine at any speed can be measured direct from the arm of the dynamometer, while the piston friction alone can be determined directly from the temperature rise of the water in the cylinder jacket.



Owing to losses by radiation, &c., it is not easy to determine accurately the absolute friction of the piston, but the relative friction as between two pistons, or any variations in piston design or rings, can be measured with extreme nicety by comparing the curves of temperature rise of the jacket water. Figs. 122, 123, and 124 show a number of such curves of temperature and time and also the total friction of the machine as a whole in terms of lb. per square inch on the piston, with various types of piston, numbers of rings, &c.

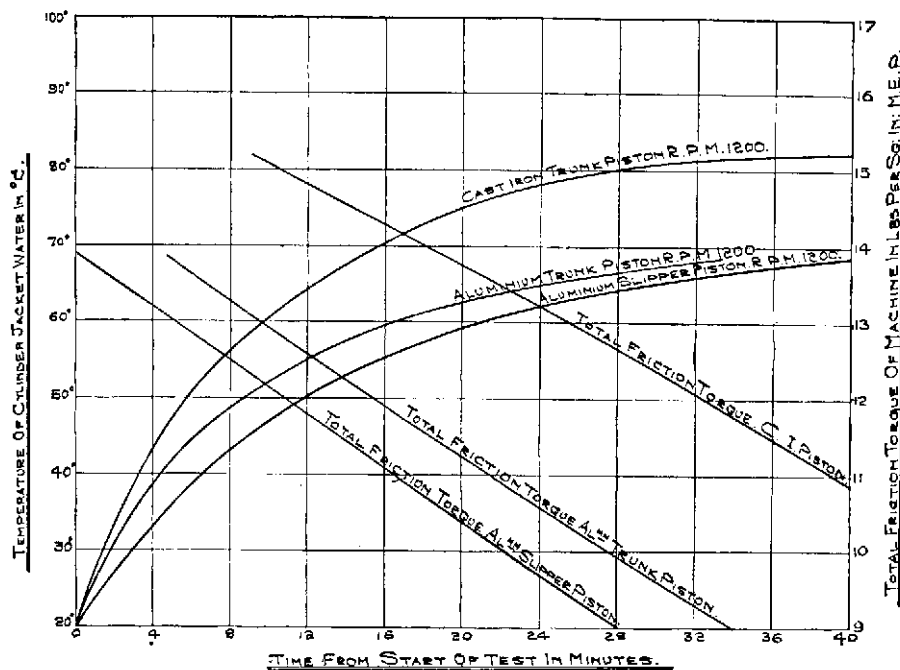


Fig. 123.—Test Readings with 3 different Pistons. Speed 1200 R.P.M. in all cases

In order both to reduce weight and to prevent distortion, it is clearly very desirable to transmit the pressure as directly as possible from the crown of the piston to the connecting-rod, and from the connecting-rod to the bearing surfaces. The customary method of transmitting the pressure from the crown through the side walls and ring grooves to the two extreme ends of the gudgeon pin has nothing to recommend it. It is clearly far better to transmit it directly from the crown of the piston to the gudgeon pin at points as near the centre as the width of the connecting-rod small end bearing will permit.

Fig. 125 shows a design of all-aluminium piston, and represents probably the lightest possible construction. In this design two main ribs transmit the load from the crown to the gudgeon pin, and from the gudgeon pin to the bearing surfaces. Also all unnecessary bearing surface has been eliminated. This type of piston has come to be known as the slipper type, and has been widely used, particularly in very high-speed engines, where its light weight, low friction loss, and effective oil-resisting properties have rendered

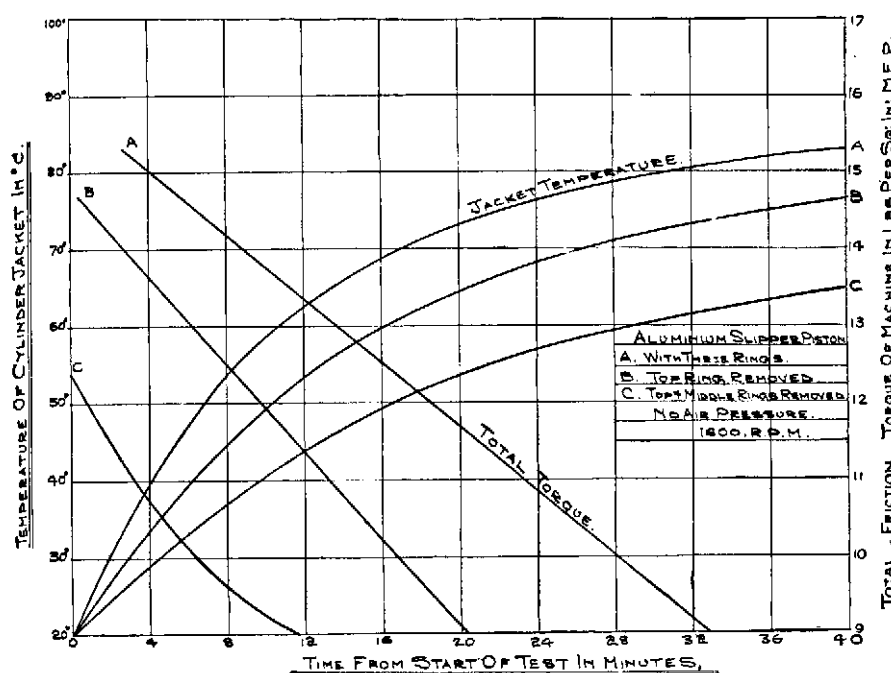


Fig. 124.—Test Readings with different Numbers of Piston Rings.  
Speed 1600 R.P.M. in all cases

it of great advantage. The example shown in fig. 125 is the largest yet made, for this single piston transmits 135 B.H.P. at 1400 R.P.M.

Fig. 126 shows an alternative design in which the slipper or bearing surfaces are of cast iron. This design has the further advantage in that the floating gudgeon pin is located by the sides of the cast-iron sleeve and requires no other means of endwise location—always rather a troublesome problem.

Fig. 127 shows yet another design, in which the aluminium alloy head is connected with the cast-iron cross-head portion by means of

the gudgeon pin, or rather by means of the bushes in which the gudgeon pin floats. This design has the advantage that it is cheaper and less fragile than that shown in fig. 126, also that the aluminium head is free to centre itself in the cylinder. The chief objection to it is that it necessitates very accurate workmanship.

**Piston Knock.**—Owing to the large clearance which must be allowed when all-aluminium pistons are used, it is very difficult to

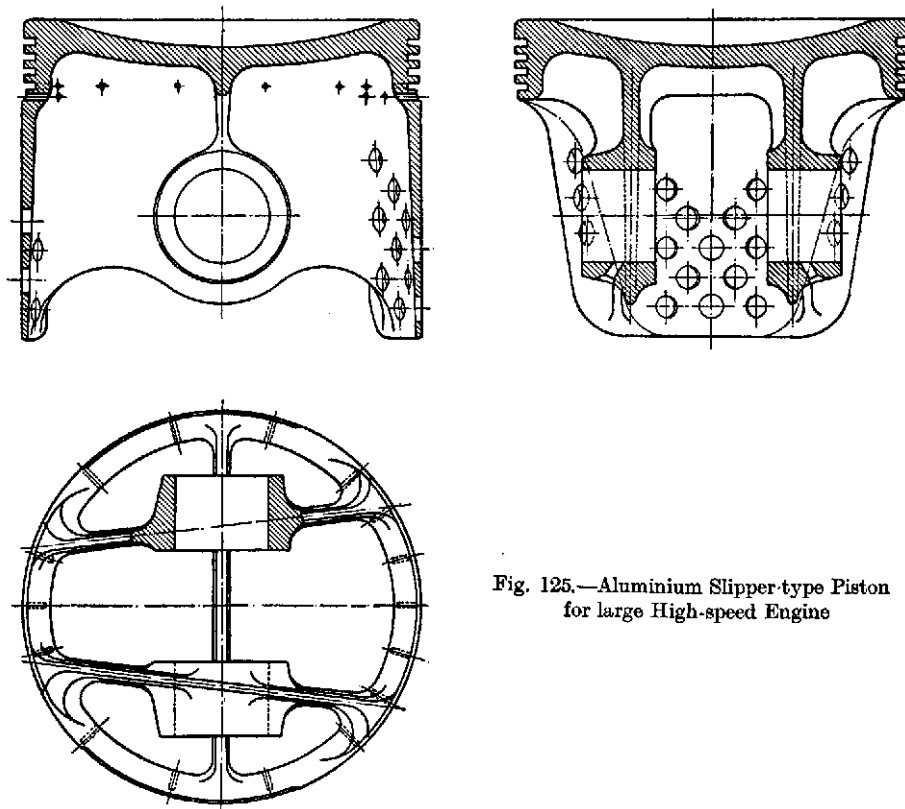


Fig. 125.—Aluminium Slipper-type Piston for large High-speed Engine

obtain silent running, for at the end of the compression-stroke the piston is thrust violently from one side of the cylinder to the other. The noise is most apparent when an engine is running slowly on a light load ; under such conditions the piston is cool, and therefore the clearance is at a maximum, also the other mechanical noises of the engine are less apparent. Various devices have been experimented with in the endeavour to overcome this troublesome noise. Some designers have used pistons in which the normal clearance is very small and the skirts are slit in order to allow some elasticity

and prevent seizures; others, again, have even gone so far as to introduce springs between the connecting-rod and piston, in order

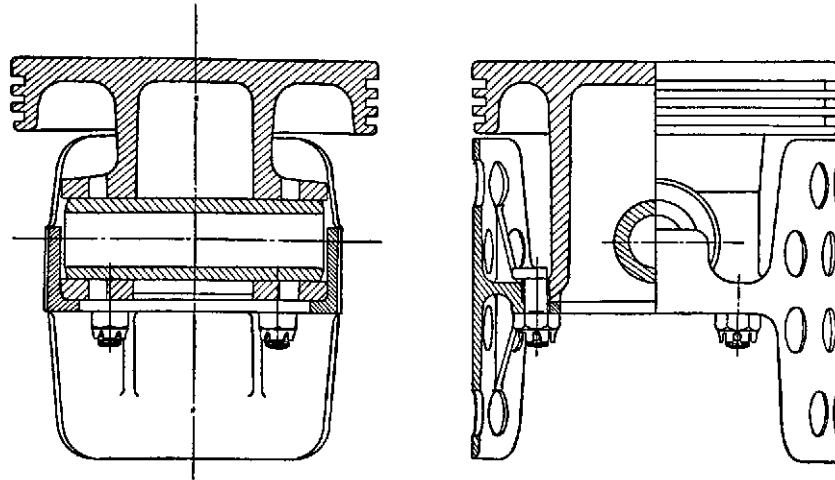


Fig. 126.—Composite Piston with Aluminium Head and Cast-iron Bearing Surfaces rigidly attached

to keep the latter bearing at all times against one wall of the cylinder. Others, again, have adopted the method of fitting the gudgeon pin out of centre in the piston so that the latter tends to tilt about the

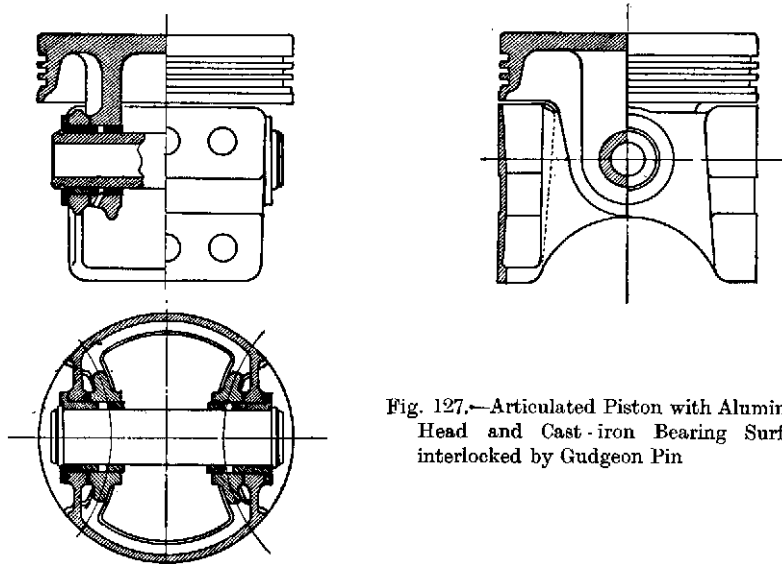


Fig. 127.—Articulated Piston with Aluminium Head and Cast-iron Bearing Surfaces interlocked by Gudgeon Pin

gudgeon-pin centre. This latter method appeared promising at first sight, but on further investigation it was found to be ineffective. The precise effect of offsetting the gudgeon pin in this manner is

illustrated in fig. 128, which shows the position taken up by the piston at various points in the cycle.

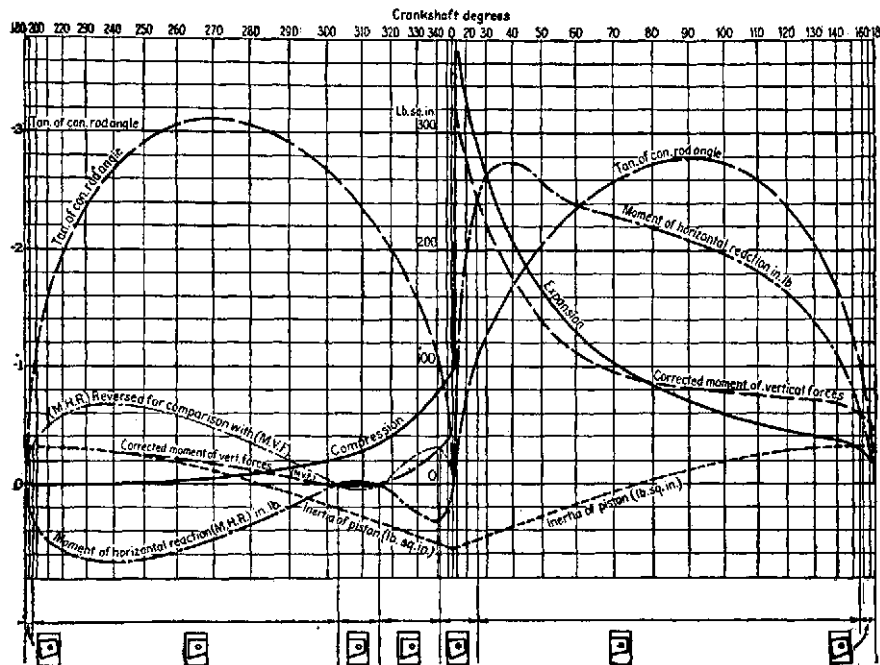


Fig. 128.—Diagram of Forces on Piston with offset Gudgeon Pin, showing effect on clearance

Fig. 129 shows a method patented by the author and applied to a slipper-type piston in which the bearing surface is severed from the crown in order to prevent the direct transmission of heat and so permit of a smaller clearance being used. This arrangement proved very successful in reducing the passage of oil past the piston rings because of the exceptionally free venting of the oil below them, but it did not permit of any appreciable reduction in clearance, for the simple reason that very little heat is transmitted from the crown to the bearing surfaces in any case; in other words, it was found that, at all events in the case of comparatively small engines, the temperature of what may be termed the cross-head portion of the piston was in any case very little in excess of that of the cylinder walls, so that insulating it from the crown had little influence on its temperature, and therefore on its expansion.

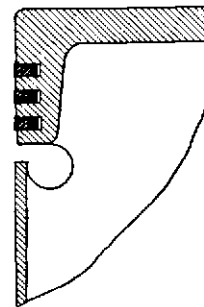


Fig. 129.—Split Slipper Piston

None of the methods described above can be said to have solved the difficulty of piston knock, or even to have gone very far towards solving it, and the author is inclined to the opinion that where extreme silence is required it is better to employ cast iron for the bearing surfaces in all except very small pistons, in which the clearance may be so small that there is little or no knock in any case, for the noise is dependent upon the absolute, rather than upon the proportionate, clearance. Experience has shown that with all-aluminium pistons fitted to a water-cooled engine of average performance it is necessary to allow a clearance on the bearing surfaces of approximately 0.002 in. per inch of diameter. As a general statement, when the total clearance exceeds from 0.005 in. to 0.006 in. piston knock becomes audible, that is to say, all-aluminium pistons up to 3 in. can be made to run silently, depending upon the lubrication and a number of other minor controlling factors; but above 3 in. diameter it is extremely difficult, if not impossible, to ensure silent running.

**Piston Rings.**—Generally speaking, piston rings do not call for much comment. With but few exceptions all high-speed internal-combustion engines employ ordinary plain concentric cast-iron rings of the Ramsbottom type. Such rings should always be ground both on the face and sides in order to ensure a close fit in the grooves, and should preferably be hammered, after being split, in order so to stress the material as to ensure a uniform pressure against the cylinder walls. The most important feature to ensure is that they fit truly against the sides of the grooves, since this determines largely the amount of oil they will pump. The top gas ring should be allowed a few thousandths clearance in order to keep it from sticking and, when the scraper ring groove is perforated, this ring also should be allowed liberal side clearance. When aluminium pistons are used there is always a tendency for the edges of the “lands” in the piston to be dragged-over by particles of grit and so to lock the rings in their grooves. This tendency can, however, be prevented by chamfering slightly the edges of the grooves. This is a small point, but it is one which should not be overlooked, for its neglect has probably done more than anything to prejudice unjustly the use of aluminium for pistons.

**Width of Ring.**—All reasoning points to the conclusion that piston rings should be made as narrow as possible so long as they are not too fragile to machine or handle. For a given radial thickness, the narrower the ring the less both the friction and the inertia

hence the lower the total pressure against and therefore the wear on the sides of the ring grooves. There, is, however, some evidence in support of the theory that an excessively narrow ring causes increased wear of the cylinder bore due to the comparative ease with which the oil film breaks down during the period the piston is at rest on top dead-centre.

**Radial Thickness.** — The radial thickness determines the spring tension against the cylinder walls. So long as this is above a certain figure there is no object in increasing the thickness, which merely involves an increase in friction and wear on the cylinder walls.

There is definite evidence to indicate that, when working, the rings are pressed out against the cylinder walls by the gas pressure behind them, and that the pressure in the ring groove is on the average something less than the mean pressure of the cycle. In fact recent experiments by the Admiralty Engineering Laboratory, in which indicator diagrams were actually taken from behind the piston rings, showed that the pressure behind the top ring followed very closely that of the gas pressure above the piston, while that behind the second ring showed a very much delayed diagram, and behind the third and others a more or less uniform pressure scaling down through the remaining pack of rings. Experience indicates that in a well-made ring a spring pressure of from 6 to 7 lb. per square inch is sufficient, and that any further pressure results merely in extra friction without any compensating advantage. The spring tension required, however, depends to some extent upon the amount of clearance between the "lands" of the piston and the cylinder walls. It is customary and proper to make the clearance of the "lands" such that they will not, under any circumstances, touch the walls of the cylinders. If, however, the clearance is too great, a considerable area of the side of the ring may be exposed to the full fluid pressure, and, as a result, the ring may be pressed so hard against the lower face of the groove that its spring tension will not overcome the friction against the side of the groove; under these conditions the ring will become locked and unable to expand against the cylinder walls, with the result that severe "blowing" will occur.

There is also very strong evidence to show that the top land should always be as deep as possible, in order to provide adequate protection for the top piston ring. In the designs of piston illustrated previously, the top land is in most instances rather too shallow, due to the fact that the available height from the gudgeon-pin centre to the piston crown was, in each case, so restricted as

to render the provision of an adequate protecting land impossible.

**Cross-head Piston.**—All the foregoing remarks refer to the open or trunk type of piston, and more particularly to that form of open piston which has come to be known as the slipper type, a form with which the author has had the most experience.

In the larger sizes of high-speed engine more particularly, the author prefers to use wherever possible a somewhat different type

which is now generally known as the cross-head piston.

In this design the two functions of an open piston—namely, to act as a piston proper and as a cross-head guide—have been separated to a far greater extent than in the case of the slipper type, until it resembles much more nearly the usual steam-engine form.

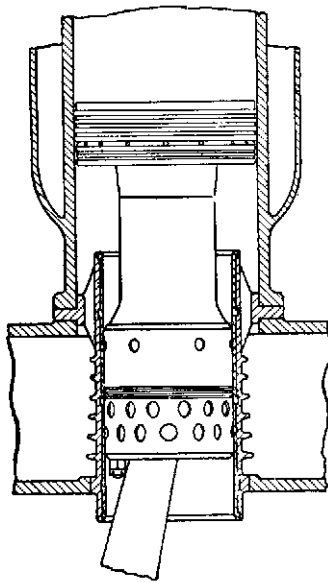


Fig. 130.—Part Section showing Cross-head Piston and Guide

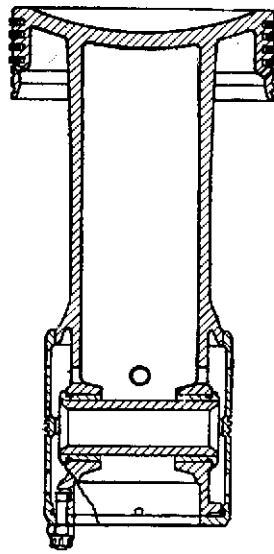


Fig. 131.—Section of Piston

The piston itself consists of an ordinary flat or concave crown carrying the piston rings and a plain light tubular stem extending from the crown of the piston to below the gudgeon pin. The lower portion of this trunk is surrounded by a steel or cast-iron sleeve, which embraces and locates the floating gudgeon pin, and constitutes the only wearing surface. This cross-head sleeve runs in a cylindrical guide, which is spigoted both into the cylinder and the crank chamber, or in some cases into the cylinder only, which is prolonged to accommodate it. A general arrangement of the piston, cylinder, and cross-head guide is shown in fig. 130, from which it will be observed that the crown of the piston serves only to carry the rings and transmit the pressure down the hollow cylindrical stem to the gudgeon pin. It does not bear upon the cylinder walls at all, and therefore requires only just sufficient lubricant to maintain the rings in good condition.



It will also be observed that in this construction the cylinder walls are cut off from all splash lubrication.

In spite of the fact that this piston is some 30 per cent heavier than the slipper type, the total friction is little more than 80 per cent that of the slipper piston and only about 60 per cent that of a piston of the ordinary trunk type.

The construction of the piston is shown in detail in fig. 131, and the photographs reproduced in figs. 132 and 133. The cross-head sleeve is an easy push fit over the lower portion of the stem, and is held in place by four small bolts; it is supported by three bearing lands, one at the middle on the gudgeon-pin centre line and one at each end.

In ordinary commercial engines this sleeve is made of cast iron, but in very high-speed engines a light high carbon steel sleeve is used. When employing



Fig. 132.—Cross-head Piston

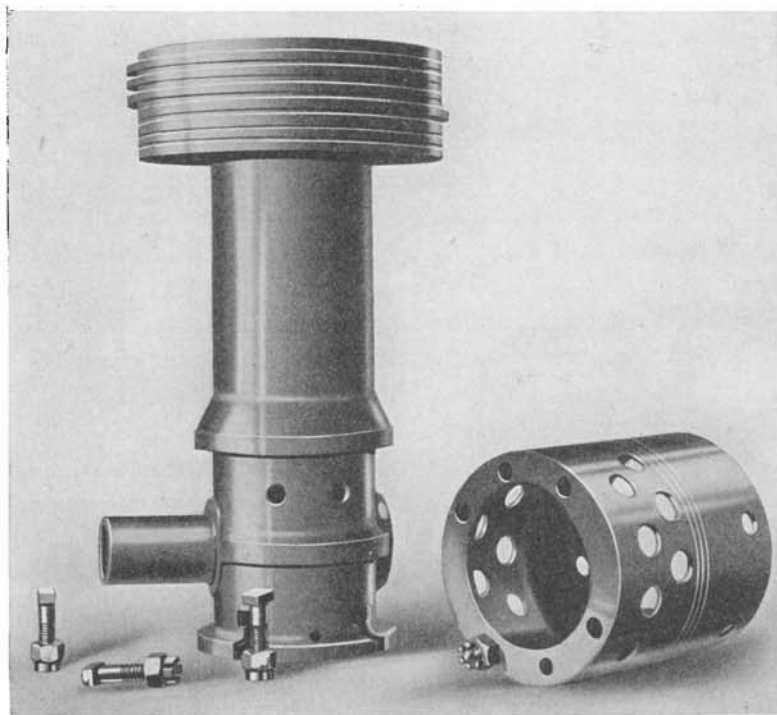


Fig. 133.—Cross-head Piston dismantled

this piston on engines having separate cylinders the guide is spigoted both into the cylinder and crankcase, and the latter is provided with a false top as shown in fig. 130 above. The space between the false top and the base of the cylinder is utilized for the circulation of air round the cross-head guide, which is ribbed for cooling. The air enters at one side and passes out to the carburettor on the other side. A portion of the air is drawn directly round the cross-head guide, and the remaining portion passes between the guide

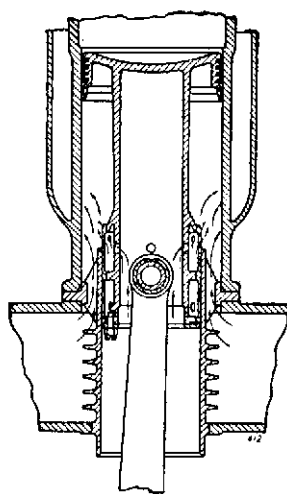


Fig. 134.—Cylinder Lubrication with Cross-head Piston

and the cylinder through slots provided for this purpose. On the upward stroke of the piston the air is drawn through these slots at a high velocity, and impinges against the crown and stem of the piston, thus effectually cooling them. On the downward stroke this heated air is discharged again into the chamber surrounding the guides, and thence into the carburettor. By this means the piston and cross-head guide are kept cool and the carburettor air is warmed.

It is found in practice that the heat abstracted from the piston and cross-head guide is just sufficient for good distribution with petrol of high boiling-point and low volatility. Tests carried out on several of the engines built for tanks with thermometers fitted in the induction piping above and below the carburettor have shown that when running on full load with an atmospheric temperature of 60° F. the air, after passing round the cross-head guides and pistons, entered the carburettor at a temperature of 130° F.

On a light load, with consequent reduced air circulation, the temperature of the air entering the carburettor rose to 150° F., and the temperature near the top of the induction pipe to 100° F., which is sufficiently high to check condensation at reduced loads. The free circulation of air through the upper portion of the crankcase tends to keep the lower portion cool, so that no oil cooling is required.

The system of cylinder lubrication is shown in fig. 134. The lower portion of the stem of the piston is provided with a few small holes, and the cross-head sleeve which surrounds it is also provided with a ring of small holes so placed that these holes are uncovered above the guide at the top of each stroke. On the upward stroke of the piston, air is drawn through slots provided in the flange of the cross-

head guide between the guide and the cylinder, and passes at a high velocity around the cross-head sleeve; in doing so, it draws a small proportion of air and oil mist from the holes in the cross-head sleeve, which are in communication with the crank chamber through corresponding holes drilled in the piston stem. The oil issuing from these holes in the form of a mist is picked up by the rush of air and sprayed over the cylinder walls while the piston is near the top of its stroke; the total quantity of oil drawn out in this manner is minute, but it is sufficient for the maintenance of the piston rings. The whole operation is similar to that of a spray carburettor in which the slots in the cross-head guide correspond to the choke tube, and the holes in the sleeve to the jets. The control of the quantity of oil delivered in this manner is governed by the area of the slots and the size or number of holes provided in the sleeve.

It will be seen that, by this means, the lubrication of the cylinder walls is continuous, that oil is only supplied to the cylinder walls in the quantity required by the piston rings, and that oil which has clung to the walls and become partially carbonized does not find its way back into the crankcase. The provision in this manner of an entirely separate system of lubrication to the cylinder allows of the use of unstinted lubrication to all the other working parts without the risk of carbonization of the piston or any tendency to smoke; also, the oil consumption is exceedingly low.

When working with kerosene or high boiling petrols, this type of piston is particularly suitable, for one of the chief troubles with such fuels is that they tend to precipitate upon the relatively cool walls of the cylinder barrel and so to pass down into the crankcase, thus contaminating the lubricating oil, and cause trouble with the bearings. With the cross-head type piston, however, any fuel which may succeed in passing the piston is trapped in the chamber surrounding the cross-head guides, from which it may be drained off before it can do any harm. The quantity of kerosene which, in practice, is drained away from this chamber is often surprisingly great, particularly when working on variable loads, often amounting to as much as from 4 to 8 per cent of the total fuel consumption of the engine, or from three to six times the oil consumption.

The advantages of this type of piston may be summarised as follows :—

(1) The lubrication is under complete control, and is independent of the crankcase lubrication; consequently the oil consumption, the tendency to carbonize both the piston and combustion chamber,

and the risk of oiling up the sparking plugs are all reduced to the minimum.

(2) The piston friction is reduced to little more than half that which obtains with an ordinary trunk piston.

(3) Owing to the fact that the cross-head and guide are relatively cool, and that both are maintained at approximately the same temperature, a very fine running clearance can safely be used, thus ensuring silent running.

(4) Since the piston itself does not bear upon the cylinder walls, an ample working clearance can be allowed without any risk of noise.

(5) The wear on the cylinder walls is reduced to a minimum, since only the piston rings bear against them and there is no side thrust.

(6) The gudgeon-pin being short, stiff, and free to rotate, and also being placed in such a position that it receives very little heat from the piston, does not wear perceptibly.

(7) The bulk of the heat from the crown of the piston and from the cross-head guide is utilized to warm the air for the carburettor, and is not transferred to the crankcase.

(8) All the working parts can be lubricated without stint and without any risk of excess of oil reaching the cylinder walls; also, the oil remains clean.

(9) In the event of any fuel condensing on the walls of the cylinder, its subsequent passage into the crankcase can be prevented absolutely.

(10) The restricted lubrication to the cylinder walls greatly reduces any tendency of the piston rings to become carbonized or gummed up.

(11) There is no tendency for the engine to become "gummed up" when cold.

The principal objections to the use of this type of piston are :—

(1) That it increases the height of an engine as compared with the use of an open-type piston, by an amount equal to about two-thirds of the piston's stroke.

(2) That it necessitates an engine designed specifically for its use, and unless separate cylinders are used it introduces difficulties in the way of alignment between the cross-head guides and the cylinder bore. These difficulties are not, however, insuperable, as is illustrated by the engine shown in fig. 135, which shows the application of cross-head pistons in the case of the engines manufactured by Messrs. Peter Brotherhood, Ltd., for tractors and marine work, in

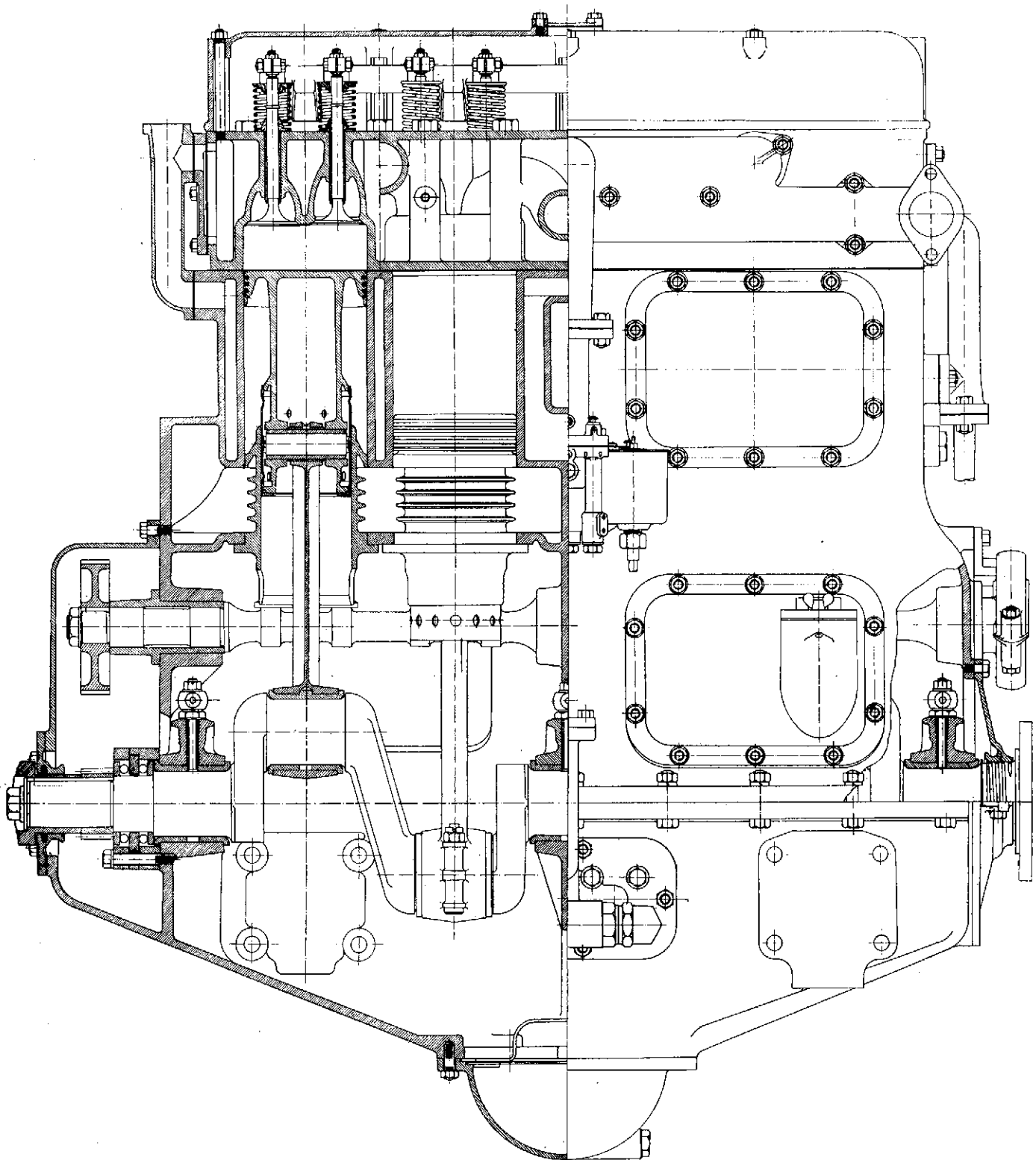


Fig. 135.—General Arrangement of  $4\frac{3}{4} \times 5\frac{1}{2}$  30-B.H.P. Brotherhood Tractor Engine

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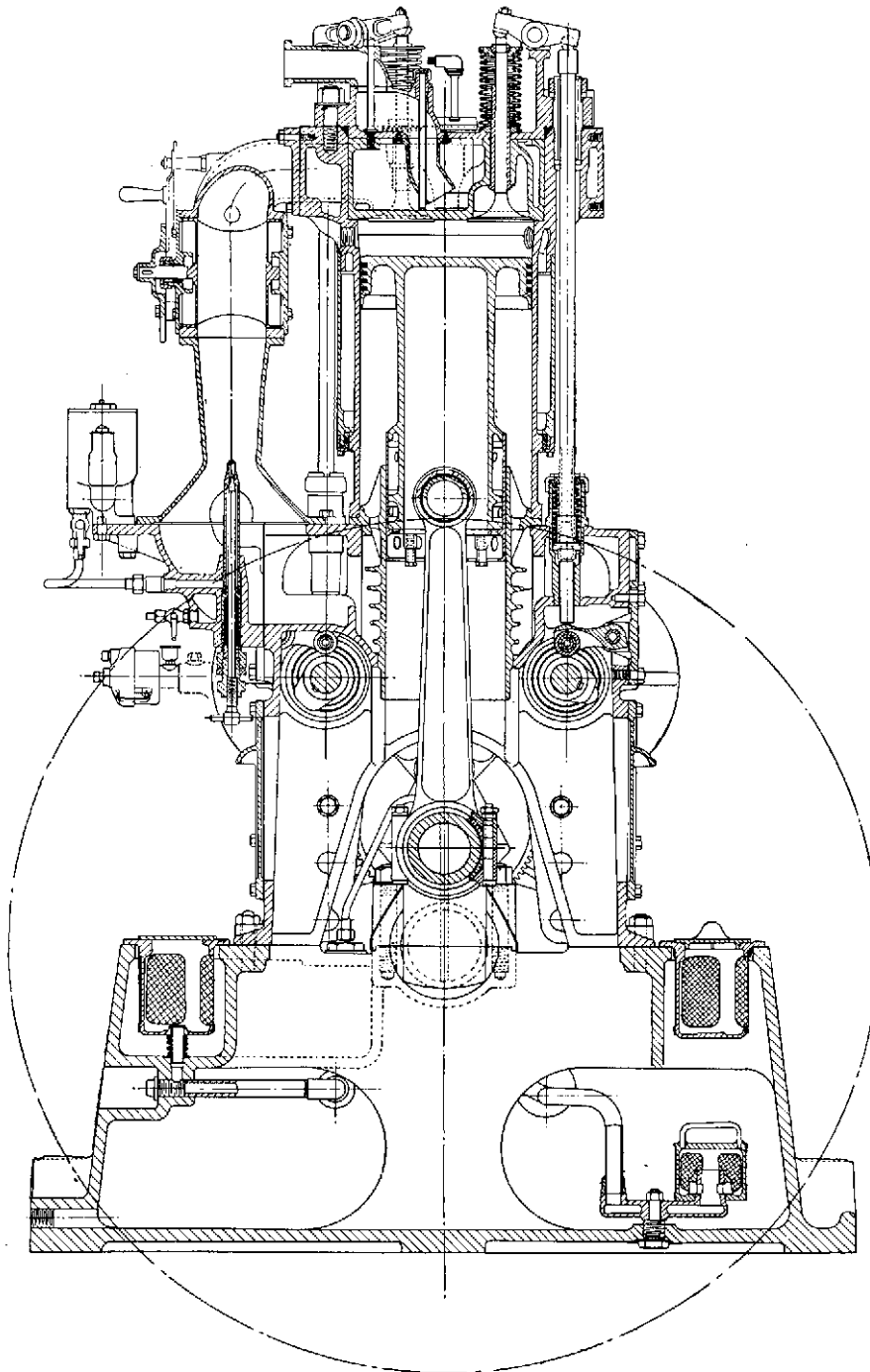


Fig. 136.—General Arrangement (Cross Section),  $3\frac{1}{2}'' \times 9\frac{1}{2}''$  Single Cylinder. Experimental Engine  
(E 246)

which the four cylinders and the upper half of the crankcase are cast in one piece.

(3) That it increases slightly both the cost and weight of an engine.

These objections are, in the author's opinion, easily outweighed by the advantages to be gained, more especially in the case of large engines, such as that shown in fig. 136, or in the case of engines using kerosene or petrol with a high final boiling point. In any case their use makes for a higher mechanical efficiency, silent running, and reduced cylinder wear and oil consumption.

## CHAPTER X

### ENGINES FOR ROAD VEHICLES

Before dealing with specific examples of motor-vehicle engines it will be well to review briefly the duties which these engines have to perform and to note along what lines further development is likely to extend.

Some twenty-five years ago the designers and manufacturers of motor-vehicle engines had need to concentrate the whole of their attention upon the one crucial problem of producing engines which would run for a reasonable period, under wholly novel conditions, without serious breakdown. This great problem overshadowed all others, and the rapidity with which it was tackled and overcome is one of the great triumphs of modern mechanical engineering. Within a space of less than ten years the motor-vehicle engine emerged from the stage of a fickle and wayward, but very fascinating, toy into a thoroughly reliable machine. Once its reliability had been established, and its charm largely exchanged for utility, the subsequent developments were mainly in the direction of refinement and increased power.

In order to provide a more uniform turning movement, and to reduce vibration, and therefore noise, the number of cylinders was increased from one to four and even to six.

The next developments were in the direction of securing greater silence in operation; these took the form of improvements in the valve gear and the elimination as far as possible not only of vibration of the engine, as a whole, but also of the vibration of the individual members of the engine.

At the same time the available speed range has steadily increased. Since owing to the inherent irregularity of torque in any type of four-cycle engine the speed range cannot usefully be reduced below a certain minimum, progress has taken the form of extending the upper end of the speed range. Developments in this direction have been much stimulated by the method of basing the taxation, of



pleasure vehicles at least, upon the diameter of cylinder used. This basis of taxation has served well, but would now be more useful if it were based on the total cylinder capacity.

With the extended use of motor-vehicles for purely utility purposes, and with the increasing cost of fuel, the most needed developments at the present time are in the direction of fuel economy, a direction in which there is still ample scope for improvement.

The motor-vehicle engine of to-day is called upon :—

- (1) To be silent under all conditions of operation.
- (2) To be self-contained and as automatic as possible.
- (3) To have as wide a range of speed as possible.
- (4) To accelerate rapidly from any speed ; in other words, it must instantly develop its maximum torque when called upon to do so, irrespective of engine speed.
- (5) To maintain a high torque at low speeds, and to do so without detonation or “ pinking.”
- (6) To be reasonably economical in fuel at all loads, and more particularly at its average load factor of from 25 per cent to 40 per cent maximum torque.

The performance of any motor-vehicle engine must be considered in reference to the vehicle to which it is fitted. We will therefore examine briefly one specific instance, namely :—

A light pleasure car weighing, fully loaded, including passengers and equipment, 3500 lb., and fitted with a wind-screen and hood. We will assume that the transmission gear is of an efficient type, that the unsprung weight is low, the weight well distributed, and that the chassis generally is as well designed throughout as the present state of the art will permit.

Unfortunately, very little accurate data is available as to the exact power required to propel a motor-vehicle at different speeds over average roads. Professor Riedler in Germany, and Chase and James in America, have made and analysed a number of dynamometer tests with the rear wheels of the car resting on rollers, but these do not always reproduce the conditions exactly. For information on this point we are compelled to fall back to a large extent upon tests carried out with accelerometers and upon accumulated experience based upon the known performance of the same engine, both on the test-bed and on the road—the latter method, though very unscientific and largely empirical, is probably the most accurate at the present time. The curve in fig. 137 gives to the nearest approximation the brake-horse-power required at the engine

flywheel to propel a 3500 lb. 4-seater car at speeds up to 80 miles per hour. It includes rolling-resistance, windage, transmission losses (on direct drive), and all other incidental losses such as wheel slip, hysteresis losses in the tyres, &c. Though purely empirical it is probably reasonably correct. For such a car the minimum size of engine of normal side-valve type which will give reasonable acceleration and hill-climbing capacity will be one of two-litre cylinder capacity, while for real comfort a three-litre engine will be preferable.<sup>1</sup>

We will consider both cases and assume that the engines are of

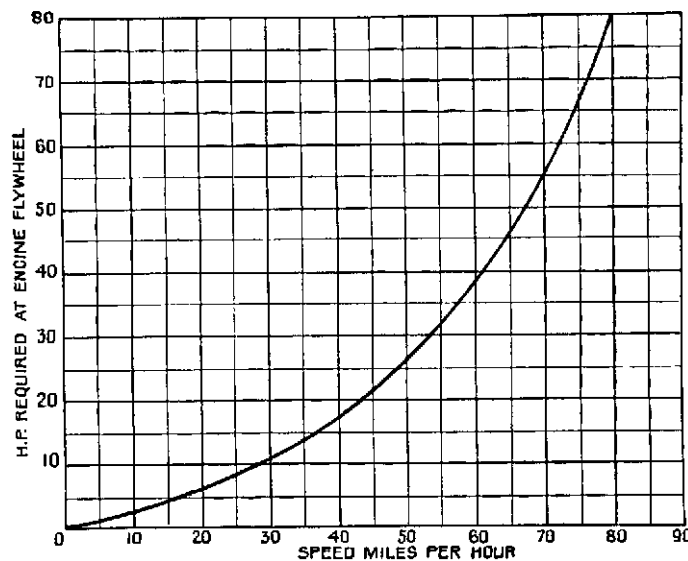


Fig. 137.—Power required at Engine Flywheel to propel Car of 3500 lb. gross Weight on the Level at varying Speeds over average Road Surfaces

the normal side-by-side valve type fitted with as efficient a form of combustion chamber as this type will permit of. Further, we will assume that the engines are designed with a view to low cost of production and ease of upkeep, that they have a reasonably low compression ratio, viz. 4.6:1, to render them capable of using inferior fuel without detonation, and generally that they are of a thoroughly orthodox type, but as efficient as possible without resorting to the use of overhead valves or to any features involving either increased cost of production, or labour in upkeep.

Figs. 138, 139, and 140 show the brake horse-power and general

<sup>1</sup> During the eight years which have elapsed since this chapter was prepared great improvements have been made, with the result that the two-litre engine of 1930 is fully the equal of the three-litre engine of 1922.

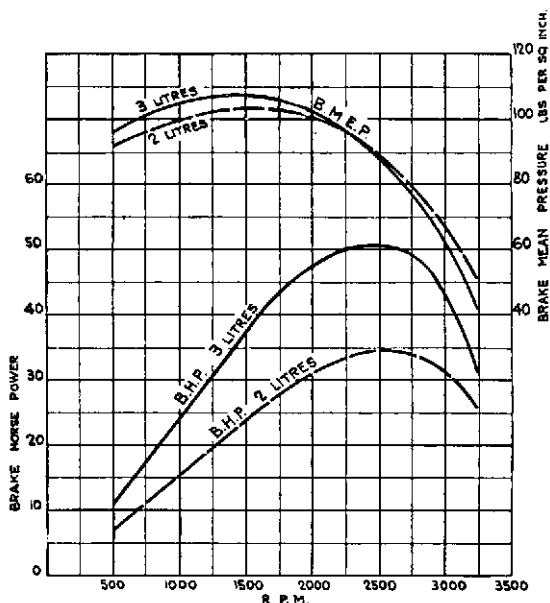


Fig. 138.—Brake Horse-power and Brake Mean Pressure of normal Two- and Three-litre Engines

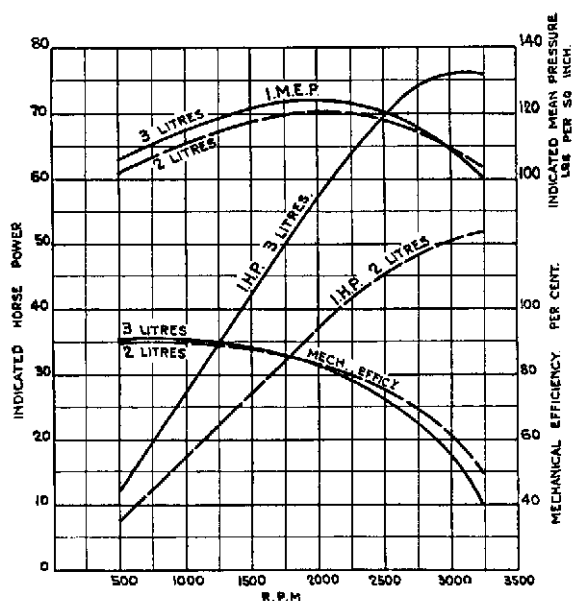


Fig. 139.—Indicated Horse-power, indicated Mean Pressure, and Mechanical Efficiency

1100 R.P.M. in the case of the two-litre, and at 880 R.P.M. in the case of the three-litre engine. If now we plot the power

performance curves which should be obtainable from such engines of two- and three-litre cylinder capacity.

Next we will assume that, in both cases, the top gear ratio is such that the maximum speed on the level is obtained when the engine is running at a speed slightly in excess of that at which maximum power is developed—this is always desirable, both on the grounds of acceleration, and in order to reduce the hysteresis losses due to irregularity in the turning moment. We will assume that three speeds are used and that the ratios in the gear box are such that the second speed is 70 per cent and the third 33 per cent of the top or direct drive. From a comparison of the power curves of the two engines and the power required on the level, as shown in fig. 137, we find that the most suitable gear ratios for the direct drive are those which give a car speed of 20 miles per hour at

curve of the two engines against fig. 137 in the above ratio, as shown in fig. 141, we find that the two-litre engine will give a maximum speed on the level of 54.5 miles per hour and the three-litre of 66 miles per hour. The margin of power at any speed over and above that required to propel the car on the level may be termed the excess power available for hill-climbing or acceleration.

Figs. 142 and 143 show the excess-power curves for the two engines on the three gears, assuming an efficiency as compared with top gear of 95 and 97 per cent respectively for the first and second speeds. Strictly speaking, the relative gear losses will be less in

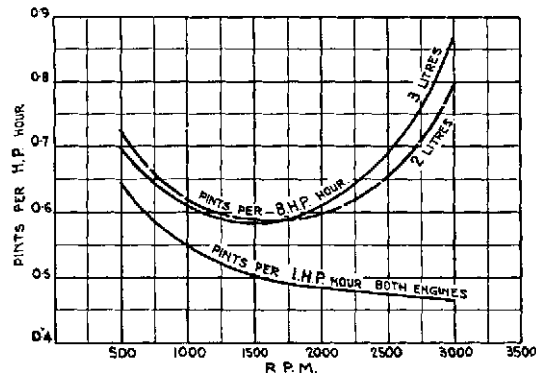


Fig. 140.—Fuel Consumption at full Torque Pints per Indicated and per Brake Horse-power Hour

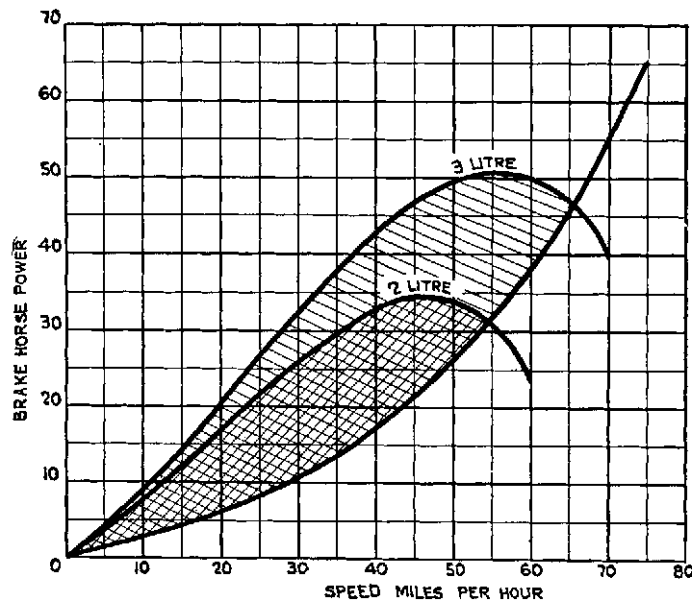


Fig. 141.—Excess-power Curves for Two- and Three-litre Engines on Top Gear

the case of the larger engine, but the difference is small and hardly worth taking into consideration. Figs. 142 and 143 show also the gradient in terms of per cent which the car will climb on each

gear and the speed at which it will climb it without gain or loss in speed.

From the curves shown in these two figures it will be seen that

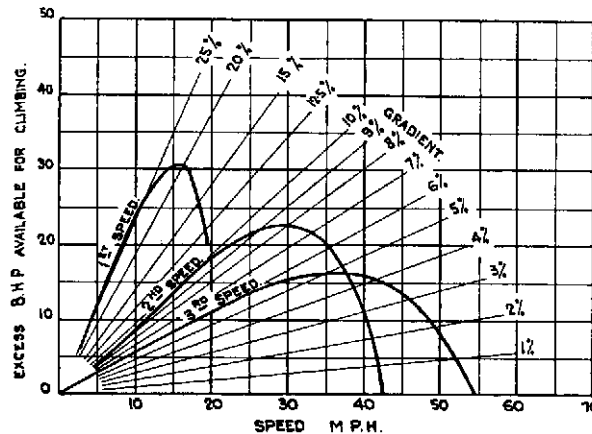


Fig. 142.—Hill-climbing Capacity, Two-litre Engine

the maximum gradient which the three-litre engine will climb without gain or loss of speed on its third or top gear is one of 8 per cent, on its second speed the maximum gradient is about 13.5 per cent, and on its bottom speed about 31 per cent. It will be noted

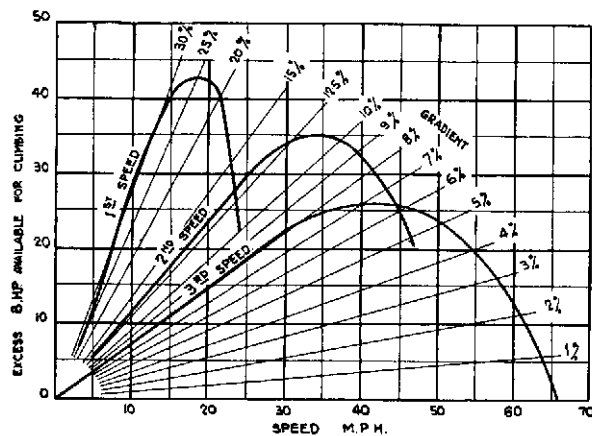


Fig. 143.—Hill-climbing Capacity, Three-litre Engine

that on a gradient of 6.2 per cent the maximum speed will be the same on either top or second speed, namely, 45 miles per hour.

In the case of the two-litre engine, the maximum gradient which the car will climb at a uniform speed on top gear is one of 6 per cent, on second speed 10 per cent, and on bottom speed about 25.5 per cent. For maximum speed in this case, gear should be changed

from top to second when the gradient exceeds 4·7 per cent, or when the speed has dropped to 38 miles per hour.

Fig. 144 gives the rate of acceleration of the car with two-litre engine, from any speed, and on the three gears, assuming that the carburation and distribution are such that the engine will respond instantly and exert its maximum torque immediately the throttle is opened—a condition, however, which is seldom reached in practice. The foregoing curves show the general performance of the car as regards ultimate speed, acceleration, and hill-climbing capacity.

We have next to consider the question of fuel consumption and the factors which control it. For this purpose we will assume that the car will be running always on its top gear, and we will examine

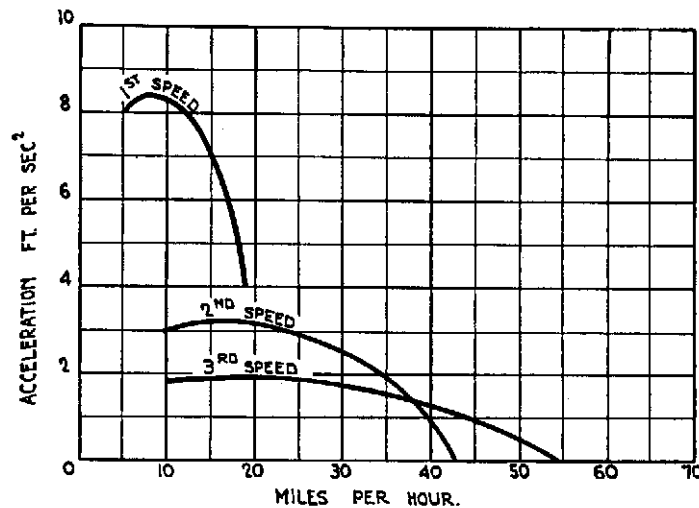


Fig. 144.—Acceleration on various Gears, Two-litre Engine

the speed range between 10 and 40 miles per hour, which covers the range of *average* speed such a car will maintain. For simplicity we will assume also that the car is running on a level road, though, in so far as fuel consumption is concerned, it makes comparatively little difference whether the road is level or undulating provided the gradients are well within the limits which the car can negotiate on top gear and that the average speed is not too low. Although, when coasting, one does not recover what is lost in climbing, yet this is very nearly compensated for by the more favourable load factor when pulling uphill.

Figs. 145 and 146 show the load factor in the case of the three- and two-litre engines at speeds ranging from 10 to 40 miles per hour, and the fuel consumption in terms of pints per B.H.P. hour at the

corresponding load factors and speeds. These figures are deduced from the mean of a large number of test results upon several engines

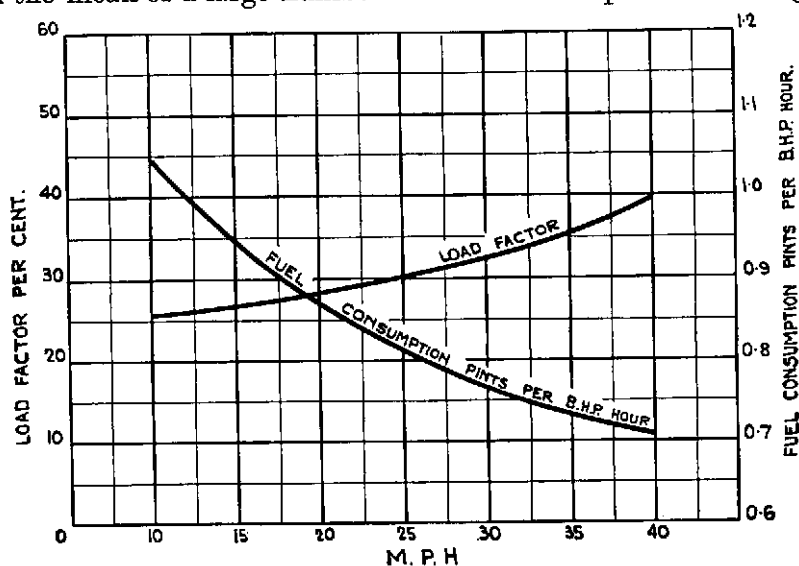


Fig. 145.—Load Factor and Fuel Consumption in Pints per B.H.P. Hour when running on the Level at average Speeds of from 10 to 40 Miles per Hour. Three-litre Engine

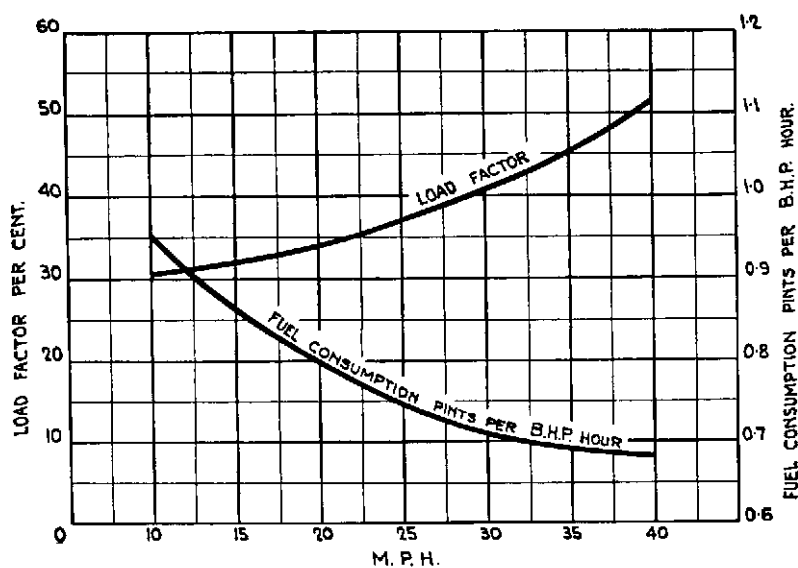


Fig. 146.—Load Factor and Fuel Consumption in Pints per B.H.P. Hour when running on the Level at average Speeds of from 10 to 40 Miles per Hour. Two-litre Engine

of the size and class under consideration with carefully adjusted carburettor and ignition settings and a reasonably good distribution system. In fig. 147 is shown the fuel consumption in terms of miles

per gallon for the two engines at average car speeds ranging from 10 to 40 miles per hour.

With normal carburation the consumption per mile is about 7 per cent greater with the larger engine at an average speed of 20 M.P.H., but the discrepancy becomes less as the average speed increases. With perfect carburation and distribution, &c., the discrepancy will become less, and at the higher mean speeds the larger engine will show, with the gear ratios selected, an actually greater fuel economy than the smaller one. In either case the larger engine will, in fact, make a better showing if the road is hilly or undulating, for it will then be able to negotiate gradients on top speed which, in the case of the smaller engine, might necessitate a change of gear.

There is another factor which also exerts a still more powerful

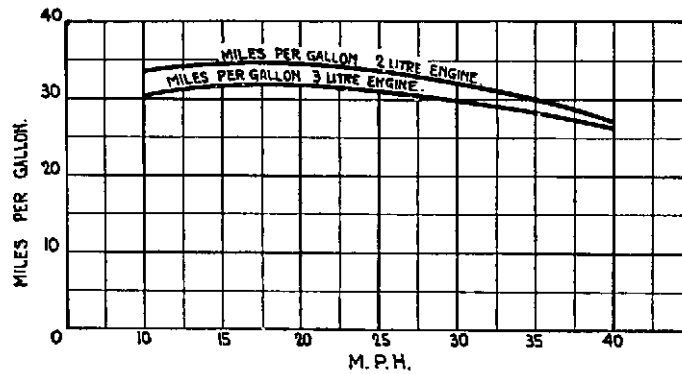


Fig. 147.—Fuel Consumption in Miles per Gallon at average Speeds varying from 10 to 40 Miles per Hour

influence upon fuel economy than carburation and distribution, and indeed upon the whole performance of the car, namely, the mechanical efficiency of the engine. This becomes the more important because of the very low load factor at which the engine operates. In the example shown, a fairly high mechanical efficiency has been assumed such as would be obtainable with light reciprocating parts and careful mechanical design. The average motor-car engine with cast-iron pistons and often excessive and ill-disposed bearing surfaces will not show by any means so high a mechanical efficiency.

It is perhaps worth while to consider the case when the car is travelling on the level at a mean speed of, say, 25 miles per hour and observe the influence of the mechanical efficiency of the engine upon fuel consumption. This speed calls for an expenditure of 8 B.H.P. at the flywheel of the engine and corresponds to an engine speed of 1100 R.P.M. in the case of the three-litre and 1375 R.P.M.



in the case of the two-litre engine : at these speeds the mechanical efficiency of the two engines has been taken as 90·5 per cent and 88·5 per cent respectively on full torque, but since the internal friction of the engine is independent of the torque, at 8 B.H.P. the mechanical efficiency will have fallen to 73·5 per cent and 73 per cent respectively. If now the mechanical losses were doubled owing to poor mechanical design, all other conditions remaining the same, the mechanical efficiency when driving the car on the level at an average of 8 B.H.P. will become only 58·2 and 58·6 per cent respectively, and the fuel consumption per B.H.P. hour will have increased from 0·81 and 0·745 to 1·02 and 0·93 pints per H.P. hour at the same load factor, but in fairness we must allow for the fact that, owing to the poorer performance as a whole, the load factor will be somewhat higher. If we take this into account we find that the consumption at a mean speed of 25 M.P.H. will be approximately 0·97 and 0·89 pint per B.H.P. hour, corresponding to a fuel consumption in terms of miles per gallon of 25·8 and 28·1 as against 31 and 33·6 for the three- and two-litre engines respectively. From these figures it will be seen that the gain in fuel economy to be obtained by a limited and perfectly possible improvement in mechanical efficiency is a very substantial one. Further, a gain in mechanical efficiency will influence not only the fuel economy but also the speed and hill-climbing capacity of the car throughout its whole working range. From such considerations we are justified in assuming that of the available scope for improvement the most important is that of reducing as far as possible the internal friction losses of the engine, and next in importance are improvements in carburation and distribution.

Unlike engines for other purposes, we may regard the pleasure-car engine as one which will never be called upon to develop high power, except for very short periods, and we have shown that the average load factor under normal running conditions is about 30 to 40 per cent in the case of the engines under consideration. Expressed in other terms, the average power required at the engine flywheel to propel a touring motor-car under normal conditions at an average speed of 25-30 miles per hour is approximately 7 H.P. per ton (unladen), while with even the most reckless driving it is almost impossible on any English main road to average 15 H.P. per ton, altogether irrespective of the maximum power of the engine.

In this connection it is interesting to note that from careful observations of fuel consumption made during the practice runs for the Isle of Man Tourist Trophy Race in 1922, the average horse-

power developed by the Vauxhall racing cars during their fastest laps, when they averaged considerably over 60 miles per hour round a perfectly clear and very hilly course, was certainly less than 50 B.H.P., even assuming that they were using the most economical carburettor setting. In view of the fact that these engines were capable of developing well over 120 B.H.P., and that the cars were naturally driven at the highest possible speed consistent with barely reasonable safety, it appears rather surprising that so small a proportion of the available power could be utilized. It shows that even when roads are cleared of all traffic and when the driver is relieved of all responsibility so far as other road-users are concerned, when he is both highly skilled and prepared to incur considerable personal risk, he is still restricted, by road conditions, to utilizing more than about 40 H.P. per ton.

Most cars at the present day show an unduly high fuel consumption, and this is to be accounted for by

(1) The mechanical efficiency of the engine being usually very low; in the one application above all others where it should be as high as possible.

(2) The form of the combustion chamber being generally inefficient, due to lack of turbulence.

(3) Defective carburation and distribution, more particularly the latter.

Recent development has been confined almost solely to the addition of various refinements, to the elimination of noise and general smoothness of operation; such lines of development are, of course, very proper, but there is a tendency for the economic fact, that the efficiency of a vehicle as a whole lies in the number of ton-miles it will run per gallon, to be overlooked. In too many cases fuel efficiency appears to have been forgotten entirely in the search for silence, in the better-class cars, and for low cost of production in the cheaper varieties. The author uses the word forgotten advisedly as against forgone, for, as it has been shown in previous chapters, fuel economy is largely a question of design and can usually be attained without adding to the cost and without the loss of other desirable features. The history of engine development has been much the same in all classes of mechanical engineering—first a struggle to attain mechanical reliability, during which stage the engine is a fascinating toy; this is generally followed by a period of intense rivalry in detail refinement to the neglect of other considerations; finally the inexorable laws of economy insist that

attention shall be concentrated on what is really the final test, namely, the amount of work an engine will do on a given quantity of fuel and on a given weight and cost of material. In the case of the motor vehicle we are probably passing from the second to the third stage of development and are beginning to realize the absurdity of, for example, loading the engine at all times with a heavy burden of frictional losses often merely for the sake of getting it to run a little slower and a little quieter when idling. As in the case of all new developments which fall into the hands of a lay public, fashion plays

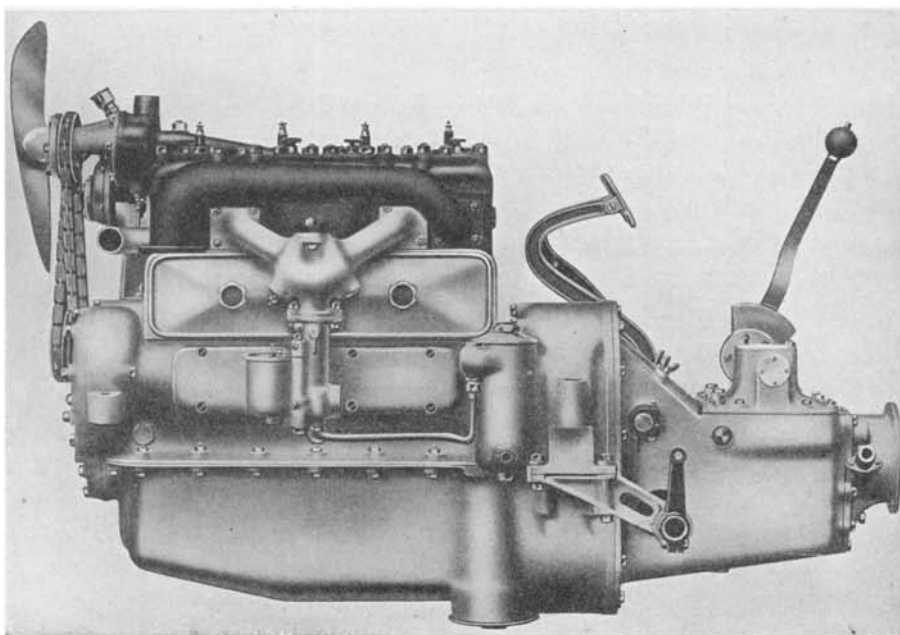


Fig. 148.—14-H.P. Vauxhall Engine

a predominant part, and fashion to-day calls for refinement in detail to the neglect of all other considerations. Ultimately utility will call for economy in operation, and the attention of designers will be concentrated upon reducing mechanical losses and improving distribution.

**The 14-H.P. Vauxhall Engine.**—The 14-H.P. Vauxhall engine illustrated in figs. 148, 149, 150, and 151 has been designed by Mr. C. E. King, Chief Engineer of the Vauxhall Company, to whose kindness the author is indebted for leave to publish the following particulars :—

It may be regarded as a typical example of a modern pleasure-car engine of the best type, designed to meet as far as possible the

dictates of fashion, and at the same time to show a performance both as regards power and fuel economy considerably above the usual average of engines of its class. It has four cylinders each of 75-mm. bore by 130-mm. stroke, giving a total cylinder capacity of approximately 2.3 litres, and is designed to drive a five-seated open touring car weighing complete with passengers and usual equipment about 3200 lb. It develops a maximum of 43.5 B.H.P. at a speed of 2600-2700 R.P.M.

Particular care has been taken to reduce, as far as possible, the internal friction losses, and also to obtain an efficient form of com-

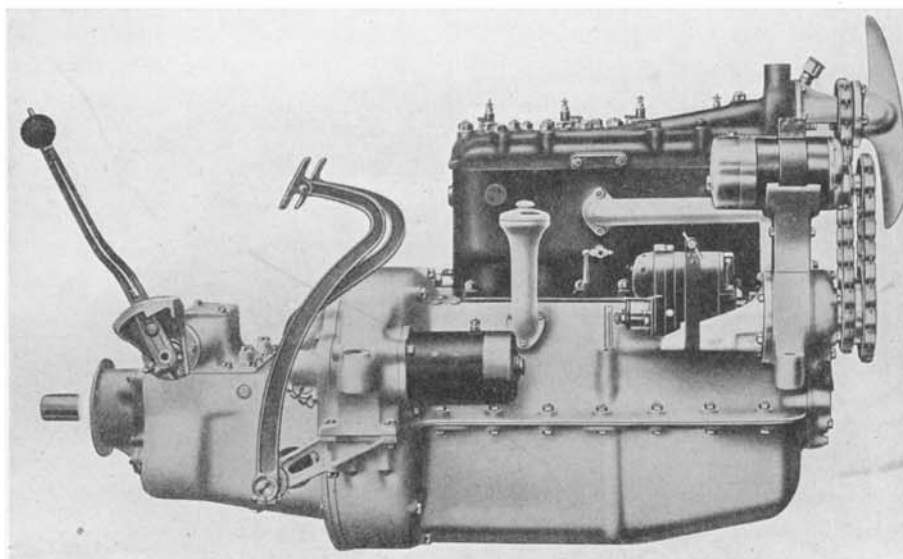


Fig. 149.—14-H.P. Vauxhall Engine

bustion chamber, with the result that the power output and efficiency are both very considerably greater than that of the average side-valve engine, particularly so at reduced loads. The details of the design are shown in figs. 150 and 151, from which it will be seen that the four cylinders are cast in one block separate from the crankcase and with a common detachable aluminium cylinder head, the combustion chamber of which is as shown in fig. 150.

The crankshaft is carried in three white-metal lined bearings and is drilled for forced lubrication to all main and crankpin bearings. The pistons are of aluminium of the slipper type, but having a complete ring formed at the base of the slippers. The gudgeon pins float freely, both in the connecting-rods and pistons, and are

located endwise by means of circlips and washers. The total reciprocating weight of each line is 1.75 lb., while the rotating weight of the connecting-rod big end is also 1.75 lb. The inlet valves have a port diameter of 1.4 inches with a lift of 0.35 inch, and the exhaust valves a port diameter of 1.31 inches with the same lift. All valves are operated by means of push rods having curved slippers.

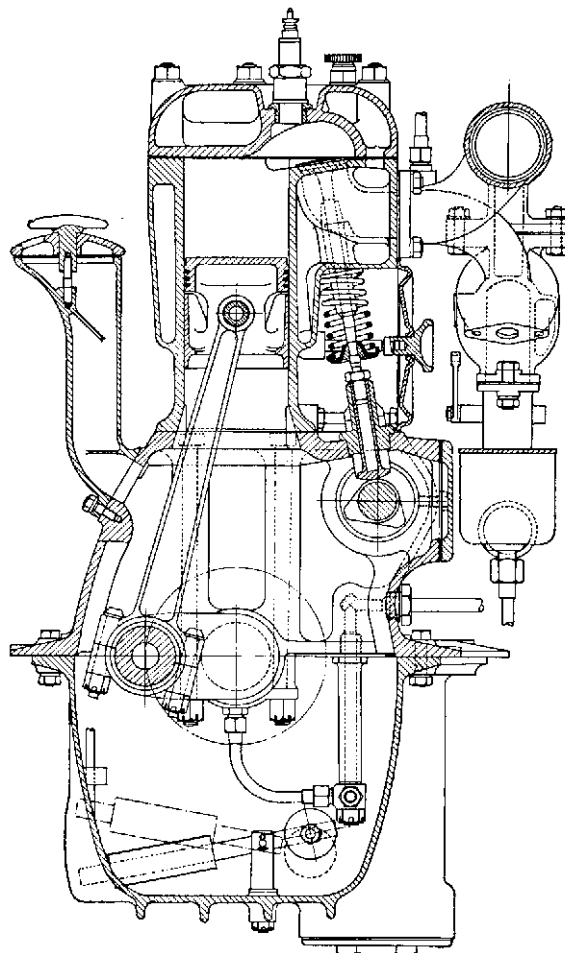


Fig. 150.—Arrangement of Cross-section, 14-H.P. Vauxhall Engine

The crankshaft has a diameter of 1.75 inches throughout, the widths of the several bearings being :

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Forward end	...	...	2.4 inches.
Centre	...	...	2.1 "
Flywheel	...	...	2.65 "
Connecting-rod	...	...	1.75 "

The ratio, therefore, of the piston area to the projected crankpin area is as 2.22:1, so that the load factor on the crankpin bearing is a very light one.

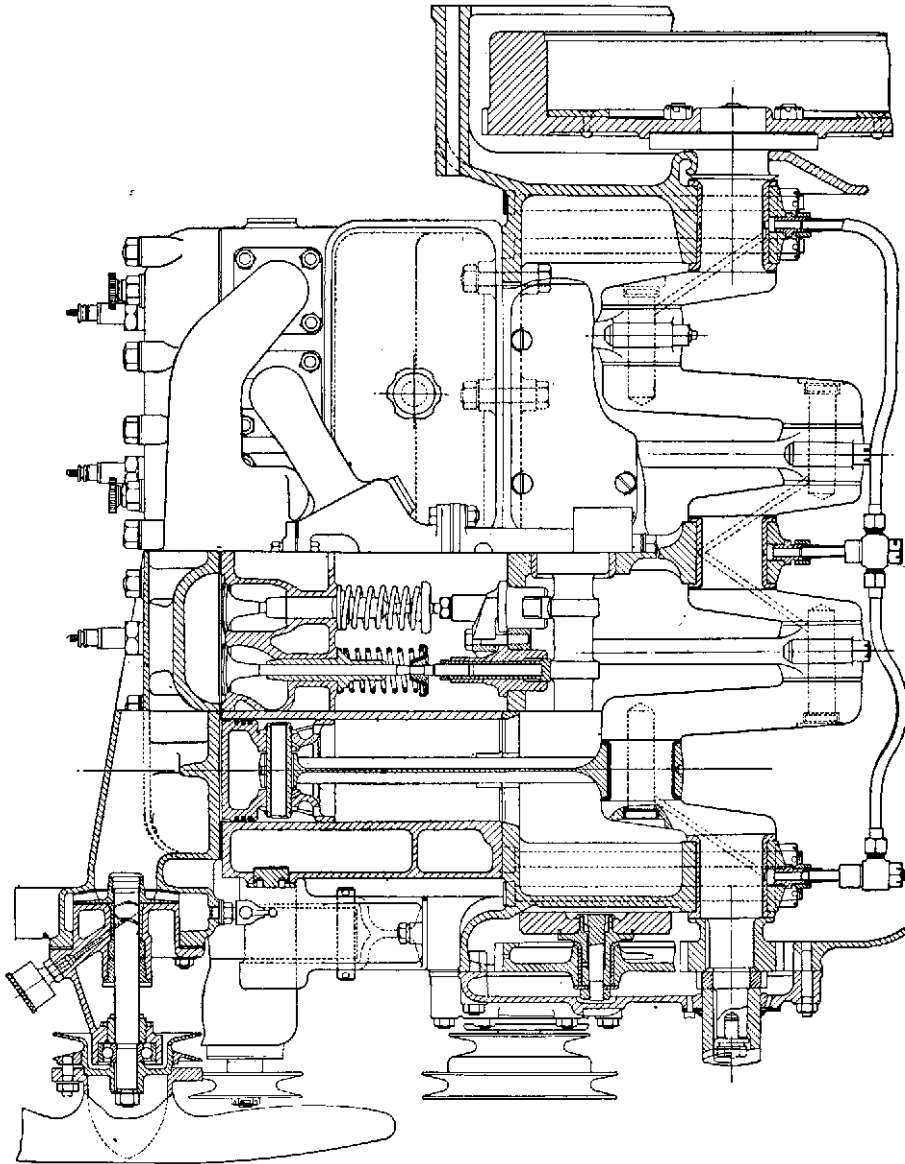


Fig. 151.—Arrangement of Longitudinal Section of 14 H.P. Vauxhall Engine. 75 x 130 mm.

The general performance of this engine, with a compression ratio of 5.1:1 and with wide-open throttle over a speed range from 750 to 2750 R.P.M., is shown in fig. 152, from which it will be seen that a brake mean pressure of 108 lb. per square inch is obtained at a

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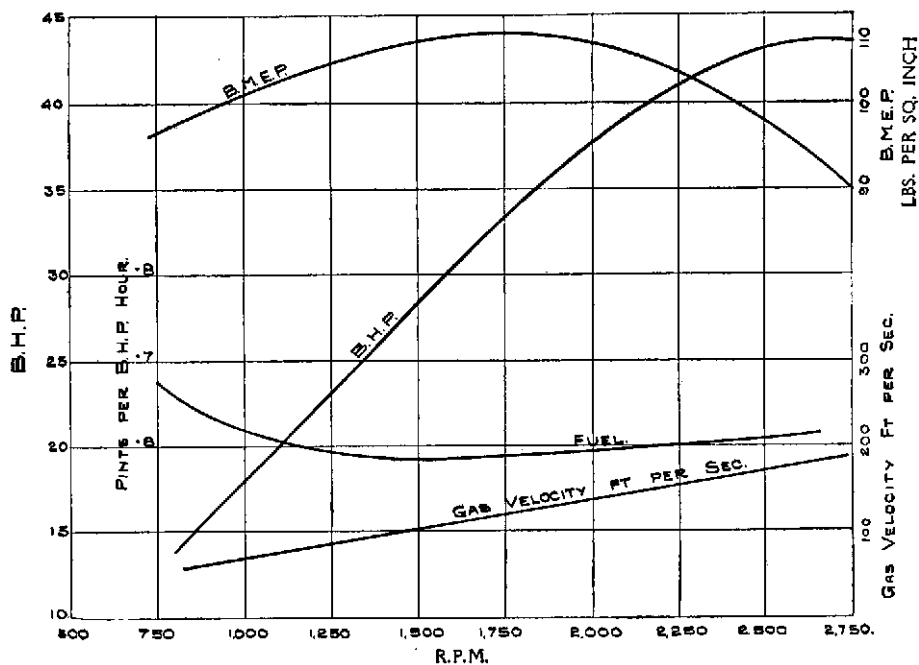


Fig. 152.—14-H.P. Vauxhall Engine. Gas Velocity, Brake Horse-power, Brake Mean Pressure, and Fuel Consumption Curves

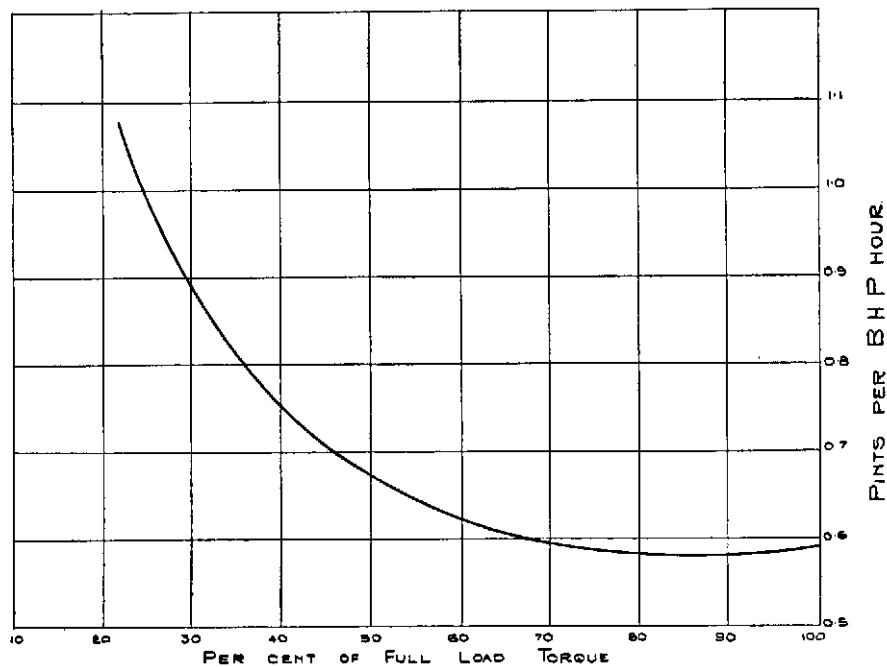


Fig. 153.—14-H.P. Vauxhall Engine. Fuel Consumption various per cent of full Load

speed of 1750 R.P.M. corresponding to a gas velocity through the inlet valves of about 120 ft. per second, the gas velocity through the induction pipe at this speed being about 175 ft. per second; this relatively high velocity is maintained in the induction pipe in order to keep the liquid particles of fuel in suspension even at reduced loads.

Fig. 153 shows the fuel consumption at a speed of 1600 R.P.M. when the power is controlled by throttling, from which it will be seen that even at 50 per cent full-load torque the consumption is less than 0.7 pint per B.H.P. hour.

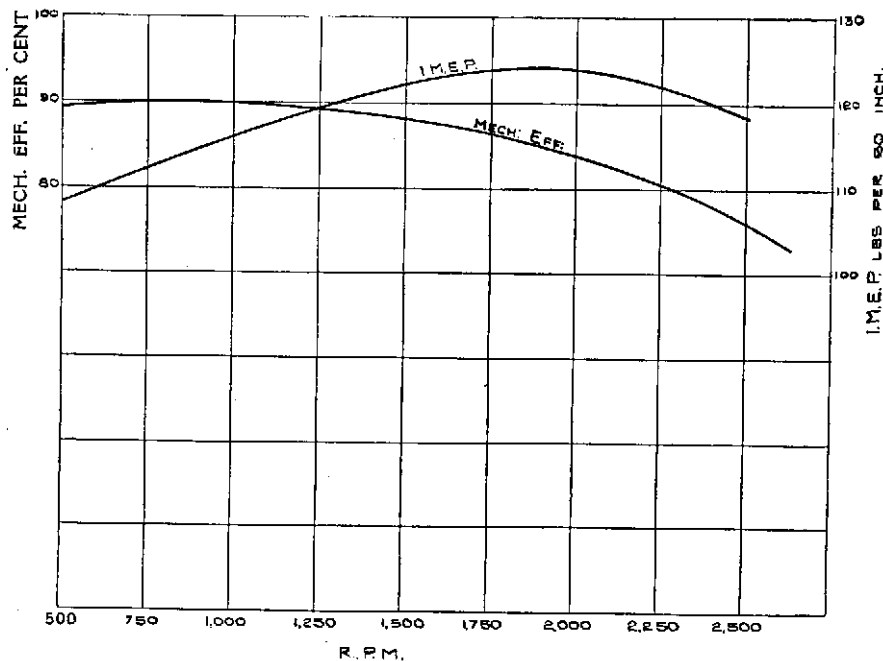


Fig. 154.—14-H.P. Vauxhall Engine. Indicated Mean Pressure and Mechanical Efficiency

No data is available as to the mechanical efficiency of this engine, but it can be estimated fairly accurately from the general design and checked back from the measured fuel consumption at reduced loads. From such deductions it would appear that the mechanical efficiency and indicated M.E.P. are as shown in fig. 154, from which it will be seen that the indicated mean pressure reaches a maximum when the gas velocity through the inlet valves is about 150 ft. per second. The indicated fuel consumption would probably reach its minimum at or about this speed, but, unfortunately, it is clear from the throttle curve that the carburettor was set to give an over-rich mixture at full throttle, so that there is no real evidence available.



The engine drives the car through a three-speed gear-box giving ratios which correspond with road speeds of 6.25, 12.5, and 21 miles per hour at 1000 R.P.M. Its maximum speed on the level is a little short of 60 M.P.H., corresponding to an engine speed, without wheel slip, of 2850 R.P.M., and its consumption averages 30 miles per gallon at an average speed of 25 M.P.H. and 29 miles per gallon at 30 M.P.H.

**Sleeve-valve Engines.**—For motor-car engines where the need for silence is great and at the same time where, owing to the low average power factor, the heat flow is small, the use of sleeve valves in place of the ordinary poppet valves appears very attractive. Such valves have the following advantages :—

- (1) Their action is, or should be, noiseless.
- (2) Their employment permits of the use of the best possible form of combustion chamber with the sparking plug centrally situated, hence the indicated efficiency should be high and the tendency to detonate at a minimum.
- (3) They require less attention than poppet valves and cannot readily be thrown out of tune by misuse.

The objections are :

- (1) That the heat flow to the cooling water is necessarily somewhat restricted, though this is not of much moment in the case of motor-car engines, more especially when a single sleeve is used.
- (2) Unless the sleeve be given an abnormally long stroke the effective port area is necessarily restricted.
- (3) The sleeve or sleeves, having a large rubbing surface, necessarily entail a higher friction loss, more particularly when a long stroke is used.
- (4) It is possible only to operate the sleeve from one side unless the whole of the operating mechanism be duplicated, which involves excessive mechanical complication and introduces grave difficulties in the way of mechanical synchronization.

In a four-cycle engine the sequence of operation is such as cannot be fulfilled by a plain reciprocating sleeve, hence it is necessary either to employ two concentric reciprocating sleeves, as in the Daimler Knight engine, or a single sleeve with a combined reciprocating and rotary motion, as in the Burt engine. A plain rotating sleeve is unsatisfactory, since a reciprocating motion of some sort is essential to prevent scoring of the sleeve and cylinder wall. It is essential also that the whole of the inner surface of the sleeve shall, at every cycle, be scraped either by the piston or the fixed cylinder head,

in order to prevent the formation of shoulders due to wear or carbon deposits, which would prove fatal to their operation.

The use of two concentric and reciprocating sleeves has the advantage that their mechanical operation is somewhat simpler, but it is very difficult to see what further advantage they can possess. The chief fault of the sleeve-valve engine, namely, the difficulty of disposing of the heat from the piston, is greatly accentuated when

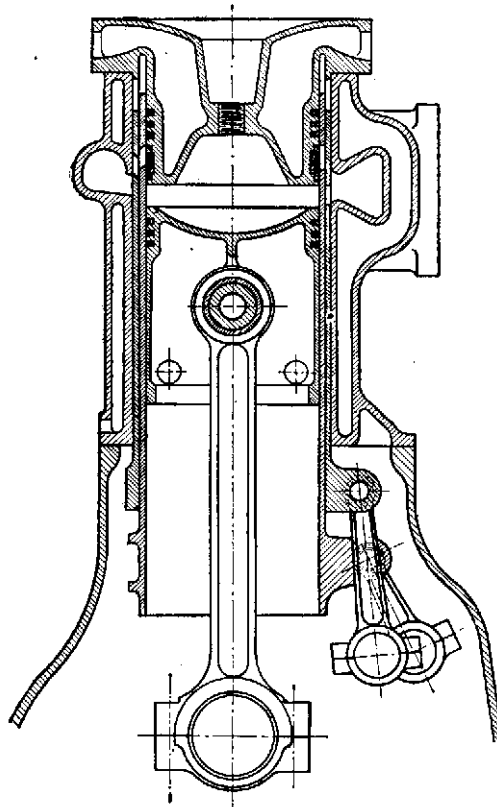


Fig. 155

two sleeves are used, as also the friction loss, which is no small item and a particularly objectionable one where the load factor is light, as in a motor-car engine.

In fig. 155 is shown diagrammatically the operation of a double-sleeve engine, from which it will be seen that the sleeves are actuated from a half-speed crankshaft connected by short rods to points at the side of each sleeve.

Figs. 156–159 show various alternative methods adopted by Burt for operating a single-sleeve valve. The method shown in fig. 156

is that used in the Picard Pictet cars and is, in the author's opinion, attractive from a mechanical point of view, but it is necessarily somewhat costly. In this arrangement two half-speed crank-shafts are employed and the sleeve is operated from the centre of

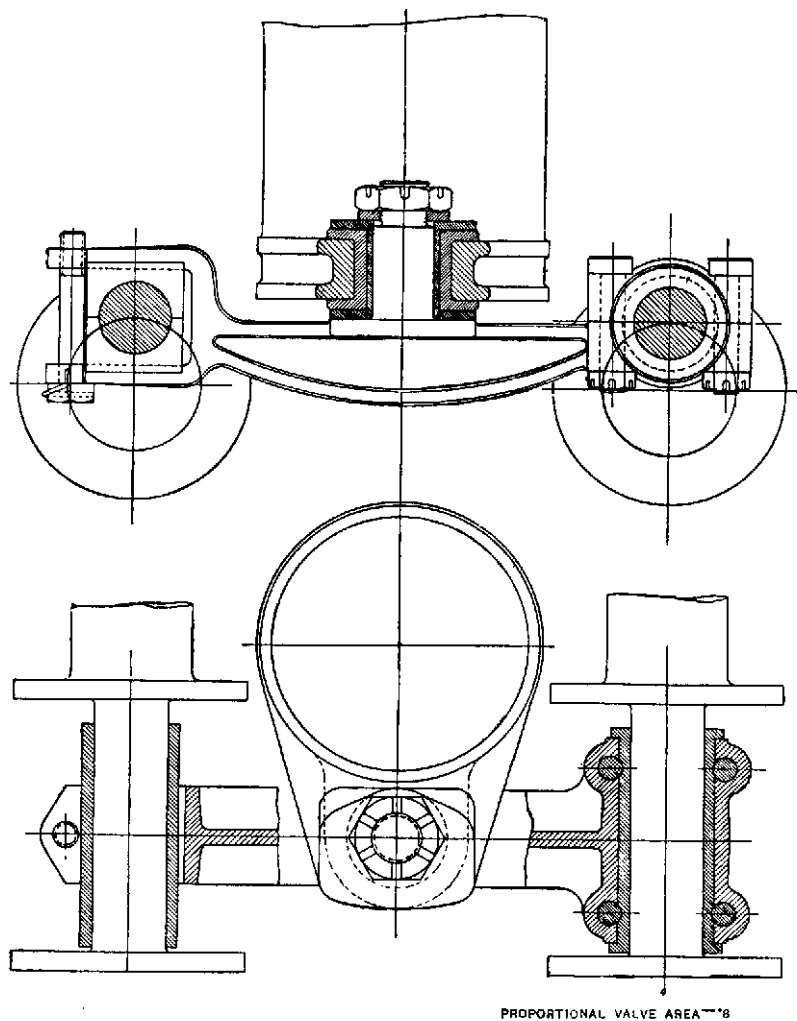


Fig. 156.

a coupling-rod connecting these shafts. It will be seen that one end of the coupling-rod is connected directly to one crankpin and the other to a second crankpin, but through the medium of a sliding block with a small amount of end play. The sliding block allows for any slight errors in synchronism as between the two half-speed shafts.

Fig. 157 shows a similar method of operation in which only one half-speed shaft is used. This is said to work well in practice, but is clearly inferior mechanically to that shown in fig. 156.

Another very attractive form is that shown in fig. 158, in which a ball-and-socket joint is used. This form has the advantage of being considerably lighter and more compact;

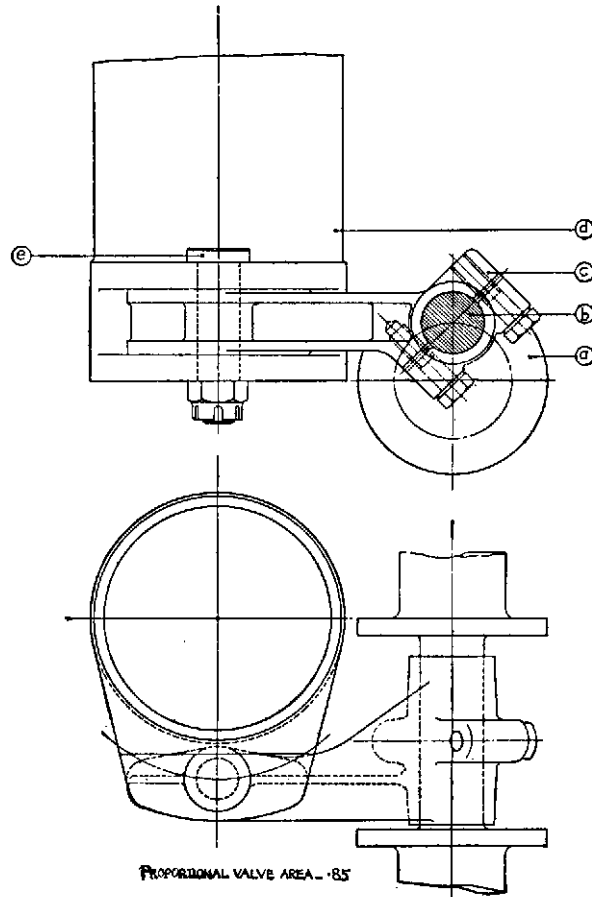


Fig. 157

also it is probably the least expensive and the most accessible. Fig. 159 shows an earlier form used in the Argyll cars, in which a reciprocating plunger is used in place of the ball socket. As in the forms 156, 157, and 159, this necessitates the use of a pin in the sleeve, and therefore both increases the radius of operation and the effective weight of the sleeve.

The author is greatly indebted to Mr. Burt for the following information and particulars as to the determination of port areas, &c., when a single sleeve is employed.

**Calculation of Ports.**—The special shape of port is adopted to give a maximum area of opening with the minimum of sleeve travel. Fig. 160 shows typical ports *a* being the ideal shape, *b* the same port

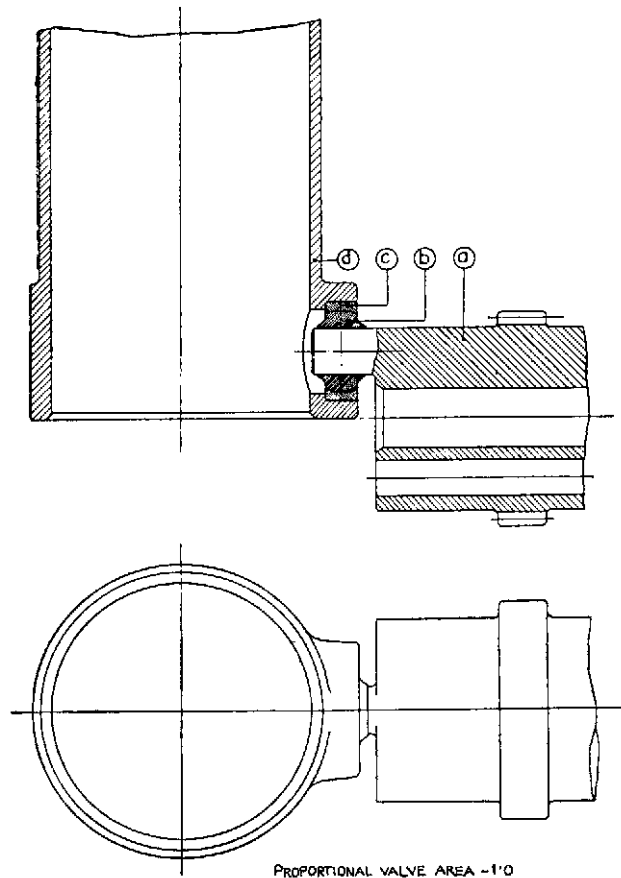


Fig. 168.

with corners rounded off to avoid interference when sleeve-valve port is passing down between two cylinder ports. The straight flank port *c* is usually adopted, as it is a better manufacturing proposition, although slightly smaller in area than type *b*, for a given valve-shaft stroke.

It is necessary to fix the following particulars before calculating single-sleeve valve ports :—

- A = Arrangement and number of ports.  
 D = Outside diameter of sleeve valve in inches.  
 C = Distance from axis of sleeve valve to axis of pivot-pin or ball-and-socket coupling in inches.  
 T = Throw of sleeve crank in inches.  
 V = Engine timing.

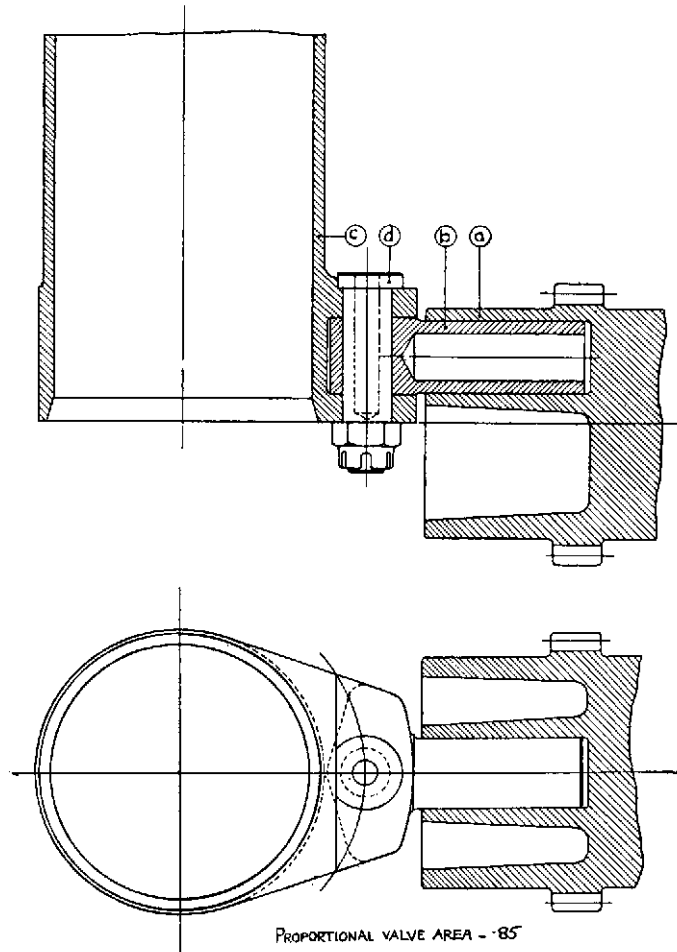


Fig. 159.

The greater the number of ports the smaller the sleeve-valve crankthrow for a given area of valve opening, thereby reducing the over-all dimensions of valve-driving mechanism and height of engine, but complicating the coring of water-passages in the cylinder castings and increasing the port cutting time. The fewer the number of ports for a similar area the greater the valve crankthrow and

over-all dimensions, but the coring is simplified with an attendant reduction of port cutting time.

The maximum inlet opening areas obtainable with various port

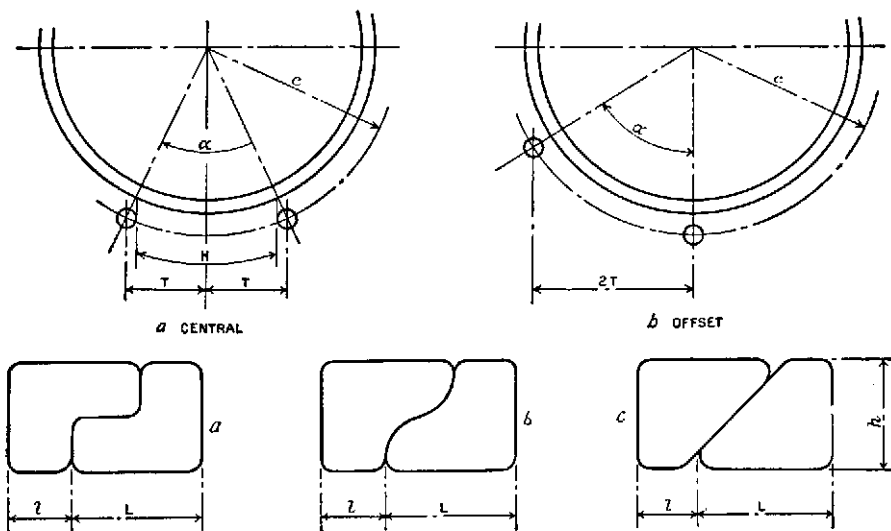


Fig. 160.—Angular Travel

settings are given in fig. 161, while fig. 162 illustrates in proportion several settings. It will be noticed that a “double purpose” port—that is, one which acts alternately as inlet and exhaust—is included in

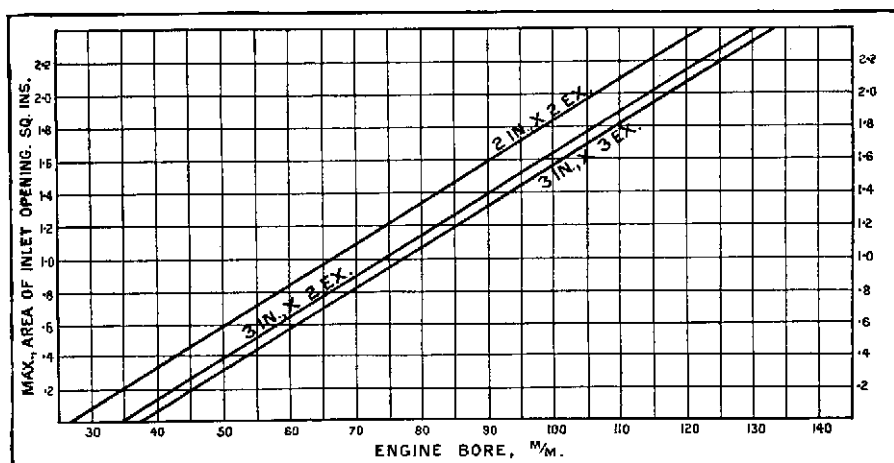


Fig. 161.—Maximum Inlet Valve Opening. Various Port Settings

each setting. This is advisable where maximum openings are desired; two single ports with a wall between would obviously use up more of the sleeve-valve circumference than a single “double purpose” port.

The sleeve valve is usually centrifugally cast of good quality grey iron, and for engines up to  $2\frac{3}{4}$ -in. diameter bore is made  $\cdot 1$  in. thick, while for engines of  $4\frac{1}{4}$ -in. diameter bore  $\cdot 125$  in. thick is quite satisfactory.  $D$  can readily be obtained by adding twice the sleeve-valve wall thickness to the cylinder bore diameter, while  $C$ , which should be kept as small as possible, is generally  $\cdot 575D$  when the ball-and-socket type of drive is used.

The throw of sleeve crank  $T$  is obtained from the number of ports in the cylinder, as given by the setting adopted, and the dimensions  $D$  and  $C$ .

$$T = \frac{\pi D \times \cdot 575 D}{D(2 \text{ No. of ports}) - 1} = \frac{1 \cdot 8 D}{(2 \text{ No. of ports}) - 1}$$

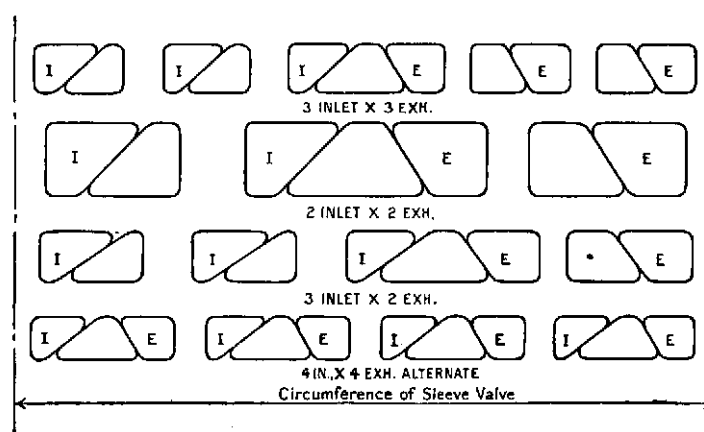


Fig. 162.—Various Port Settings

This gives the maximum throw possible with this type of actuation and may be reduced within limits according to the type of engine under consideration. Maximum throws for various settings and engine bore diameters are given graphically in fig. 163.

The valve timing  $V$  has now to be settled, and in common with poppet valves it varies according to the type of engine.

The actual calculation of ports will be best understood by taking an example. Assume a high-speed engine of 68-mm. bore, the desired maximum opening area of the inlet ports being 1.0 sq. in. approximately, so that we have

$A = 2 \text{ inlet} \times 2 \text{ exhaust}$ , for it will be seen from fig. 161 that this setting gives the required area.

$D = 68 \text{ mm.} + \cdot 2 \text{ in.} = \text{say } 2 \cdot 9 \text{-in. diameter.}$

$C = \cdot 575 \times 2 \cdot 9 = 1 \cdot 66 \text{ in., say } 1 \cdot 65 \text{ in.}$



$$T = \frac{1.8 \times 2.9}{(2 \times 4) - 1} = .74 \text{ in., say } .7 \text{ in.}$$

$\alpha$  = Angular travel of sleeve valve (see fig. 160).

$$\sin \frac{1}{2}\alpha = \frac{T}{C} \text{ when } T \text{ is central} = \frac{.7}{1.65} = .42424 \text{ or } 25^\circ 6';$$

$$\therefore = 25^\circ 6' \times 2 = 50^\circ 12'.$$

$$\text{When } T \text{ is offset } \sin \alpha = \frac{2T}{C}.$$

Referring to fig. 164:

$$H = \text{Horizontal travel} = \frac{\alpha \pi D}{360} = \frac{50^\circ 12' \times 3.1416 \times 2.9}{360} = 1.275''.$$

$$L = \text{Length of port} = H - \text{cover} = 1.275'' - .05'' = 1.225''.$$

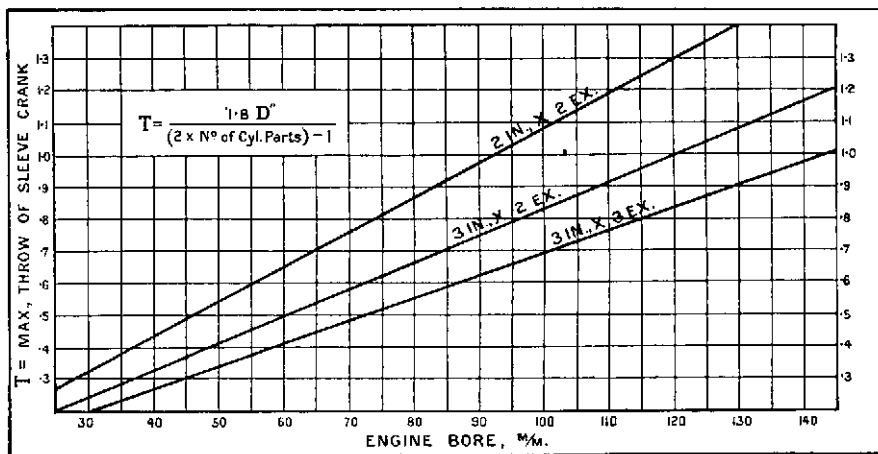


Fig. 163.—Maximum Sleeve-valve Crankthrow. Various Port Settings

Minimum cover = .04". This is usually arranged so as to bring  $L$  to an even figure.

$W$  = Minimum space between ports (this does not apply to alternate ports as shown in setting  $d$ , fig. 162) =  $H + \text{cover}$   
 $= 1.275'' + .05'' = 1.325''.$

$$h = \text{Height of ports} = T + (T \sin \beta) = .7 + (.7 \sin 22\frac{1}{2}^\circ) = .97''.$$

$$4 \text{ ports at } 1.225'' \text{ long} = 4.9''$$

$$2 \text{ spaces at } 1.325'' = 2.65''$$

$$2 \text{ spaces at } .775'' = 1.55''$$

Circumference of sleeve valve = 9.1".

$$li = \text{Inlet port tail} = \frac{H}{2} - \left( T \sin \delta \frac{.5D}{C} \right),$$

$$= .6375 - \left( .7 \sin 7\frac{1}{2}^\circ \times \frac{1.45}{1.65} \right) = .6375 - .08 = .5575''.$$

$$l_e = \text{Exh. port tail} = \frac{H}{2} + \left( T \sin \nu \frac{.5D}{C} \right)$$

$$= .6375 + \left( .7 \sin 15^\circ \times \frac{1.45}{1.65} \right) = .6375, .1595 = .797".$$

With straight-sided ports the flank angle can be solved as follows (see fig. 164):—

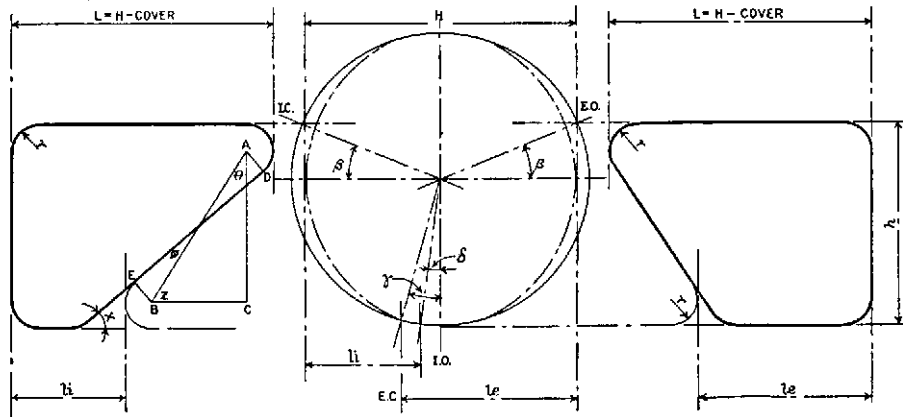


Fig. 164.

$$AC = h - 2r, \quad \tan \theta = \frac{BC}{AC}$$

$$r = \text{corner radius usually } 1/8", \quad AB = \frac{BC}{\sin \theta}$$

$$AD \text{ and } EB = r, \quad \sin \phi = \frac{r}{\frac{AB}{2}}$$

$$BC = L - (l + 2r), \quad z = 90^\circ - \phi.$$

$$\text{Flank angle} = X = z - \phi.$$

In fig. 165 the crankshaft timing diagram for the example worked

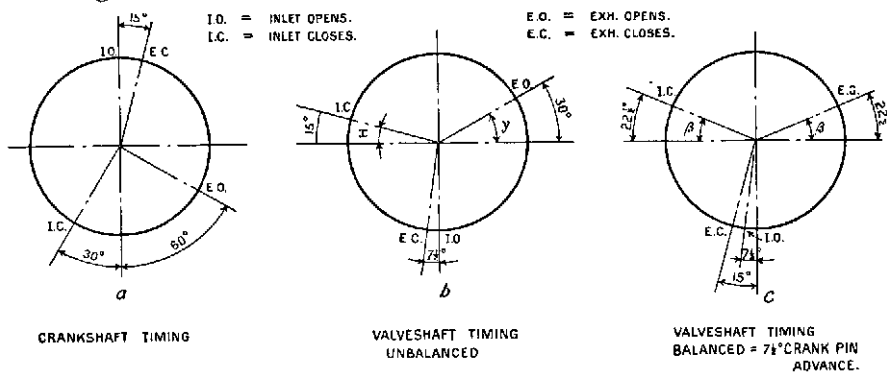


Fig. 165.—Timing Diagrams

is shown at *a*, while the same timing transferred to the valveshaft is indicated at *b*, the crankshaft being at TDC, while the valveshaft crankpin is at BDC. It is obvious that with this valveshaft setting

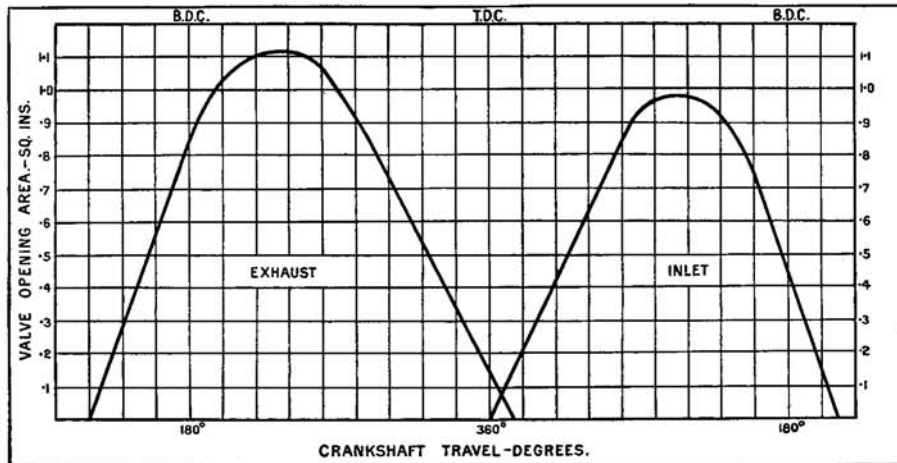


Fig. 166.—Valve-opening Diagram. 68-mm. Bore Engine ;  $C=1.65"$ ,  $T=.7"$

a relatively large opening to exhaust would be obtained owing to the greater height of exhaust port. To overcome this, and in order to make the machining of ports a simpler operation, inlet and exhaust

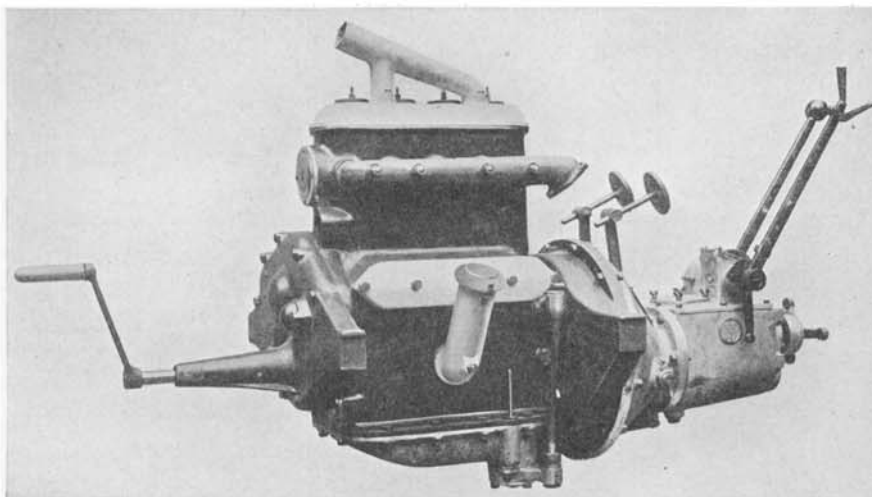


Fig. 167.—Four-cylinder Burt Single-sleeve Valve Engine

ports are made of equal height. This is made possible by the setting of valveshaft crankpin in advance of its BDC and in relation to the crankshaft until the angles  $x$  and  $y$  are equal. The amount of offset

is found by  $\frac{x+y}{4}$ , and the result for the example taken is shown in fig. 165 at *c*; the corresponding port-opening diagram being as shown in fig. 166.

Fig. 167 shows an example of a small four-cylinder motor-car engine of 68-mm. bore and 103 mm. in which single-sleeve valves actuated by the ball-and-socket mechanism are employed; while

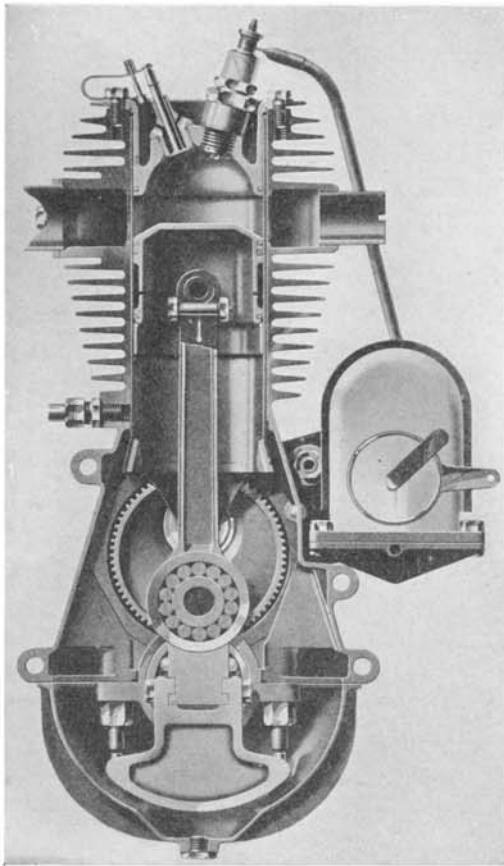


Fig. 168

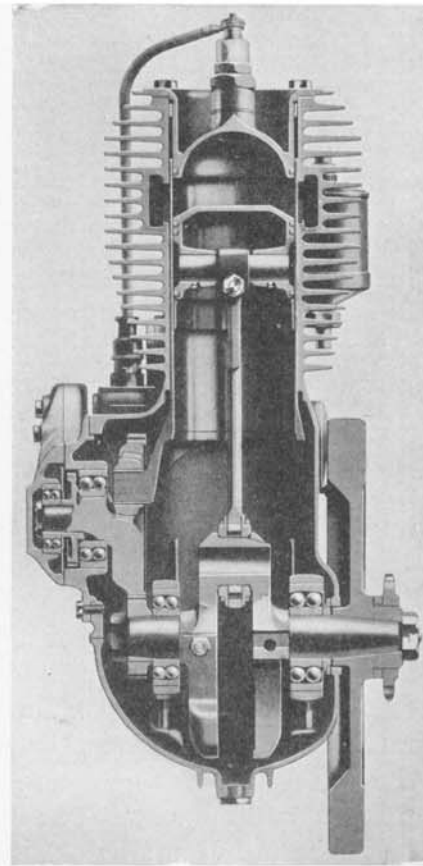


Fig. 169

Barr & Stroud Engine with Burt Single Valve

figs. 168 and 169 show a remarkably neat design of motor-cycle engine built by Messrs. Barr & Stroud, and incorporating the same features.

**Racing Cars.**—The practice of motor-car and cycle racing has been one of the most valuable stimulants to the design of efficient internal-combustion engines, for the racing engine operates under conditions of severity such as are met with in no other field, with

the result that weaknesses which would develop in the course of years under ordinary conditions of service are shown up in as many minutes. The rapid progress which the high-speed internal-combustion engine has made during recent years is due to the stimulus of motor racing, and that to an extent which few people fully realize.

In the production of a racing engine the designer has full liberty to employ every means known to him to obtain the highest possible power output regardless of any other consideration, except that during more recent years it has become the practice to limit the cylinder capacity of racing engines, a restriction which has proved of undoubted benefit.

There is a popular impression that because the racing engine no longer resembles the actual article used in touring cars its value, from an educational point of view, has been lost; this, however, is a sheer fallacy: the racing engine operates on the same cycle and under the same conditions, except that they are much more severe, as the ordinary touring-car engine, and the lessons learnt from its behaviour are just as applicable to the intelligent designer as though the engines were identical.

From an educational point of view it is probably desirable that the racing engine should differ from the touring model, for by its difference—

(1) Higher speeds and therefore more strenuous conditions of test are obtained.

(2) The racing engine of to-day is providing lessons for the future also, and not only for the immediate present.

Again, there is a popular but wholly mistaken belief that the racing motor-car engine, though powerful, is not “efficient,” and that since fuel economy does not enter into consideration such engines teach us nothing about this important question. For an engine to be powerful it must be efficient in every respect—that is to say, it must convert the highest possible percentage of the heat energy available from the combustion of every pound of air into useful work at the flywheel. If, as may sometimes be the case, the air is super-saturated with fuel, that is the carburettor’s not the engine’s fault, for with good carburation the racing engine will show the highest possible thermal efficiency reckoned on the fuel also.

The engine illustrated in figs. 170–177 is one of several constructed by Messrs. Vauxhall Motors Ltd. for their racing cars for the 1922 season. It is of three-litres capacity, and develops, the author believes,

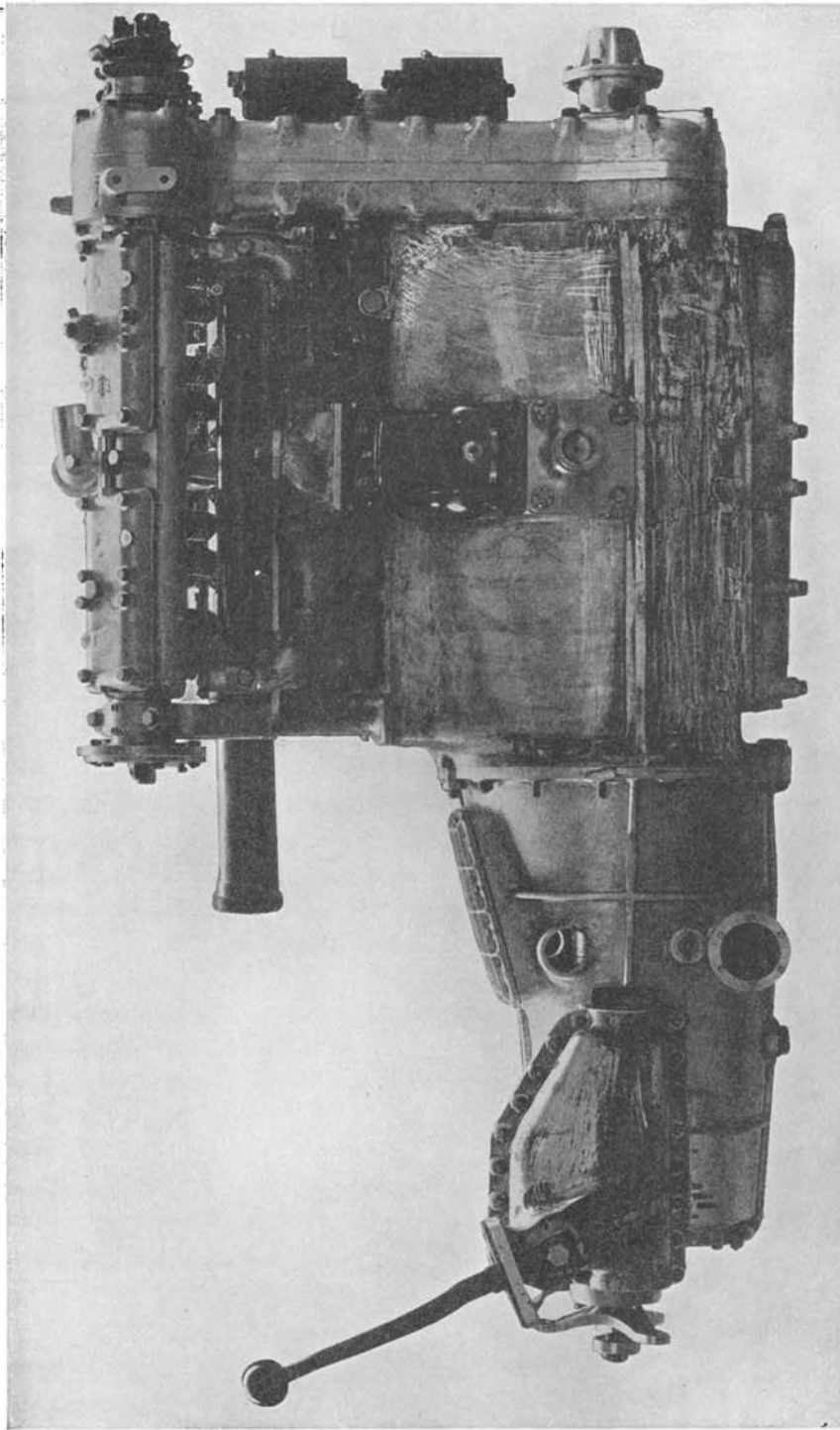


Fig. 170.—Vauxhall Three-litre Racing Engine

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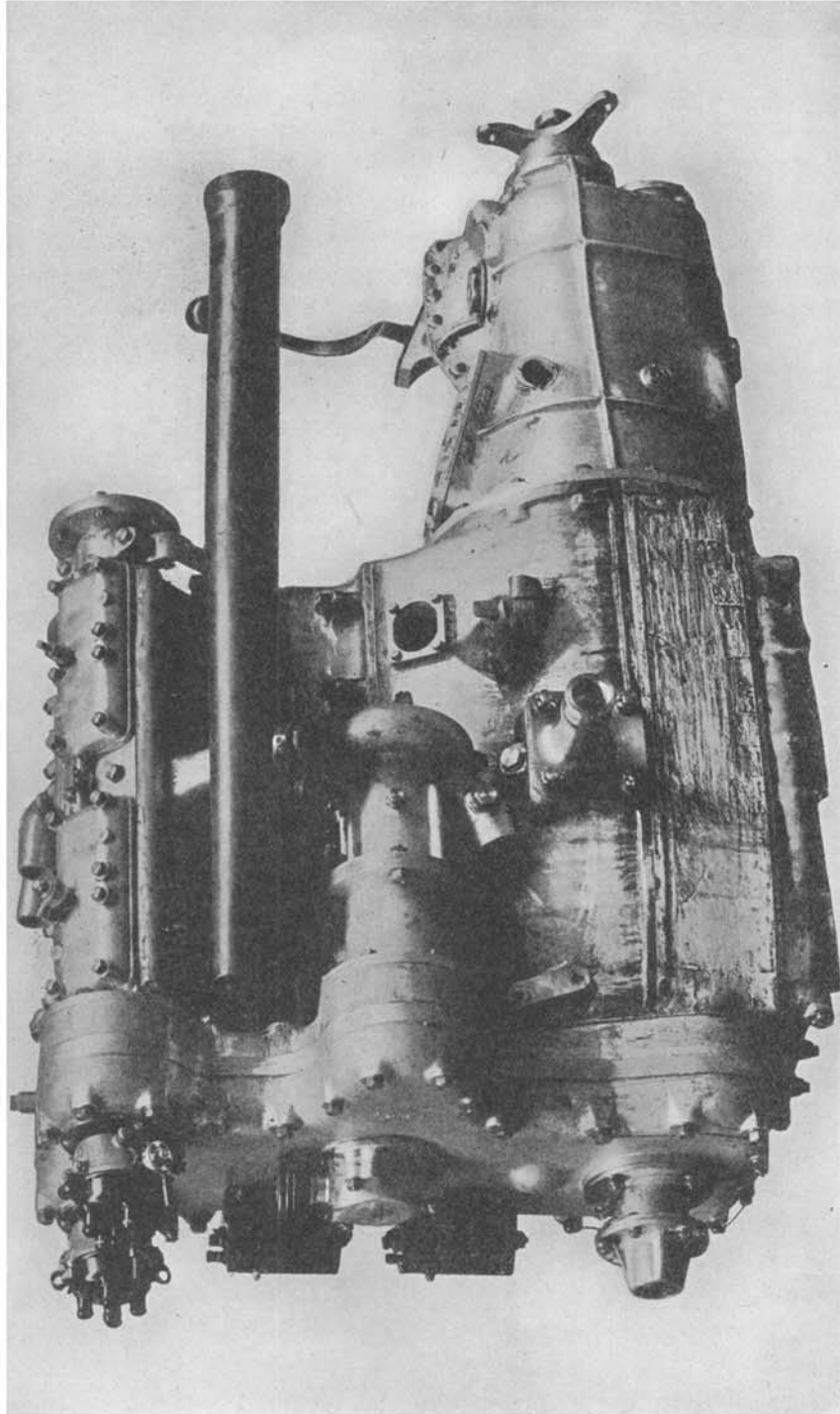


Fig. 171.—Vauxhall Three-litre Racing Engine

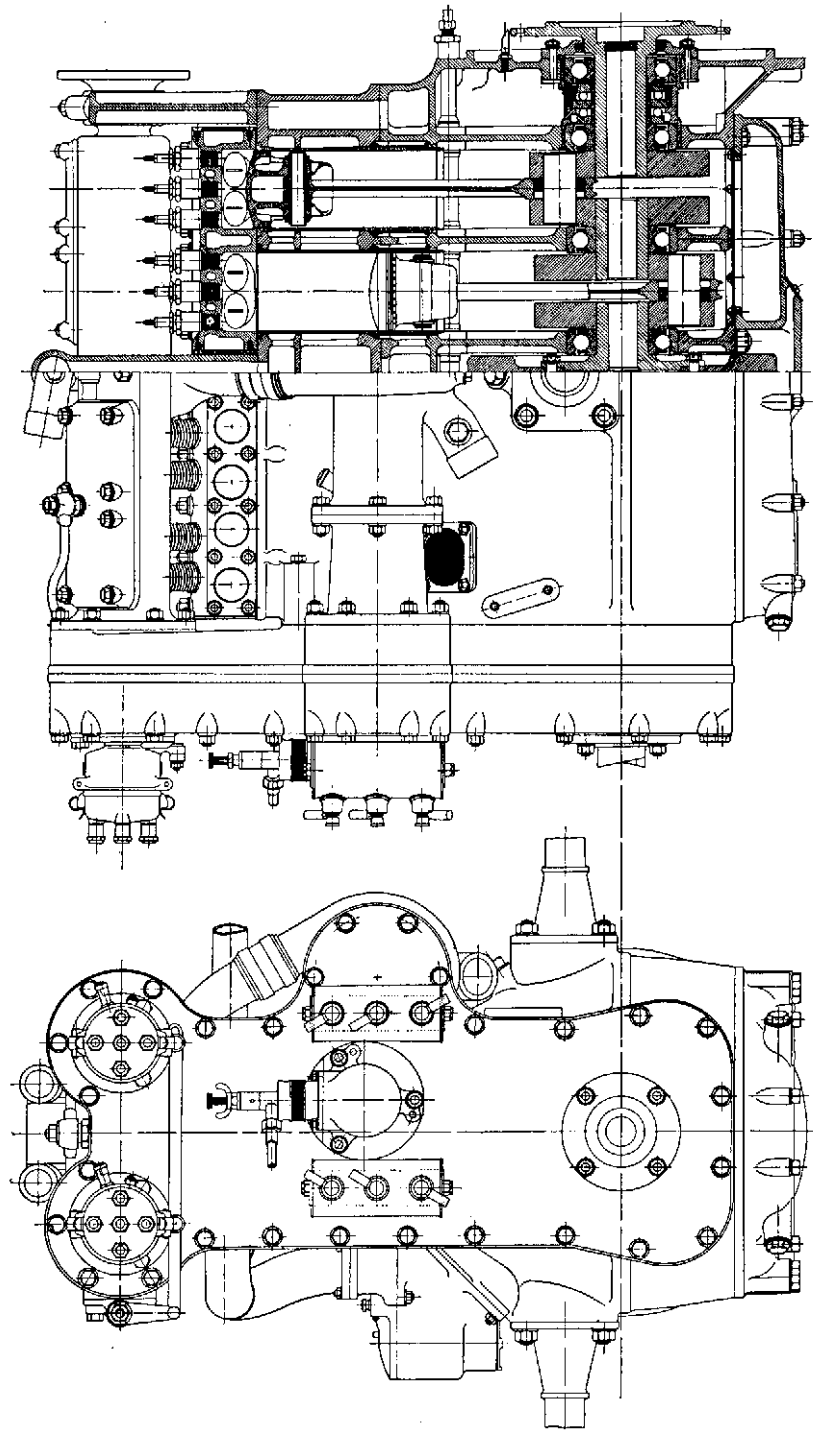


Fig. 172.—Vauxhall Three-litre Racing Engine



the highest power output ever yet obtained from an engine of this size.  
The main features aimed at in the design of this engine were :

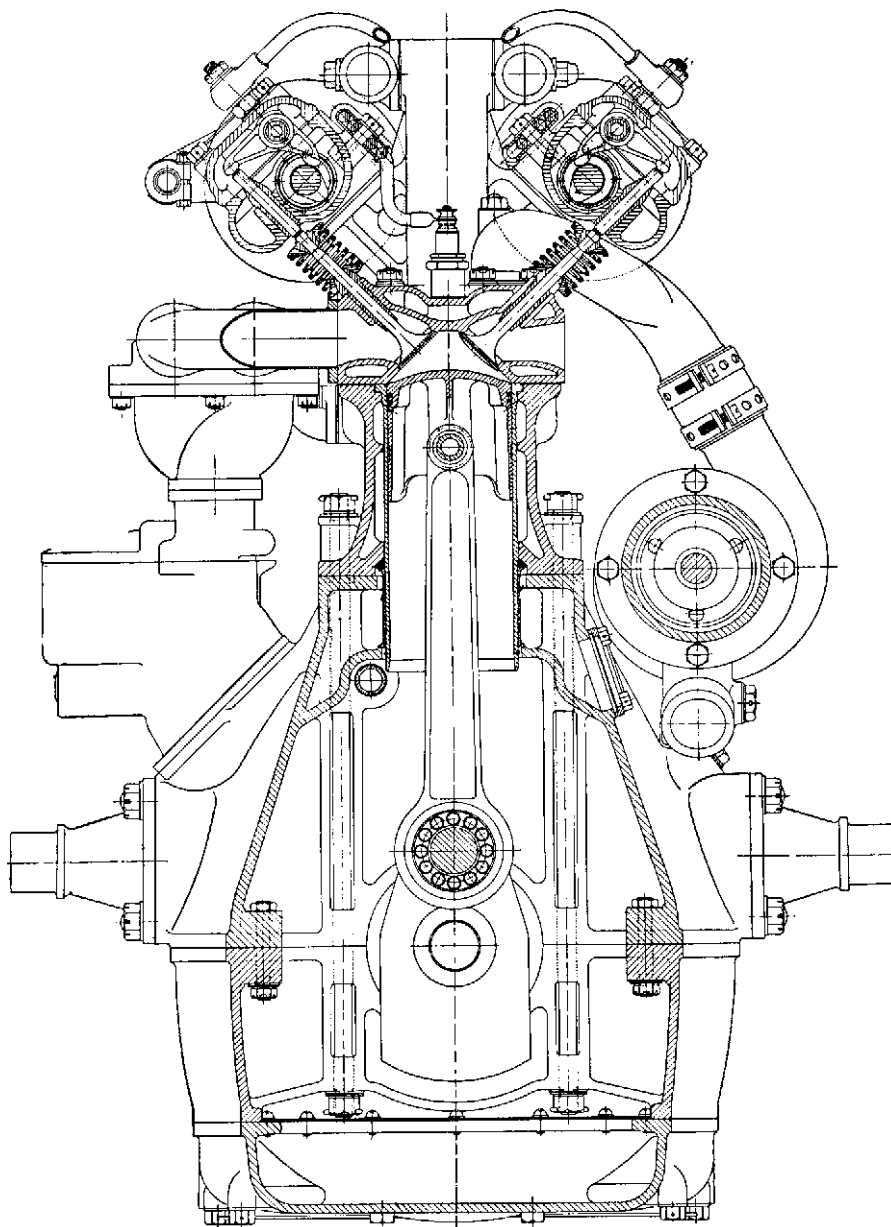


Fig. 173.—Vauxhall Three-litre Racing Engine

(1) To obtain the maximum possible thermal efficiency, with a view to getting the utmost possible power output from the available air.

- (2) To ensure the maximum of structural rigidity.
- (3) To avoid crankshaft torsional vibration at any speed of which the engine was capable.
- (4) To obtain the highest possible mechanical efficiency.
- (5) To obtain a high volumetric efficiency.
- (6) To provide a form of connecting-rod big-end bearing which

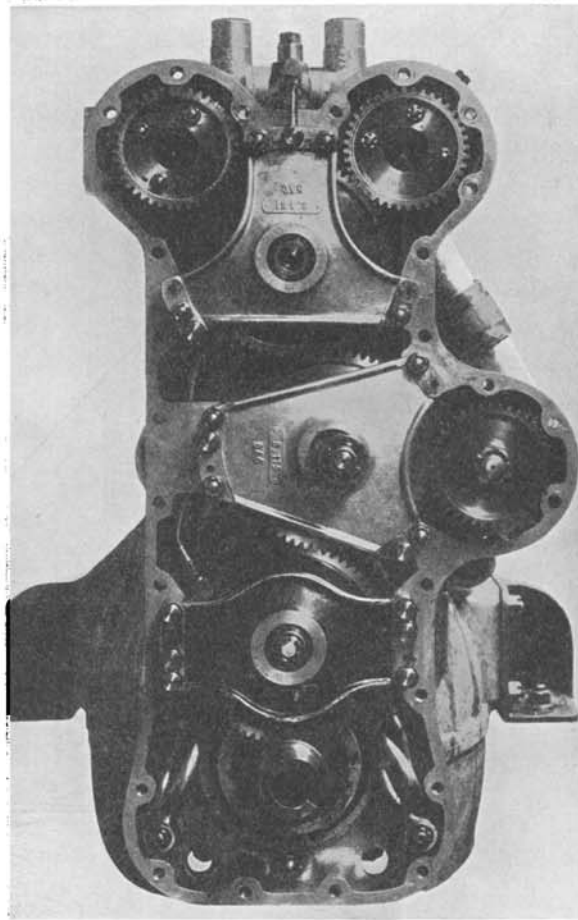


Fig. 174.—Vauxhall Three-litre Racing Engine : Auxiliary Drive

should be capable of withstanding continuous running at a mean speed of well over 4000 R.P.M.

The steps taken to meet these conditions were :—

- (1) In order to obtain the highest possible thermal efficiency the combustion chamber was made of the shallow pent-roof type with the sparking fitted centrally in the cylinder head. The maximum

distance from the sparking plug points to the farthest point in the combustion chamber is only 1.9 inches. In addition to the central plug, provision was made for the fitting of two other plugs, one on either side, to be operated synchronously from a single low-tension contact breaker. These additional plugs were intended rather as a standby in case of failure of the central plug; they were, in fact, never used.

(2) In order to ensure the maximum of structural rigidity the crankcase is made as deep as possible and of a barrel shape, with the maximum cross-section at the centre; the cylinder water-jackets were cast in one piece and rigidly attached to form an additional

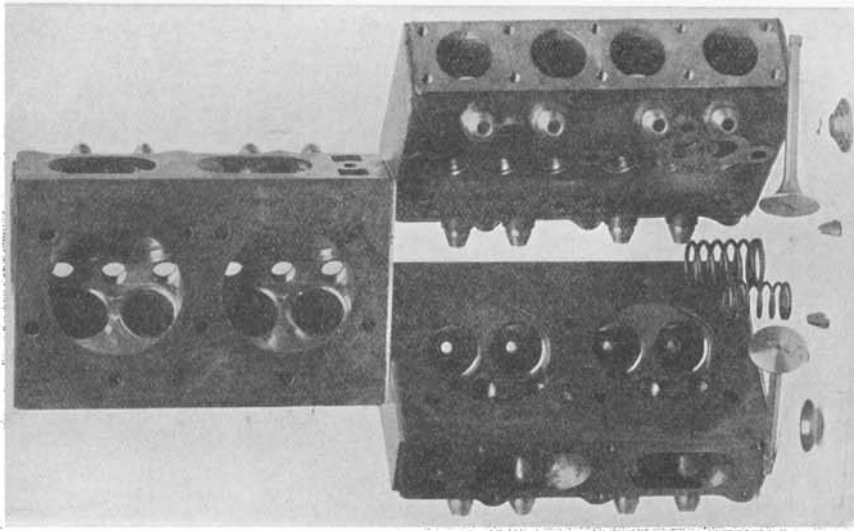


Fig. 175.—Vauxhall Three-litre Racing Engine: Cylinder Heads and Valves

girder, while through bolts extend from the cylinder block to the very bottom of the crankcase, thus forming a structure of extreme rigidity both as regards torsion and bending.

(3) With a view to eliminating torsional vibration, the flywheel is mounted in the centre of the crankshaft so that the maximum length subject to torsion is reduced to about 8 inches. The shaft is, in fact, made in the form of two entirely separate two-throw cranks, each provided with flanges between which the flywheel is bolted. This arrangement, although very unorthodox, proved most successful, no trace of torsional vibration being observed at any speed at which the engine could be run.

(4) With a view to obtaining the highest possible mechanical

efficiency, pistons of the slipper type are employed and the cylinder liners are maintained at a high temperature in order to lower the viscosity of the lubricant adhering to them. This latter is accom-

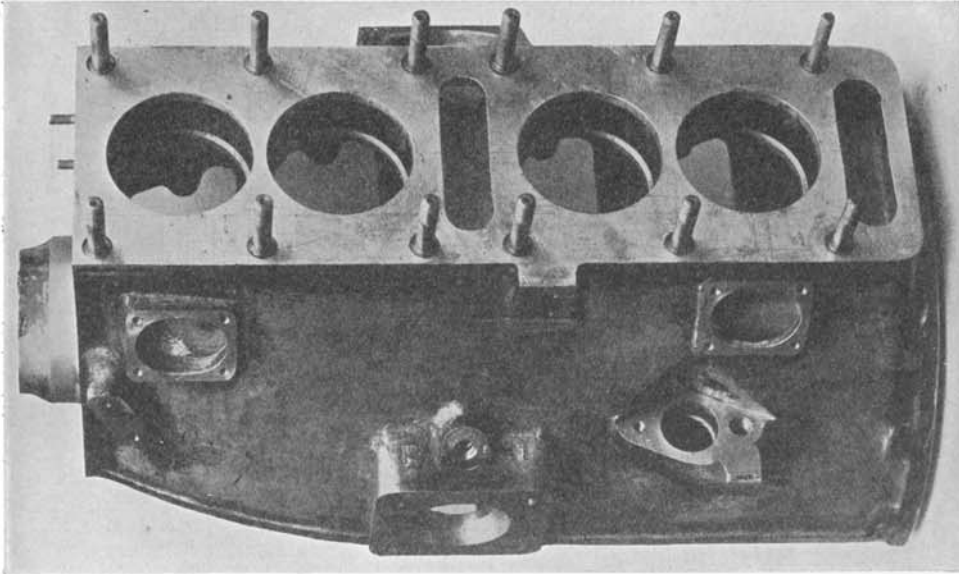


Fig. 176.—Vauxhall Three-litre Racing Engine : Crankcase

plished by isolating the lower part of the liners from the main water circulation, so that the cooling water surrounding them is left practically stagnant. For the rest, the use of ball and roller bearings wherever possible contributed to reducing the friction losses to the

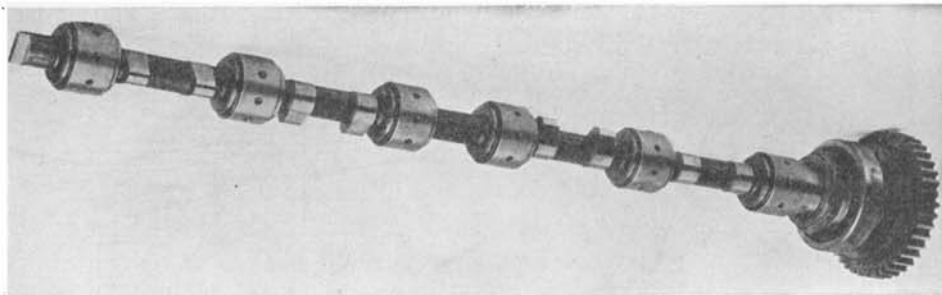


Fig. 177.—Vauxhall Three-litre Racing Engine : one of the Camshafts

lowest possible limit and at the same time obviated the necessity for any form of oil cooling—always a troublesome problem.

(5) In order to obtain a high volumetric efficiency the induction system was divided so as to avoid any overlapping of the suction strokes. The central pair of cylinders were fed from one carburettor

and the outer pair from a second and entirely independent carburettor. In this manner use could be made of the kinetic energy of the gases flowing in the induction pipes thoroughly to fill the cylinders without any risk of one cylinder robbing another, as must inevitably occur when all four cylinders draw from any one induction manifold, owing to the overlapping of the opening period of the inlet valves.

The engine has four cylinders, each of 85 mm. bore and 132 mm. stroke, and was designed with a view to running continuously at from 4000 to 4500 R.P.M. with short periods up to 5000 R.P.M. In order both to provide structural rigidity and to enclose the central flywheel the crank chamber is of barrel shape and is mounted in the chassis by trunnions attached to the sides. The cylinder block consists of an aluminium casting forming the water-jacket into which loose steel liners are fitted, with rubber rings to ensure water-tightness. The cylinder heads are cast in pairs in hard bronze and call for no particular comment. The valves, of which there are four in number to each cylinder, are comparatively small with a high lift, and the valve gear generally is designed to operate at a speed of 5000 R.P.M. The inlet valves are heavily masked, and by this means it is possible to employ a comparatively low rate of acceleration and therefore to use very light and lightly stressed valve springs. The two camshafts are carried in aluminium housings supported from the main cylinder block at their centre and at either end. They run in plain bearings with cast-iron floating bushes. The cams themselves are of very small diameter in order to reduce to the minimum the rubbing velocity; they are of the plain tangent flank form. The cam followers are of the plain curved slipper type, with a short straight push rod interposed between the follower and the valve itself. The camshafts are driven by means of a chain of spur gears, the intermediate pinions of which are carried in separate spider housings to permit of the meshing being correctly adjusted.

(6) Experience with bearings, both in actual engines and under separate tests, had shown that, even under the most favourable conditions as to lubrication, plain white-metal lined bearings could not be relied upon for the connecting-rod big-ends, because no matter how much oil be circulated through the bearings, nor how thoroughly it be cooled, there was little hope of getting rid of the heat generated by friction at a rate sufficient to keep the temperature of the bearing material within safe limits. Further, in order to reduce vibration and ensure structural rigidity, it was essential

to keep the cylinder centres as close together as possible, and this limited the permissible width of bearing surface both on the crankpins and main crankshaft journals.

In view of these considerations it was apparent that if a continuous mean speed of over 4000 R.P.M. was to be maintained some form of crankpin bearing other than a plain white-metal lining would have to be employed. The choice lay between (a) the use of a floating bush between the crankpin and connecting-rod, under forced lubrication, and (b) a roller bearing. Both these would necessitate the use of some form of built-up crankshaft with case-hardened crankpins, since neither the floating bush nor the roller race could be split, nor could the weight of a split big-end bearing be tolerated. Of the two alternatives the floating bush would require continuous lubrication under pressure, while the roller bearing could be used with splash lubrication. Since for the same and other additional reasons, it was essential to employ ball or roller bearings for the main journals, the provision of continuous lubrication under pressure to the crankpins became a very difficult problem, and it was decided therefore to adopt the second alternative and employ roller bearings. The method of building up the crankshaft was another problem, and after much consideration it was decided to employ a completely built-up crank, consisting of plain parallel pins on to which the crank-cheeks were shrunk as in marine and large gas-engine practice.

The crankshafts throughout were made from plain mild steel with the pins case-hardened. For the connecting-rod big-end bearing, it was decided to use a double row of short rollers located in a one-piece bronze cage, while the hardened eye of the connecting-rod itself formed the outer race. This is further stiffened by means of two circular webs. Like the crankshafts, the connecting-rods are of plain low-carbon case-hardened steel.

**Lubrication.**—Two oil pumps of the oscillating valveless plunger type are provided, both of which are operated from one of the idle wheels of the gear train. One pump draws oil from the oil sump and delivers it to an oil gallery running the full length of the crankcase, provided with four jets playing oil on to each of the crankthrows. The second pump delivers oil under a pressure of about 25 lb. per square inch to the camshafts, the oil being distributed through the hollow fulcrum pins of the valve rockers; from the camshaft casings, the oil drains back by gravity to the crankcase.

**Cooling.**—The cooling water is circulated by means of a centri-

fugal pump running at one-half engine speed. From the pump the water passes around the upper end of the cylinder liners, the lower parts of which are partitioned off in order to maintain the water more or less stagnant, and so to permit of the bearing portion of the liners attaining rapidly a fairly high temperature. From the upper deck of the cylinder block the water passes to the cylinder heads in parallel, and so returns to the radiator. In order both to cool the crankcase and slightly to heat the air on its way to the carburettors, the air supply to the engine is drawn through the upper part of the crankcase and around the exposed lower ends of the cylinder liners.

The compression ratio used is 5·8 : 1. It had originally been proposed to employ a much higher compression ratio, and to run on a special fuel mixture, but owing to the difficulty of providing an efficient form of combustion chamber with any higher compression ratio, and at the same time avoid any risk of the pistons striking the valves should these accidentally stick in the full open position, it was considered safer to employ a lower ratio, at which, owing to the short-flame travel from the sparking plug, ordinary good quality petrol can be used without detonation.

In general, though the engine is designed throughout to run at very high speeds, it contains no extremes either of design or material. The whole of the crankshaft, the connecting-rods, and the gudgeon pins are of straight low-carbon mild steel. Neither the connecting-rods nor the pistons are particularly light. The cams are of the plain tangent flank form, free from any concave surfaces, the acceleration of the valve gear is low, and the valve springs are very lightly stressed. In short, the engines were designed throughout with a view to reliability both in manufacture and in running, and an ample margin of safety provided for.

The performance of one of these engines which underwent prolonged testing on the test-bench is shown in fig. 178, from which it will be seen that a maximum of 129 B.H.P. is reached at a speed of 4500 R.P.M., the brake and indicated mean pressures at this speed being 124 and 159 lb. per square inch respectively, and the mechanical efficiency 78 per cent. It will be observed also that the highest indicated mean pressure was obtained at a speed of about 3700 R.P.M., showing that the combination of induction pipe design and valve setting was such as to give maximum over-all efficiency at this speed. With a combustion efficiency of 34·75 per cent this corresponds to a volumetric efficiency at N.T.P. of 80·3 per cent, a figure which is in very close agreement with that obtained from the author's

variable compression engine under similar conditions as to temperature, and at a speed of 1750 R.P.M., which is the most efficient speed for this particular engine. This figure also is in very close agreement with readings taken of the compression pressure when motoring, which at 4000 R.P.M. was found to be 139 lb. per square inch, indicating that the cylinders were filled up to very nearly full atmospheric pressure at this speed. The mechanical efficiency was arrived at from a very large number of tests by Morse's method on pairs of cylinders separately, on individual cylinders, and by motoring tests. The three methods showed exceptionally close agreement over the whole range of speed.

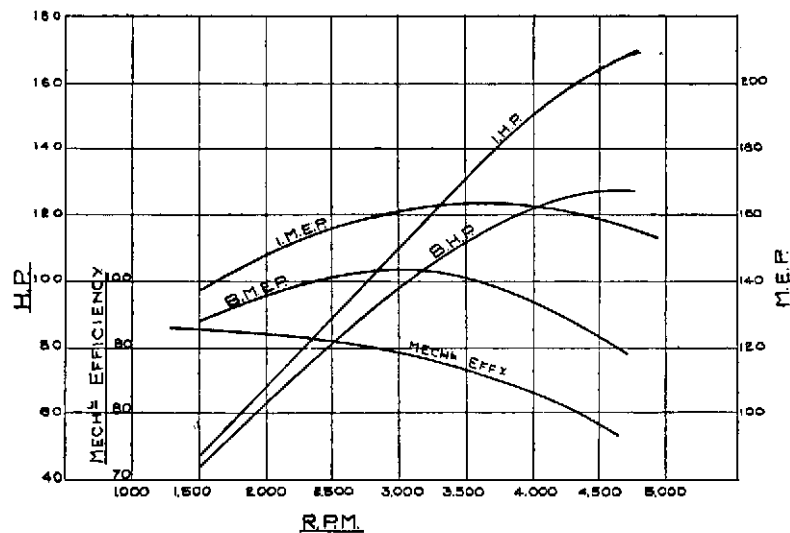


Fig. 178.—Vauxhall Three-litre Racing Engine

The performance of the engines is almost exactly what might be anticipated from an analysis of the general design based on the data given in the preceding chapters, and, since it conforms so closely, it may be of interest and perhaps of some use to engine designers to recapitulate such data in so far as it applies to these particular engines. It may also, the author hopes, help to dispel the still prevalent superstition that some mystery enshrouds the performance of racing motor-car engines, whose behaviour is, in fact, perfectly normal in every respect.

The leading dimensions of these engines are as follows :—

Bore	...	...	...	...	...	...	...	85 mm. = 3.34 inches.
Stroke	...	...	...	...	...	...	...	132 mm. = 5.2 "
Compression	...	...	...	...	...	...	...	5.8 : 1.



Area of piston	...	...	...	...	...	8.75 square inches.
Swept volume of cylinder	...	...	...	...	...	45.5 cubic inches.
Weight of reciprocating parts (per cylinder)	...	...	...	...	...	1.7 lb.
"	"	"	(per sq. in. of piston area)	...	...	0.195 lb. per sq. in.
Number of valves	...	...	...	...	...	4—2 inlet, 2 exhaust.
Diameter of valve ports (inlet)	...	...	...	...	...	=1.34 inches.
"	"	"	(exhaust)	...	...	=1.30 "
Lift of all valves	...	...	...	...	...	=0.354 "
Effective area through inlet valves	...	...	...	...	...	=2.55 sq. in.
(Inlet valves masked 0.050 inches of travel.)						
Ratio piston area to effective inlet port area	...	...	...	...	...	=3.44 : 1.

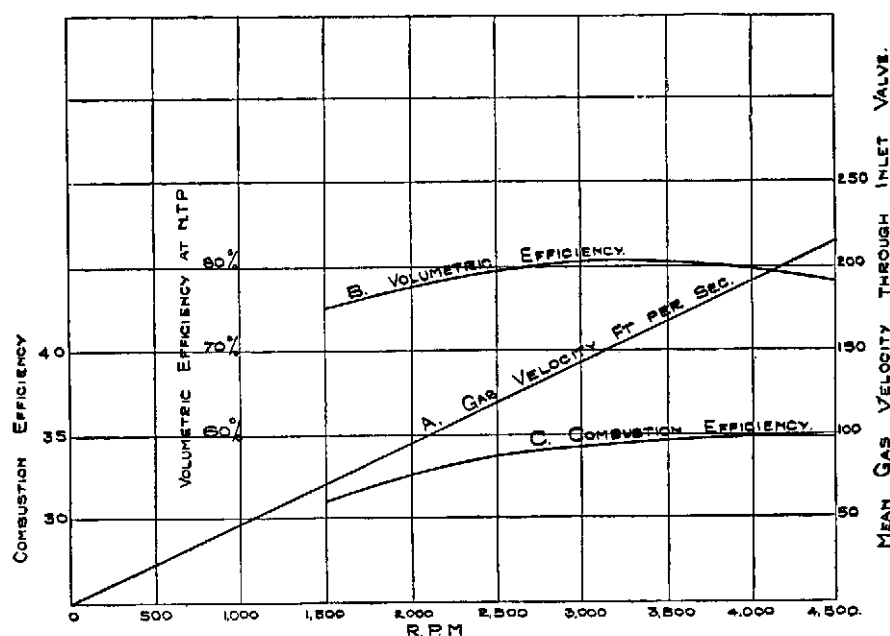


Fig. 179.—Vauxhall Three-litre Racing Engine

The chart, fig. 179, curve A, shows the mean gas velocity through the inlet valves at speeds varying up to 4500 R.P.M., and curve B shows the volumetric efficiency which corresponds with these mean gas velocities allowing for the latent heat of evaporation of the fuel, assuming that the minimum of preheating were used and that there were no undue wiredrawing in the induction pipe or carburettors, both of which conditions apply in this case. Curve B is arrived at by calculations such as those given in Chapter II, by deduction from various test results, and finally by direct air measurements taken on various engines with similar combustion chambers and similar gas velocities through the inlet valves. The falling off

in volumetric efficiency at the lower speeds is due merely to the late closing of the inlet valves, which results in the rejection of some of the combustible mixture during the early part of the compression stroke; while the falling off again at very high speeds is due to wiredrawing or insufficient valve area. With a combustion chamber of the form used giving the maximum of turbulence, and with central ignition, the efficiency reckoned on the air consumption will, at high speeds, be approximately 69 per cent of the air-cycle efficiency for the compression ratio (*vide* Chapter IV).

The air-cycle efficiency corresponding to a compression ratio of 5.8:1 is 50.5 per cent, so that the combustion efficiency will be  $\frac{69}{100} \times \frac{50.5}{100} = 34.8$  per cent at the highest speeds, when turbulence is at a maximum and direct heat loss at a minimum. As the speed is reduced turbulence becomes less, owing to the lower entering gas velocity, and the direct heat loss, though of comparatively small consequence, will also increase, with the result that combustion efficiency may be expected to vary approximately as the curve C, fig. 179. The values given in this curve are taken from those found on the author's variable compression engine under almost exactly similar conditions.

From the curves B and C and the known heat of combustion per standard cubic inch of mixture, as given in Chapters I and II, the indicated mean effective pressure can be arrived at directly by multiplying together the volumetric efficiency, the heat of combustion per standard cubic inch (47.8 ft.-lb. in this instance)  $\times 12 \times$  the combustion efficiency; thus at 3000 R.P.M. the indicated mean pressure will be  $0.807 \times 47.8 \times 12 \times 0.343 = 159$  lb. per square inch. Similarly at speeds from 1500 to 4500 we find that the theoretical indicated mean effective pressure should be—

R.P.M.	Volumetric Efficiency per cent.	Combustion Efficiency per cent.	Indicated mean Pressure lb. per sq. in.	Observed I.M.E.P. lb. per sq. in.
1500	75.0	31.0	133.4	138.5
2000	77.6	32.6	145.1	148.0
2500	79.7	33.9	155.0	156.0
3000	80.7	34.3	159.0	161.0
3500	80.7	34.6	160.4	162.5
4000	79.9	34.7	159.2	162.2
4500	78.5	34.8	156.9	159.0
5000	77.0	34.8	153.7	...

From the above it will be seen that the agreement between the estimated and the observed figures is so close as to prove that the engine is behaving normally in every respect; it is, in fact, so close as to indicate a certain share of coincidence, for neither calculation nor measurement could be exact to within closer limits than 1 per cent, leaving scope for a variation of 3 lb. per square inch between the observed and the calculated figures, assuming that all the premises on which the latter were based were strictly accurate. No tests were made above 4500 R.P.M. or below 1500.

The losses in any internal-combustion engine may be divided up into—(1) Piston friction. (2) Fluid pumping losses. (3) Bearing friction and auxiliary drives.

Of these piston friction constitutes always by far the largest proportion, and all more recent tests appear to indicate that for a piston of more or less normal design and proportions, and for normal conditions as to lubrication and jacket temperature, the piston friction in terms of lb. per square inch on the piston head may be arrived at with a fair degree of approximation by the empirical formula—

$$\left\{ \frac{\text{Mean fluid pressure including compression}}{4} + \frac{2 \text{ Mean inertia pressure}}{3} \right\} \div 10 + 2.0.$$

From such a formula we find that the piston friction expressed in terms of mean pressure on the piston at every fourth stroke (i.e. expressed on equal terms to the useful mean pressure) should be—

Speed R.P.M.				Piston Friction (lb. per sq. in. M.E.P.)
1500	..	..	..	7.7
2000	..	..	..	8.6
2500	..	..	..	10.0
3000	..	..	..	11.4
3500	..	..	..	13.1
4000	..	..	..	15.1
4500	..	..	..	17.3

The fluid pumping losses are dependent, assuming a normal valve setting, as in this case, and no other serious obstruction to the flow of gas into or out of the cylinders, upon the mean gas velocity through the valves and more particularly through the inlet valves. Fig. 180 shows the observed fluid pumping losses at different gas

velocities: from this curve, and that of gas velocity given in fig. 179, we find that the fluid pumping losses in this particular engine again expressed in terms of mean pressure should amount to—

Speed R.P.M.	Fluid Pumping Losses. (lb. per sq. in.)
1500 .. ..	2.0
2000 .. ..	2.3
2500 .. ..	2.9
3000 .. ..	4.0
3500 .. ..	5.4
4000 .. ..	6.9
4500 .. ..	8.6

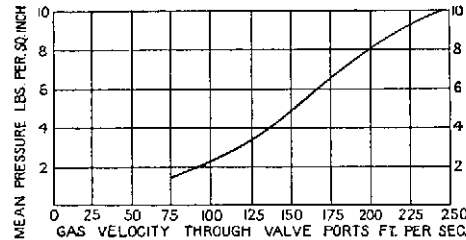


Fig. 180.—Mean Pressure of Suction Loop for different Gas Velocities with Normal Valve Setting and Poppet Valves.

Finally, there remains the friction of the bearings and the power absorbed by the auxiliary drives: these latter consist of a long train of gears to operate the overhead camshafts, the camshafts themselves in plain floating bearings, a large water circulating pump, two oil pumps, a small air-compressor for fuel supply, and the ignition gear. No direct measurement was taken of the power absorbed by these auxiliaries, and in the absence of actual data we must fall back on analogy from tests on other engines more or less similarly equipped. From such an analogy it may be assumed that the loss due to all these sources will range in more or less a straight line from the equivalent of 3 lb. per square inch at 1500 R.P.M. to about 5 lb. at 4500 R.P.M.

From the above it will be seen that the total of fluid and frictional losses may be estimated at—

Speed R.P.M.	Total Losses. (lb. per sq. in. M.E.P.) estimated.	Observed Losses.
1500	12.7	10.9
2000	14.2	12.6
2500	16.5	15.2
3000	19.3	18.6
3500	22.7	22.7
4000	26.6	28.8
4500	30.9	35.0

Fig. 181 shows in full lines the estimated friction losses as arrived at in the above tables, and in dotted line, the observed losses as

arrived at by motoring and by tests with the ignition cut off from various cylinders. It will be observed that while the general slope of the curve of mechanical and other losses does not agree very closely with the estimated curve, yet the mean value throughout the whole range of speed is in very fair agreement.

A number of readings of fuel consumption using the petrol referred to—as sample A in Chapter I—were taken at different speeds and loads, using in all cases an economical carburettor setting: that is to say, about 10 per cent weak as against the 20 per cent rich mixture employed for the attainment of the utmost possible power output. The fuel consumption readings therefore were taken with

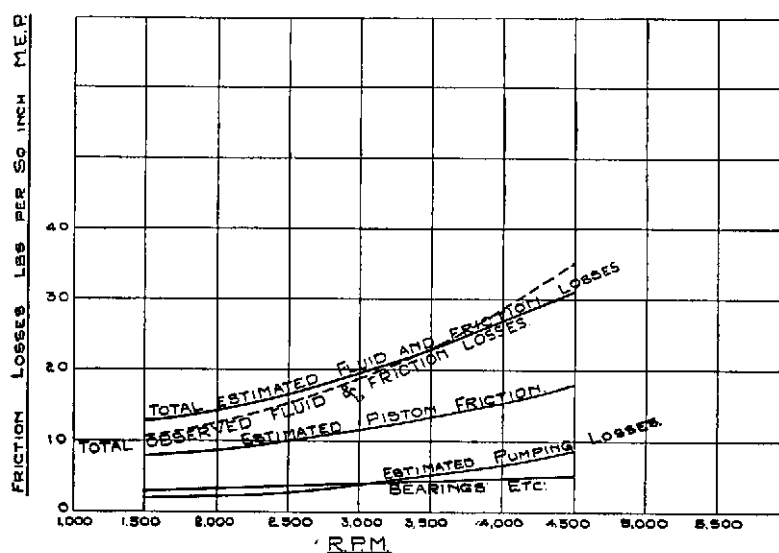


Fig. 181.—Estimated and observed Fluid and Friction Losses

the carburettor so adjusted as to reduce the maximum power by about 6 per cent. These tests gave the results shown in figs. 182 and 183, in terms of pints per hour per indicated and per brake horsepower. This particular petrol has a corrected calorific value (including the latent heat of evaporation of the liquid) of 18150 B.T.U.s per pint.

It will be seen that at 3000 R.P.M. the fuel consumption on full load is only 0.45 pint per B.H.P. and 0.395 pint per I.H.P. hour, corresponding to a brake thermal efficiency of 31.2 per cent and an indicated thermal efficiency of 35.4 per cent, a figure actually slightly in excess of the computed combustion efficiency, while at 66 per cent full load torque the observed fuel consumption at about

this speed was only 0.48 pint per B.H.P. hour, a figure unequalled even at full load by any touring-car engine. These figures should

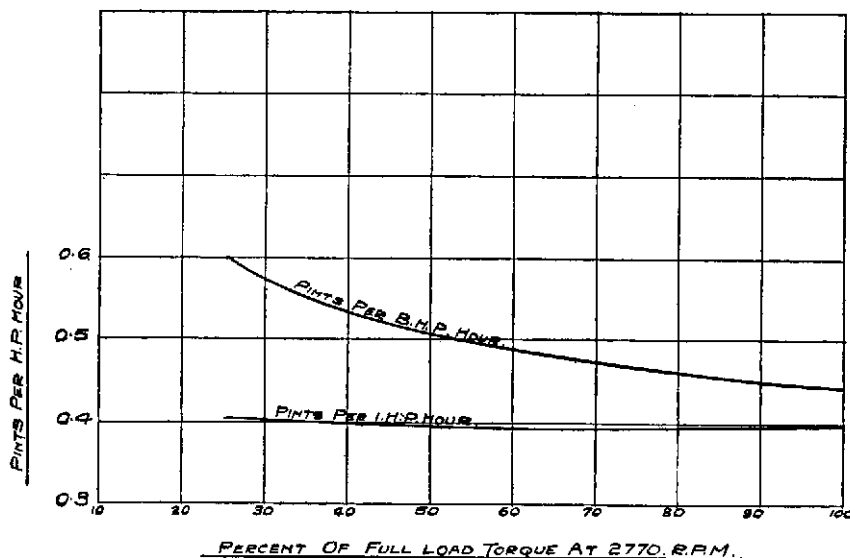


Fig. 182.—Three-litre Vauxhall Racing Engine  
Curve showing Consumption on Throttle at 2770 R.P.M., and most Economical Mixture Strength. Fuel, Petrol. Oil, "Shell," L.R.O.

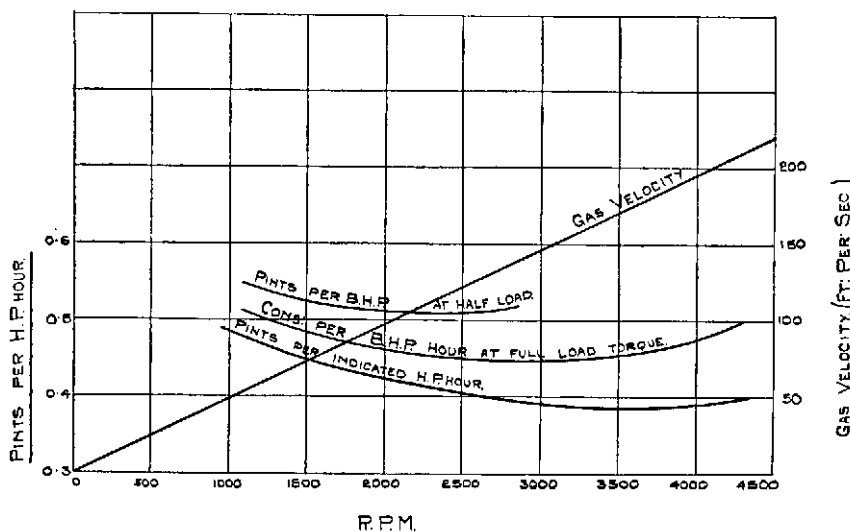


Fig. 183.—Three-litre Vauxhall Racing Engine  
Curve showing Variation in Fuel Consumption with Speed. Most Economical Mixture in all cases. Fuel, Petrol. Oil, "Shell," L.R.O.

go far to dispel the theory that a racing-car engine is essentially extravagant in fuel. The readings of fuel consumption when the

load was reduced by throttling down to one-third full load torque are rather striking.

Thus at 2770 R.P.M. the fuel consumption on petrol was found to be—

B.H.P.				Fuel Consumption. (Pints per B.H.P. hour.)
80	...	...	...	0.455
70	...	...	...	0.465
60	...	...	...	0.480
50	...	...	...	0.505
40	...	...	...	0.535
30	...	...	...	0.565
20	...	...	...	0.610

At 1950 R.P.M. the fuel consumption was—

B.H.P.				Fuel Consumption. (Pints per B.H.P. hour.)
60	...	...	...	0.470
50	...	...	...	0.480
40	...	...	...	0.495
30	...	...	...	0.525
20	...	...	...	0.570
15	...	...	...	0.605

It will be noted that when developing 20 B.H.P. at 1950 the gross consumption is only  $20 \times 0.570 = 11.4$  pints per hour. With the gear ratio used 1950 R.P.M. corresponds to a road speed of 48.5 M.P.H., a speed which calls for an expenditure of just about 20 B.H.P. reckoned at the engine shaft, so that the consumption in miles per gallon with an economical carburettor setting even at this high mean speed should be  $48.5 \times \frac{8}{11.4} = \text{about } 33.9$  miles per gallon.

A brake thermal efficiency of 31.2 per cent is, the author believes, the highest ever yet achieved by any engine running on petrol. Incidentally it is worthy of note that the indicated thermal efficiency reckoned from the fuel consumption, when using a weak mixture, corresponds very closely with the calculated combustion efficiency, showing that there can be practically no loss of unburnt fuel by irregular distribution or indeed from any other cause. It is interesting also to note that the fuel consumption per I.H.P. hour is exactly the same as that obtained in the single cylinder variable compression engine (described in Chapter I) when running on the same fuel, at the same compression ratio, and at the same gas velocity through

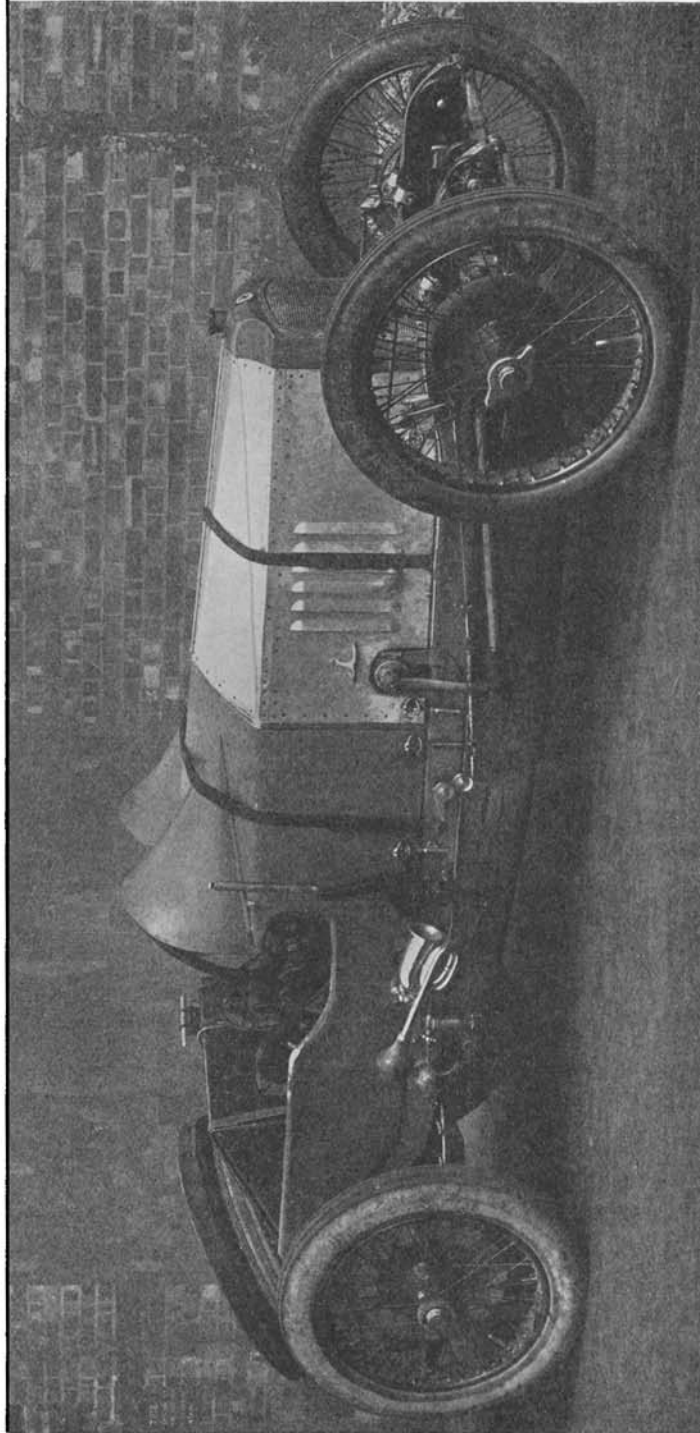


Fig. 184.—Vauxhall Three-litre Racing-car



the valves, but the brake thermal efficiency of the racing engine is considerably greater owing to the higher mechanical efficiency of the latter.

The author has dealt at considerable length with this particular engine, because the results obtained from it serve admirably to emphasize that a racing-car engine is nothing more or less than a highly efficient internal-combustion engine designed throughout on purely scientific lines, and whose behaviour is, from a thermodynamic point of view, perfectly normal in every respect.

Fig. 184 shows a photograph of one of the racing-cars fitted with the engine described above. Fully equipped, as shown in the photograph, the car weighs with driver and mechanic 2700 lb.; in this condition and with a gear ratio giving 25 miles per hour at 1000 R.P.M. it is capable of a maximum speed on the level of 115 miles per hour, corresponding to an engine speed exclusive of wheel slip of 4600 R.P.M., the actual engine speed being probably about 4800 R.P.M.

It is interesting to note that one of these cars after nearly eight years' use, and when fitted with a super-charger, won a number of competitions against much more modern racing cars, and in 1929 established a new record for the annual Shelsley Walsh hill climb for cars of unlimited size or power.

## CHAPTER XI

### AERO-ENGINES

Of all the applications of the Internal-combustion Engine, it is to aircraft in particular that high efficiency in its widest sense is most essential. The aero-engine must be efficient not only in relation to the fuel it consumes, but in every possible respect, including the material of which it is constructed, and it is therefore primarily to the aero-engine that most of the considerations in the preceding chapters have been directed.

Although it is only a very few years since the first power-driven aeroplane succeeded in leaving the ground, yet in this very short space of time the aero-engine has passed through several phases of its development.

In its earliest stages of development the one controlling factor was weight, then, as the aeroplane improved and longer flights were contemplated, the weight, not of the engine itself, but with fuel, oil, etc., for a protracted flight became the primary consideration, and extreme lightness of the engine alone, began to give way to some extent before economy in fuel and oil consumption, and reliability.

The Great War broke out during a very early stage in the development of aircraft, but the importance of the airship and aeroplane for military and naval purposes became so obvious that development was stimulated to an extent which has probably never before occurred in any branch of engineering.

The beginning of the war found Germany, alone of all the nations concerned, with any form of considered policy in regard to the type and line of development to be pursued. France possessed comparatively, a very large number of aeroplanes propelled by every conceivable type of engine, including air-cooled, water-cooled, fixed radial and rotating radial, four-, six-, eight-, and twelve-cylinder stationary types, in fact a heterogeneous collection representing examples of every conceivable type, but apparently without any

policy as to which types to perpetuate for immediate military purposes. Our own country possessed very few aeroplanes at all, and still less experience. Such few engines as we did possess were propelled for the most part by a miscellaneous collection of French and German engines, a few by the R.A.F. Vee-type air-cooled engines, and one or two other more or less experimental English-designed engines of the straight-line six or Vee type. America, by watching the trend of events for over two years as a neutral, had ample opportunity to frame a policy, and decided, on entering the war, on the development of a twelve-cylinder Vee-type engine, embodying the proved features of the best Vee-type engines in use at the time. Though eventually a very satisfactory engine, its development, despite the fact that she had unlimited experience placed at her disposal, took too long, and the engine did not appear in time to play any appreciable part in hostilities, almost all the American aeroplanes in actual service during the war being equipped with engines of European design.

Germany, from the start, decided to restrict development almost entirely to the six-cylinder straight-line water-cooled engine, on the grounds that this type of engine, though heavy, would give the maximum of reliability and fuel economy and permit of the largest production with limited manufacturing resources. Her policy was probably right, even as events turned out, and would certainly have been right had the war proved, as she undoubtedly expected, to be of short duration.

We in England had, before the war, given so little attention to aviation that we had no experience upon which to frame a policy of any kind at the start, hence we were forced to adopt the only course possible and purchase or produce every engine we could lay our hands on, regardless of type, until we had gained the necessary knowledge and experience to enable us to proceed independently. Despite this heavy handicap it is not a little to the credit of British engineers and scientists that, by the Armistice, we had the largest production, and had ourselves evolved probably the most efficient designs, both of engines and aircraft, of any of the countries concerned.

The progress of the war very soon indicated that several entirely different types of aircraft would be needed ; for example :

(1) A very fast, but small and light fighting machine, capable of rapid manœuvring and of climbing to high altitudes, but not required for long sustained flight.

(2) An observation aeroplane for spotting for artillery and generally reconnaissance work, to be capable of attaining high altitudes, and of long-sustained flight, but not necessarily very fast.

(3) A large bombing machine capable of carrying heavy loads and of flying great distances without replenishment.

The first type required an engine of high power at any altitude, light weight, and short over-all length. Economy of fuel and oil was, however, of secondary importance, since such machines were not normally expected to carry out long flights.

The second type required an engine of high power at high altitudes only and high fuel economy. Since such machines always climbed to a high altitude before crossing the enemy's lines, and could therefore afford to climb slowly, the power output at or near the ground was of little importance, provided it was sufficient for safety in taking off.

The third type required an engine of the highest possible economy, both in fuel and oil, and a very high power output at or near ground level in order to enable it to take off with the heaviest possible load. Since such machines were used almost entirely by night, performance at very high altitudes was not required. If capable of leaving the ground at all at the start, they would, by the time they had reached their objective, have attained a sufficient altitude to ensure reasonable security against anti-aircraft fire from the ground.

For all these purposes Germany decided to compromise with a single type of six-cylinder straight-line engine of between 160 and 300 B.H.P., a typical example of which is shown in fig. 185, though towards the closing stages, finding that she was being outclassed by the Allies, who were employing specialized engines for each class of machine, she began to show signs of departing from this policy.

For the first class of machine the lightest and shortest possible engine was required, and this undoubtedly would have been met by the air-cooled fixed radial had any country succeeded in producing a really successful example of this type and of sufficient power. No such example was produced before the Armistice, although all the allied countries made strenuous attempts to do so. Machines of this class were therefore fitted either with air-cooled rotating engines or with water-cooled eight-cylinder Vee type.

For the second class, namely, the reconnaissance machine, the eight- and twelve-cylinder Vee type and the six-cylinder straight line were used. The latter was, however, never held in particular favour by the Allies, despite its inherent reliability.

For the heavy bombing machines, the Allies used the same engines as for reconnaissance work, for it was not until the last stages of the war that the military importance of this class of machine was appreciated, when several engines of from 500 to 800 H.P. of the twelve-cylinder Vee type were developed for the purpose.

The requirements of aerial transport in peace time call for an engine of considerable power, capable of getting off with heavy loads, but not required to climb to very high altitudes, nor to cover very great distances without replenishment of fuel. In the former respect the conditions are somewhat similar to those obtaining in

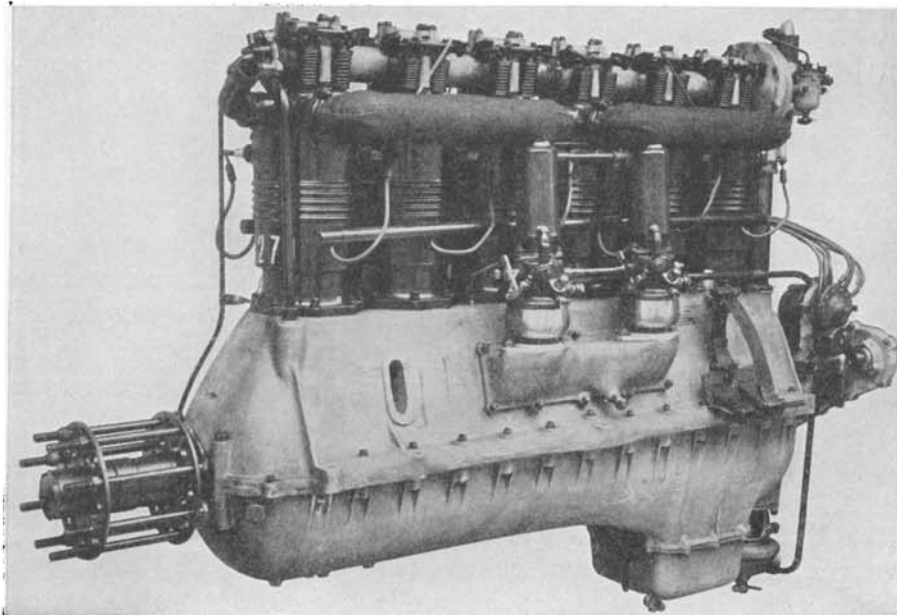


Fig. 185.—Typical Modern Six-cylinder German Aero-engine (1918)

the case of bombing machines, except that the need for economy in fuel and oil is not so insistent.

Both for bombing machines and, more particularly for commercial transport, reliability is of great importance, and, since the machines are heavily loaded and fly in a relatively dense atmosphere, the load factor on the engine is very much higher than in other types of aircraft, so that the engines are operating under much more strenuous conditions. Finally, we must consider the seaplane or flying boat, in which a very high power output is required momentarily when getting off from the water.

It is popularly supposed that an aeroplane engine operates

normally at or near its full load, but this is very far from being the case, for it must be remembered that at an altitude of about 20,000 ft. the density of the atmosphere is but little more than half that at ground level, so that, even though the throttle be wide open, the weight of charge taken into the cylinder is only about half the normal, and the conditions are very much akin to throttling down to half torque on the ground, for both the pressures and the heat flow are reduced to nearly one half. It is only when leaving the ground and climbing for the first few thousand feet that the load factor is at all high.

An average modern single-seater fighting machine (1920) will climb 10,000 ft. in about seven minutes, that is to say, within seven minutes of leaving the ground the machine is in air at an absolute pressure of only 10.6 lb. per square inch, and the indicated horse-power is only 72 per cent of that developed at ground level, so that the power output is very rapidly reduced, even though the throttle be kept wide open. It is only in the heavy bombing machine and in aerial transport that the engine is called upon to work "all out" at low altitudes, and therefore under severe conditions as regards pressures and heat flow.

Discussion still rages as to the lightest form of aircraft engine consistent with sound mechanical design; and the dimensional theory has been used and abused to an unwarranted extent. The dimensional theory is applicable only when all dimensions are strictly proportional, which they can never be. According to such a theory the lightest possible engine will be the type with an infinite number of pistons connected to a minimum number of cranks. This holds good only so long as small pistons, cylinders, &c., can be made proportionately as thin as larger ones, which is out of the question in the sizes in view, and also all the auxiliary structure and mechanism can be reduced in proportion.

The nearest practical approach to this theoretically ideal form is the air-cooled radial engine with seven or nine cylinders disposed radially round a single crank, or double the number round two cranks. It does not pay to increase the number of cylinders beyond nine, because they then crowd too closely round the crankcase, necessitating a larger crankcase and longer connecting-rods for the same stroke. Though very attractive on paper, this form of engine has certain inherent defects.

(1) The loading on the single crankpin is excessive, and necessitates very special treatment. Moreover, this loading being due

almost entirely to centrifugal and reciprocating forces, does not ease off appreciably as the density of the air is reduced.

(2) The distribution of fuel and air in uniform proportions to an odd number of cylinders disposed radially is no easy problem.

(3) The valve gear is troublesome, and being so widely scattered it is practically impossible to enclose or lubricate it.

Apart from its light weight, the fixed radial engine has several important advantages which go far to balance its inherent defects.

(1) It lends itself admirably to air cooling, since every cylinder has equal advantages, and all have their combustion heads projecting well into the slip stream from the propeller.

(2) It is very short, and therefore particularly attractive from the point of view of rapid manœuvring.

(3) Its general shape and ease of attachment to the fuselage of an aeroplane are points very much in its favour.

During the war numerous attempts were made to produce such an engine, but without much success, owing to the defects named above, but since the Armistice at least two successful engines of this type have been produced, notably the Bristol Jupiter engine of 380 B.H.P. and the Armstrong Siddeley Jaguar of 350 B.H.P.

The difficulties as regards crankpin loading and distribution can both be obviated by the employment of a fixed crankshaft with the cylinders rotating round it, and by feeding the fuel and air through the crankcase as in the Gnome, Le Rhone, Bentley, and other engines. This form was widely used before and during the earlier stages of the war, particularly by France. For relatively small powers up to about 200 B.H.P. it is satisfactory, but the windage resistance becomes very serious and the gyroscopic effect due to the large rotating mass very troublesome, when the size is increased beyond this limit. A compromise between these two types wherein both the cylinders and crankshaft rotate in opposite directions has been suggested, and several engines have actually been built, notably one by Siemens and Halske in Germany, and an experimental engine of about 250 B.H.P. by Messrs. Ruston and Hornsby, built to the designs of Mr. A. E. L. Chorlton.

After the single-crank radial the next stage is the fan type, such as the well-known Napier Lion, in which three pistons operate on each crank, and the Maltese cross type with four cylinders per crank. It is usual to make both these types with blocks of four cylinders, making twelve or sixteen in all, though some few examples have been built with six-throw cranks, making eighteen and twenty-

four cylinders respectively. Figs. 186 and 187 show the Napier Lion engine, an example of this type which has proved particularly successful.

The next type is that in which two pistons are coupled to each crank, generally known as the Vee-type engine. This type has usually either four or six cranks, and therefore eight or twelve cylinders. To the Vee-type class belong most of the successful engines used by the Allies during the war, and probably also at the present day, though more recent developments have brought both the single-crank radial and the straight-line into prominence. The

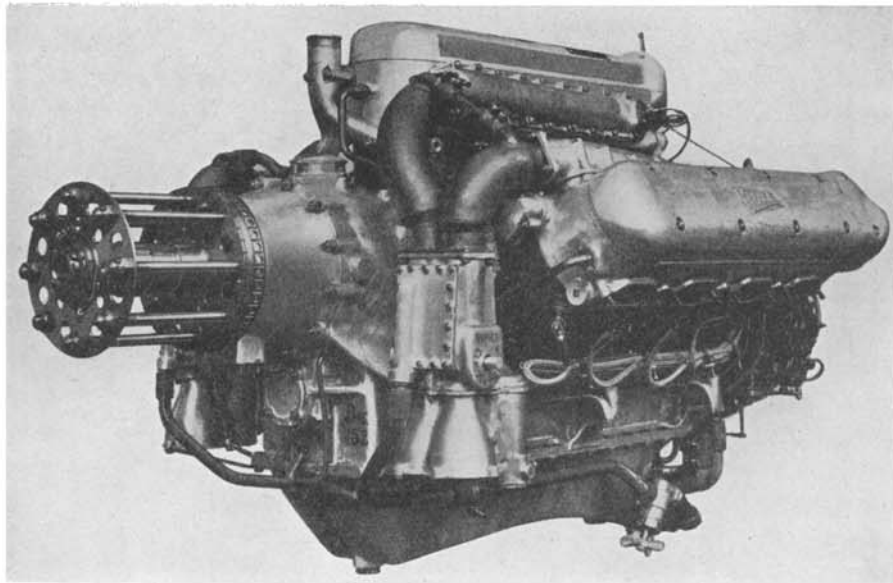


Fig. 186.—Napier Lion Engine

Rolls-Royce Eagle and Falcon engines, the Hispano Suiza, and the R.A.F. were among the most successful examples of this type used during the war, while the Rolls-Royce Condor of 550 B.H.P., the Liberty of 400 B.H.P., and the 600 H.P. Fiat represent excellent examples of more modern development.

Finally, we have the plain straight-line six-cylinder engine with one piston operating each crank ; a type which has been vehemently condemned by the supporters of the dimensional theory, but which was used practically throughout by the Germans, and to a considerable extent by the Allies also, e.g. the Siddeley Puma and the Beardmore, both of which engines did admirable work and competed very favourably with the other types.



Controversy as to the lightest type of engine will probably continue to rage indefinitely, since so many conflicting and often indeterminate factors have to be taken into account, viz. reliability, fuel and oil consumption, &c. On the grounds of reliability there can be no doubt but that for equal excellence of design and workmanship the advantage lies with the straight-line six-cylinder, since the load factor on its bearings is considerably the

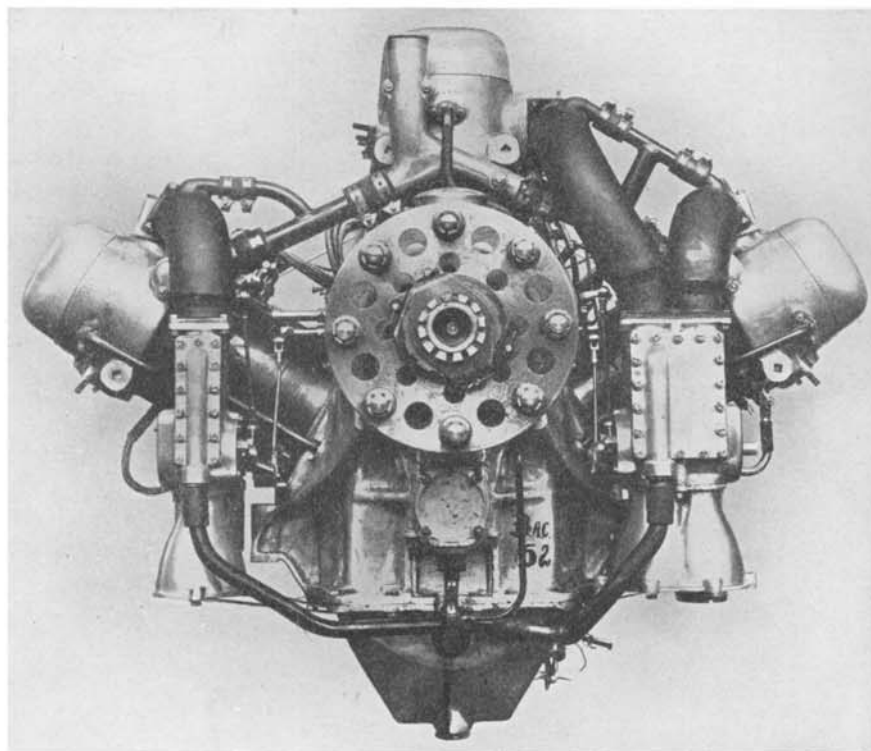


Fig. 187.—Napier Lion Engine, End View

lightest, the stresses are for the most part simple and direct, and can be dealt with by simple and direct means; also the auxiliary gear, upon which the reliability of the engine so largely depends, is reduced to the minimum. On the score of fuel efficiency it has again, for equal excellence of design and workmanship, all the advantage, having only two carburettors and an inherently good form of distribution, the losses due to defective carburation and distribution can, with a given amount of superintendence, be kept lower than with any other type.

Finally, since the load factor on its bearings is the lowest, less oil

need be circulated for cooling purposes, and consequently less is thrown on the cylinder walls and consumed. On all these counts, therefore, the straight-line six has advantages which go far to compensate for the extra material in the crankshaft and crank chamber as compared with other types. It is, of course, impossible to evaluate the factor of reliability and to equate it in terms of weight, though, clearly, reliability is always worth some pounds in initial weight.

Again, in comparing the weights of various engines the efficiency of the propeller is sometimes overlooked. One of the essential requirements of any aero-engine is that it shall turn the propeller at its most efficient speed, and this, in the case of large and heavily loaded machines, is a comparatively low one. Hence the engine must either turn at a relatively low speed, 1200–1400 R.P.M., or must be geared down, if a high over-all efficiency is required. Experience has shown that the weight of reduction gearing very nearly balances the increased weight of engine required to develop the same power at the lower speed, while here again the factor of reliability looms largely, since reduction gears, at the best of times, are a source of weakness, the more so as the ratio of reduction is increased. If we require the highest over-all efficiency, from the fuel burnt to the thrust of the propeller, and assume a reasonable duration of flight, we find that the over-all weight of the power unit, together with its fuel or oil, becomes, in practice, virtually a function of the piston speed, and this almost irrespective of number or disposition of cylinders or of the use or otherwise of reduction gearing.

**Air- or Water-Cooling.**—Here again a great deal of controversy rages as to which is the more desirable. The water-cooled engine starts with the heavy handicap of a radiator and water connections, involving considerable additional weight, and, what perhaps is even more serious for military purposes, much greater vulnerability; but against these defects must be offset a very large advantage on the score of reliability, and the ability, owing to the lower cylinder temperature, both to consume less oil, to employ a higher compression, and therefore to obtain a lower fuel consumption. It is not proposed to deal, at any length, with the pros and cons of air-*versus* water-cooling, but it is probably sufficient to point out that the radial engine, by reason of the disposition of the cylinders and its relation to the slip stream from the propeller, offers the most ideal case for air-cooling, and, the author is tempted to think, the only case for it.

The great virtue in water-cooling appears to lie not so much in the fact that the mean temperature of the cylinder can be kept fairly low, though this, of course, is of advantage, but rather in the fact that water boils at a comparatively low temperature, and has a very high latent heat, so that in the event of local overheating due to any cause, the water will boil locally, and the heat abstraction will be intensified to an enormous extent. Thus in a water-cooled engine we may safely assume that under no circumstances whatever (except bad design involving steam-locks) can any part of the outside surface of the cylinder wall exceed a temperature of  $212^{\circ}$  at ground level.

In the air-cooled engine we have no such advantage. Although by very careful design, &c., the mean temperature of the cylinder may be kept down to perfectly safe limits, we have nothing but the conductivity of the metal to ensure against the development of hot spots, such as are liable to be brought about by slight leakage, by detonation, or by slight distortion, with consequent increase of local friction or leakage.

At the best of times, the conductivity of any metal is but a poor substitute for the latent heat of evaporation of water, and it can compare only when the metal walls are made excessively thick, which is impracticable on the ground of weight. While, therefore, the air-cooled cylinder may be kept cool enough to run very satisfactorily so long as everything is in perfect condition, it has not, and probably never can have, anything like the factor of safety of the water-cooled cylinder.

For military purposes, and more especially for fighting machines, the advantages on the score of reduced vulnerability, freedom from freezing, and reduced weight, are all very important ones and may outweigh the disadvantages of air-cooling, but it is, in the author's opinion, very doubtful whether air-cooling will oust water-cooling for commercial or long-distance military flight.

A great deal of extraordinarily valuable research work on the subject of air-cooling was carried out during and since the war by the late Major G. Norman and Professor A. H. Gibson at the Royal Aircraft Establishment, and some remarkably fine results were obtained with experimental cylinders, having thin but well-ribbed steel barrels with very thick aluminium cylinder heads cast on to them, and with steel valve seats cast in.

By the use of very thick aluminium, which is an excellent con-

ductor, the heat can be spread fairly uniformly over the cylinder head and out to the deep and thick radiator flanges, with the result that under normal working conditions local over-heating can be avoided and results obtained, comparable with those of a water-cooled cylinder.

The lack of recuperative power, which in the author's opinion is the weakness of the air-cooled engine, applies also, though to a lesser extent, to those water-cooled engines in which there is a double metal wall through which the heat has to pass before reaching the cooling water.

In fig. 194 is shown a part-sectioned cylinder block of the Hispano Suiza water-cooled engine, in which it will be seen that a composite construction is used consisting of a complete aluminium cylinder into which is screwed a steel thimble forming both the liner and cylinder head, and that the valves seat directly on to this steel thimble. This form has many important advantages from a constructional point of view, but it is open to the objection that the heat has to pass through two separate thicknesses of metal, in contact only by screwing, before reaching the cooling water. Given good fitting this suffices for the normal rate of heat flow,

but it has a very much reduced margin of safety for dealing with excessive rates of flow such as occur when detonation is set up, &c.

**Cylinder Construction.**—In an aircraft engine it is necessary always to provide some form of composite cylinder construction, because the limitations of weight deny the use of the normal construction in which the outer jacket is cast in one with the cylinder liner and of the same material. Resort must, therefore, be had to composite built-up forms, and much diversity of

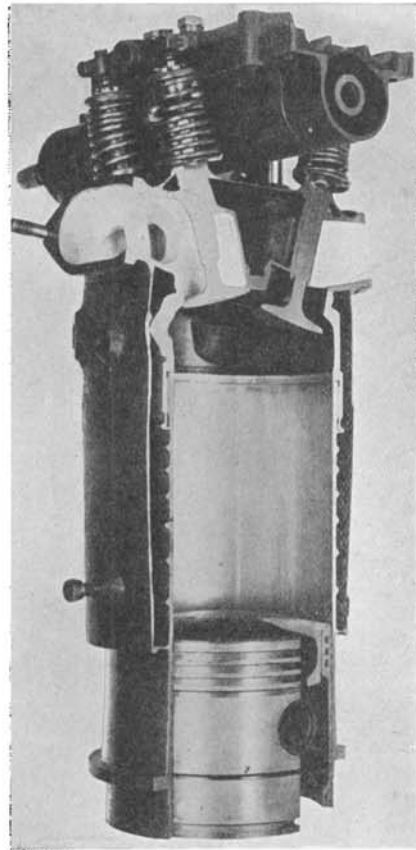


Fig. 188.—Cylinder Construction used in the 260 H.P. Mercedes Engine

opinion prevails as to the relative merits and demerits of the different forms in use.

Fig. 188 shows the cylinder construction used in the 260 B.H.P. German Mercedes engines such as were fitted to the large Gotha bombing planes used during the war for long-distance bombing raids. This engine proved to be quite exceptionally reliable.

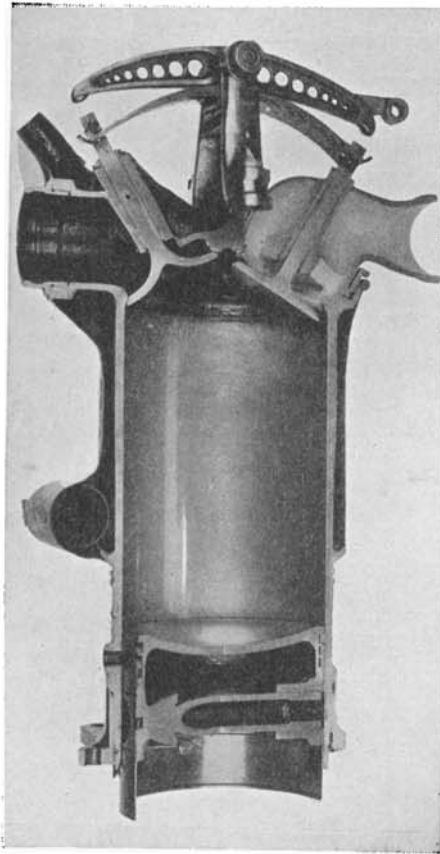


Fig. 189.—Austro-Daimler Cylinder

In this case a high-carbon steel barrel is screwed into a pressed or cast-steel head, the bottom edge of which is spun over one of the flanges of the barrel and welded to it in order to ensure against any possible leakage down the thread. A light built-up sheet-steel jacket is then welded over all. This form of construction has proved a very reliable one, provided always that the welding has been skilfully accomplished. It is open to the objection that it requires a good deal of specialized plant and special skill in welding, especially

in dealing with the sparking plug losses and the water connections. Given, however, the requisite skill and plant, it is certainly a very satisfactory method. The long and very well-cooled guides for the exhaust valves are an excellent feature and deserve special notice.

Fig. 189 shows the form of construction used in the early Austro-Daimler, and later in the 120 and 160 B.H.P. Beardmore six-cylinder engines. In this design the cylinder barrel and head are cast in one piece in cast iron, but the inlet valve is fitted in a separate detachable housing held in place by an annular locking nut. The lower end of the cylinder barrel is screwed externally to receive a steel flange for the holding-down bolts.

The water-jacket is of electro-deposited copper; this is formed in place on a wax matrix, which is subsequently melted out. Much experience and great precautions are needed to ensure a uniform deposition of copper and thorough adhesion to the cast iron. In this case also, once the plant is available and the necessary experience has been gained the method is very satisfactory.

Fig. 190 shows the form adopted by the Maybach Company in Germany for their 300 B.H.P. six-cylinder engines used in the later Zeppelin airships and, during the latter phases of the war, in many of the larger aeroplanes.

In one form of this construction the cylinder head, together with the whole of the water-jacket, is of cast iron, and a high-carbon steel barrel is screwed and sweated, but not welded, into the head, the lower end of the jacket being sealed by means of a rubber ring. In another form, only the cylinder head and the upper portion of the jacket are of cast iron, the jacket of the barrel being a very light seamless steel tube also screwed to the cylinder head in the same manner as the liner.

The construction shown in fig. 191 is that used by the Benz Company for all their aero-engines. In this case the whole of the cylinder barrel, cylinder head, and holding-down flange are cast in one piece in cast iron over which a light pressed steel jacket is electrically welded direct on to the cast iron. The welding of such a thin steel jacket to a relatively thick cast-iron body is no easy problem, but it has been met satisfactorily by this company. Attention should be called to the use in these engines of a special support from the crown of the piston to the gudgeon-pin in order to transmit the load as directly as possible to the connecting-rod.

The construction shown in fig. 192 is that adopted by Messrs. Rolls-Royce, and subsequently employed in the Liberty and several

other engines. In this case the cylinder barrel and head are forged in one piece, and the inlet and exhaust valve elbows are screwed and welded into place, the whole body being subsequently covered by a light pressed steel jacket, welded over all.

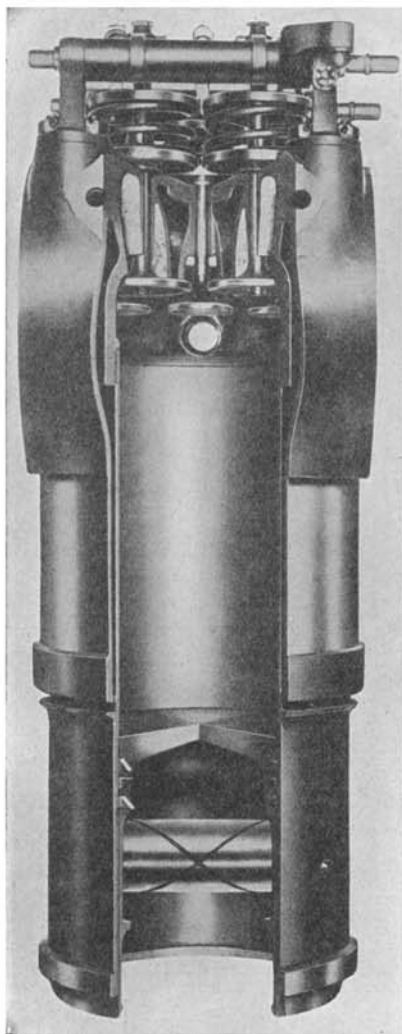


Fig. 190.—Zeppelin Airship Engine Cylinder  
(Maybach)

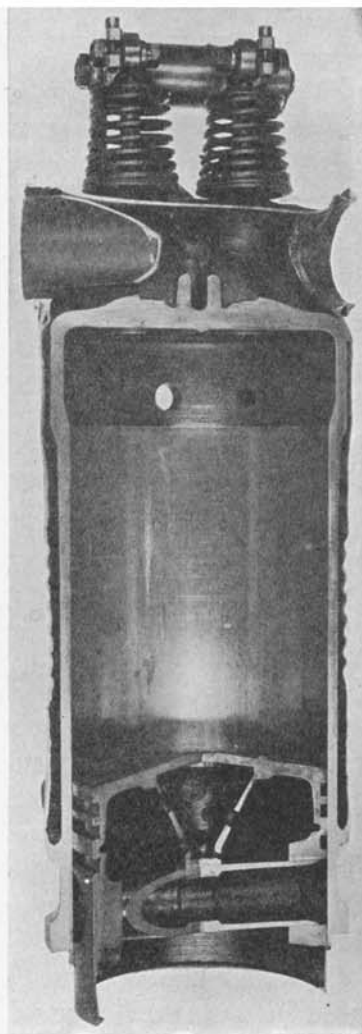


Fig. 191.—Benz Cylinder

Fig. 193 shows the cylinder construction used in the earlier Sunbeam engines, in which complete blocks of cylinders were cast in iron together with their cylinder heads and the upper portion of the water-jackets; to save weight the whole of the sides of the casting below the valve outlets were removed and replaced by light sheet

metal plates. In some of the later Sunbeam engines such as the 200 B.H.P. Arab engine, the complete cylinder block was cast in aluminium with thin steel liners shrunk in.

Fig. 194 shows the construction of the Hispano Suiza cylinder block already referred to, in which a complete steel thimble, forming

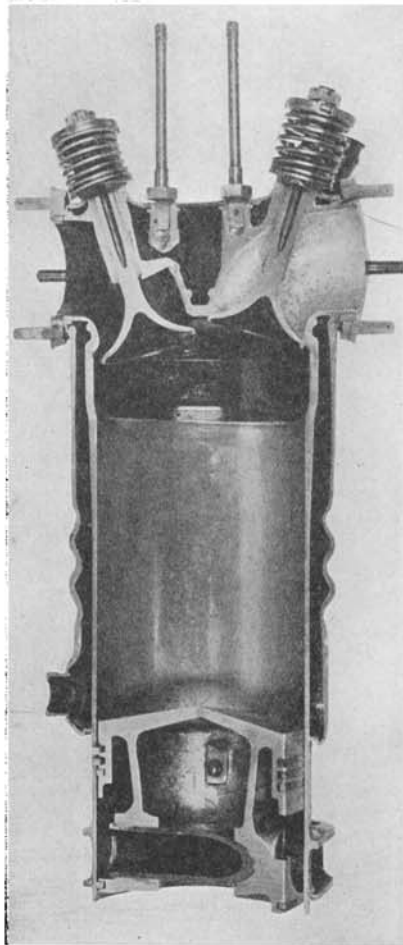


Fig. 192.—Rolls-Royce Cylinder

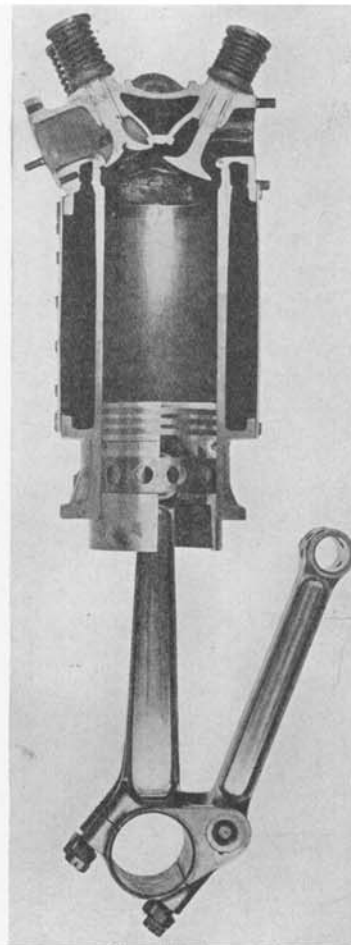


Fig. 193.—Sunbeam Cylinder

both the cylinder liner and valve seats, is screwed directly into an aluminium cylinder block.

In the Siddeley Puma engine shown later in fig. 204, blocks of three-cylinder heads, together with the upper portion of the water-jacket, are cast in aluminium with pressed-in bronze valve seats. Into these are screwed, for a short length only, thin steel cylinder barrels. The lower portions of these exposed steel liners are



enclosed by means of light and very thin die-cast aluminium jackets which are bolted direct to the cylinder head casting, while the lower joints consist of rubber-packed stuffing glands.

The form of construction shown in the previous chapter in connection with the three-litre Vauxhall racing-car engine, has been used in several experimental aero-engines, and has been adopted

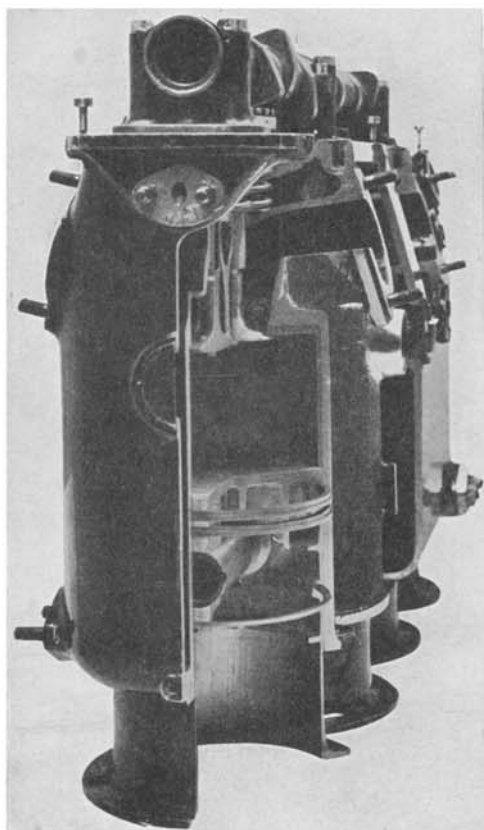


Fig. 194.—Hispano Suiza Cylinder

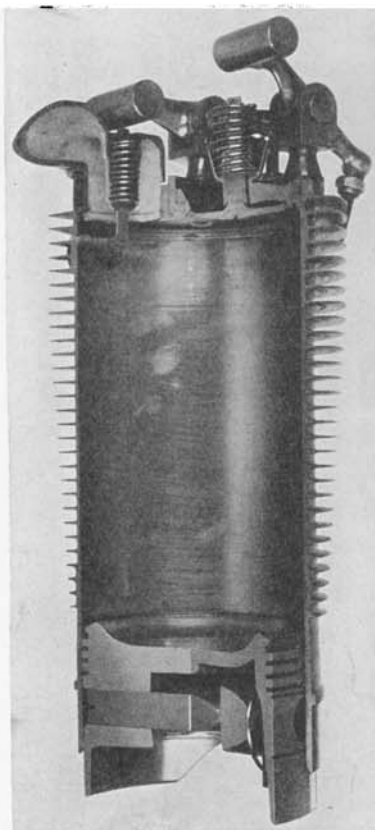


Fig. 195.—Clerget Cylinder

by Messrs. Beardmore in the large 750 B.H.P. six-cylinder aero-engine built by that firm.

It has in the author's opinion much to recommend it, not the least being its extreme simplicity and ease of manufacture.

Turning now to air-cooled engines the problem becomes somewhat different, since weight is no longer the sole consideration, high conductivity being at least equally, if not more important. In engines with rotating cylinders the cooling conditions are very favourable, and in such engines it is usual to employ plain steel

cylinders machined throughout from a single forging as shown in fig. 195, which is the form employed in the Clerget rotating engine, and fig. 196, which shows a section of the Gnome single valve engine.

Fig. 197 shows a section of the Le Rhone cylinder. In this case the whole of the cylinder and head is machined from a single piece of steel, but a very thin cast iron liner about 1 mm. in thickness is

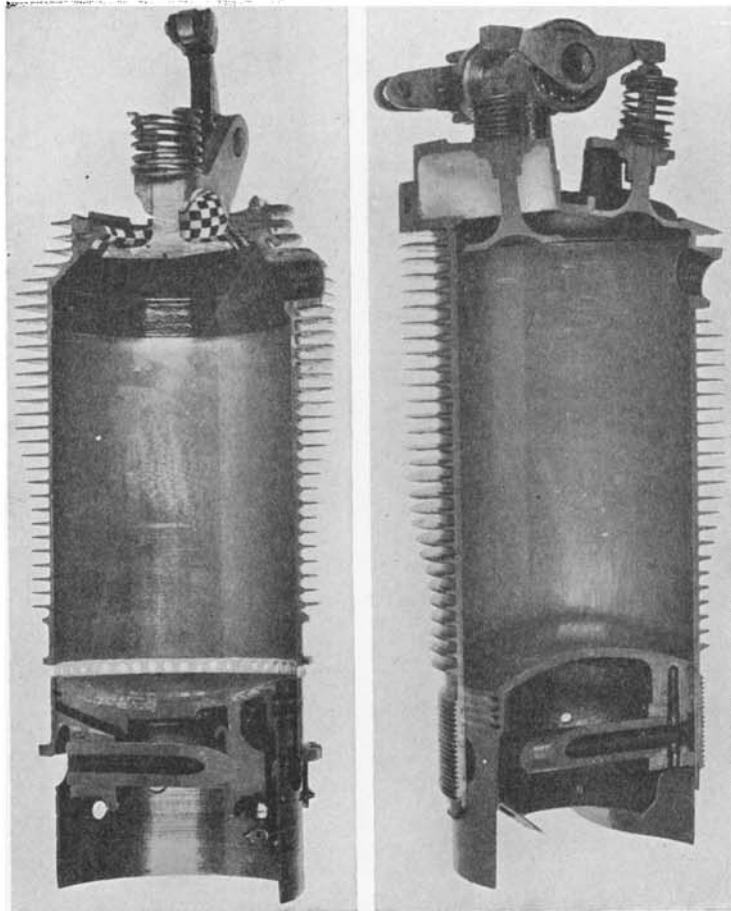


Fig. 196—Gnome Cylinder

Fig. 197.--Le Rhone Cylinder

pressed in. This construction is curious, and the author has never been able to discover for what reason it has been adopted.

In the B.R. 1 and B.R. 2 rotating cylinder engines, fig. 198, the cylinder barrel consists of a hard steel liner surrounded by a thin and light ribbed aluminium jacket. The cylinder heads in this engine are detachable and are of steel, also a curious construction.

In the case of fixed cylinder engines the problem of cooling

becomes more difficult and resort had to be made to more complicated constructions.

Fig. 199 shows an experimental R.A.E. cylinder for fixed cylinder air-cooled engines. This consists of a thick aluminium casting with deep ribs and with steel valve seats cast in position. A thin steel liner is shrunk in as shown in the photo of the sectioned cylinder.

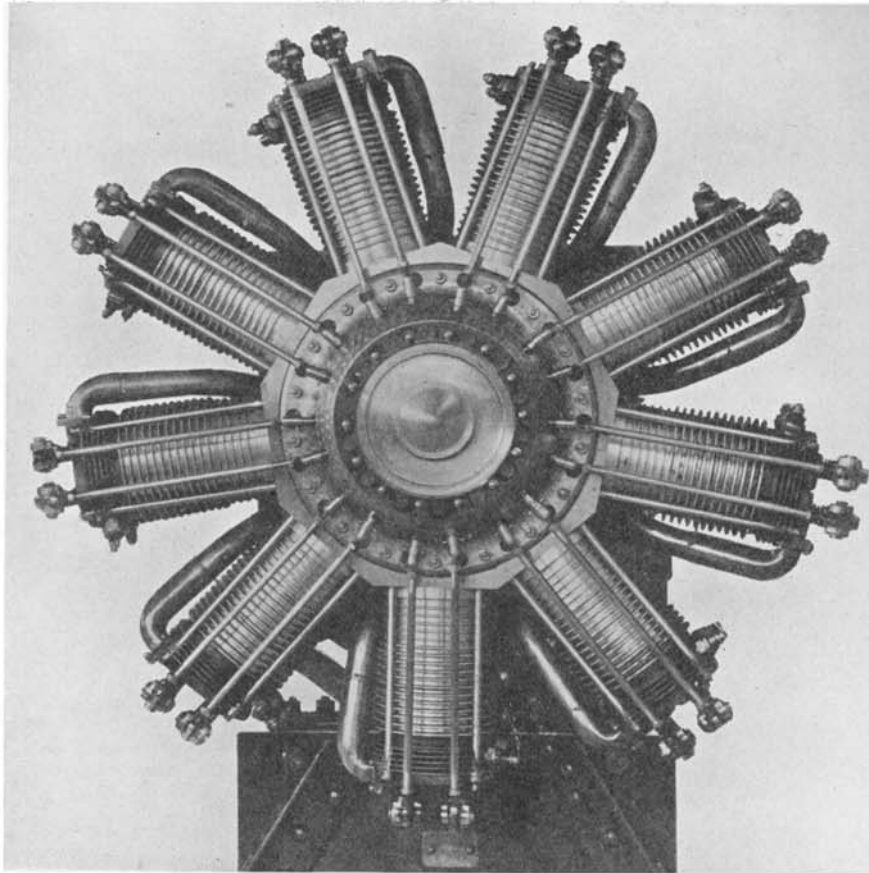


Fig. 198.—9 Cylinder B.R. Rotating Engine

This proved satisfactory for a time, but contact between the liner and cylinder body gradually failed, resulting in overheating of the liner. As explained previously, this form of cylinder construction was subsequently abandoned in favour of one consisting of an aluminium cylinder head cast on to a plain ribbed steel barrel.

In the Bristol Jupiter engine shown in fig. 200 the whole of the cylinder barrel and head are of steel, but a cast aluminium poultrice containing the inlet and exhaust valve elbows and heavily

ribbed, is attached to the flat steel head. This form is simple to manufacture, and has the advantage that if contact between the cylinder head and aluminium poultice is impaired by warping, it can be restored by scraping the surfaces.

As stated previously, no satisfactory fixed radial air-cooled engine of adequate size was developed during the period of the war despite the most strenuous efforts in this direction. Since the war,

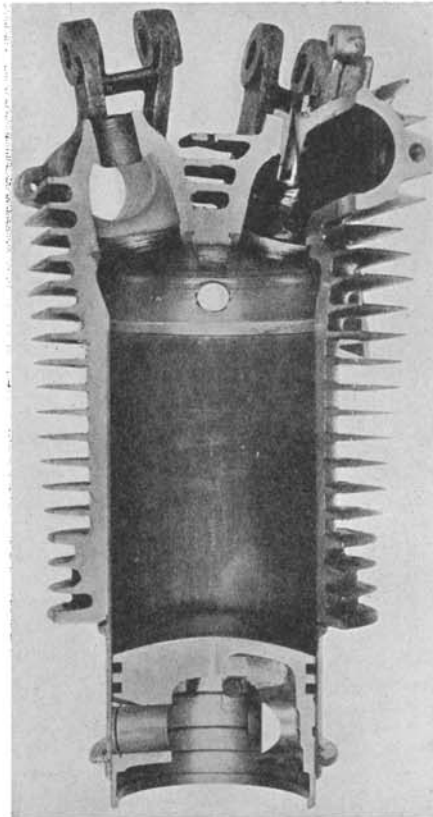


Fig. 199.—Experimental R.A.E. Air-cooled Cylinder

however, two such engines have been developed, namely, the Bristol Jupiter and the Siddeley Jaguar, figs.199-203.

The former is a single crank nine-cylinder engine developing a normal power output of 380 B.H.P. at 1575 R.P.M. The cylinders are of 5.75-inch bore with a stroke of 7.5 inch. The normal compression ratio of this engine is 5 : 1, and it will run continuously for long periods at a brake mean pressure of 109 lb. per square inch with a consumption of 0.535 lb. of petrol and 0.048 lb. of oil per B.H.P.

hour, making a gross consumption of 0.583 lb. per B.H.P. hour. The weight of the complete engine is 729 lb. or 1.92 lb. per horsepower. With fuel and oil for a run of six hours the gross weight (exclusive of tanks) is 2075 lb. or 5.46 lb. per horse-power.

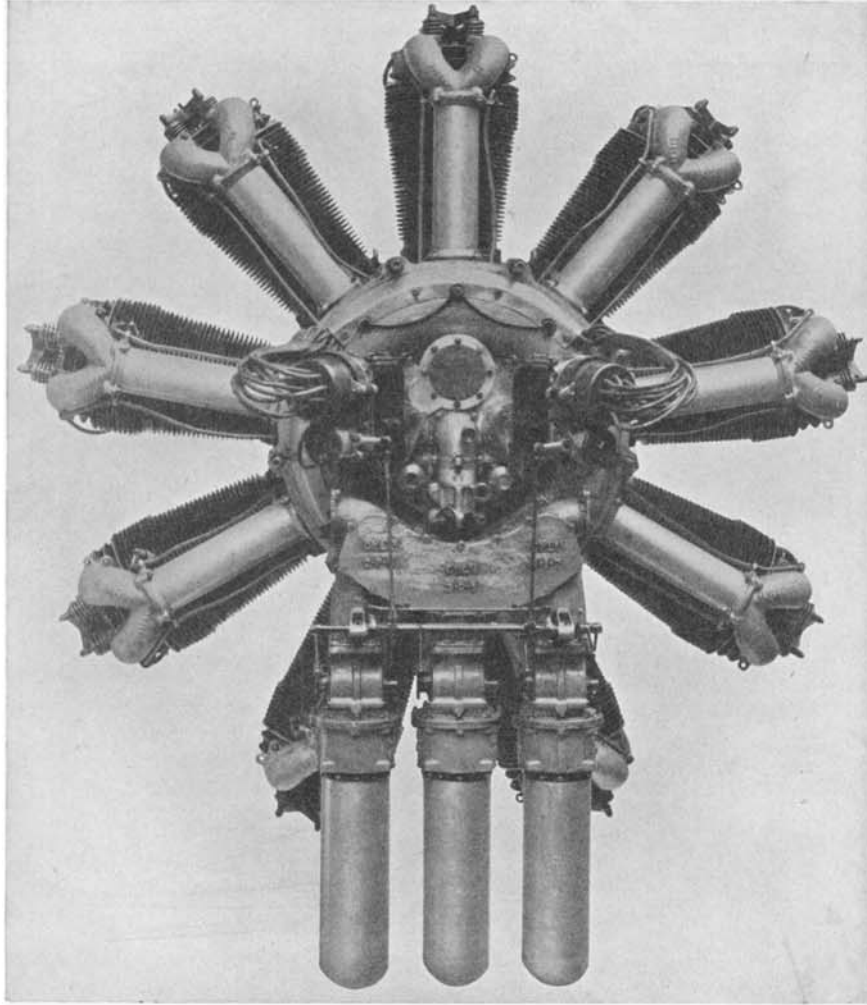


Fig. 200.—400 H.P. Bristol Jupiter Engine

In this engine a single master connecting-rod is used to which all the remaining eight pistons are connected by means of articulated rods, the pins for which are carried by a pair of flanges on the split master connecting-rod. A plain white metal-lined bearing is used for the master rod, and beyond lavishing the utmost care in the fitting and lubrication of this bearing, no special design precautions

are taken to deal with the very heavy loading, due to the combined centrifugal and inertia pressures from the nine pistons and rods all operating on a single pin.

One of the greatest difficulties encountered with air-cooled engines is that of dealing with the expansion of the cylinder and the resulting increase in the valve motion clearances when hot. In the case of the Bristol Jupiter engine this has been dealt with most

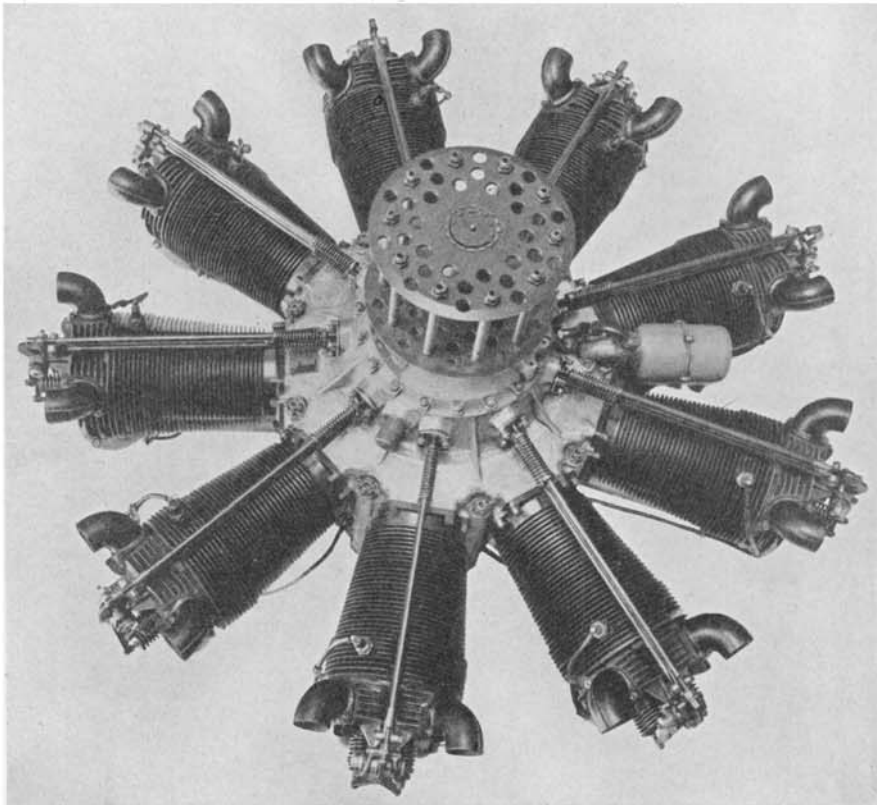


Fig. 201.—400 H.P. Bristol Jupiter Radial Engine

effectively by means of a very ingenious compensating device developed by Mr. Raymond Morgan. Briefly, this consists in the provision of a movable fulcrum pin for the overhead valve rockers which is controlled by a fixed rod attached at one end to the crankcase, and at the other to a hinged cradle carrying the valve rockers. This control rod being subject to the same temperature conditions as the operating rods maintains the same relative length, with the result that as the hot cylinder expands it tends to draw the cradle



down, and so to retain the same tappet clearance at all cylinder temperatures. These control rods can be seen on the forward end of the engine (see fig. 201).

The Siddeley Jaguar shown in figs. 202 and 203 is a fourteen-

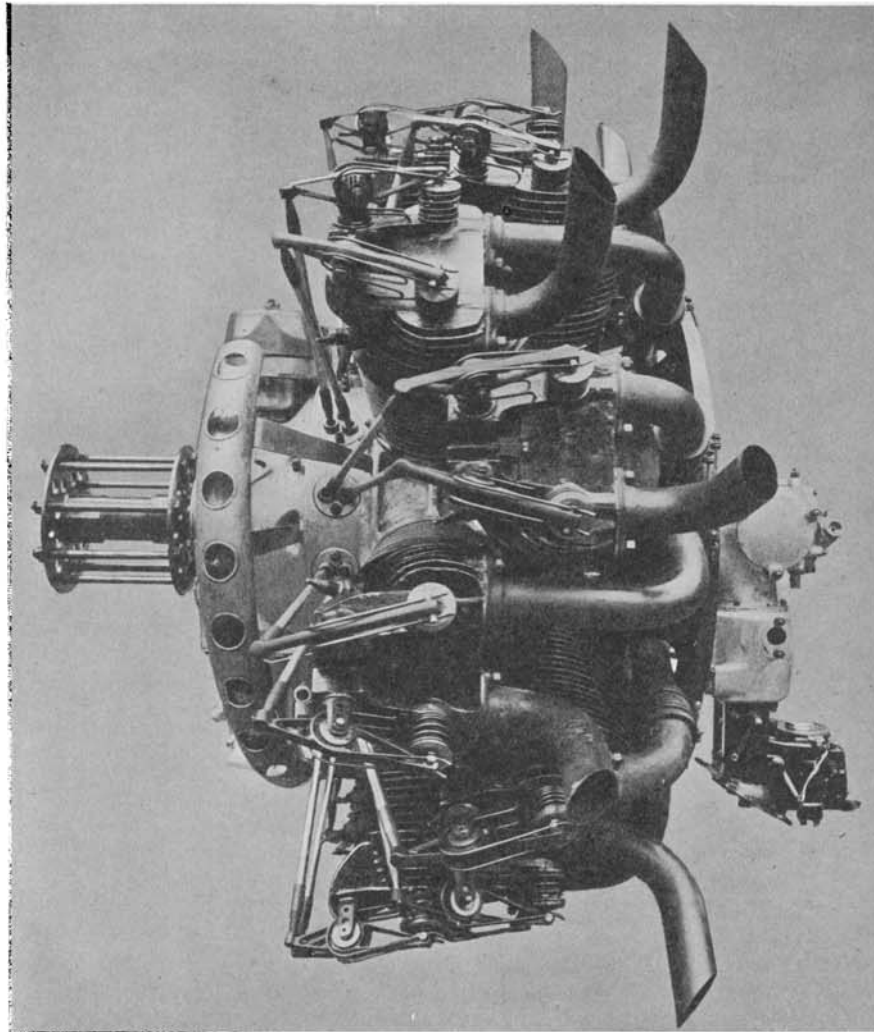


Fig. 202.—Siddeley Jaguar Engine

cylinder two-crank radial, and develops a normal power output of 350 B.H.P. at a speed of 1500 R.P.M.; the cylinders are five-inch bore and five-and-a-half-inch stroke.

As in the case of the Bristol Jupiter this engine also has been developed since the Armistice, though in both cases the earlier

stages of development were in progress during the war. The manufacturers give the fuel and oil consumption of this engine as 0.525 and 0.027 lb. per B.H.P. hour respectively, a gross consumption of 0.552 lb. per B.H.P. hour. The weight of the engine alone is given as 710 lb. or 2.03 lb. per horse-power. The gross weight with fuel and oil for six hours' flight (exclusive of tanks), works out at 1870 lb. or 5.35 lb. per horse-power. In this engine, the cylinder barrels are of steel as in the Jupiter, but the cylinder heads are aluminium

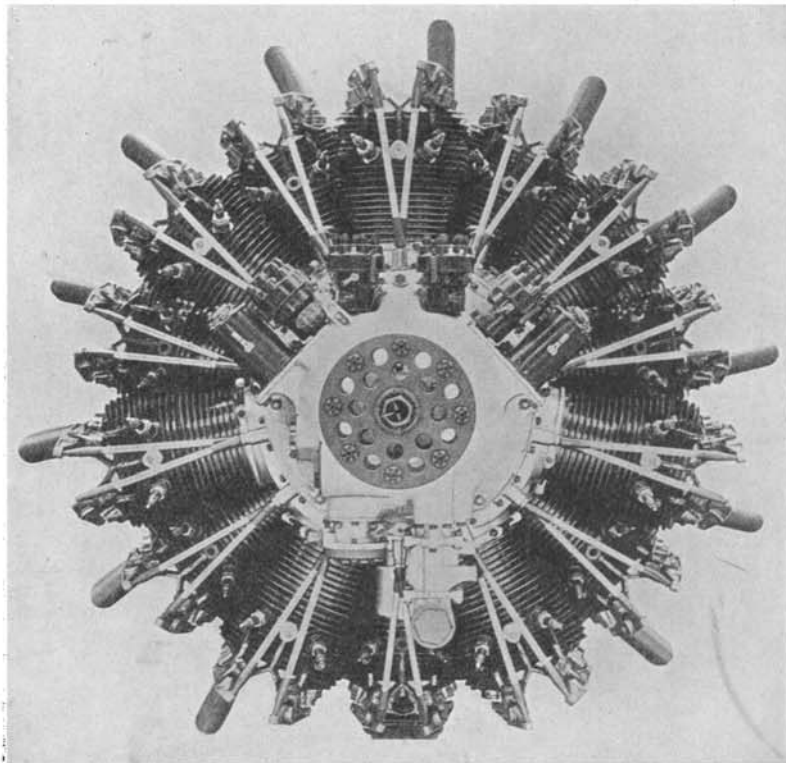


Fig. 203.—Siddeley Jaguar Engine

castings screwed on to the steel barrels. Ignition is by high tension coil and battery, and a small dynamo is provided for charging the accumulator.

The engine shown in figs. 204 and 205 was known during the war as the Siddeley Puma, and is a development of the B.H.P. engine designed by Messrs. Beardmore and Major Halford of the R.A.F. It has six cylinders each of 145 mm. bore and 190 mm. stroke with a normal power output of 240 B.H.P. at a speed of 1400 R.P.M. With a compression ratio of 5 : 1 the fuel consumption is 0.5 lb. and



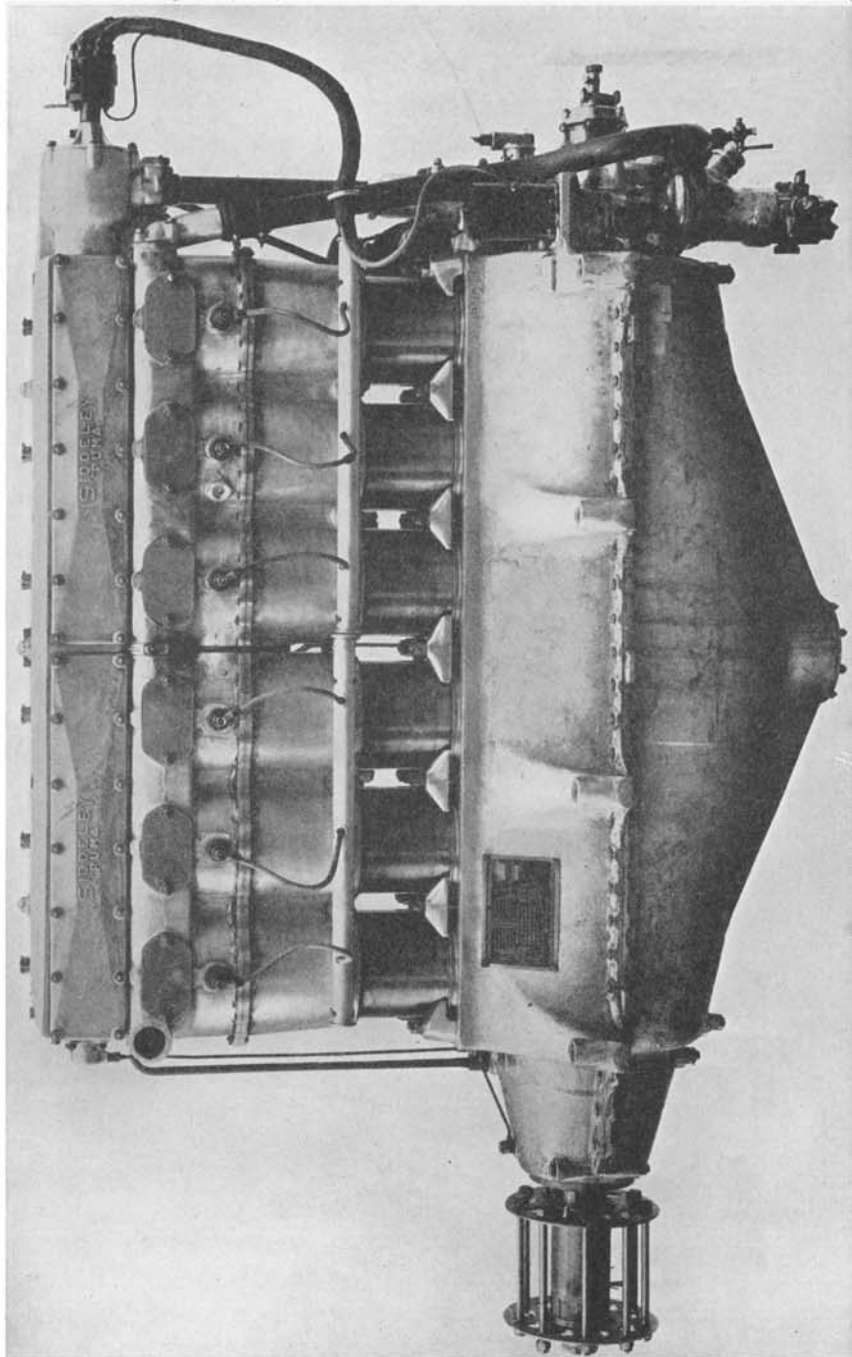


Fig. 204.—Siddeley Puma Engine

the oil consumption 0.03 lb. per B.H.P. hour, a gross consumption of 0.53 lb. per B.H.P. hour. The weight of the engine complete with radiator and cooling system is 781 lb. or 3.25 lb. per horse-power.

The gross weight with fuel and oil for six hours (exclusive of

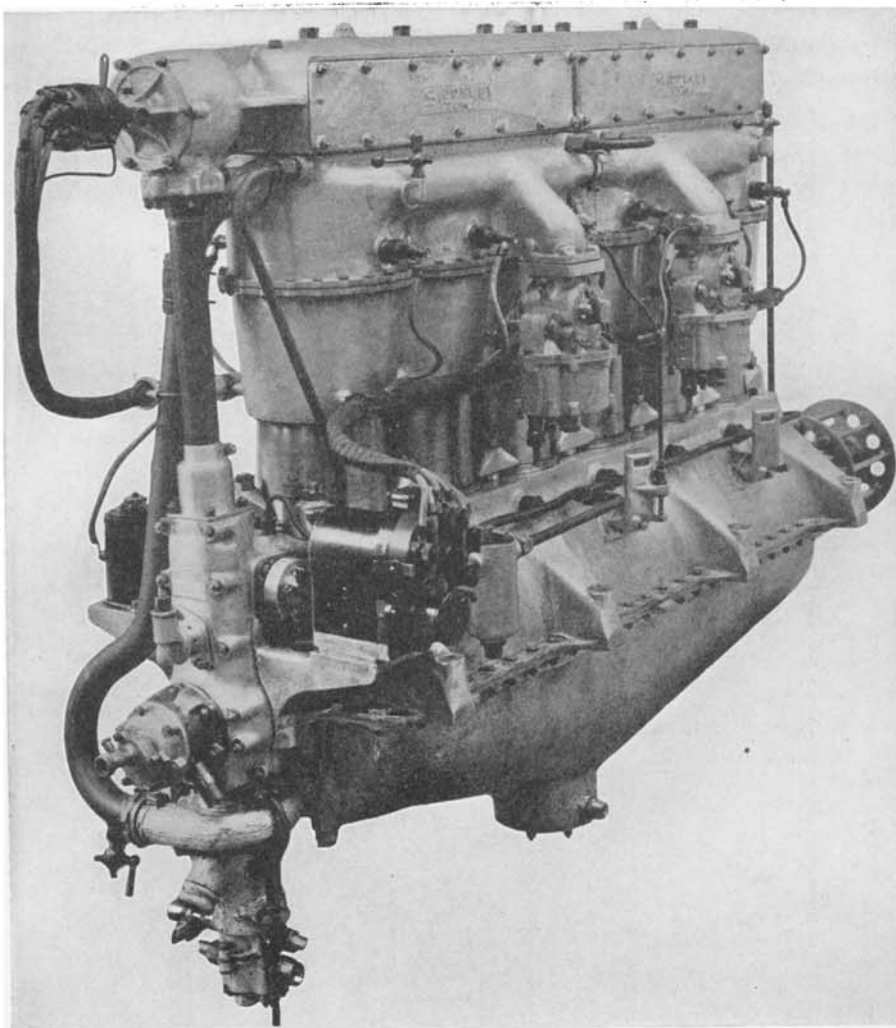


Fig. 205.—Siddeley Puma Engine

tanks) amounts to 1443 lb. or 6.01 lb. per horse-power. This engine is essentially a plain and straightforward piece of design, simple alike in manufacture and in handling.

The engine shown in figs. 186 and 187, also in the sectional drawings figs. 206 and 207, is the Napier Lion. It has twelve cylinders

each of five-and-a-half-inch bore and five-and-one-eighth-inch stroke arranged in three groups of four cylinders operating on

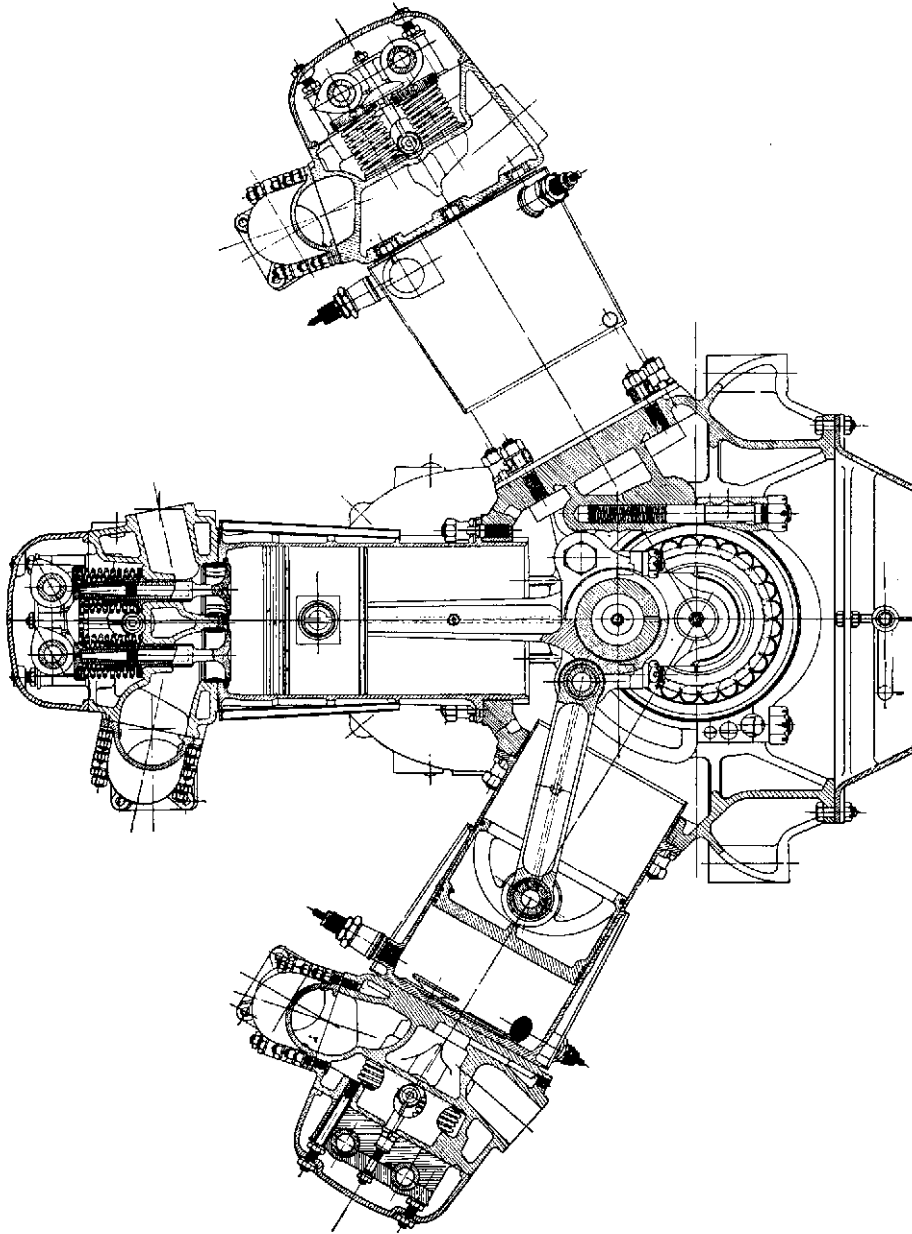


Fig. 207.—Section of Napier Lion Engine

a plain four-throw crank. It develops 450 B.H.P. at its normal crankshaft speed of 2000 R.P.M. Its weight complete with radiator, cooling system and speed reduction gearing is only 1134 lb. or

2.52 lb. per horse-power. Its fuel and oil consumption per B.H.P. hour are given as 0.495 and 0.022 lb. respectively.

The gross weight of the engine complete with all necessary gear and fuel and oil for six hours' flight is 2534 lb. or 5.63 lb. per horse-power.

In fig. 208 is shown the 350 B.H.P. Rolls-Royce Eagle engine, a twelve-cylinder Vee type, having cylinders of four-and-a-half-inch bore and six-and-a-half stroke, with a normal crankshaft speed of 1800 R.P.M. This engine, which was developed by Messrs. Rolls-Royce during the war, proved to be undoubtedly the most satis-

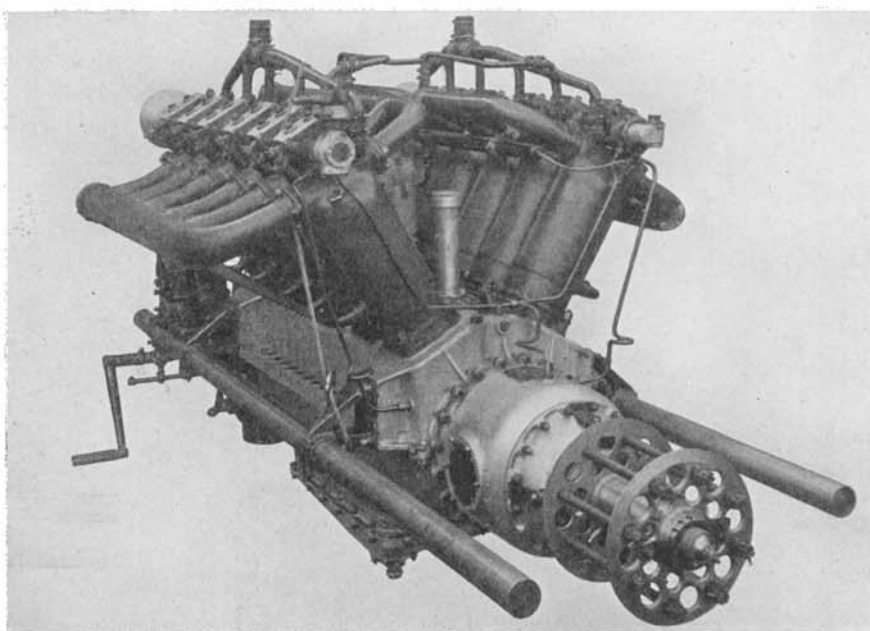


Fig. 208.—Rolls-Royce Eagle

factory and reliable engine in the hands of the Allies, and was of great value, not only on account of its magnificent performance, but perhaps even more because of its encouraging effect on the *moral* of the Allied pilots. Official records compiled in France during the war show that the average number of hours flown by these engines between overhauls was 103.2, or very nearly double that of any other aero-engine used in the British service. This engine, also, is of interest because it is at once probably the most complicated and quite the most reliable engine yet built for aircraft.

Its weight complete with epicyclic speed reduction gear, radiator

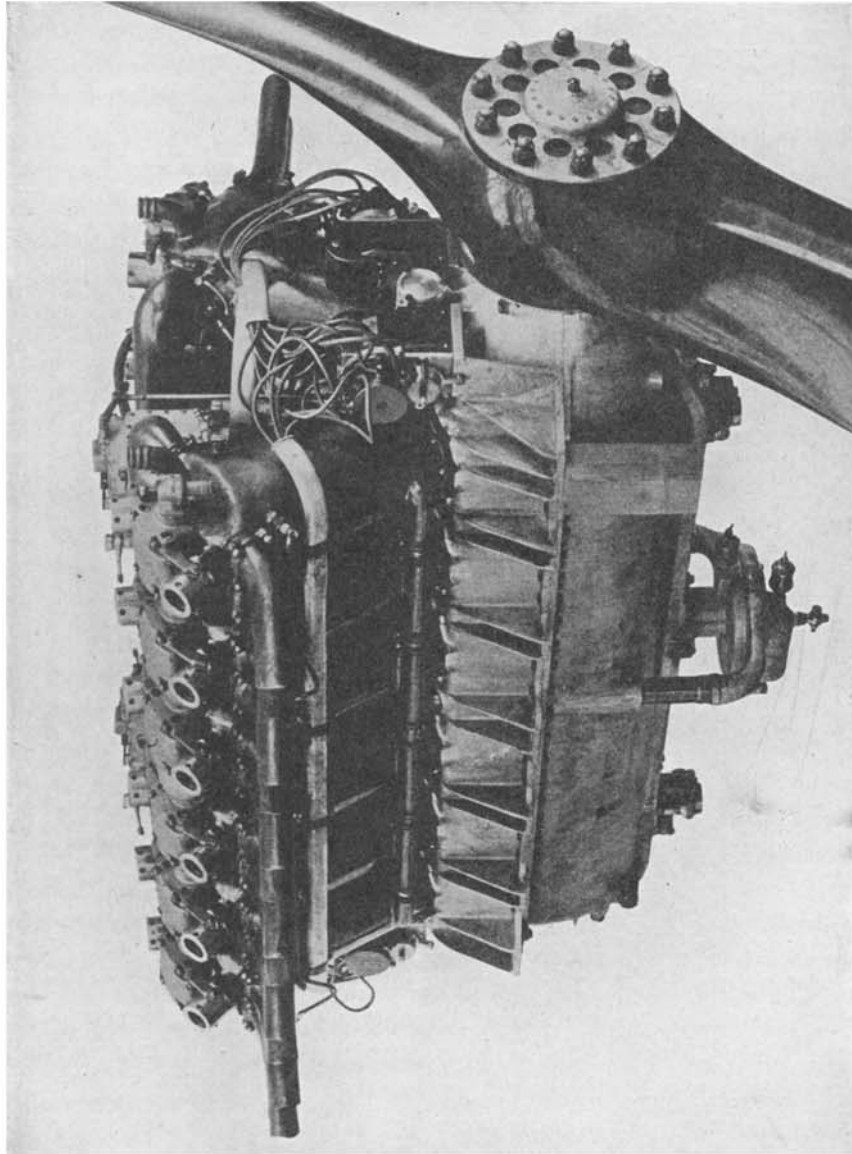


Fig. 209.—600 H.P. Fiat Engine

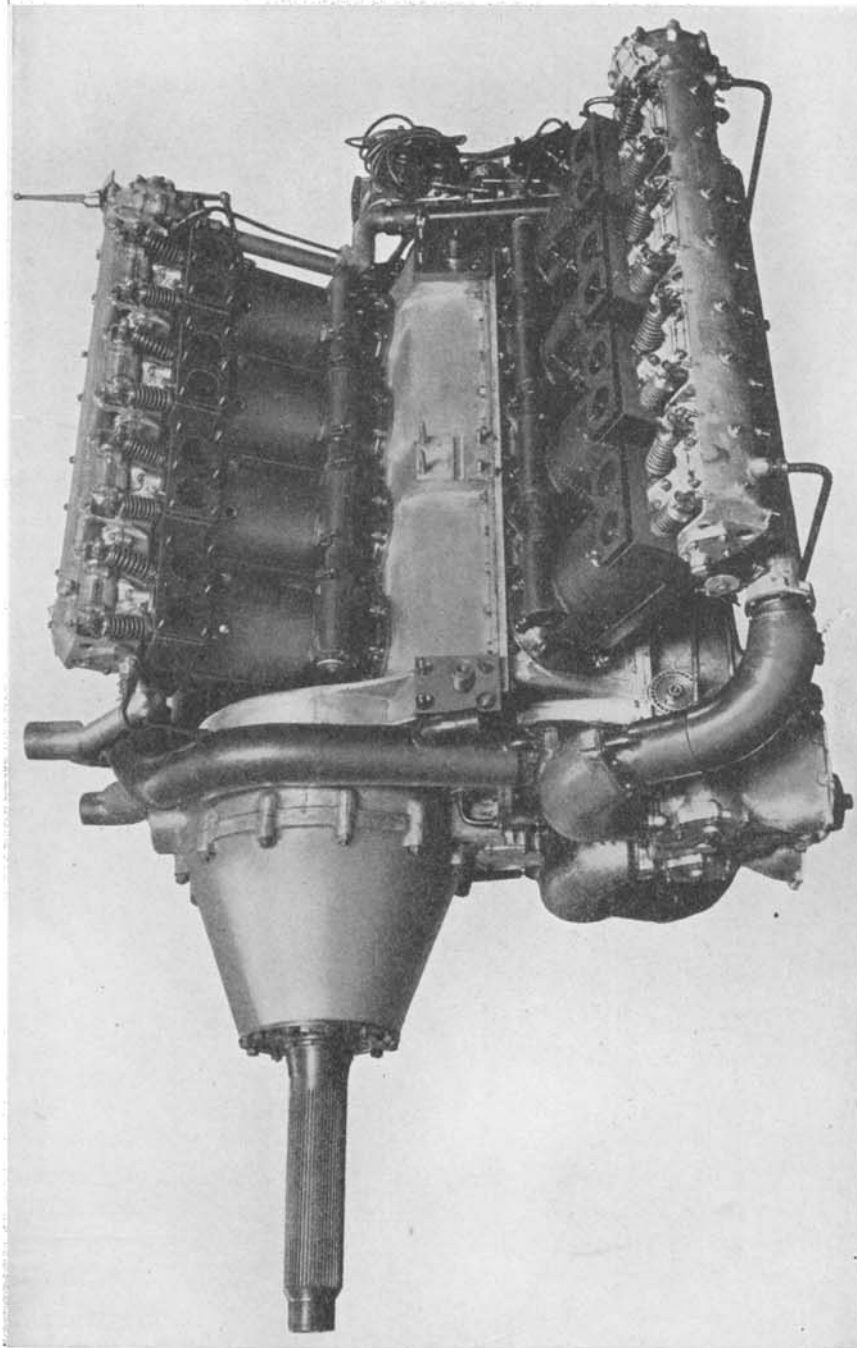


Fig. 210. —1000 H.P. Napier Cub Engine

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cooling system, &c., is 1177 lb. or 3.37 lb. per horse-power. The fuel and oil consumption are given as 0.50 and 0.028 lb. per B.H.P. hour respectively, so that the weight complete for a six-hour flight is 2287 lb. or approximately 6.5 lb. per horse-power.

In fig. 209 is shown the 600 B.H.P. Fiat twelve-cylinder Vee engine, which also may be taken as a fairly typical example of the

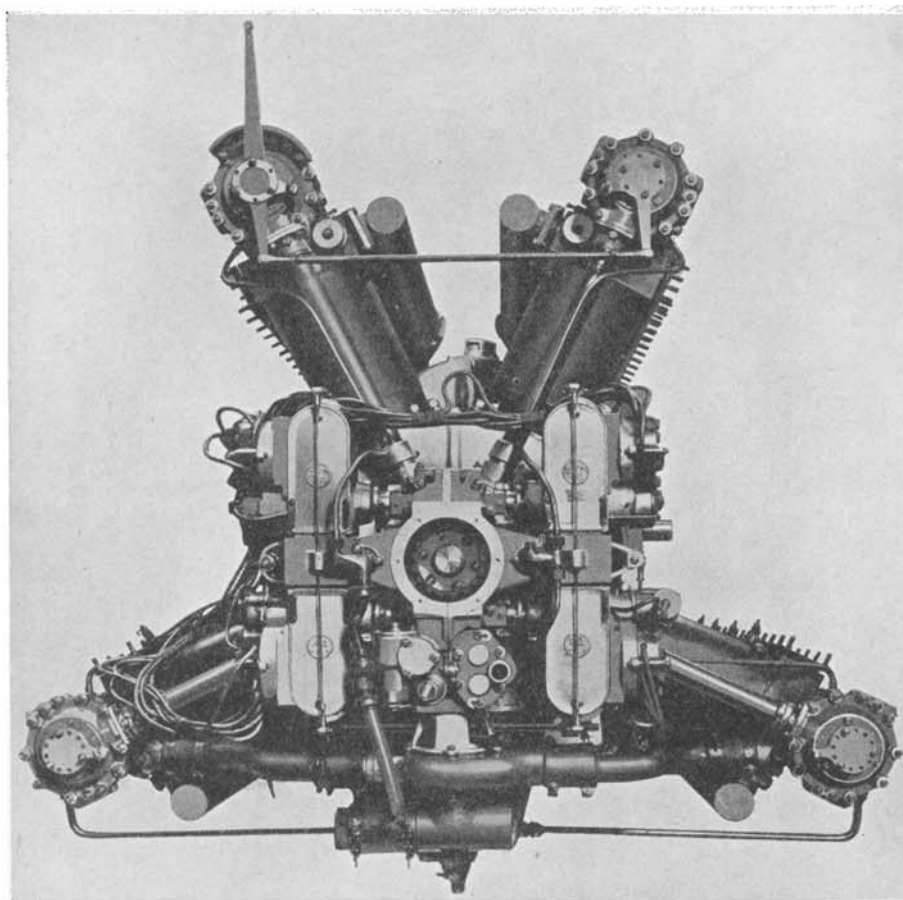


Fig. 211.—1000 H.P. Napier Cub Engine

type of engine developed by the Allies during the latter phases of the war. In figs. 210 and 211 are shown photos of the 1000 H.P. Napier Cub engine, probably the largest engine which has yet flown successfully.

The engine shown in fig. 212 is the 500 B.H.P. Benz engine; it is of particular interest because it marks the departure from the apparently settled German policy to adhere to the straight line six-



cylinder engine for military purposes. During the closing phase of the war, the Germans evidently began to find that their policy could not be adhered to in face of the very large engines which were

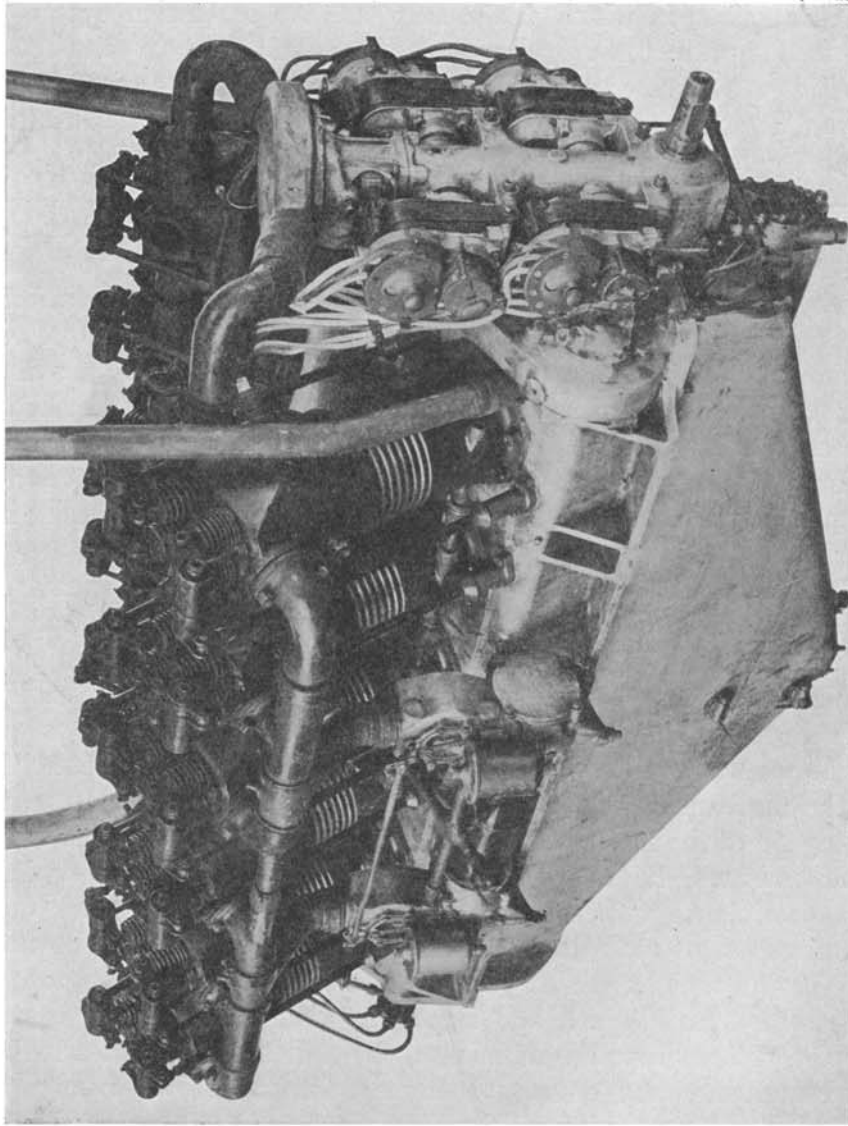


Fig. 212.—500 H.P. Benz Engine

being developed by the Allies, and this and a few other similar engines in course of development during 1918 bear evidence that they contemplated paying the Allies the compliment of following their lead.

**Aero-engines for High Altitudes.**—As a broad general-



ization the means of retaining the power output of aero-engines at high altitudes may be divided into two groups, one in which the power output is maintained by maintaining, or nearly maintaining, ground level density of the charge in the induction system, and the other, in which the density in the induction system is not so maintained, but the relative power output at the lower densities is increased by increasing the expansion ratio and so getting more useful work from a given weight of charge. In other words, the former system aims at operating with an artificially dense atmosphere at high altitudes, and the other caters for an artificially attenuated atmosphere near the ground, advantage being taken of the lower density to employ a greater expansion ratio and to obtain thereby an increase in thermal efficiency. The former system affords a means of maintaining the power output at any altitude at present attainable, though at some cost in efficiency. The latter provides only for a limited maintenance of power ; but, on the other hand, it affords a considerable gain in fuel economy.

In addition to these two general systems there are also certain possible compromises between the two, which will be considered later.

With regard to the first method, that of increasing artificially the density in the induction system, this can best be accomplished by the use of a turbo-blower driven either mechanically from the main engine, by a separate engine, or by means of an exhaust driven turbine. Such a system has the advantage that the full ground level torque can be maintained at almost any height, for the limit is set solely by the mechanical strength of the engine and by its capacity for getting rid of the heat generated in a highly attenuated atmosphere. It necessitates, however, the use of a variable pitch propeller. It is possible to obtain, at high altitudes, an actual power output in excess of that developed at ground level, for it is obvious that if the weight of air per cycle be maintained, the torque also will be maintained, and since the external resistance to the rotation of the propeller diminishes, the engine will run at a higher speed and therefore develop a higher power output, even after deducting the power required to drive the turbo-blower. Under such conditions, however, the flow of heat to the cylinder jackets, &c., is increased, while the capacity of the radiator or cooling fins is reduced, at all events while climbing, owing to the reduced density of the surrounding air, and, though the lower air temperature tends to balance this to a limited extent, it is necessary to provide a much

larger radiator. At first glance this system of direct supercharging appears to afford the simplest and easiest solution of the problem, but on closer examination it will be found to present many difficulties.

In the first place, the efficiency of the best turbo-blowers, though relatively high, is actually only about 55 per cent to 60 per cent, so that the power absorbed by them is a very considerable item, especially when considered in terms of fuel consumption. In the second place, such blowers, whether driven mechanically or otherwise, must necessarily run at a very high speed, generally from 20,000 to 30,000 R.P.M., and this, in itself, introduces very serious mechanical difficulties.

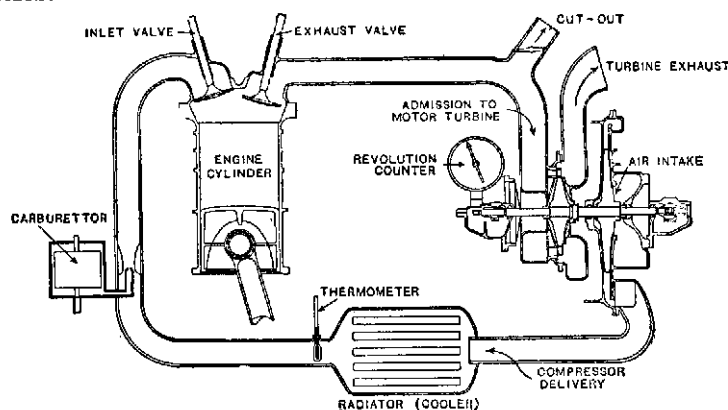


Fig. 213.—Arrangement of Turbo-compressor Supercharger

While the earlier experiments with supercharging were, for the most part, carried out with mechanically-driven blowers, opinion in this country appeared at one time to be veering in favour of exhaust-driven blowers such as those shown in figs. 213, 214, and 215, partly because of the many difficulties encountered in the actual mechanical drive, and partly because the exhaust-driven turbo-compressor can the more readily be adapted to meet the varying conditions as regards atmospheric density, since its speed is not directly dependent upon that of the main engine. So far as purely mechanical problems are concerned, it is doubtful whether it is easier to operate an exhaust-driven turbine at, say, 30,000 R.P.M. in an atmosphere of exhaust products at a temperature about 1100°–1200° F. than it is to drive a clean and cool air blower by suitable gearing at the same speed. The over-all efficiency of a combined exhaust turbine and blower cannot very well be determined, but at best it must be low, for, on purely mechanical grounds, it cannot

operate at a very high temperature or at a sufficiently high speed, with the result that a serious proportion of the work done in

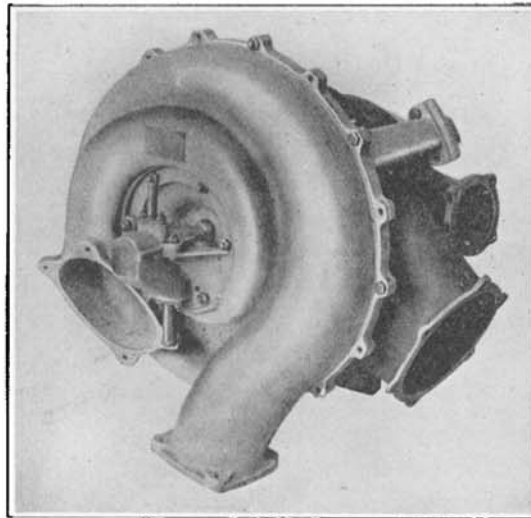


Fig. 214.—Turbo-compressor for Supercharging Aircraft Engine

compressing the air must appear as back pressure on the pistons of the main engine. Further, the resistance to the free flow of the exhaust products results, even at the best efficiencies so far attained, in the retention in the combustion chamber of hot residual products at a pressure considerably in excess of that of the entering charge (actually the best results so far obtained show an increased back pressure of about 3 lb. per square inch

above the air pressure). Also the very serious problem of cooling the air after compression is still further aggravated by the addition, by

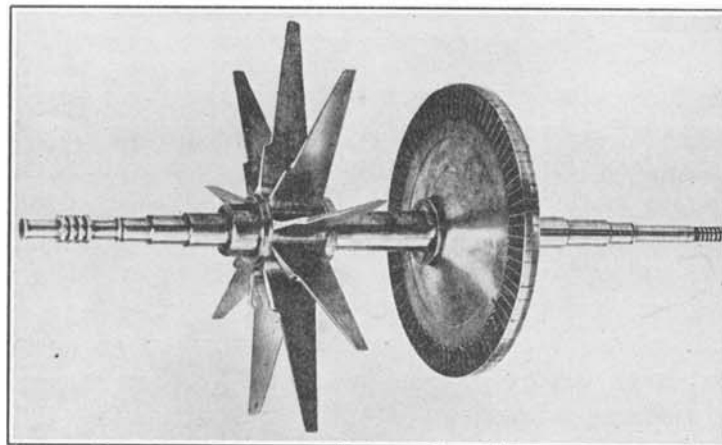


Fig. 215.—Rotor for Turbo-compressor for Supercharging Engine

conduction, of some heat from the exhaust turbine. At the best of times the removal of a large amount of relatively low temperature heat is a troublesome problem, involving large radiating surfaces

and therefore increased weight and head resistance. The exhaust-driven blower has, however, one outstanding advantage over the mechanically driven, namely, that the variation in the impeller speed at different densities is effected automatically. With regard to mechanical driving, the chief difficulties which arise are those due to cyclical changes in the angular velocity of the tail end of the crankshaft, from which the blower is usually driven; to sudden changes in the mean speed of the crankshaft, due to throttling down or opening up suddenly; and to faulty alignment, due to the blower not being built as an integral part of the crankcase. Most of these difficulties can probably be overcome by the provision of suitable dampers, flexible couplings, &c.

When, as in some of the large German machines, the blower is driven by means of a separate engine devoted to that one task alone, most of the mechanical difficulties disappear, and although at first sight it may seem very cumbersome, costly, and heavy to use a separate engine, yet there is a good deal to be said in favour of it, at all events in the case of large installations.

Direct supercharging is probably the only possible means of obtaining any really large increase in power at high altitudes, and, as such, it is extraordinarily valuable; but, however it be applied, it is neither simple nor easy. For very high altitudes, it is probably quite as important to apply supercharging to the aviator as to the engine; and when supercharging is employed, the possibility of enclosing both the pilot and the engine in a light pressure tight casing is worth considering seriously, for both are equally in need of oxygen.

The alternative method of increasing the output of aero-engines at high altitudes, by increasing the compression-expansion ratio, aims more particularly at an increase in fuel economy rather than in power, for the weight of charge taken into the cylinder per cycle is not increased, but, on the other hand, more useful work is obtained from a given weight of air, since it is expanded further. This gives an increase both in power output and fuel economy, though the increase in the former is comparatively small as compared with that obtainable by supercharging. There are, however, a good many advantages in connection with this system. With ordinary fuels, the limit of compression and expansion is set by the tendency of the fuel to detonate and ultimately to pre-ignite. This depends mainly upon the chemical constitution of the fuel, but it depends also, as has been explained, upon the maximum flame temperature, the pressure of compression, upon the form of combustion chamber,

and the position of the ignition plug therein. For a fuel of any given chemical constitution the tendency to detonate will become less as the altitude is increased, for both the temperature and pressure of compression will be reduced, as also the maximum flame temperature, which will be reduced in sympathy. It is found that, while the ordinary aviation petrol will tend to detonate at any compression ratio in excess of about 5 : 1 at ground-level density, at about 12,000 feet a compression ratio of 7 : 1 may be used with, at least, equal freedom from detonation. Actual experiments on a variable compression engine have proved that increasing the ratio of compression or expansion from 5 : 1 to 7 : 1 increases the indicated thermal efficiency from 32 per cent to 37·5 per cent, a gain of 16·5 per cent, which corresponds very closely indeed with the theoretical figure predicted by Tizard and Pye. When running with a ratio of 7 : 1 the heat stresses are somewhat reduced, and although the maximum pressure on the pistons is higher, both the pressure and temperature of the gases leaving the exhaust valves are substantially lower—a very important consideration from the point of view of reliability.

The principal difficulty in the way of employing a very high compression engine for high-altitude work lies in operating such an engine at or near ground level. This is so serious a difficulty that, unless some heroic means be adopted, it becomes almost impossible to leave the ground at all. There are several possible ways of attacking the problem. Among these are:

(1) By throttling when at ground level, in order to reduce both the pressure of compression and the maximum flame temperature, the latter because of the greater relative proportion of inert exhaust products to fresh charge, and because of the reduced density generally.

(2) By holding the inlet valve open during a portion of the compression stroke, so that while the expansion is retained, the compression temperature and pressure are reduced.

(3) By adding inert exhaust products, in order both to reduce the maximum flame temperature and the maximum pressure.

(4) By using a special fuel mixture at or near ground level.

With the exception of the last named, all these methods have the disadvantage that they reduce the available power at or near ground level, even when compared with a normal engine having a compression ratio of 5 : 1.

The method of throttling a high compression engine at or near

ground level is very undesirable—not only is it dangerous, but if anything approaching a 7 : 1 compression ratio is used with ordinary aviation petrol, the power output available is not nearly sufficient. The curves, fig. 216, show the maximum indicated mean pressure obtainable with, in this case, a somewhat inferior aviation petrol detonating normally a compression ratio of 4·85 : 1. The compression ratio was gradually raised and the throttle closed just sufficiently

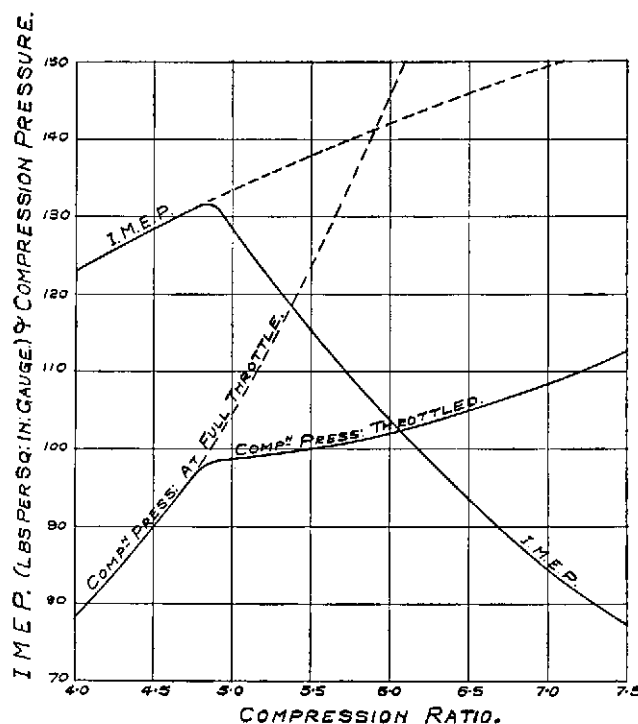


Fig. 216.—Curves showing I.M.E.P. and Compression Pressure with varying Compression and Throttling

to avoid detonation. It will be seen that at a compression ratio of 7 : 1 the available indicated mean pressure is only 85 lb. per square inch, corresponding in this case to a brake mean pressure of 70 lb. per square inch. This probably would not nearly suffice to raise the machine from the ground.

By the use of variable closing inlet valves in order to vary the compression ratio, somewhat better results can be obtained for various reasons, but even so, the weight of charge is considerably reduced and an extra mechanical complication is added. The method has, however, some substantial indirect advantages, and, as

compared with throttling, it is much safer and yields a somewhat greater power output at or near ground level.

By the addition of cooled exhaust gases, detonation can be suppressed and the maximum pressures reduced, at the least expense in power output of any of the three methods yet considered, but for a compression ratio of 7 : 1 the quantity of exhaust products required is so large that they have an adverse influence on the thermal

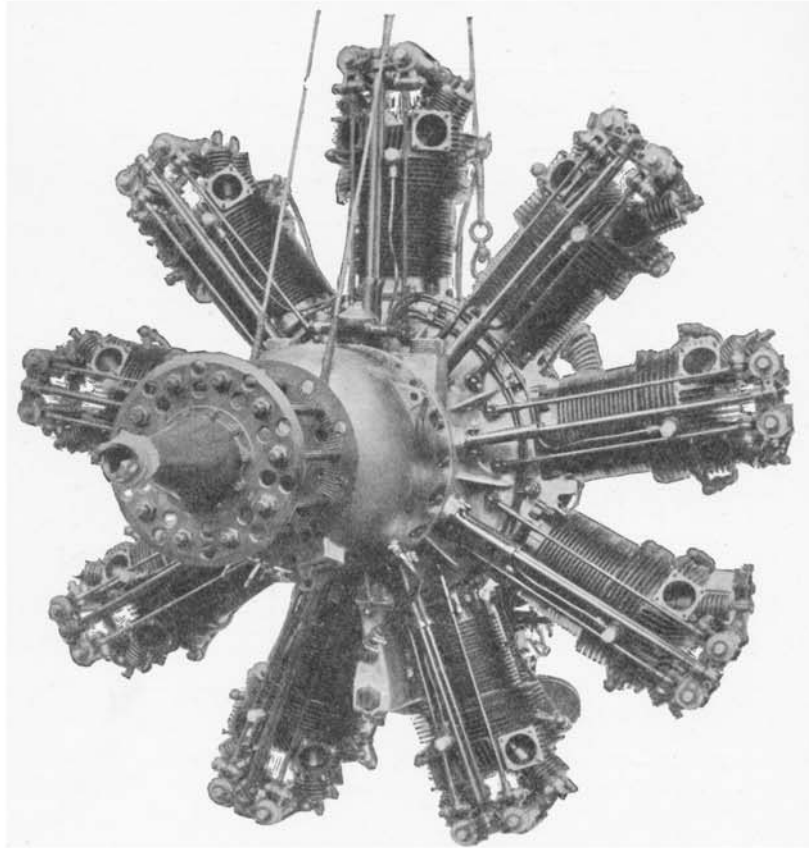


Fig. 217.—The Geared Bristol "Jupiter" Aero Engine

efficiency as well as on the power output. Also it appears essential that they shall be thoroughly cooled before admission to the carburettors or induction system, and this in itself is sometimes rather troublesome. It is, however, in all probability the best of the three methods considered.

By suitable treatment of the fuel, such as by the addition of toluene, &c., detonation can be eliminated entirely and the full power obtained at ground level, provided that the engine can with-

stand the excessive pressures involved. At first sight this might appear the simplest and best method, but on investigation it is very doubtful whether it is really practicable, because to withstand the very high maximum pressures involved by the use of a compression ratio of 7 : 1 the whole of the engine, and especially the reciprocating parts, must be strengthened and the weight increased

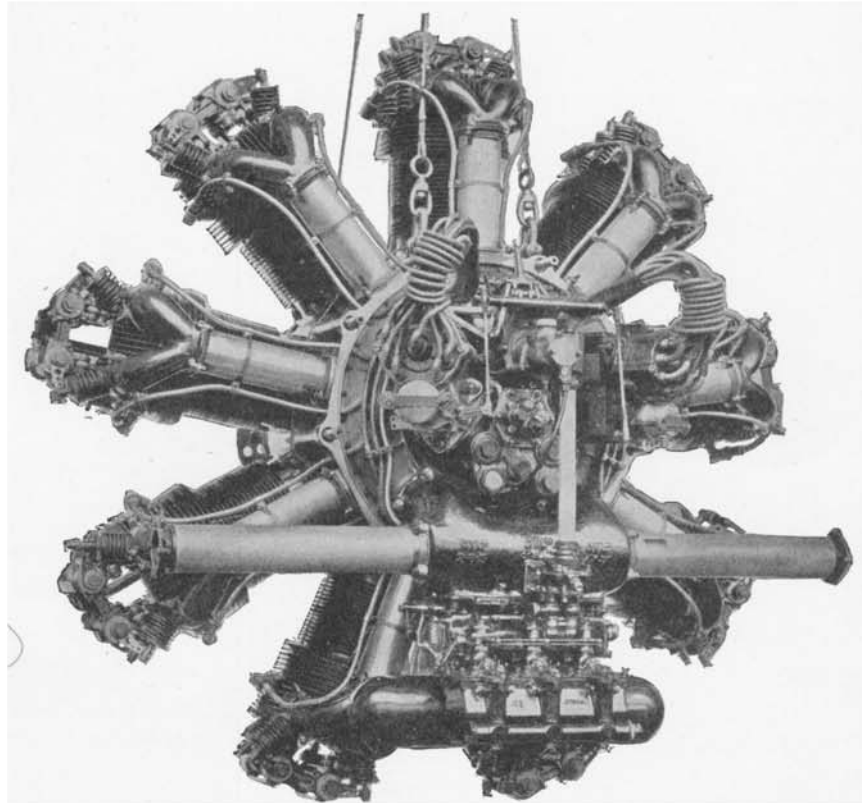


Fig. 218.—Rear View of the Bristol "Jupiter" Engine

very considerably. The use, however, of a fuel of lower flame temperature and higher latent heat, so that neither the temperature nor the maximum pressure are increased appreciably, such, for example, as an alcohol-acetone mixture, would appear very hopeful.

Since this chapter was prepared—in 1922—great progress has been made, but it has been restricted almost entirely to detail improvement. The three engines described in this chapter, namely, the Napier Lion, the Bristol Jupiter, and the Siddeley Jaguar, are still in manufacture, and are still the favourite engines both in the British



Air Service and for commercial transport. There has, however, been considerable development work in the way of small engines for light aircraft of which the most notable is the four-cylinder air-cooled Cirrus engine as fitted in the well-known de Havilland Moth. So far, however, as the larger and more powerful engines are concerned, one firm only, namely, Rolls-Royce, have developed a new type, while the other three makers mentioned above have contented

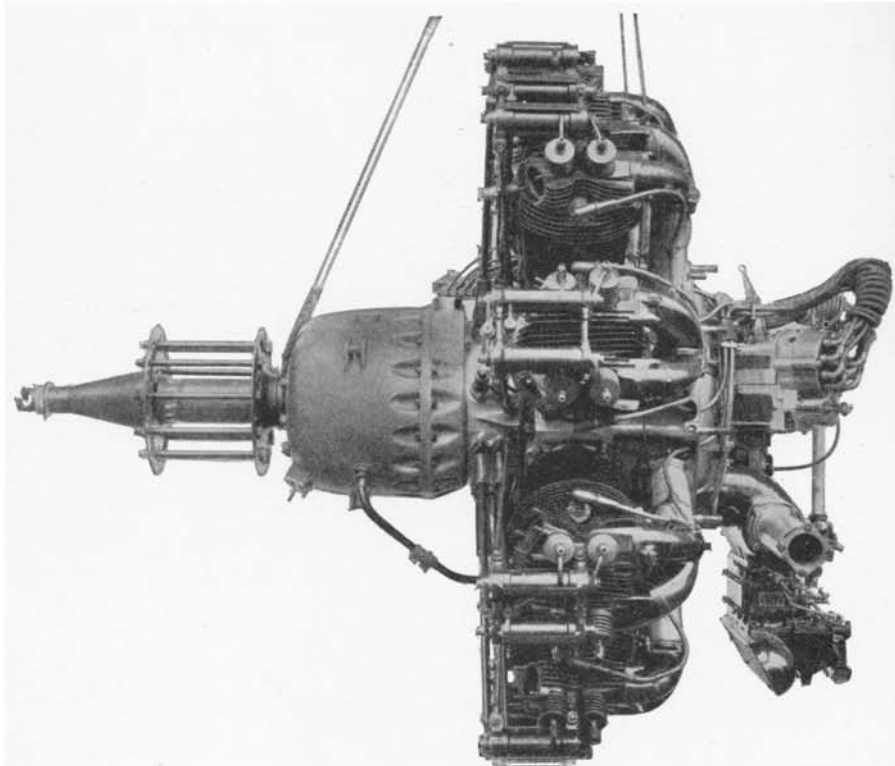


Fig. 219.—The Geared Bristol "Jupiter "

themselves with detail improvements to a type of engine which, so far as its general design is concerned, has remained substantially the same for ten years. By dint of painstaking development work in regard to detail design, the performance of these engines has, during this period, been increased by about 40 per cent, and their reliability, as judged by the period between overhauls, by something like 300 per cent. This has been attained, not by any radical changes, but solely by minute attention to detail and application of the progress which has been made in metallurgy, and in the technique of manufacture.

Since progress has resulted in the ability to run at higher crankshaft speeds, the need for a reducing gear has become all the more insistent, and to-day even radial engines are being so fitted. The greatly increased reliability of air-cooled engines has, to a large extent, robbed the water-cooled engine of any advantage it once held on this score, while the reduced vulnerability and freedom from "plumbing" are strong arguments in favour of the air-cooled engine. Where very high speeds are required the water-cooled engine, however, still reigns supreme, on account of its low head resistance, and it appears to be gaining a new lease of life from the employment

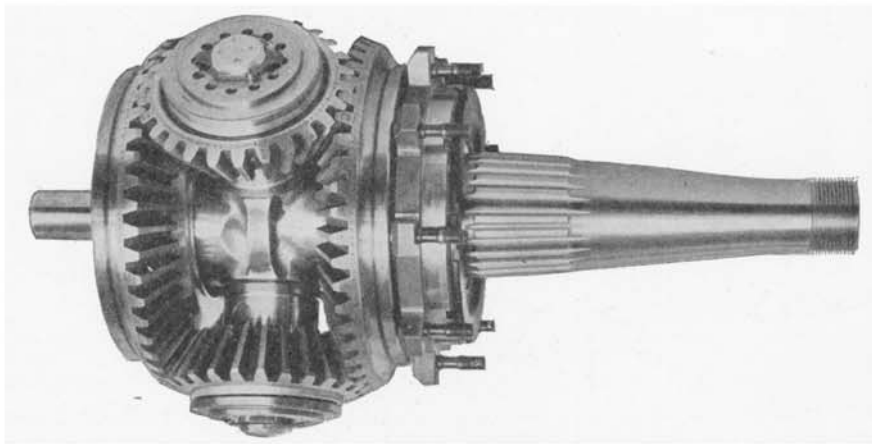


Fig. 220.—"Jupiter" Engine—Reduction Gear

of evaporative or steam cooling, in which the radiator becomes a steam condenser. Since the temperature difference between it and the surrounding air is greater, a considerably smaller radiator will suffice; moreover, since the pressure in the cooling system is maintained very slightly below atmospheric pressure it follows that the system is much less vulnerable.

The use of exhaust-driven superchargers has, as the author anticipated, proved disappointing, but gear-driven blowers of the high-speed centrifugal type are coming into extensive use. Such blowers are usually of the single-stage type fitted between the carburettors and the cylinders and geared to run at from ten to twelve times engine speed. Under such conditions, ground-level density can be maintained in the induction system and ground-level power developed up to an altitude of from 8000 to 12,000 ft. In air-cooled cylinder construction the use of forged Duralumin or Y-alloy cylinder heads screwed and shrunk on to a steel barrel has replaced

the early cast-iron or steel cylinders, while in water-cooled engines there is a tendency to avoid welding as far as possible in the cylinder

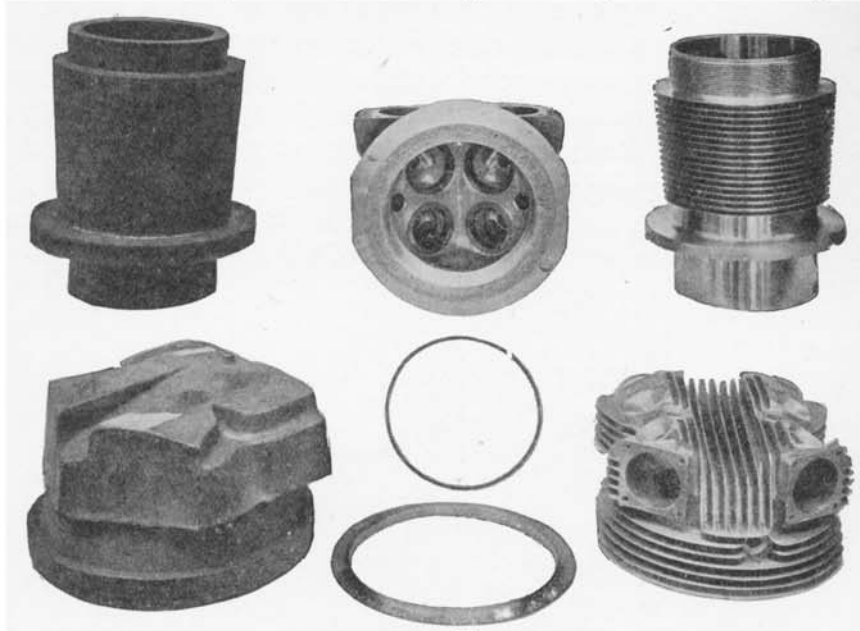


Fig. 221.—“Jupiter” Engine Cylinder and Engine Head

construction, to employ cast aluminium “block” cylinder heads with the valve seats screwed and shrunk into place and separate steel liners with aluminium water jackets.

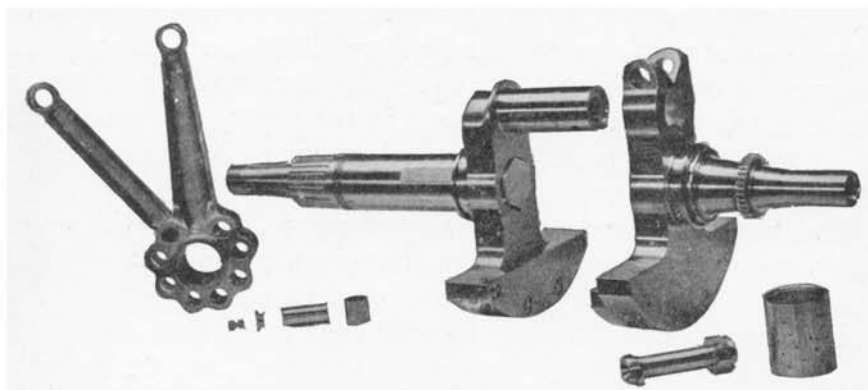


Fig. 222.—Crankshaft parts of the Bristol “Jupiter” Engine

The 1929 Bristol “Jupiter” engine, photos of which are shown in figs. 217, 218, and 219, will be seen to differ but little in external appearance from the 1921 version shown in figs. 200 and 201, but while

the cylinder dimensions are unaltered, the power output has been increased from 380 B.H.P. at 1575 to 525 B.H.P. at 2000 R.P.M. In view of the high speed, a 2:1 reduction gear of the concentric bevel type, as shown in fig. 220, has been added, and the total weight,

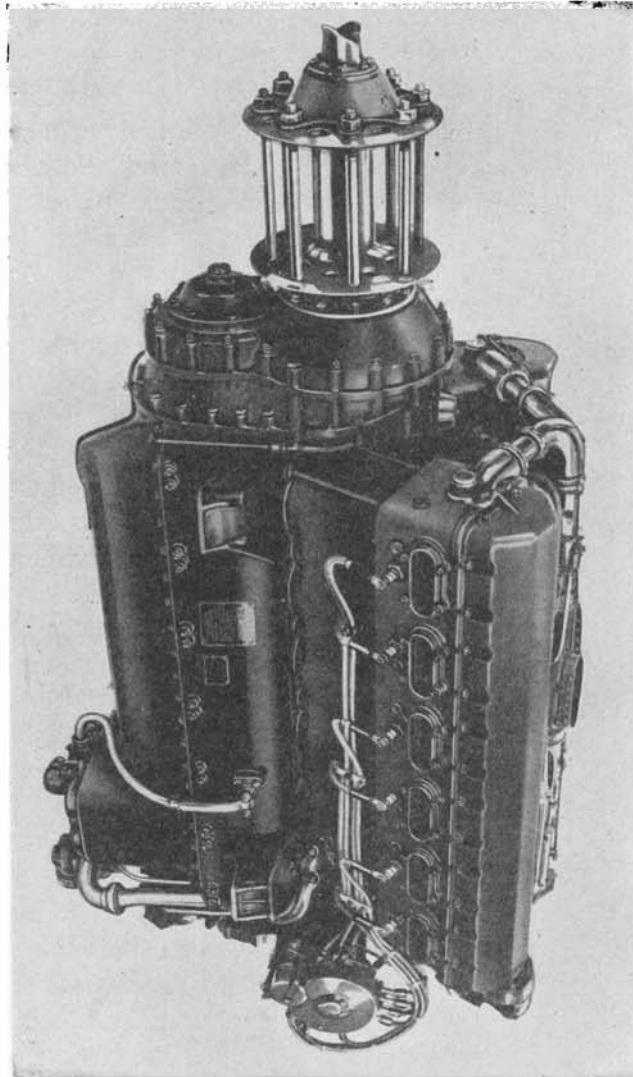


Fig. 223.—Rolls Royce "F" Engine

including that of the reduction gear, is only 880 lb. or 1.68 lb. per horse-power. The same engine without reduction gear and running at 1700 R.P.M. develops 480 B.H.P. with a total weight of 720 lb. or 1.56 lb. per horse-power. The principal changes which have brought about this large increase in performance are:—

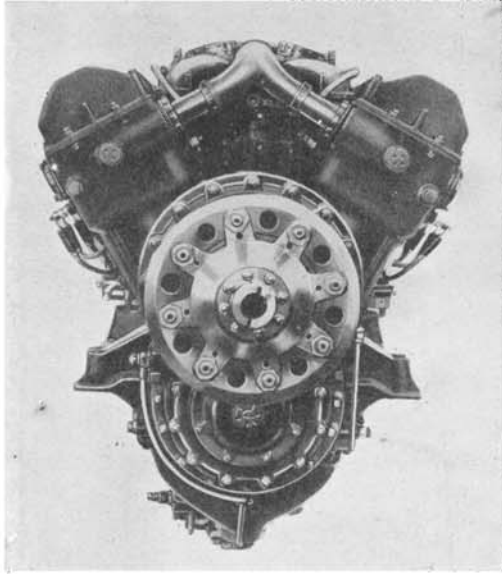


Fig. 224.—Rolls-Royce "F" Engine

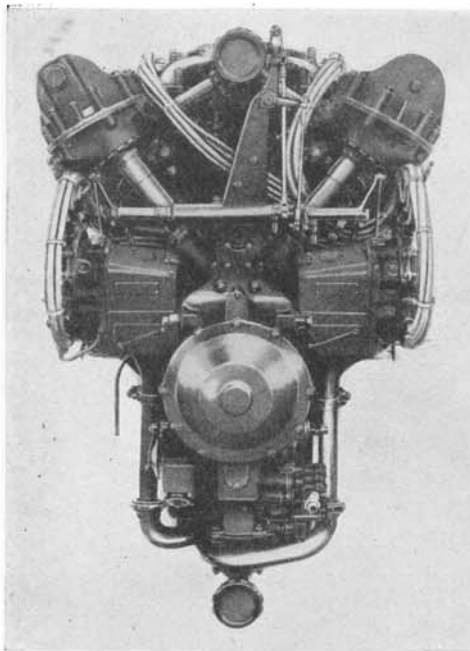


Fig. 225.—Rolls-Royce "F" Engine

(1) The flat poulitice-type cylinder head with vertical valves has been changed for an all-aluminium pent-roof head with inclined valves, thus giving better cooling, better volumetric efficiency, and better thermal efficiency, and allowing of a somewhat higher compression ratio being employed. To this change therefore is due the higher mean effective pressure. Fig. 221 shows the latest cylinder and head construction.

(2) The one-piece crankshaft with split big-end bearing has been exchanged for a built-up crankshaft allowing of an unsplit big-end eye for the connecting-rod, and making possible the use of a floating bush big-end bearing, a change which has made possible the large increase in speed from 1575 to 2000 R.P.M. Fig. 222 shows the latest design of crankshaft and connecting-rod assembly.

For high altitude work a gear-driven centrifugal fan is disposed in the rear-end cover of the crankcase, and driven through concentric gearing in which are incorporated slipping clutches to deal with velocity changes due to acceleration, or torsional vibration. The

capacity of this supercharger is such as to provide nearly ground-level density at an altitude of 12,000 ft.

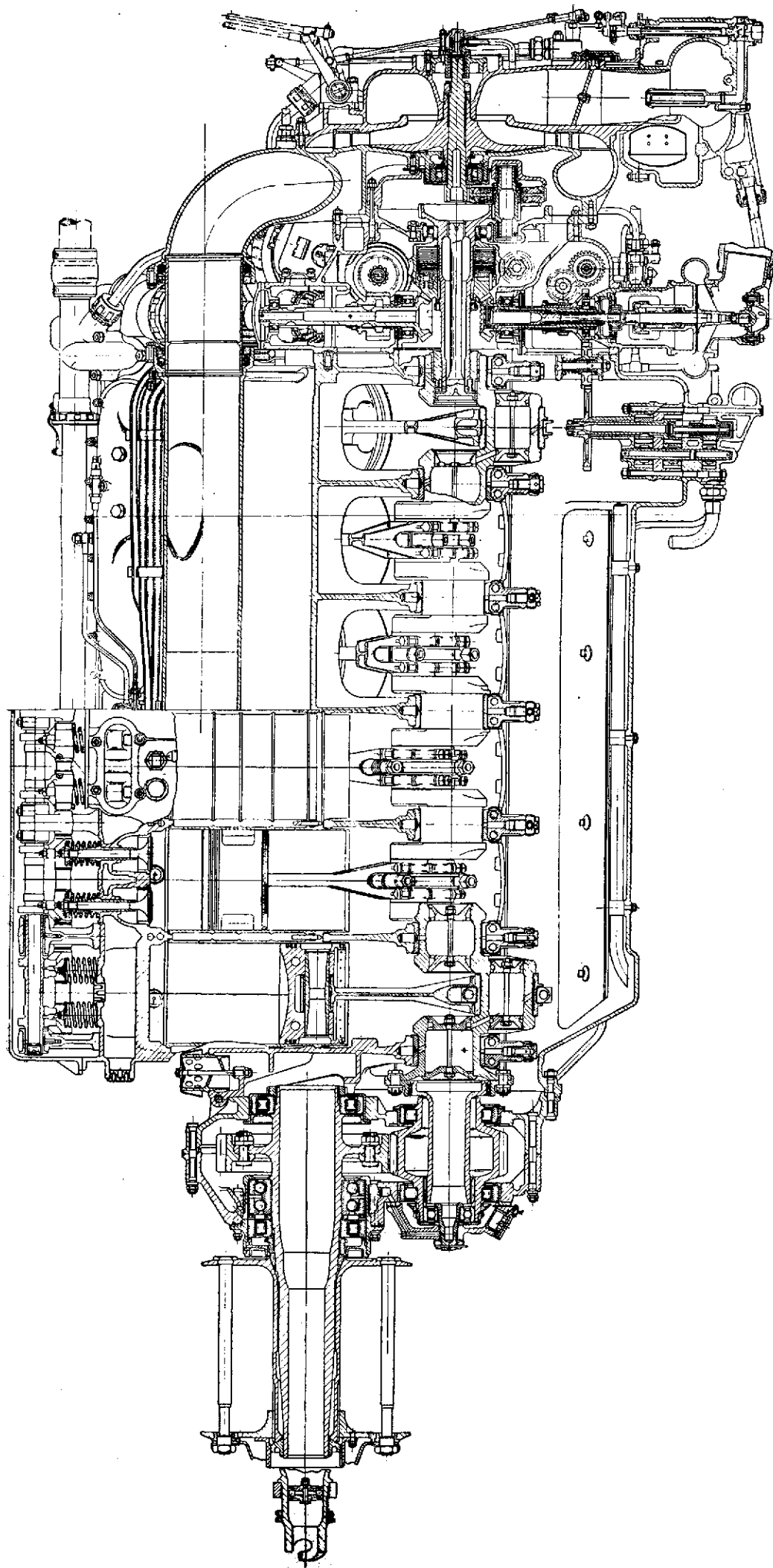


Fig. 226.—Longitudinal Section of Rolls-Royce Supercharged "F" Engine

Facing p. 352.

Apart from the above-mentioned changes other improvements which have been incorporated are:

(1) The use of a solid-forged in place of a cast aluminium alloy crankcase.

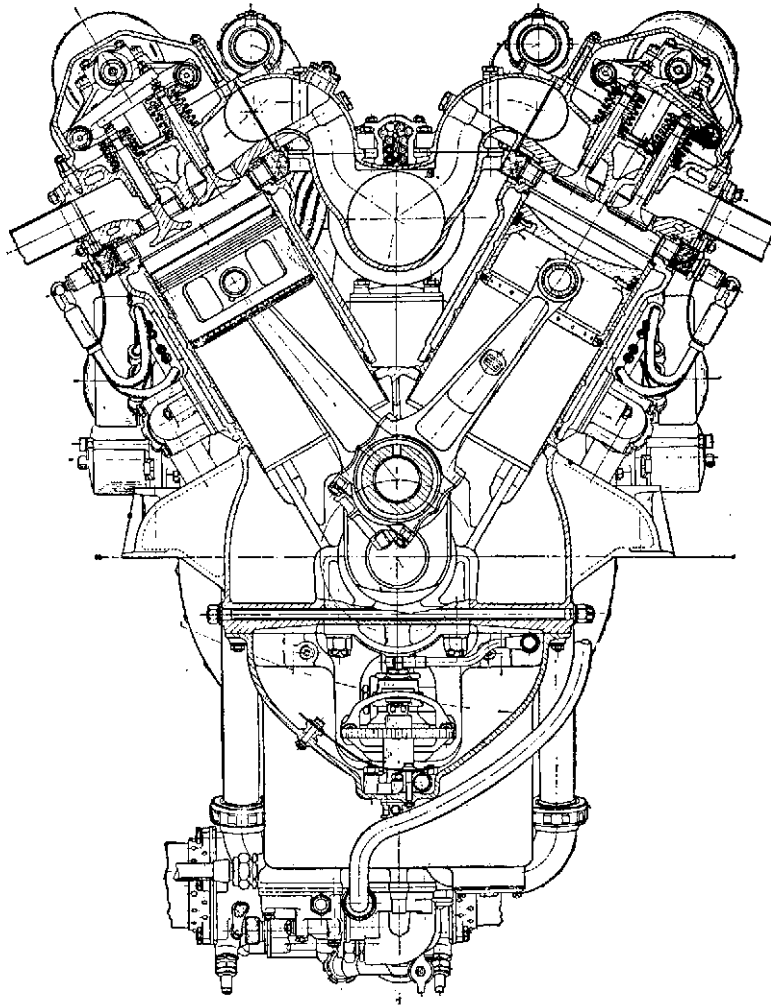


Fig. 227.—Cross Section of Rolls-Royce "F" Engine

(2) A general re-arrangement of the auxiliary drives at the rear-end of the crankcase to allow of accommodating the supercharger when required and generally to facilitate mounting in the latest types of aircraft.

In figs. 223, 224, 225, 226, 227 and 228 are shown photos and sectional drawings of the latest Rolls-Royce "F" type engine as used

(E 246)

in the most recent day bombing and fighting machines. A larger but generally similar version of this engine was fitted to the seaplane

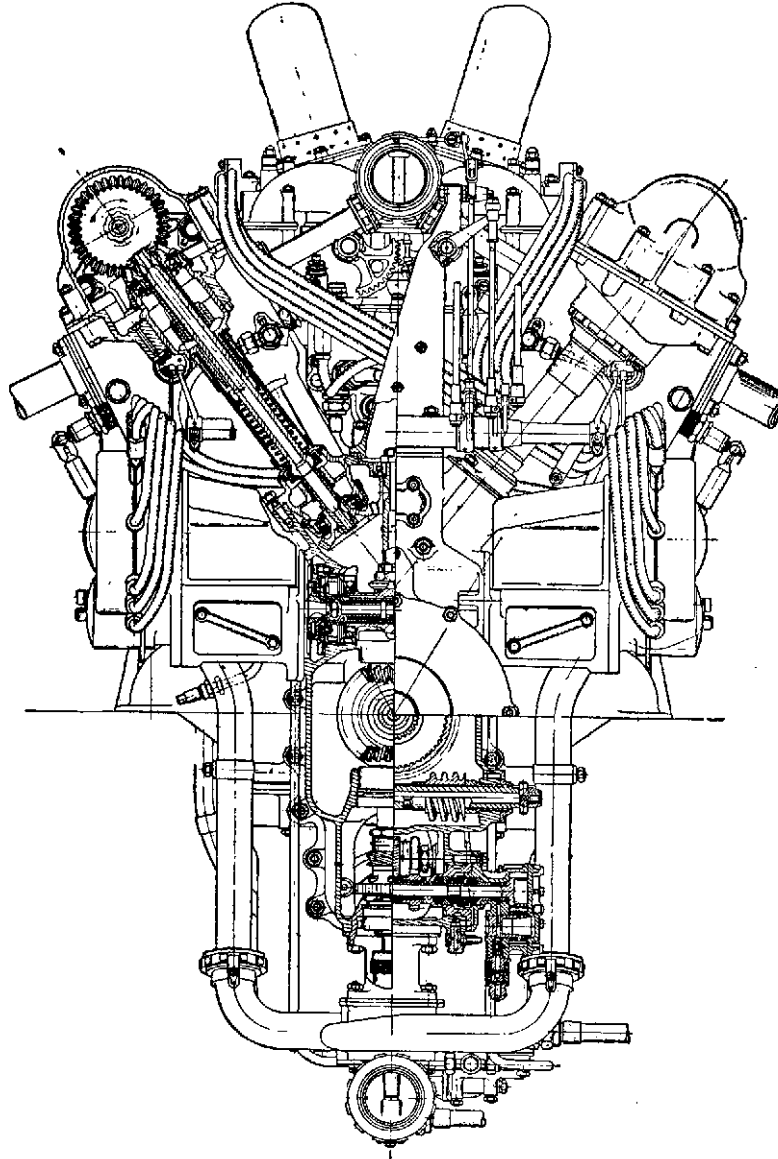


Fig. 228.—Rolls-Royce "F" Engine—Auxiliary Driving Gear

which won the Schneider Trophy in 1929 at a mean speed of 328.6 miles per hour, and the world's speed record of 357.7 miles per hour. The "F" engine has twelve cylinders each of 5-in. bore by  $5\frac{1}{2}$ -in.



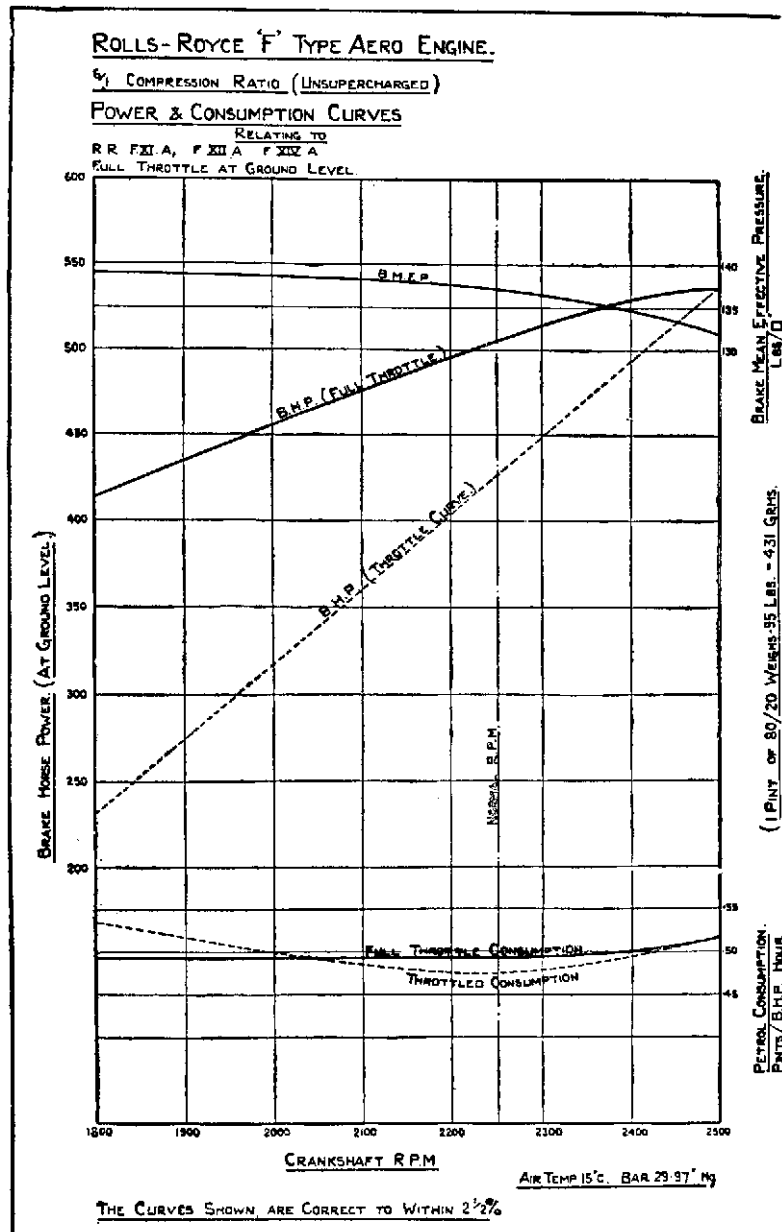


Fig. 229

stroke, and is fitted with or without a gear-driven supercharger of the centrifugal type. As a normally aspirated engine used with a compression ratio of 6.0 : 1, it develops a maximum of 540 B.H.P. at 2500 R.P.M. of the crankshaft. The power, M.E.P., and consump-

tion curves of the unsupercharged engine are shown in fig. 229. The "dry" weight of the engine is 865 lb. without, and 900 lb. with, the supercharger.

**Cylinder Construction.**—The six-cylinder heads of each block, together with the water jackets of the cylinder barrels are cast in a single piece in aluminium alloy; the valve seats, four per cylinder, are of bronze screwed into the aluminium head. The cylinder barrels themselves are separate thin open-ended steel tubes flanged and spigoted to the crankchamber and held in position by a series of long bolts running from well down in the crankchamber to the top of the cylinder head casting. They are thus clamped between the head and the crankcase. The joints between the cylinder barrels and the head are sealed by means of soft aluminium washers, while rubber rings are employed to seal the open end of the water jackets. The long through bolts already referred to pass through the water jacket, but are surrounded by aluminium tubes expanded into place. The complete cylinder block assembly is shown in fig. 230.

**Crankchamber.**—The crankchamber calls for no particular comment, beyond reference to the excellence of its design—though perhaps somewhat shallow it is very well ribbed and its torsional rigidity is enhanced greatly by the addition of horizontal bolts passing from side to side and through the lower bearing caps; these serve the dual purpose of tying the crankcase well below the gash formed by the crankshaft bearings and, at the same time, they nip the sides of the bearing caps and so prevent any tendency of the latter to "work". The bearing caps themselves are Duralumin forgings and the bearings mild steel shells lined with a very thin facing of white-metal.

**Connecting-rods.**—It will be seen that the connecting-rods are not articulated as in the Napier and most radial engines. One rod of each pair is forked, and bears directly on each crankpin; the second runs on the outside of the first. In the case of the forked rod the big-end bearing is of the so-called marine type, that is to say, the end of the connecting-rod is flat—though relieved in the centre—and separate blocks are fitted for the two halves of each bearing—a design which avoids any concentration of load immediately under the shank of the rod. This assembly is shown in detail in fig. 231.

**Pistons.**—The pistons which are machined from the solid are of forged aluminium alloy with three gas rings above the gudgeon pin and a single scraper ring near the bottom of the skirt.

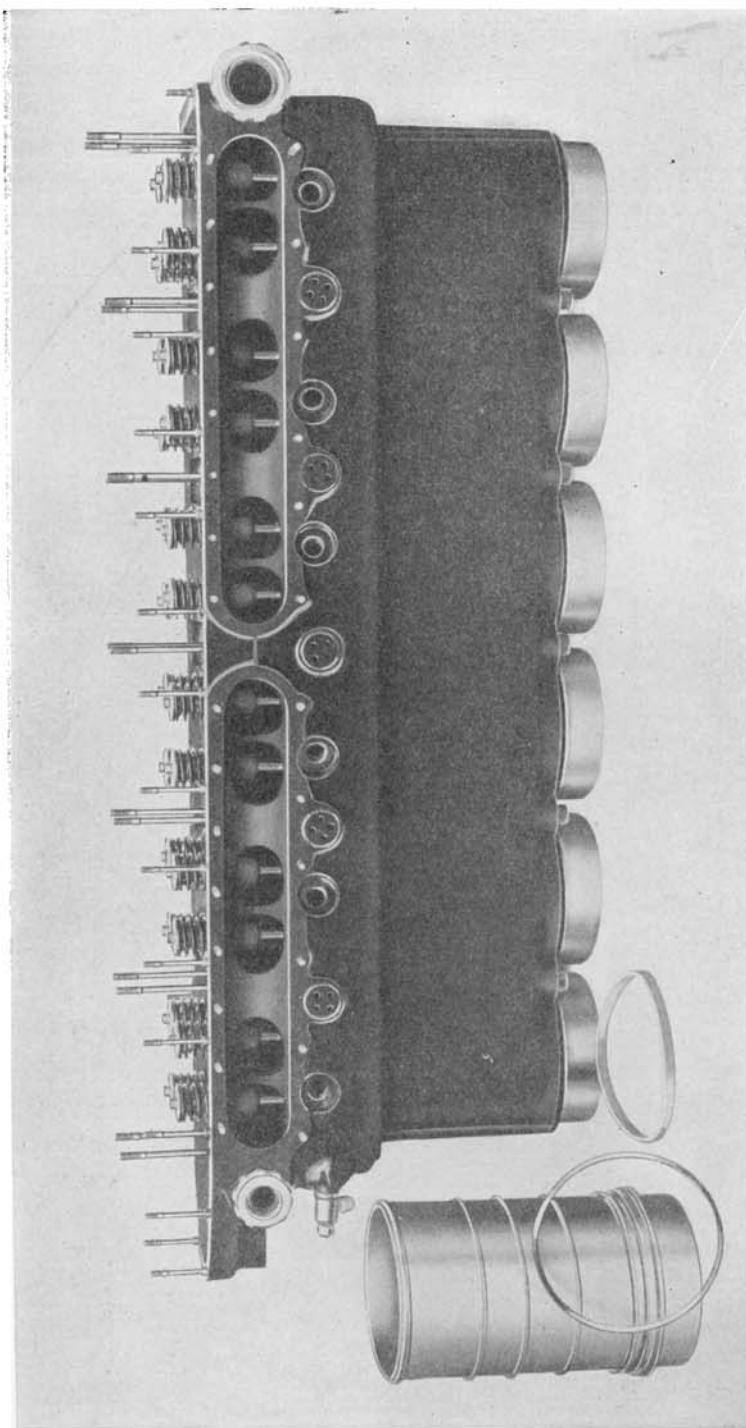


Fig. 230.—Rolls Royce 'F' Engine—Aluminium Cylinder Casting

**Crankshaft.**—The crankshaft calls for no particular comment except that it is a model of good proportions and of design generally.

**Reduction Gear.**—The reduction gear is particularly interesting and is again a fine example of the designer's skill. It will be seen that the driving pinion is carried entirely independently of the crankshaft, and is thus insulated from the slight deflections in the latter. The drive from the crank to this pinion is taken through a

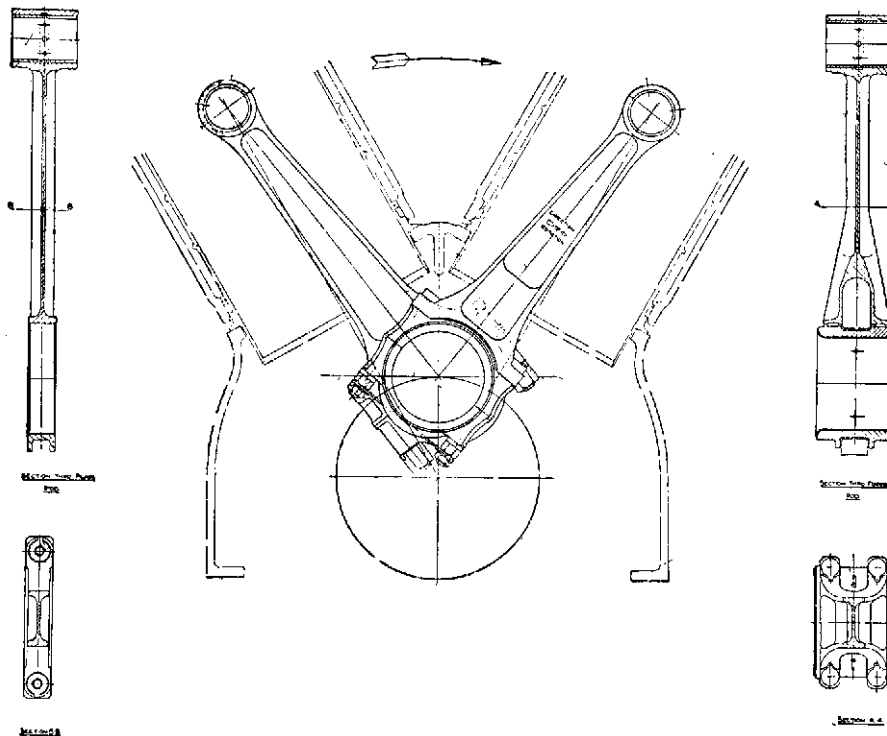
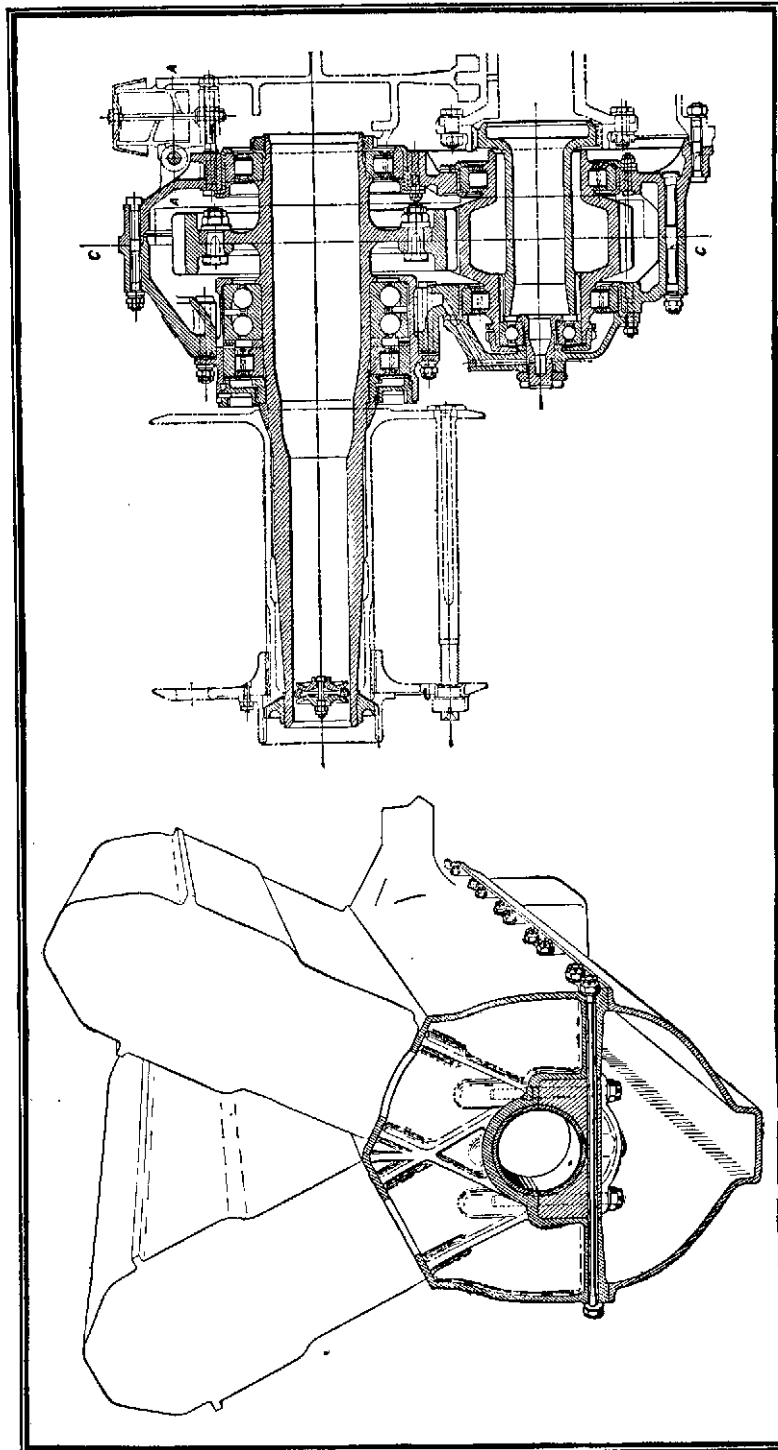


Fig. 231.—Rolls-Royce "F" Engine—Connecting-rod details

short floating cardan shaft which engages with the forward end of the pinion by means of splines and with the crankshaft by means of a toothed ring. By this means the correct alignment between the pinion and the driven member can be maintained entirely independent of the crankshaft, see fig. 232.

The camshafts and auxiliary mechanism are all driven from a single bevel wheel at the rear end of the crankshaft. Here again this wheel is not mounted on the crank itself but is supported in a separate bearing. As in the case of the driving pinion it engages with the crankshaft through the medium of a separate shaft drive, but in this case the shaft is made of small diameter in order to be



Main Reduction-gear Box

Fig. 232.—Rolls-Royce "F" Engine

Crankshaft Main Bearing

flexible torsionally. The outer end of this shaft is connected through a friction damper clutch to a hollow sleeve which limits the rotation of the flexible shaft to a predetermined amount before the sleeve

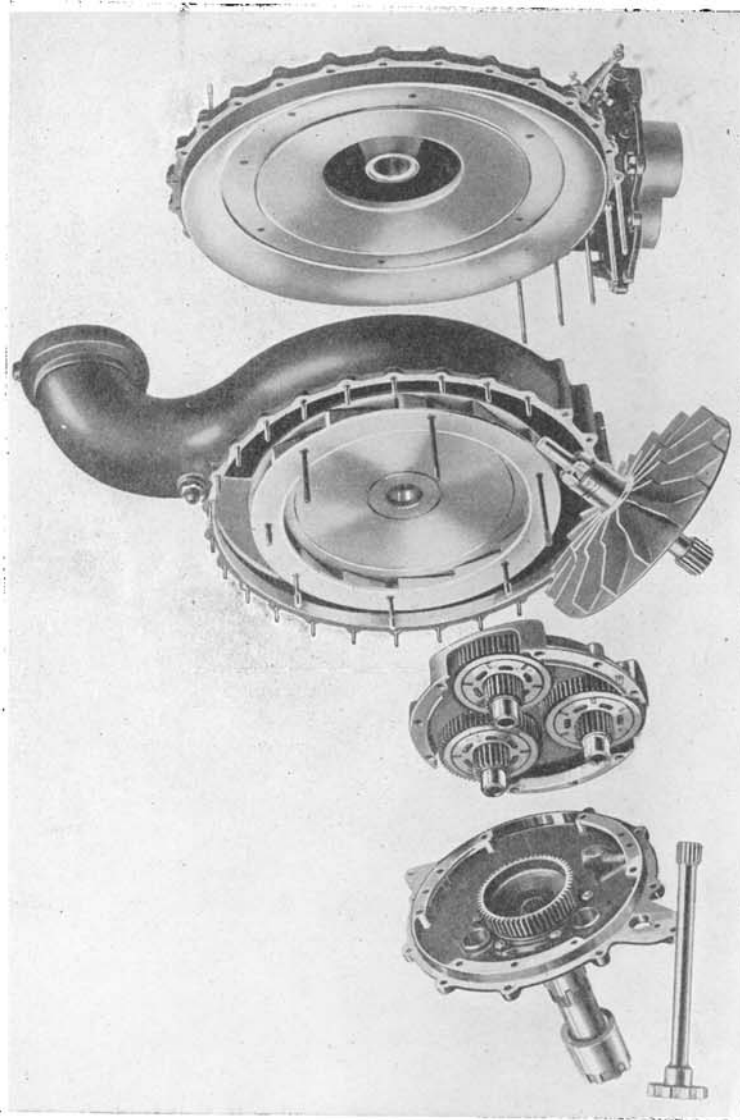


Fig. 233.—Rolls-Royce "F" Engine—Supercharging Parts

takes up the drive. By this means all the auxiliary mechanism is insulated from any twitching of the rear end of the crank. This is a most desirable feature at any time, and becomes all the more so when a gear-driven supercharger is combined with the rest of the auxiliary mechanism.

The super charger when fitted to the F-type engine is of the single-stage centrifugal type driven through a three-star back gearing with slipping clutches arranged in each of the larger pinions to deal with rapid acceleration. In this engine, as in most modern English engines, the supercharger is placed between the carburettors and the cylinders, thus avoiding the necessity for putting pressure on the carburettor with all the balancing problems which this involves. The gear ratio between the crankshaft and the impellor is either 1 : 6·9 for moderate pressure giving normal ground-level power up to 3000 feet, or 1 : 10 for full supercharge, giving normal power up to 11,500 feet. Fig. 233 shows details of the supercharger itself and of its driving mechanism.

## CHAPTER XII

### HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

Although the conditions applying to an engine for tanks are somewhat specialized owing to the peculiar nature of the service required of them, yet, apart from certain features, the following examples may be taken as fairly typical of the class of large high-speed heavy-duty engine developed during the War. Unlike most other heavy transport duties, the engines for tanks were called upon to run for comparatively long periods under very heavy loads, the average load factor when travelling across rough country being over 80 per cent as compared with the 35-45 per cent load factor of ordinary motor lorry engines ; again, the engines ran always at their governed speed, which ranged from 1200 to 1350 R.P.M. and averaged about 1250 R.P.M., corresponding to a normal piston speed of 1560 ft. per minute, or about double the average piston speed of motor lorry engines.

Owing to the very large amount of dust and mud imported into the tank by the creeping tracks, the engine was always smothered in dirt or dust, and for this reason it was very desirable totally to enclose the crankcase and to eliminate breathers or any other form of ventilation. Further, they were required to use inferior fuel, and in many cases received only the most scant and unskilled attention.

Owing to the severe gradients which the tanks were capable of negotiating, the engine was frequently required to operate at an angle of over  $35^{\circ}$  to the horizontal, as shown in the photographs, figs. 234 and 235, which show a tank climbing out of a deep trench, while fig. 236 shows some of the other duties expected of a tank. Further, it was laid down by the authorities that under no circumstances should the engines show smoke from the exhaust. These two conditions necessitated the adoption of special measures both as regards the lubrication and the piston design.

By reason of the low priority under which tanks and their equipment were constructed until the very last phase of the War, only the



cheapest and most easily worked materials could be used. The allowance of aluminium available was so small that it sufficed only for the



Fig. 234.—Tank at about Maximum Climbing Angle



Fig. 235.—Tank Climbing with Assistance of Unditching Gear

pistons and induction pipes, while the use of high tensile steel was entirely banned.

## HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS 365

The standard 150 H.P. type is shown in the photos, figs. 237-240, and 255, and in the drawings, fig. 241. Six separate cylinders



Fig. 236.--Tank crossing wide River

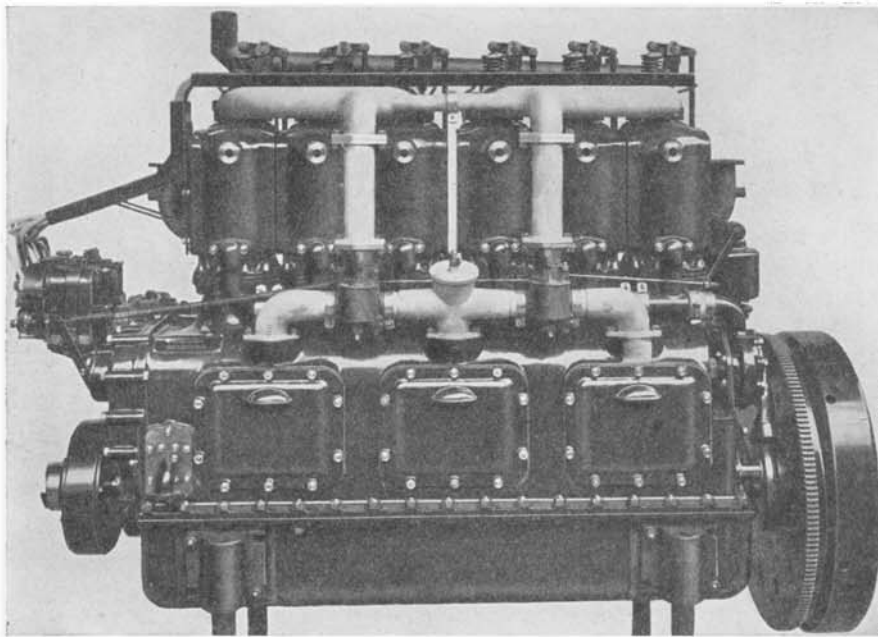


Fig. 237.—150 H.P. Tank Engine, Carburettor Side

are employed, each of  $5\frac{5}{8}$  in. bore and  $7\frac{1}{2}$  in. stroke; the water jackets are arranged with large openings at the sides, which are covered

with screwed-on sheet steel doors. This form of construction, in addition to facilitating the foundry work, allows of the cylinder centres being brought very close together, thus reducing both the over-all length, which was very limited, and the bending moment on the crankcase, due to the two opposing couples formed by each group of three pistons.

The cooling water is delivered to the bottom of the water jacket on the side remote from the valves, and the outlet is arranged between the two sparking plug bosses on the opposite side of the

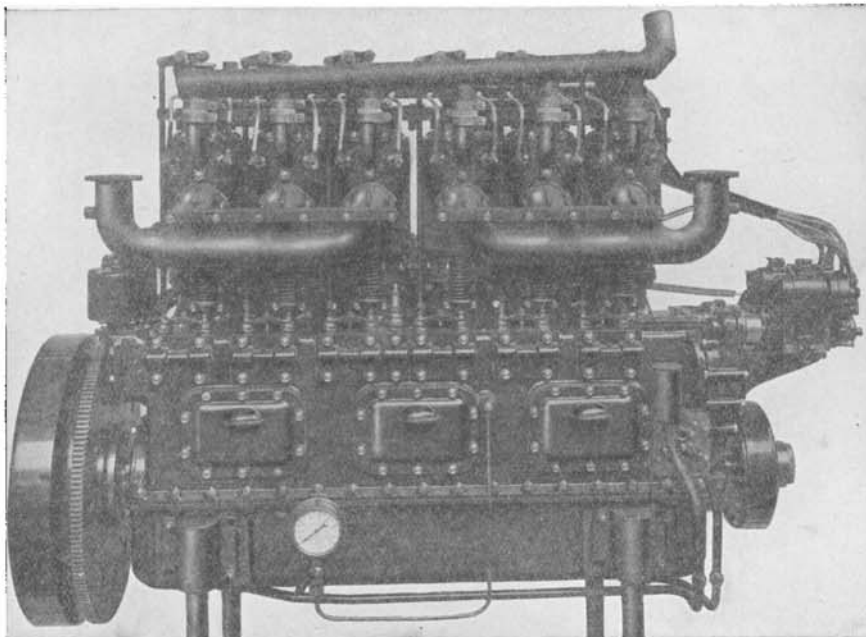


Fig. 238.—150 H.P. Tank Engine, Exhaust Side

cylinder, the object being to ensure a rapid circulation of water round the sparking plugs.

Provision is made in the cylinder heads for fitting compressed-air starting valves, although this system of starting the engines was never employed.

So far as the exhaust valves are concerned, there is nothing very special to record. Care was taken to ensure the best possible cooling of these by providing a wide seating with an ample supply of water all round, and by using a valve stem of large diameter to conduct the heat away. The valve is cooled by carrying the water as close as possible up to the head of the valve, and also by the use of a valve guide of phosphor bronze, which is an excellent conductor of

heat. There is one feature, however, in connection with the exhaust valves which perhaps calls for comment—that is, they are of 3 per cent nickel steel, case-hardened all over. The object of this treatment was twofold :

- (1) Although, of course, the head of the valve does not remain

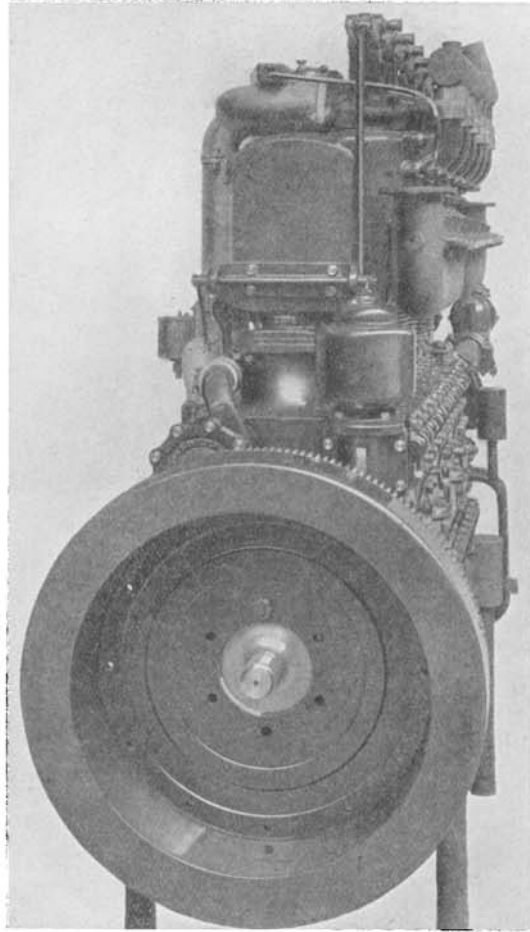


Fig. 239.—150 H.P. Tank Engine, Flywheel End

hard, the carbonized surface resists pitting, with the result that the seating lasts much longer, and grinding-in is seldom necessary.

- (2) The case-hardened stem renders possible the use of a phosphor-bronze valve guide without risk of tearing or seizing.

The connecting-rods are mild-steel stampings of normal design. The only point for comment is in the length of the rods, which are 16 in. between centres, giving an  $l/r$  ratio of 4.26 : 1. The principal

reason for the employment of these long rods lies in the fact that it was anticipated that a four-cylinder unit of this engine would be required at a later date, as indeed proved to be the case, and the shorter rods which it would have been possible to employ on a six-

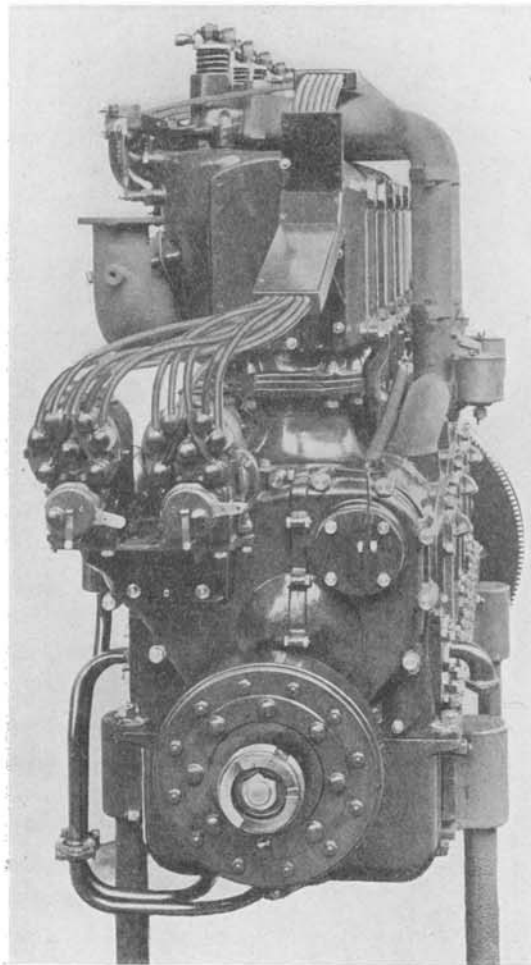


Fig. 240.—150 H.P. Tank Engine, Magneto End

cylinder engine would have been a great disadvantage, in a four-cylinder engine, on account of the secondary disturbing forces.

The top half of the crankcase, or column, is an iron casting of an average thickness of  $\frac{3}{8}$ -in. The general design is clearly shown in the general arrangement drawings, fig. 241, and the function of the false top to the crankcase has already been explained in the chapter dealing with piston design, &c. Inspection doors are fitted

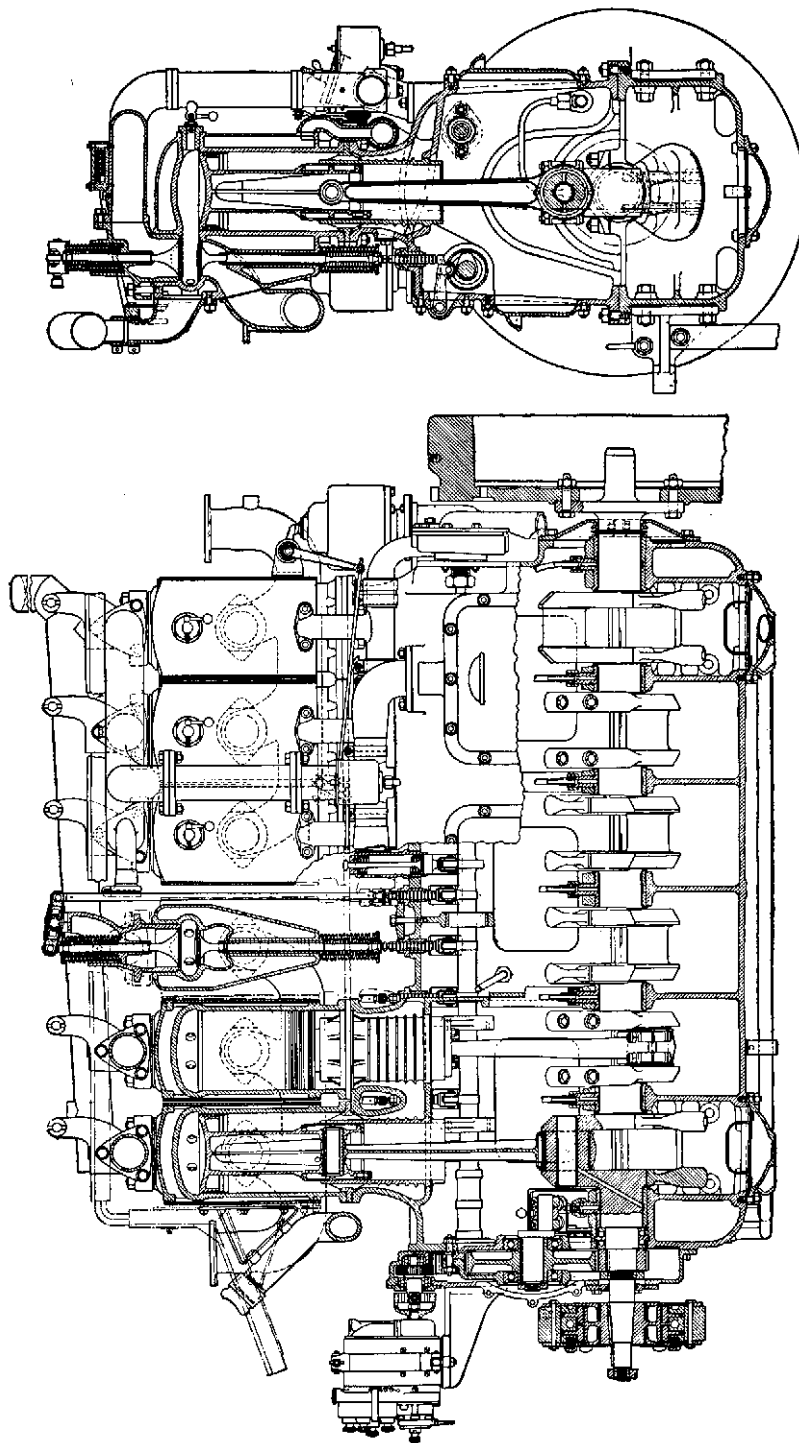


Fig. 241.—General Arrangement of 150 H.P. Engine, Longitudinal and Cross Sections

( E 246 )

on both sides of the column, and the construction is such that it is possible to remove the connecting-rods, &c., through the inspection doors (fig. 242).

The crankshaft is mounted on seven plain bearings carried in the cast-iron bed-plate; the bearing caps are mild-steel stampings; and the white-metal-lined "brasses" are located in the bearing caps

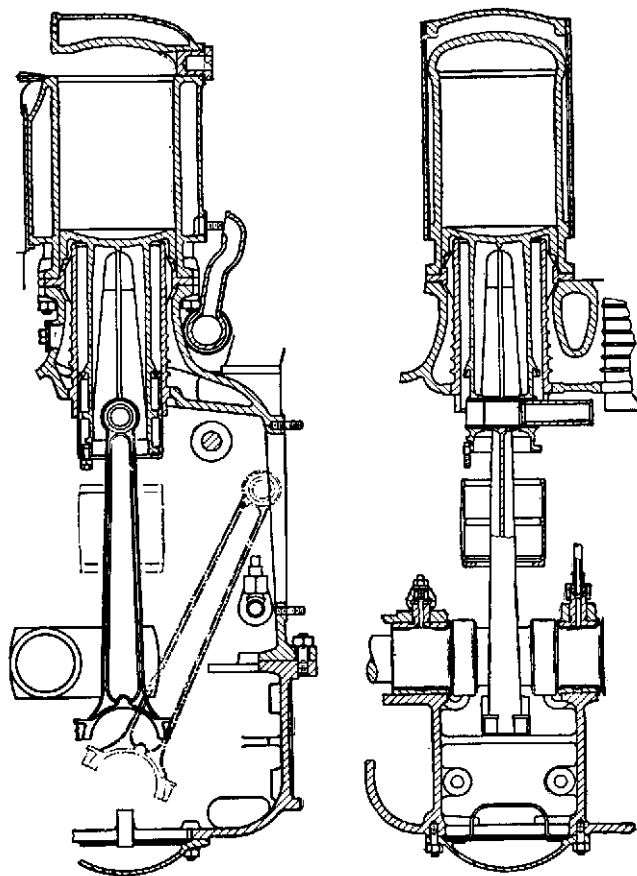


Fig. 242.—Connecting-rod, Dismantling Diagram

in order to allow of the removal of both halves of the journal bearings, should this be found necessary, without disturbing either the bed-plate or the crankshaft. Fig. 243 shows the arrangement of the lubrication connections to the journal bearings, and it will be seen that the oil pipe is attached directly to an extension piece cast integrally with the top half of the bearing brass. The extension piece passes through a hole drilled in the steel bearing cap, thus serving to locate the bearing shell. This method of construction

has the advantage that there is less tendency for oil to leak round between the bearing shell and its housing, and so insulate the brasses. The bearings thus dispose of their heat the more readily.

The crankshaft is a mild-steel forging, the principal dimensions of which are given in the table at the end of this description. Owing to the restrictions as regards the length of the engine, the available area of bearing surface was severely limited, and the difficulty of providing adequate bearing areas was still further increased by its being necessary to employ material for the crankshaft of very low surface hardness and having therefore very poor wearing properties. In apportioning the bearing surface between the connecting-rod and journal bearings in the original design, a higher load factor was allowed on the journal bearings and particularly the centre bearing, since this factor could be reduced, if found necessary, by the addition of balance weights.

The arrangement for the oil supply to the big ends is orthodox.

The arrangement of expanded-in tube in the crank-pin as shown did not prove altogether satisfactory in service, for it was found that there was a tendency for the annular space to become choked in course of time.

The tube was therefore discarded and replaced by the usual arrangement of two end plugs retained in position by a single through bolt.

The flywheel is an iron casting 26 in. in diameter, and is bolted to a flange formed solid with the crankshaft. A Lanchester vibration damper is attached to the forward end of the crankshaft in order to damp out any torsional vibration.

In order to allow of the engine operating satisfactorily when tilted through large angles, the lubrication is on the "dry base" system; that is to say, the oil supply is not carried in the bed-plate, but in a separate oil tank. Three oil pumps of the valveless plunger type are fitted, all three of which are driven from a single crank-pin, which in turn is driven by the intermediate timing gear wheel. The general arrangement of the oil pumps and their driving gear is shown in fig. 244. The centre pump circulates the oil through

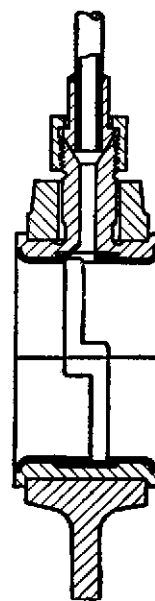


Fig. 243.—Crankshaft Journal Detail



the various bearings, and the two scavenge pumps collect and return the used oil to the external oil tank. Each of the scavenge pumps is connected up to one of the small oil sumps which are provided at each end of the bedplate. The lubrication pipework is shown in fig. 245, which also illustrates the oil relief valve at the flywheel end of the main oil lead. In the original design, the scavenge pump suction pipes were arranged externally on the grounds of accessibility of the pipe joints, but in the Mark V Tanks, in which these engines were principally used, the joints were not accessible when the engine

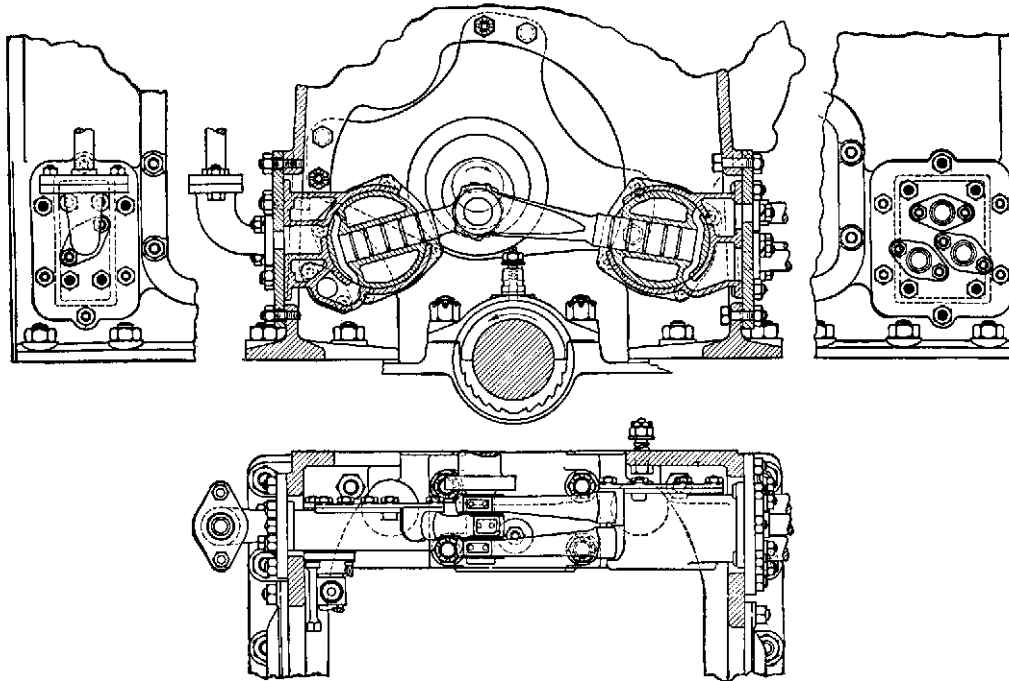


Fig. 244.—Oil Pump Arrangement

was mounted in position. Later engines were therefore fitted with the suction pipes inside the bedplate.

Owing to restriction in width, it was necessary to place all the auxiliaries at the ends of the engine. The auxiliaries to be provided for were as follows: Two magnetos, three oil pumps, two governors, water-circulating pump, and air-pressure pump. The arrangement of the various auxiliary drives will be seen from the illustration, and fig. 246 shows these diagrammatically. In the original design two governors were provided, one to limit the maximum speed of the engine, and the other to open the carburettor throttles directly the

engine speed fell below 400 R.P.M. The object of the second governor was to prevent accidental stoppage of the engine. It was, however, found to be unnecessary, and only the high-speed governor was retained.

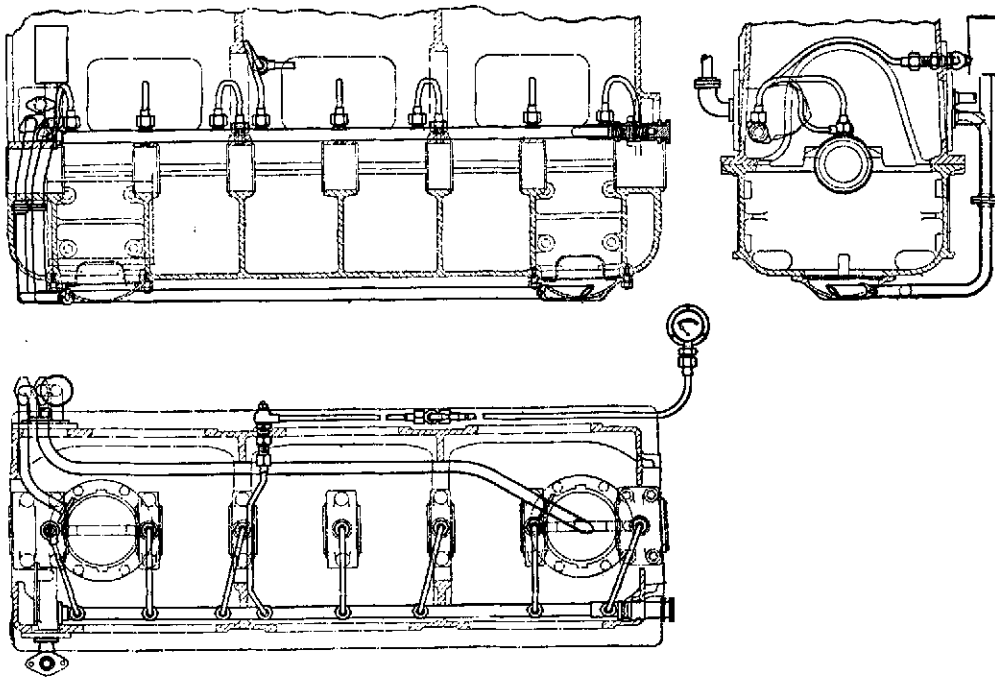


Fig. 245.—Oil Pipe Arrangement

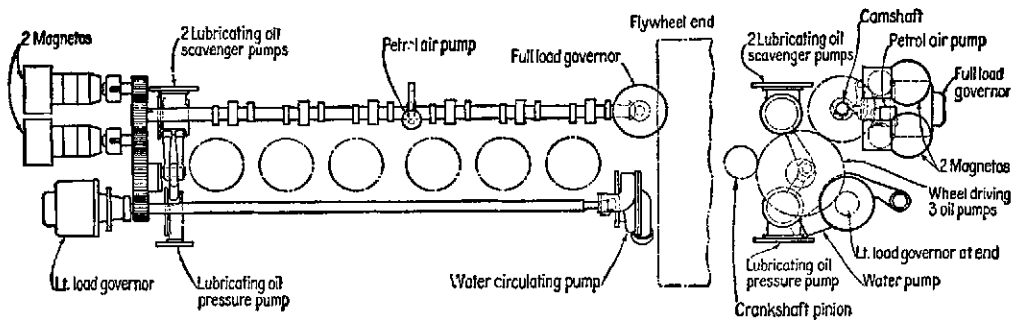


Fig. 246.—Auxiliary Drive Diagram

Fig. 247 shows the arrangement of the water-pump drive. The pumps were designed and made by the Pulsometer Engineering Co., and their performance is shown in the curve, fig. 248.

The intermediate timing gear wheel is mounted on ball bearings carried in a cast-iron spider bolted up to the front wall of the column;

a form of construction which facilitates the correct meshing of the timing gear. All three oil pumps are driven by a small disc crank keyed to the hub of the intermediate wheel.

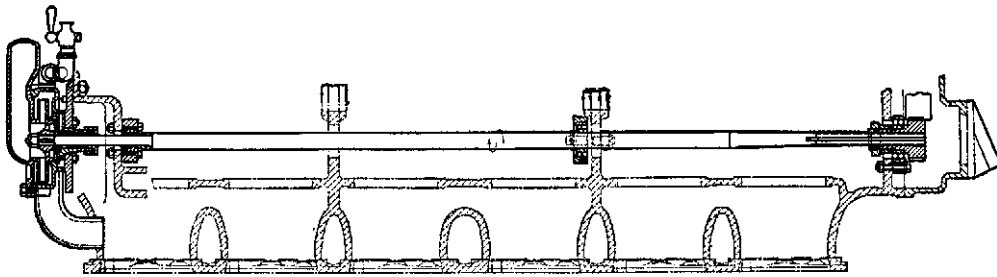


Fig. 247.—Water Pump Drive Arrangement

Fig. 249 shows the general arrangement of the governor; it is a miniature of that used by Messrs. Mirrless, Bickerton & Day, Ltd., for their large Diesel engines.

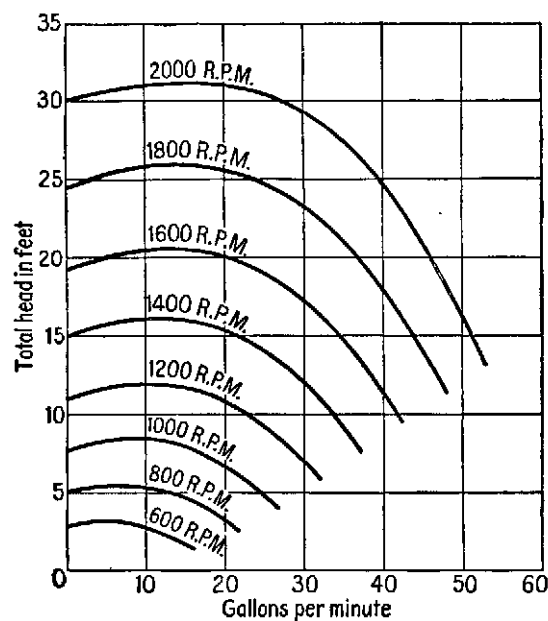


Fig. 248.—Performance Curves of Water Circulating Pump

Two 55-mm. vertical Zenith carburettors are fitted, the whole of the air supply to which is taken from the chambers surrounding the cross-head guides; the method of warming the air supply to the carburettors has already been described in connection with the piston

construction. A hand-adjusted cold-air valve is fitted between the two carburettors for use in very hot weather.

All engines were required to pass the following tests before acceptance :

- (1) A full-load test of two hours' duration, during which the power

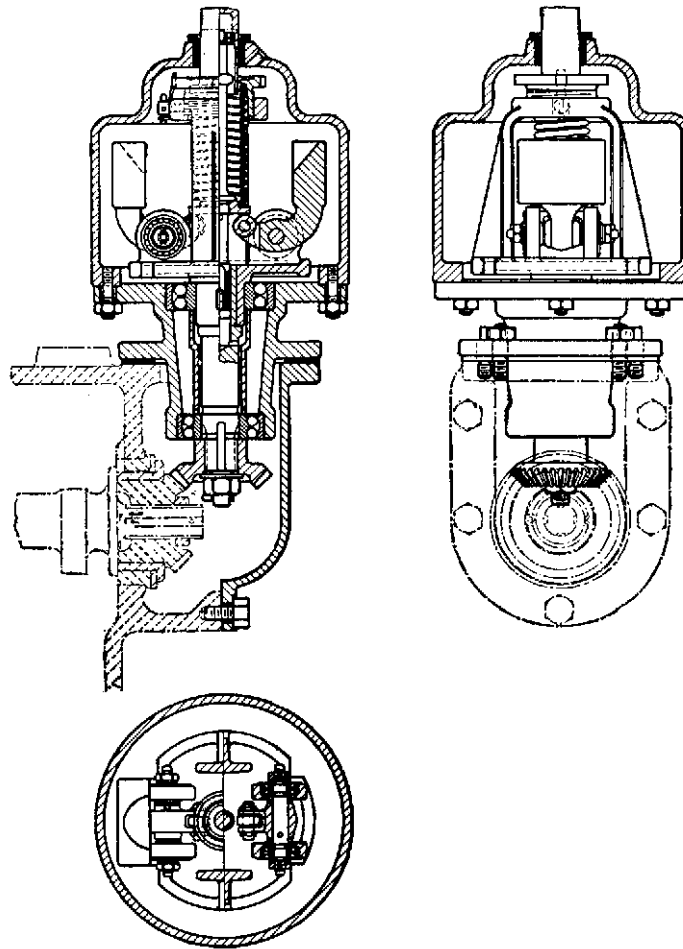


Fig. 249.—Vertical Governor Arrangement

must not fall below 150 B.H.P. at 1200 R.P.M. During this test the fuel and oil consumption was not to exceed 0·7 pint (petrol) and 0·02 pint (oil) per B.H.P. hour.

- (2) The above test to be followed by a run of ten minutes at 1600 R.P.M. and not less than 150 B.H.P.

- (3) Governor tests.

(4) A low-speed torque test, when each engine was required to develop not less than 55 B.H.P. at 400 R.P.M.

(5) The first engine by each maker and thereafter one in every

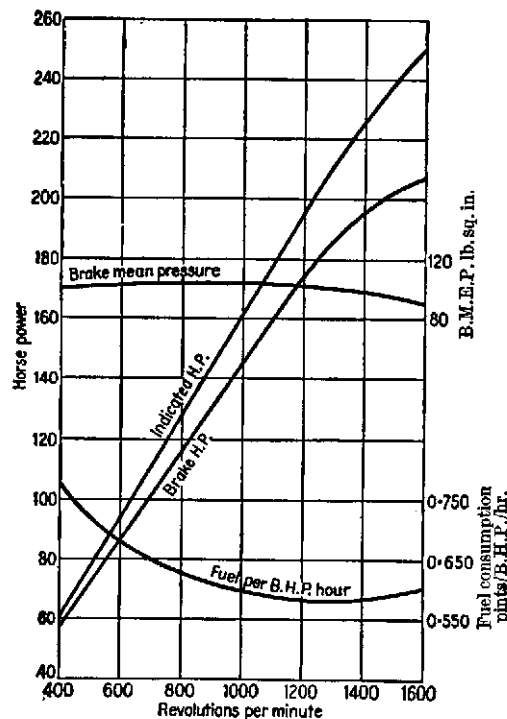


Fig. 250.—Performance Curves, 150 H.P. Engine

fifty, as selected by the Inspector, were submitted to the following additional tests :—

(a) A continuous full-load run of fifty hours, during which the

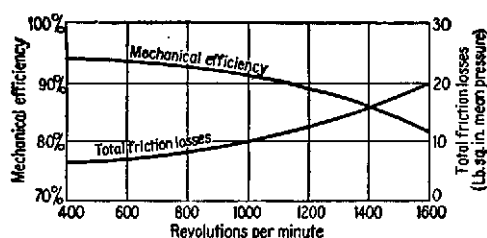


Fig. 251.—Mechanical Efficiency and Friction Losses, 150 H.P. Engine

conditions as to power, fuel and oil consumption were identical with those of the ordinary two-hour full-load test (1).

(b) A tilting test, the engine to be mounted on a tilting table and tilted through an angle of  $35^\circ$  first in one direction and then in the

other. When tilted at this angle the engine was required to be run

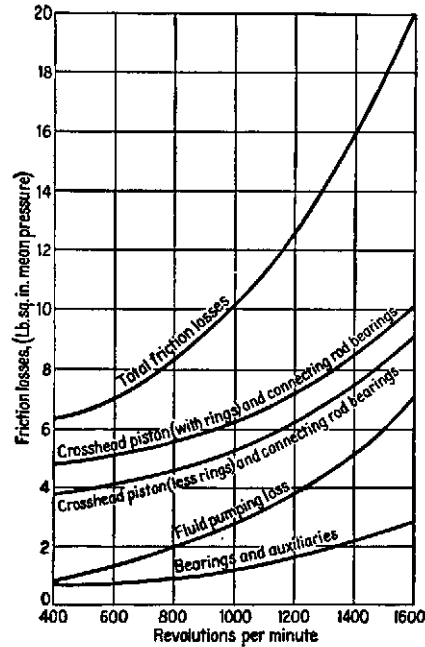


Fig. 252.—Mechanical Loss Curves, 150 H.P. Engine

for ten minutes at about 400 R.P.M. and no load; after this period the throttle was to be thrown wide open, when the engine must open

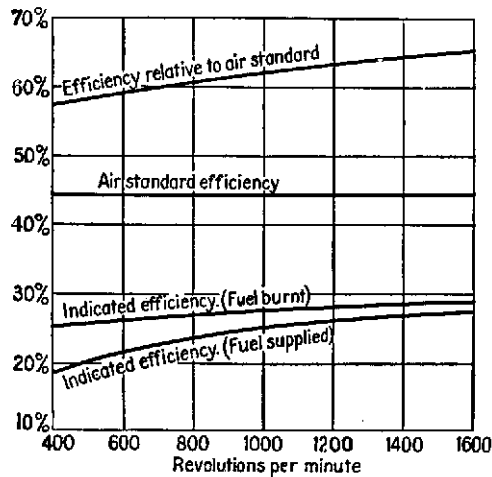


Fig. 253.—Efficiency Curves, 150 H.P. Engine

up firing regularly on all six cylinders without showing any smoke and without any oil leaking out of the base chamber (see fig. 264).

The useful life of a fighting tank was at first so short that an endurance between overhauls of 100 hours was considered ample, but, as might be expected, the engines in actual practice were called upon for an endurance of very much more than the 100 hours originally specified, and at least four instances were reported of engines having run 1400 hours at full speed without requiring or receiving any overhaul beyond the ordinary routine adjustments. Moreover, being of a convenient size and speed, they were used very largely for driving

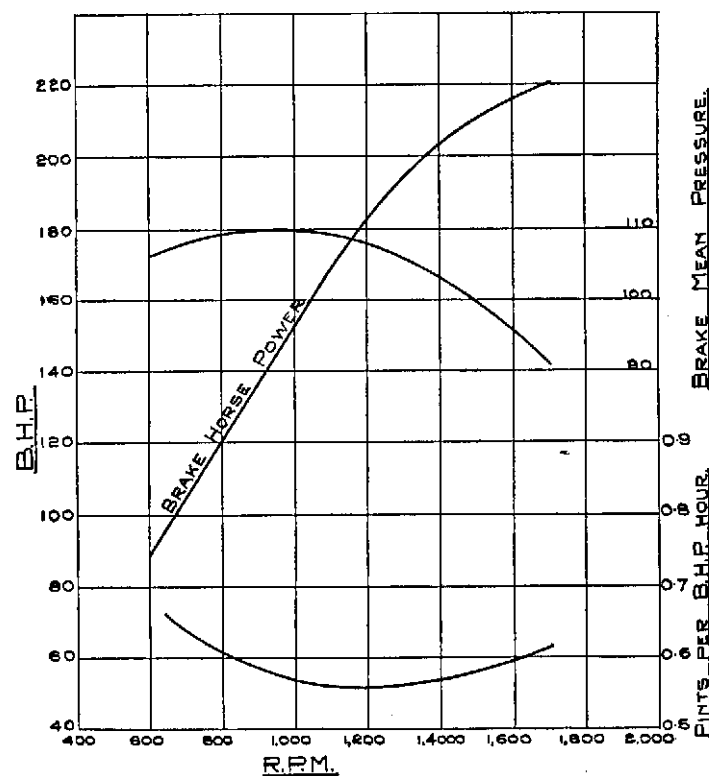


Fig. 254.—Power and Consumption, 150 H.P. Engine

electric generators for supplying light and power to large camps, field workshops, &c., in which service their hours of running were naturally very much longer.

The preceding curves (figs. 250–253) give in full the average performance of these engines. Fig. 250 shows the indicated and brake horse-power, also the brake mean pressure developed at speeds ranging from 400 R.P.M. up to 1600 R.P.M. The brake horse-power and torque curves are the mean of a large number of tests carried out by the different engine-makers, and may be taken as fair average

results. Fig. 254 shows the results obtained from a particularly good example after the conclusion of its 50-hour full-power test. Fig. 252 shows the mechanical losses, which were determined in detail and with considerable accuracy by means of a swinging field dynamometer. All engines on completion of their official full-power run were motored for a short period to determine their mechanical efficiency, and the

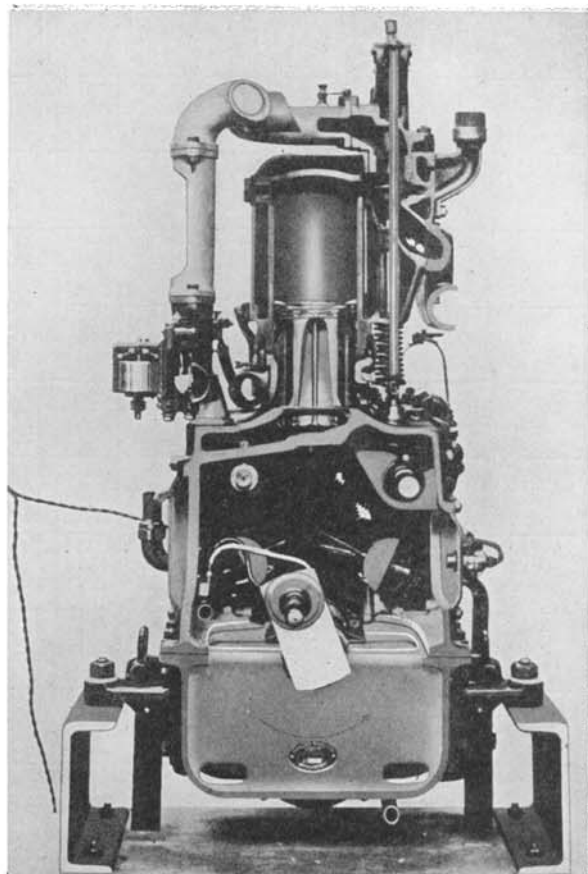


Fig. 255.—Sectional Model of 150 H.P. Engine

total mechanical losses were found to agree very closely with the sum of the several detail losses shown in the above curve. Further, in a few instances, tests for mechanical efficiency were carried out by the method employed by Morse, of cutting out one cylinder at a time while the engine is running on full load. These tests also showed very close agreement. All the test sheets show that the mechanical efficiency, as arrived at by the motoring test, was remarkably uniform



over a wide range of engines, a variation of 1 per cent in the mechanical efficiency figure being very exceptional.

Fig. 253 shows the thermal efficiency and the efficiency relative to the air standard; two efficiency curves are shown—(1) based on the fuel burnt, and (2) based on the fuel supplied. The efficiency based on the fuel supplied is calculated directly from the known fuel con-

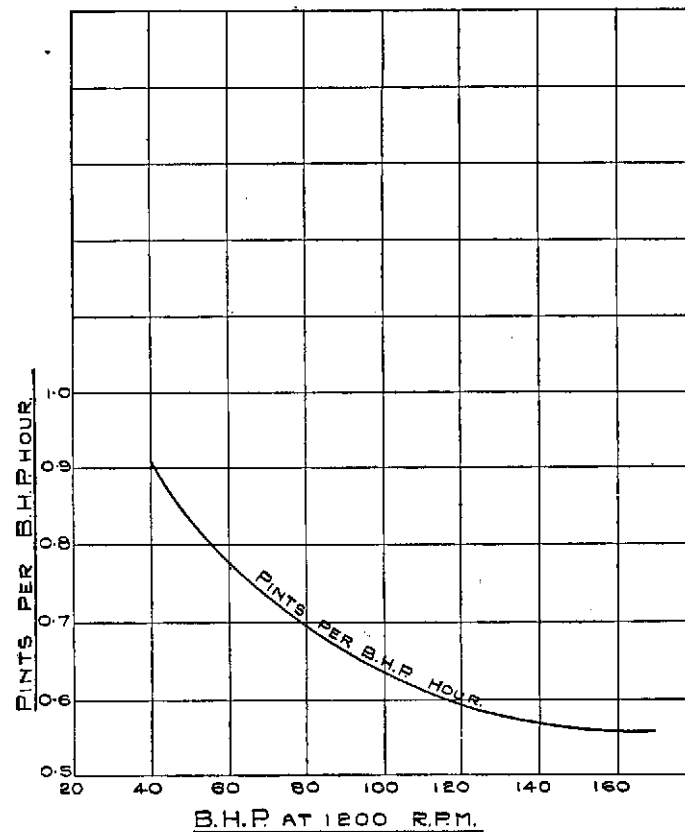


Fig. 256.—Throttle Curve, 150 H.P. Engine

sumption. The efficiency based on the fuel burnt is arrived at by calculating back from the mean pressure actually obtained in the cylinder, and the difference between these two curves represents the loss due to imperfect carburation and distribution.

Fig. 254 shows the power and consumption at full throttle and varying engine speeds.

Fig. 256 shows the fuel consumption at varying loads when running on the governor at speeds ranging between 1200 and 1300 R.P.M.

Fig. 257 shows in detail the cam formation and valve timing.

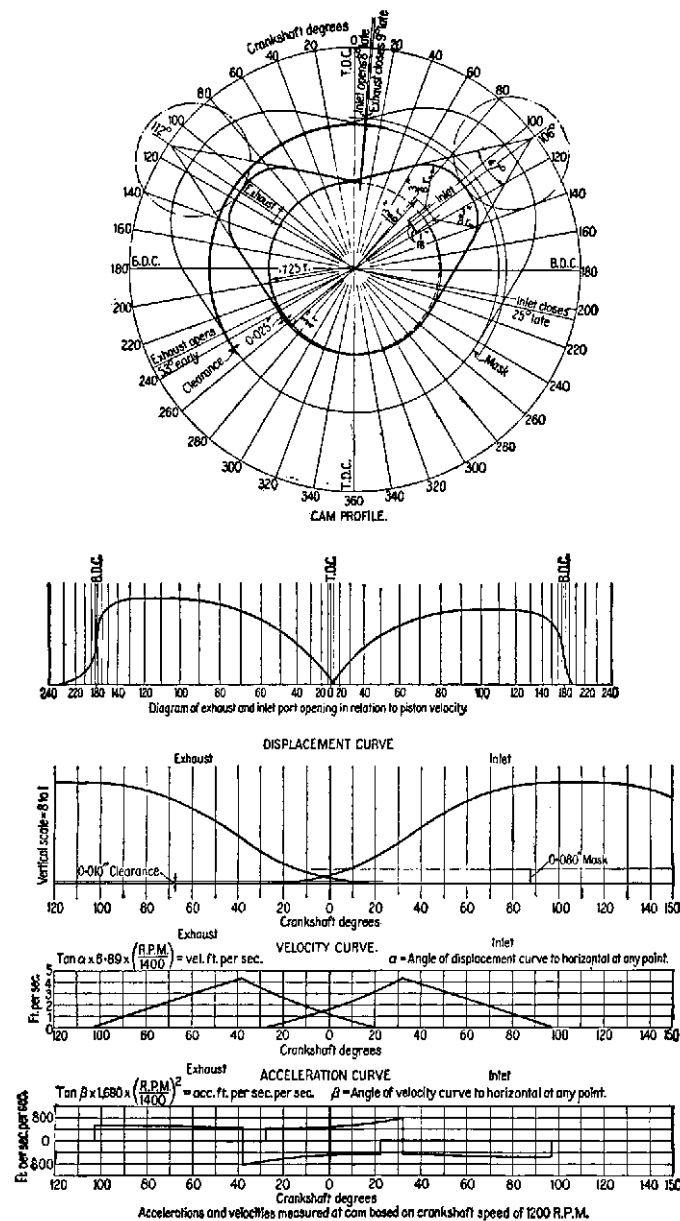


Fig. 257.—Cam Details

The following table taken from a sample test-sheet gives the heat distribution :—

## CALIBRATION TEST, 150 H.P. TANK ENGINE

Duration of test.—Ten hours.

Fuel.—Shell spirit (specific gravity 0.725).

Lower heating value of fuel, 18,600 B.Th.U.s per lb.

Air standard efficiency.—44.4 %.

Mechanical efficiency.—87 %.

## MEAN RESULTS OF LAST EIGHT HOURS OF TEST.

Brake horse-power	..	..	..	..	..	162.9
Fuel (lb./b.h.p.-hour)	..	..	..	..	..	0.554
Brake thermal efficiency	..	..	..	..	..	24.7 %
Indicated horse-power	..	..	..	..	..	187.0
Indicated thermal efficiency	..	..	..	..	..	28.4 %
Relative efficiency (per cent of air standard)	..	..	..	..	..	64.0 %
Heat loss to jackets (B.Th.U.s per hour)	..	..	..	..	..	418,000
Heat to indicated work	..	..	..	..	..	28.4 %
Heat to cooling water	..	..	..	..	..	24.9 %
Heat to exhaust, radiation, etc.	..	..	..	..	..	46.7 %

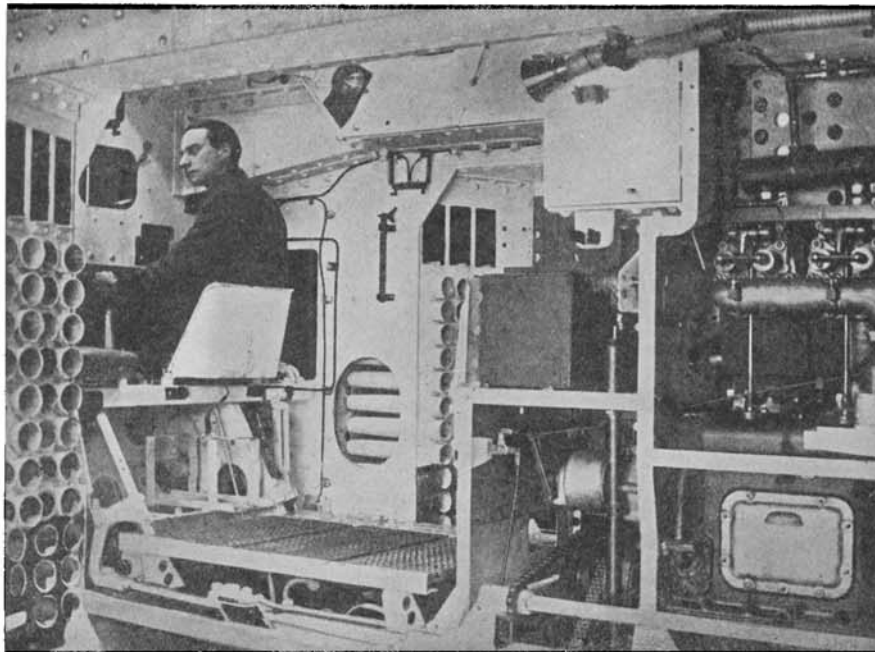


Fig. 258.—Interior of Mark V Two-star Tank

Fig. 258 shows the installation of the engine in the Mark V two-star tank, while fig. 259 gives an exterior view. (In this model the larger 225 H.P. engine was used.)

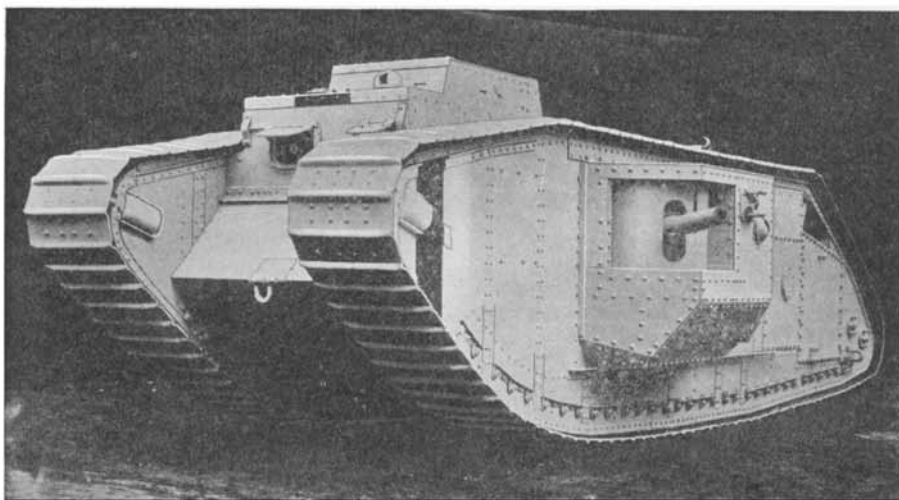


Fig. 259.—No. 106, Mark V Two-star Tank

The leading dimensions of these engines and general data are given in the following tables :—

Number and arrangement of cylinders ..	Six, vertical, separate.
Bore .....	5·625 in.
Stroke .....	7·500 in.
Stroke/bore ratio.....	1·323 : 1.
Area of one piston .....	24·85 sq. in.
Total piston area of engine .....	149·10 sq. in.
Swept volume of one cylinder .....	186·40 cu. in.
Total swept volume of engine .....	1118·40 cu. in.
Volume of clearance space .....	55·90 cu. in.
Compression ratio .....	4·34 : 1.
Normal b.h.p. and speed .....	165 b.h.p. at 1200 r.p.m.
Piston speed .....	1500 ft./min. (25·0 ft./sec.).
Indicated mean pressure .....	110·6 lb. sq. in. (187·5 i.h.p.).
Mechanical efficiency .....	88 %.
Brake mean pressure .....	97·3 lb. sq. in.
Fuel consumption .....	0·636 pint/b.h.p.hr. (s.g. 0·730). 0·580 lb./b.h.p.hr. (18,600 B.T.U./lb.).
Brake thermal efficiency .....	23·6 %.
Indicated thermal efficiency .....	26·8 %.
Air standard efficiency .....	44·40 %.
Relative efficiency .....	62·0 % (fuel burnt). 60·4 % (fuel supplied).

#### GAS VELOCITY, VALVE AREAS, ETC.

*Gas Velocity (ft. per sec.)—*

Choke tube .....	353·0.
------------------	--------

Carburettor body .....	168.8.
Vertical induction pipe .....	165.2.
Induction manifold .....	156.3.
Inlet port .....	109.6.
Inlet valve .....	168.3 (e). 130.8 (t.).
Exhaust valve .....	148.25.
Exhaust port .....	140.3
Exhaust branch pipes .....	140.3.
Exhaust manifold .....	126.6.
<i>Cross Sectional Area (sq. in.)—</i>	
Choke tube .....	1.760.
Carburettor body .....	3.680.
Vertical induction pipe .....	3.760.
Induction manifold .....	3.976.
Inlet port .....	5.672.
Inlet valve .....	3.690 (e). 4.750 (t.).
Exhaust valve .....	4.190.
Exhaust port .....	4.430.
Exhaust branch pipes .....	4.430.
Exhaust manifold .....	4.910.
<i>Diameter (in.)—</i>	
Choke tube .....	1.496 in. (38 mm.).
Carburettor body .....	2.165 in. (55 mm.).
Vertical induction pipe .....	2.1875 in.
Induction manifold .....	2.250 in.
Inlet port .....	2.6875 in.
Exhaust port .....	2.375 in.
Exhaust branch pipes .....	2.375 in.
Exhaust manifold .....	2.500 in.
Weight of piston, complete with rings and gudgeon-pin, etc. ....	7.25 lb.
Weight per sq. in. piston area .....	0.292 lb.
Weight of connecting-rod, complete with bearings, etc. ....	10.70 lb.
Total reciprocating weight per cylinder ..	10.82 lb.
Weight per sq. in. piston area .....	0.435 lb.
Length of connecting-rod .....	16.00 in.
Ratio connecting-rod/crank throw .....	4.27 : 1.
Inertia pressure, top centre .....	82.2 lb./sq. in. piston area.
Inertia pressure, bottom centre .....	51.0 lb./sq. in. piston area.
Inertia pressure, mean .....	33.3 lb./sq. in. piston area.
Weight of rotating mass of connecting-rod ..	7.13 lb.
Total centrifugal pressure .....	1094 lb.
Centrifugal pressure, lb. per sq. in. piston area .....	44.4 lb./sq. in.
Mean average fluid pressure, including compression .....	43.0 lb./sq. in.
Total loading from all sources, lb./sq. in. piston area .....	109.5 lb./sq. in.

# HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS 385

Diameter of crank-pin .....	2.875 in.
Rubbing velocity .....	15.04 ft./sec.
Width of big-end bearing .....	2.25 in. (effective).
Projected area of big-end bearing .....	6.47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing .....	3.84 : 1.
Mean average pressure on big-end bearing	421 lb./sq. in.
Load factor on big-end bearing .....	6330 lb. ft./sec.
<i>Inlet Valve (one per cylinder)—</i>	
Outside diameter .....	2.875 in.
Port diameter .....	2.6875 in.
Width of seating .....	0.0937 in.
Angle of seating .....	45°.
Radius under valve head .....	1.9375 in.
Diameter of valve stem .....	0.4995 in. in guide ; 0.4985 in. below guide.
Lift of valve (total) .....	0.5625 in.
Life of valve (effective) .....	0.4375 in.
Length of valve guide .....	4.375 in.
Clearance, valve stem in guide .....	0.0005 in.
Over-all length of valve .....	8.375 in.
No. of springs per valve .....	Two.
Free length of spring .....	(a) 5.875 in. (b) 5.8125 in.
Length of spring in position, no lift	(a) 4.000 in. (b) 3.8125 in.
Mean diameter of coils .....	(a) 1.703 in. (b) 1.120 in.
Gauge of wire .....	(a) No. 6 B.W.G., 0.203 in. diam. (b) No. 11 B.W.G., 0.120 in. diam.
Valve tappet clearance, cold .....	0.002 in.
Weight of valve, complete with spring, etc. ....	1.6234 lb. { Valve bare, 0.8120 lb. Two springs, 0.6718 lb. Spring cap, etc., 0.1406 lb.
Weight of spring, bare .....	0.6718 lb. per pair.
Inlet valve opens (deg. on crank) ...	{ Valve leaves seat 29° early. Valve opens 8° late.
Inlet valve closes (deg. on crank) ...	{ Valve closes 25° late. Valve back on seat 59° late.
Material for valve .....	3% nickel steel stamping, case-hardened.
<i>Exhaust Valve (one per cylinder)—</i>	
Outside diameter .....	2.6875 in.
Port diameter .....	2.375 in.
Width of seating .....	0.156 in.
Angle of seating .....	45°.
Radius under valve head .....	0.875 in.
Diameter of valve stem .....	0.5595 in. in guide ; 0.5475 in. below guide.
Lift of valve .....	0.5625 in.
Length of valve guide .....	7.125 in. (effective).
Clearance, valve stem in guide .....	0.003 in.

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Over-all length of valve.....	12.8125 in.
No. of springs per valve .....	Two.
Free length of spring .....	(a) 5.875 in. (b) 5.8125 in.
Length of spring in position, no lift..	(a) 4.000 in. (b) 3.8125 in.
Mean diameter of coils .....	(a) 1.703 in. (b) 1.120 in.
Gauge of wire .....	{ (a) No. 6 B.W.G., 0.203 in. diam. (b) No. 11 B.W.G., 0.120 in. diam.
Valve tappet clearance, cold.....	0.010 in.
Weight of valve, complete with spring, etc. ....	2.0934 lb. { Valve bare, 1.281 lb. Two springs, 0.6718 lb. Spring cap, etc., 0.1406 lb.
Weight of spring, bare .....	0.6718 per pair.
Exhaust valve opens, degrees on crank	53° early.
Exhaust valve closes, degrees on crank	9° late.
Material for valve .....	3% nickel steel stamping, case- hardened.
<i>Connecting-rod—</i>	
Length between centres .....	16.00 in.
Ratio connecting-rod/crank throw ..	4.27 : 1.
Little-end bearing, type.....	Chilled phosphor-bronze bush fixed in rod.
Little-end bearing, diameter .....	1.250 in.
Little-end bearing, length .....	1.625 in.
Little-end bearing, projected area ...	2.032 sq. in.
Big-end bearing, type .....	Bronze shell, lined white metal.
Big-end bearing, diameter .....	2.875 in.
Big-end bearing, length .....	2.6875 in.
Big-end bearing, projected area ....	6.47 sq. in. (effective).
Ratio piston area/projected area of big-end bearing .....	3.84 : 1.
Number of big-end bolts .....	Four.
Full diameter .....	0.500 in.
Diameter at bottom of threads .....	0.4375 in.
Total cross sectional area at bottom of threads .....	0.602 sq. in.
Total load on bolts, at 1200 r.p.m. ..	2659 lb.
Total load on bolts, at 1600 r.p.m. ..	4795 lb.
Stress per sq. in., at 1200 r.p.m. ....	4490 lb./sq. in.
Stress per sq. in., at 1600 r.p.m. ....	7965 lb./sq. in.
<i>Crankshaft—</i>	
Length of complete shaft .....	66.00 in.
Cylinder centres .....	7.250 in.
Cylinder centres (centre pair) .....	8.000 in.
Outside diameter of crank-pin .....	2.875 in.
Inside diameter of crank-pin .....	1.4375 in.
Length of crank-pin .....	2.750 in.
Outside diameter of journals .....	2.875 in.
Inside diameter of journals .....	Solid.
Number of journal bearings .....	Seven.

Length, flywheel-end bearing .....	4.000 in.
Length, forward-end bearing .....	2.875 in.
Length, centre bearing.....	2.875 in.
Length, intermediate bearing .....	2.125 in.
Width of crank webs .....	3.500 in.
Thickness of crank webs .....	1.1875 in.
Radius at ends of journal and crank-pins .....	0.250 in.
Diameter of drilled oil-ways in shaft .....	0.250 in.
Weight of complete shaft .....	182 lb., includes thrust race pinion, starting clutch, nuts, etc. (no balance weights).
Material .....	Mild-steel forging.

It will be interesting and perhaps helpful to review the defects which revealed themselves in these engines. With over 4000 engines in service it is possible to discriminate between mere accidents and epidemic troubles.

**Combustion Chamber.**—The design of the combustion chamber, though excellent from the point of view of turbulence, and therefore of power output and efficiency, proved, as in the light of present knowledge might have been expected, rather bad from the point of view of detonation, despite the comparatively low compression ratio. Originally it was intended to operate the tanks only on aero-spirit, but later they were required to use the lowest grade of war spirit—an American fuel consisting almost entirely of the heavier fractions of the paraffin series. With this fuel detonation became severe when the engines were pulled down to a low speed with wide-open throttle.

**Pistons.**—The first few engines were fitted with sand-cast pistons in 88 per cent aluminium and 12 per cent copper alloy. These castings sometimes gave trouble owing to porosity of the metal at the point where the hollow trunk joins the head of the piston. Many pistons were rejected on this score, but a number in which the porosity did not appear on the surface and therefore was not detected by inspection were fitted to engines, and some of these broke away at this point, but since the connecting-rod was not released by such failures little or no further damage resulted therefrom. This defect was remedied completely by employing pistons cast in metal dies in all subsequent engines.

**Cross-head Guides.**—These were at first made of bronze lined with white metal, and proved quite satisfactory. Owing to the scarcity, or alleged scarcity, of bronze, the use of this material was eventually forbidden by the authorities, and cast iron was therefore



substituted. The cast-iron guides gave a good deal of trouble owing to distortion after machining, and to meet this, since there was no time for "ageing" or annealing, it became necessary to allow rather a large working clearance, which gave rise to noise. Later the same copper-aluminium alloy used for the pistons was employed also for the guides, and this proved extremely satisfactory—quite equal to the white-metal-lined guides as regards wear, and better in so far as, since the clearance increased with temperature, it was safe to work with a very close fit.

**Exhaust Pipes.**—The exhaust manifolds and pipes radiated so much heat that it was found necessary, for the comfort of the tank crew, to jacket them with an air jacket through which a circulation of cold air was induced by means of a fan driven from the engine.

**Crankshafts.**—Owing in part to the small bearing area necessitated by the limits on the over-all length of the engine, and in part to the use of dead soft steel, the wear in the journals, and in particular the centre journal, was rather severe. So far as the actual fighting tanks were concerned, this was of little consequence, because the crankshaft easily outlasted the rest of the machine; but in the case of the tanks used for training, which ran all day and every day, and of electric-generating sets, this wear became troublesome, for it was generally necessary to regrind the crankshaft and fit new bearings after from 500 to 700 hours' running at full speed and 80 per cent load factor. It was therefore decided to fit balance weights on all engines destined for other than fighting tanks, a procedure which reduced the rate of wear to less than one-third, but which, by lowering the natural periodic speed of the shaft and by increasing the intensity of the oscillations, threw a heavy duty on the torsional vibration damper.

Apart from wear, the soft mild-steel crankshafts were very reliable indeed, and, out of over 4000 engines, no single instance of broken crankshaft was reported.

**Vibration Dampers.**—These were at first identical with those fitted to the Daimler engines. So long as the crankshafts were unbalanced, the duty on these dampers was very light indeed, and they gave no trouble. But so soon as balance weights were fitted and the dampers were called upon to function, the thin plates used in them soon cut away through the castellations and the damper wore out completely in about 50 hours' running. As the result of this experience the dampers were re-designed by Messrs. Gardner & Sons; two thick cast-iron plates with good substantial castellated bosses

being fitted in place of the rest of thin steel plates : these proved perfectly satisfactory, the wear, even after 1200 hours' running with balance weights, being reported as almost negligible.

**Lubrication System.** — As explained previously — two scavenging pumps exhausted the oil, one from either end of the necessarily very shallow bedplate, and delivered it to a filter tank placed some 3 feet above the level of the crankshaft. A third force pump drew oil from the filter tank and delivered it to all the crankshaft bearings. All three pumps were operated from a single crank at the forward end of the engine, and had the same stroke ; but the scavenging pumps were of slightly larger bore, and so could exhaust more rapidly than the force pump could deliver. It was found, however, that under certain conditions when the forward part of the engine was tilted up at the extreme angle the suction pump drawing from the after end failed to keep pace, with the result that oil gradually accumulated in the after end of the crankcase till it eventually ran out of the flywheel bearing. This was found to be due to the fact that while the force pump had a positive head of about 3 feet in its favour the scavenge pump had, under extreme conditions, a suction head of about 3 feet, while the length of suction pipe of about 5 feet was subject to a pulsating flow. Under these conditions the volumetric efficiency of the force pump exceeded that of the scavenge pump by more than the difference in volumetric capacity. This difficulty was cured completely by the provision of an air vessel on the suction side of the scavenge pump, thus maintaining a uniform flow in the suction pipe.

Again, it was found that, when the engine stopped in a certain position, oil could leak back from the filter tank through the bearings into the crankcase, sufficient in time to flood the latter. This occurred only when the machine was left standing for several days in very hot weather. To obviate this defect a cock was fitted in the oil pipe leading to the force pump, and, in order to render it impossible to start the engine with the oil supply cut off, this cock was combined with the magneto earthing switch.

Apart from these two minor difficulties, which were easily overcome, the lubrication system worked well ; and although the scavenge pumps were considerably above the highest oil level, and under normal conditions their suction inlets also were above, yet they never failed to pick up the oil at once so soon as the engine was tilted and one or other of the suction inlets was drowned.

**Valve Spring Caps.**—In order to economize machining opera-

tions the valve spring caps were pressed out from sheet steel, and with a view to reinforcing the conical portion and prevent any risk

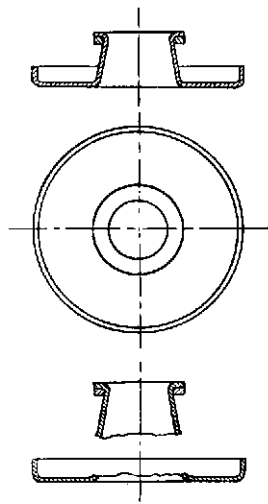


Fig. 260.

of its being drawn over the taper wedges a steel wire ring was embodied in the pressing, as shown in fig. 260. In practice these pressed spring caps used to fail after a period ranging from 200 to 600 hours, the failure invariably occurring at the junction between the conical portion and the flat retaining face, as shown. The failure was due in part to fatigue and in part to the fact that, in spite of all precautions, the pressing was generally sharply nicked at this point. Though no doubt with a little more care in manufacture satisfactory pressings could have been made, it was decided, in view of the urgency, to employ spring caps turned from steel bar, which proved quite satisfactory.

**Valve Gear.**—With a view to eliminating noise and wear in the valve gear the base circle of the cams was ground slightly eccentric in order to permit of a large working clearance at the time when it was essential that the valves should be closed, and gradually to take up this clearance preparatory to opening. This method is very satisfactory as a rule, but it is liable to abuse. In practice it was at first found very difficult to prevent the mechanics in charge from adjusting the clearance regardless of the position of the cam, with the result that the valves were lifted slightly from their seats long before the correct opening period. So far as the inlet valves were concerned no trouble arose, but a few cases of burnt-out exhaust valves were found to be due to this cause. The difficulty was eventually overcome by the issue of very full instructions for the adjustment of valve clearances and by rigorous inspection. Once the correct adjustment was fully understood, trouble with valves became almost unknown, and there is little doubt but that the immunity from trouble and the quiet running obtained were largely due to the use of cams with eccentric base circles. Apart from these troubles, of which all but the tendency to detonate and the heat radiated from the exhaust were quickly and easily overcome, these engines behaved very well. The system of hermetically sealing the crankcase and drawing all the air through the false top proved most effective in keeping the crankcase cool and the working parts clean.

The cross-head type pistons proved thoroughly effective both in eliminating the usual troubles with large aluminium pistons, in preventing smoke, and in yielding a very high mechanical efficiency.

With the steady growth in size and armament of tanks it became necessary to provide still larger engines, and another six-cylinder engine of the same general type but embodying all the experience gained with the 150 H.P. engine was next designed and produced. This engine is shown in the drawings and photos: figs. 261, 262, 263, 264, 265. Although about 800 of these engines were completed, yet owing to delays in tank production none of them saw any active fighting. Sufficient experience was, however, gained both

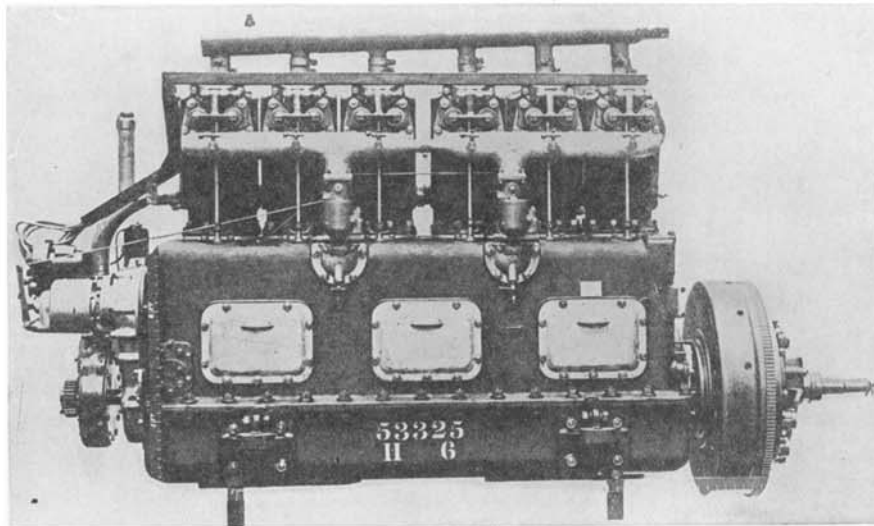


Fig. 261.—225 H.P. Engine, Carburettor Side

on the test-bed and subsequently in tanks to prove that these larger engines were a very great improvement over the earlier type.

Rated normally at 225 B.H.P., they had a bore and stroke of  $6\frac{3}{4}$ " and  $7\frac{1}{2}$ " respectively and developed 260 B.H.P. at 1200 R.P.M.

In order to obviate the defects of the 150 H.P. engines the following modifications were made:—

The combustion chamber was made in the form of a compact rectangular chamber with the ignition plugs placed as nearly as possible in the centre, as shown in fig. 265. This resulted in complete immunity from detonation under all conditions, so much so that the engines could be loaded down to two or three hundred R.P.M. with wide-open throttle without the least trace of detonation even

on the lowest grade of American petrol, and this despite the fact that the cylinders were larger and the compression ratio as high as in the 150 H.P. engines.

The valves were duplicated and placed horizontally with the inlets on one side and the exhaust on the other. This arrangement

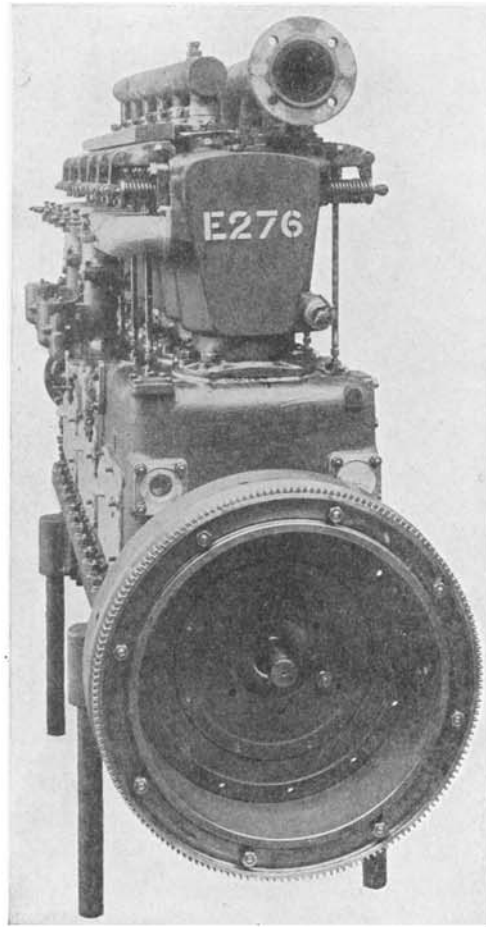


Fig. 262.—225 H.P. Engine, Flywheel End

of valves permitted of the exhaust ports being turned up vertically so that short straight exhaust pipes could be taken direct through the roof of the tank to a manifold fitted outside. Thus the length of exhaust pipe inside the tank was reduced to the very minimum, while all bends, &c., were avoided. The inlet valves were fitted in separate cages which could easily be removed without disturbing any of the pipe work, and the exhaust valves could then be

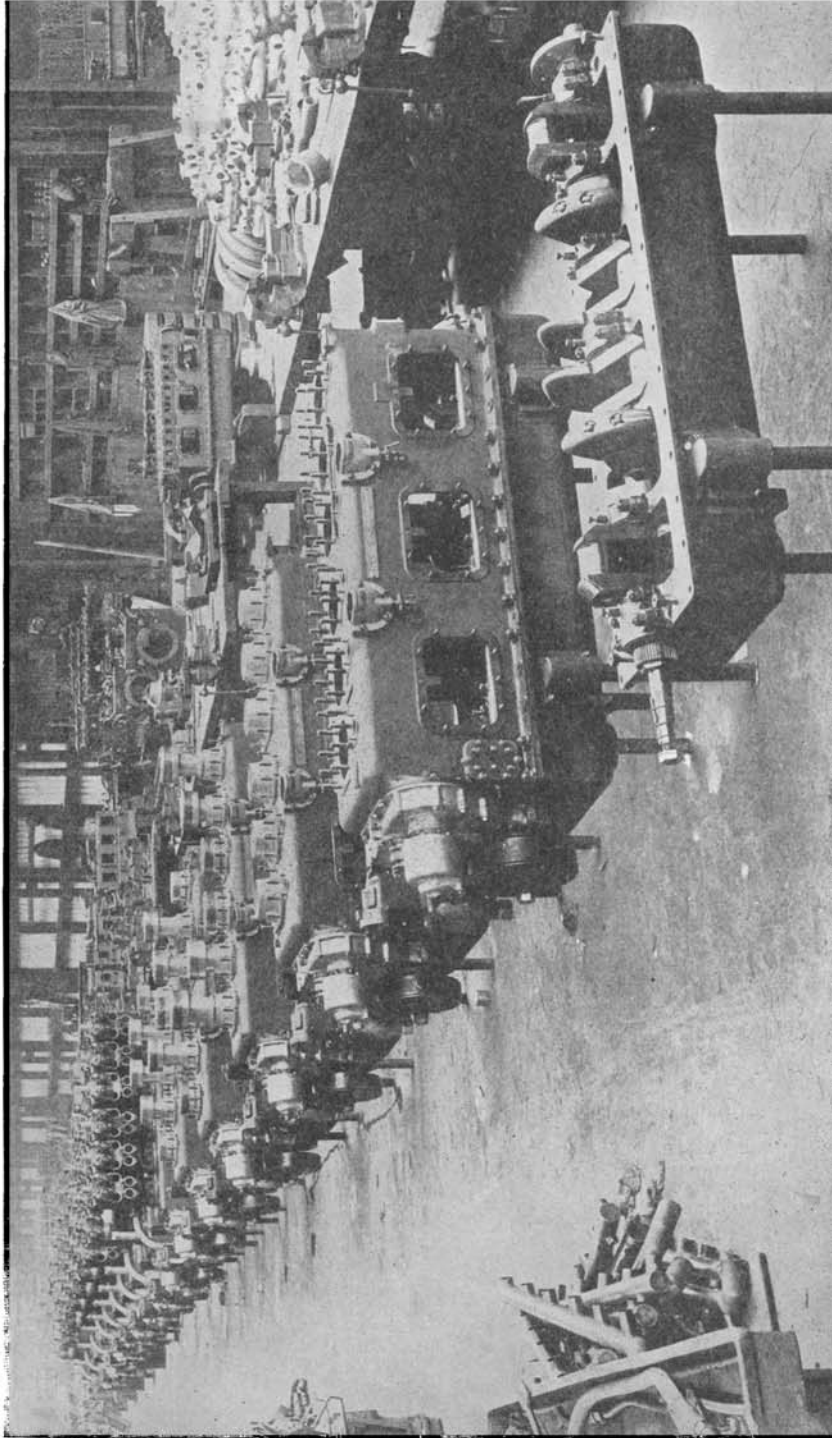


Fig. 263. —225 H.P. Tank Engines in course of erection at Messrs. Gardner's Works

withdrawn through the openings left by the removal of the inlet valves.

Die-cast pistons were used from the very start, and no single instance of piston failure was ever reported. There being no rigid restrictions as regards length, more liberal bearing surfaces could

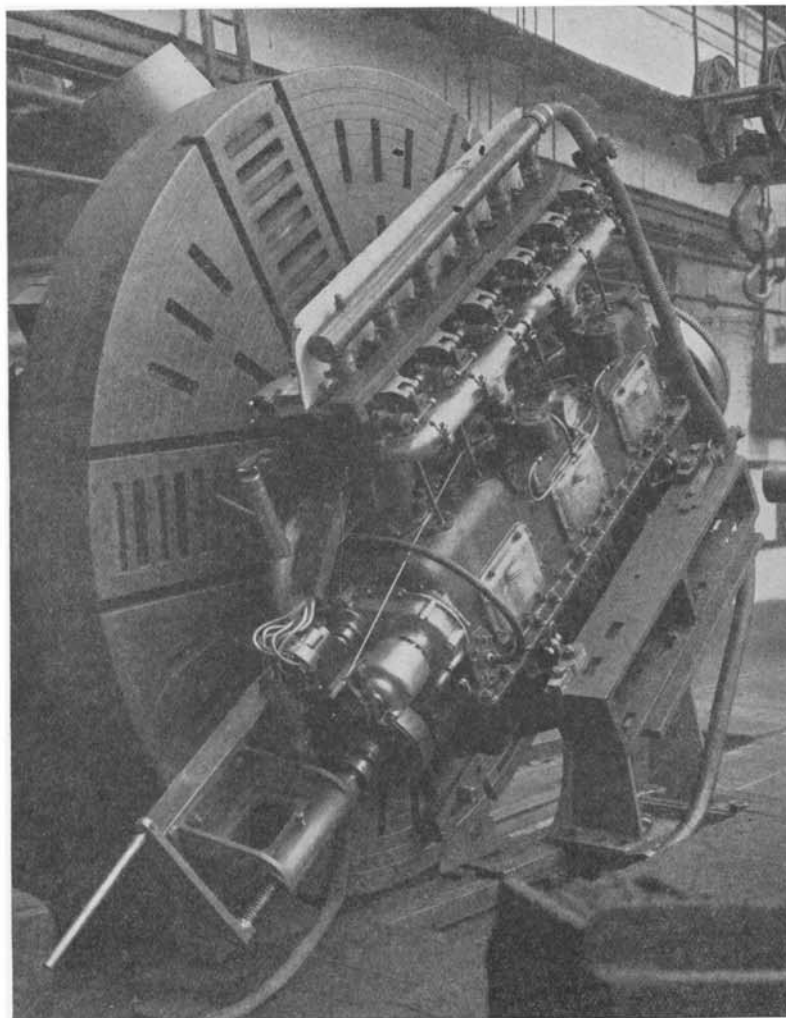


Fig. 264.—225 H.P. Engine, mounted for Tilting Test

be and were provided, particularly as regards the crankshaft centre bearing. Balance weights and Messrs. Gardner's revised design of torsional vibration damper were fitted in all cases.

The crankshafts were made from 0.35 carbon steel, and under these circumstances proved practically immune from wear. The

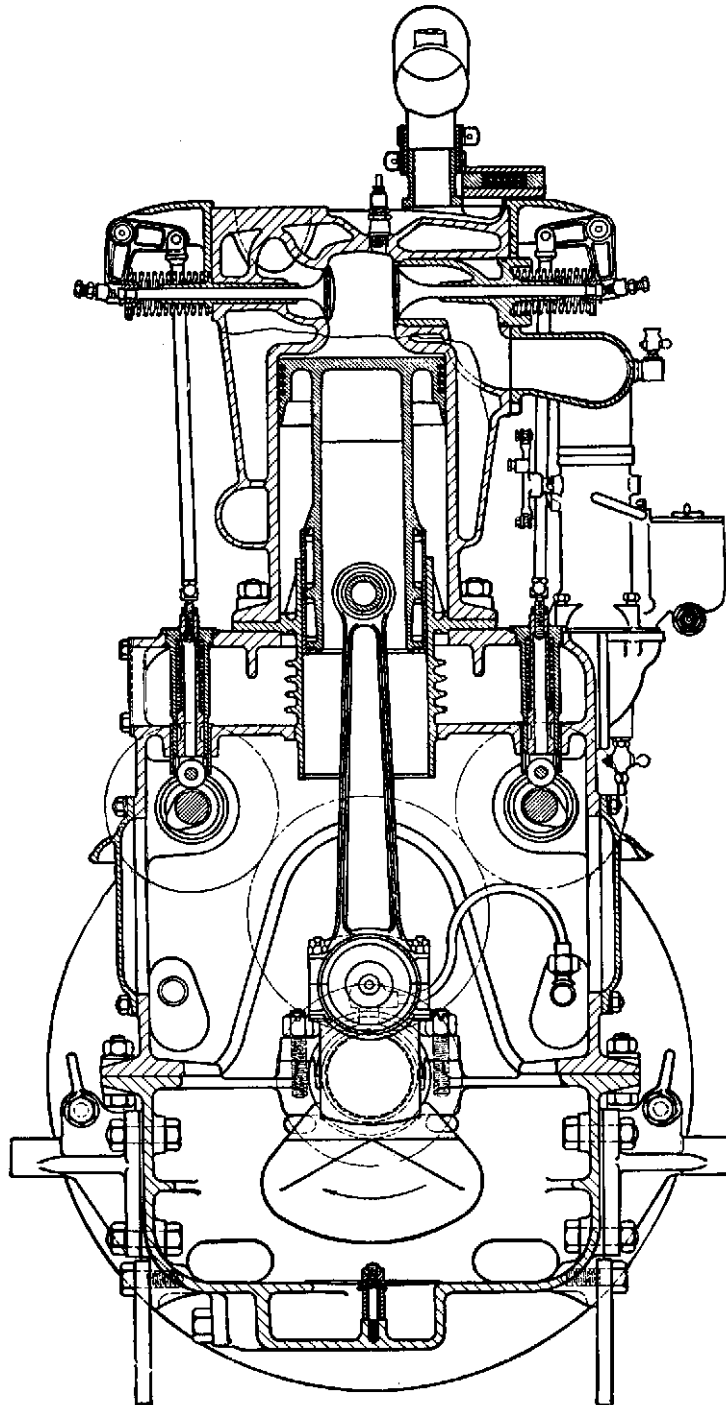


Fig. 265.—225 H.P. Tank Engine, Sectional Elevation



other defects which manifested themselves in the first batches of 150 H.P. engines were, of course, obviated by adopting the expedients explained above.

Although these larger engines never saw actual fighting service, yet they were submitted to very severe tests both in tanks and on the test-bed, the latter including full-power runs of 200 hours' duration. As may be supposed, with the experience of the smaller engines available before the design was commenced, these engines were an improvement on the 150 H.P. type in almost every respect. A very large number of them have since been converted to run on gas, and are now in use in electric-power stations in various parts of the country. Unfortunately, however, their compression ratio is too low for efficient running on town or producer gas.

## CHAPTER XIII

### HIGH-SPEED DIESEL ENGINES

During recent years much attention has been devoted to the problem of so-called high-speed Diesel engines, that is to say, engines using relatively heavy and high boiling fuels such as cannot be vaporized outside the cylinder, and have therefore to be injected into the cylinder in the liquid form, and which depend for ignition on the temperature of compression.

The low fuel cost coupled with the very high thermal efficiency obtainable throughout the whole range of load renders such an engine very attractive in fields where fuel cost is a major item and where the mess and smell inseparable from these heavy petroleum products are not a serious nuisance. In certain applications also, e.g. marine and aircraft, the great reduction in fire risk resulting from the use of a fuel of high flash point is a further very important consideration in favour of the heavy-oil engine.

Ever since the Diesel engine first appeared attempts have been made in various countries to produce a high-speed version, that is an engine operating at piston speeds in excess of, say, 1500 ft. per minute, but until comparatively recently these have met with little success. It was not until the Royal Aircraft Establishment, under the ægis of the Air Ministry, tackled the problem that any really substantial advance was made. To the Royal Aircraft Establishment is due the credit for having first shown the world, some eight years ago, that a really high-speed heavy-oil engine could be produced which would not only run at a piston speed of well over 2000 ft. per minute, but that even at this high piston speed, a higher brake mean pressure and a higher overall efficiency could be obtained than had ever been recorded previously by any slow-running Diesel engine of any size or type. Although during the ensuing eight years intensive development work has been carried out both on the Continent and in America, yet the results obtained by the Air Ministry some eight years ago have yet to be surpassed.

Modern developments in the way of high-speed Diesel engines have tended to depart widely from the slow-speed type and to follow much more closely the petrol engine, not only in mechanical design, but also as to the thermo-dynamic cycle on which they operate, that is to say, the idea of working on a constant pressure cycle, never strictly adhered to even in the slow-speed, has in the high-speed engine, gone entirely by the board, and in its place is substituted a cycle resembling much more nearly the constant volume cycle of the petrol engine. The constant pressure cycle or even any near approximation to it can be employed only in relatively slow-speed engines, and then only when the fuel is injected by means of blast air, since this method alone combines the requisite degree of pulverization, of penetration, and of turbulence. It is not, however, a desirable cycle for small high-speed engines, for its overall efficiency is relatively low, and its principal advantage, namely a lower maximum pressure, is of little import when applied to a small piston. Moreover, even the air injection engine may, owing to slight derangement, develop occasionally very high maximum pressures, and must in consequence be made robust enough to deal safely with them should they occur.

In any type of Diesel engine, whether of high or low speed, the major problem is that of bringing the fuel and air within the cylinder in contact with one another, and at the same time of maintaining a high relative speed between the fuel particles and the air in order constantly to sweep away the products of combustion as they are formed, and to replace them with fresh air. The jet or jets of fuel must penetrate far enough through the dense atmosphere in the combustion chamber to reach the most outlying air, and at the same time the spray must be fine enough to spread out on all sides. This condition can be accomplished fairly readily when high-pressure blast air is used for injecting the fuel, since the air stream then serves both to pulverize the fuel and, by creating additional and violent turbulence, to distribute the finely pulverized fuel more or less uniformly throughout the combustion chamber. With so-called solid injection, i.e. without the aid of air blast, the problem is far more difficult and the conditions conflicting. On the one hand, in order to reach the outlying air, we must have a fairly "hard" or penetrating jet; on the other hand, to give adequate dispersion of the particles of fuel, we require a "soft" or pulverizing jet. Thus solid injection starts with the disadvantage of having two apparently irreconcilable conditions to fulfil and it is in the attempts to fulfil

or circumvent these conditions that so much ingenuity has of late been displayed. No complete reconciliation has yet been found, but compromises have been attained which render the solid injection at least as efficient as the air injection engine, and thus render possible the light high-speed self-contained heavy-oil engine.

As stated previously, the thermo-dynamic cycle adopted in the high-speed Diesel engine as it is being developed to-day (1929) resembles much more nearly that of the petrol engine than of the conventional Diesel engine.

Before proceeding further, it is interesting to compare the two cycles.

(1) In the petrol engine, the whole of the fuel and air are intermixed very thoroughly not only before ignition but even before admission to the cylinder.

In the Diesel engine, air only is admitted to the cylinder during the suction stroke, and the fuel is injected at or just before the end of compression, hence the process of intermixture and of ignition are co-existent.

(2) In the petrol engine, ignition is brought about by the passage of a spark across the electrodes of the ignition plug; this leaves behind it a minute thread or nucleus of flame which spreads slowly at first, but with rapidly increasing velocity, to the envelope of mixture immediately surrounding it.

Under the conditions of indiscriminate turbulence which exist in any practical engine, the flame once fully established is torn and spread throughout the cylinder with the result that the whole cylinder contents are inflamed with a rapidity many times in excess of that which would occur under stagnant conditions. The combustion process with spark ignition may be regarded as taking place in two distinct stages, first the building up of a self-propagating nucleus of flame, and second the spread of that flame throughout the working fluid. The first phase occupies an appreciable interval of time, more or less independent of engine speed, during which there is no perceptible rise of pressure. The actual time occupied by the building up of this flame nucleus depends upon:

- (a) The chemical nature of the fuel.
- (b) The mixture strength and therefore the flame temperature itself.
- (c) The temperature of the combustible mixture at the time of passage of the spark and therefore that of the surrounding envelope.

(d) The pressure of the combustible mixture at the time of ignition, for this controls the intimacy between the inflamed and uninflamed particles.

The second phase, namely the spread of flame throughout the main body of the combustible mixture depends upon:

- (a) The shape and form of the combustion chamber.
- (b) The degree and nature of the turbulence whose function it is to spread the flame rapidly from zone to zone.

This second phase is a mechanical one pure and simple, and since turbulence increases pro rata with increase of speed, it follows that the time occupied by this stage varies with the speed of the engine and decreases as the engine speed is increased.

In the petrol engine then we may consider the combustion process as taking place in two distinct phases, the first of which may be termed the delay period occupied by the building up of a substantial nucleus of flame; during this period there is no perceptible rise of pressure. Under any given set of conditions as to fuel, mixture strength, compression pressure, and temperature, this delay period occupies a substantially constant period of time irrespective of engine speed; therefore, as the speed increases the ignition timing must be advanced to compensate for the "delay".

The second phase, that of rapid pressure rise is determined by mechanical design and turbulence; under any given set of conditions its period is constant in terms of crankangle but not of time. The author prefers to express this stage in terms of increase of pressure in pounds per square inch per  $0^\circ$  of crankangle. The two periods of course merge into one another, and as explained in Chapter IV it is the manner in which they merge which determines primarily the roughness or smoothness of running.

If the speed of a petrol engine be doubled, the delay period in terms of crankangle will be doubled, or nearly so, but the period of pressure rise will remain the same. Thus fig. 266 shows typical diagrams on a time basis from a petrol engine running at 1000 R.P.M. (above) and 2000 R.P.M. (below). In these diagrams the compression expansion line (when ignition is shut off) is shown dotted. It will be seen that as between the upper and lower diagrams the angular movement from the passage of the spark until any appreciable rise of pressure occurs is approximately doubled, being of the order of  $10^\circ$  and  $20^\circ$  respectively, while the pressure rise occupied approximately  $10^\circ$  crankshaft degrees in each case and amounts in this instance to 35 lb. per square inch per  $0^\circ$  of

crank-angle. The completion of Stage II marks the end of the combustion process so far as the petrol engine is concerned.

In the case of the Diesel engine with solid injection a stream of liquid fuel is injected into highly compressed and highly heated air. In this case the combustion process may be regarded as taking place in three distinct phases.

(1) A delay period, during which fuel is admitted but either no ignition takes place or the ignition is restricted to some very

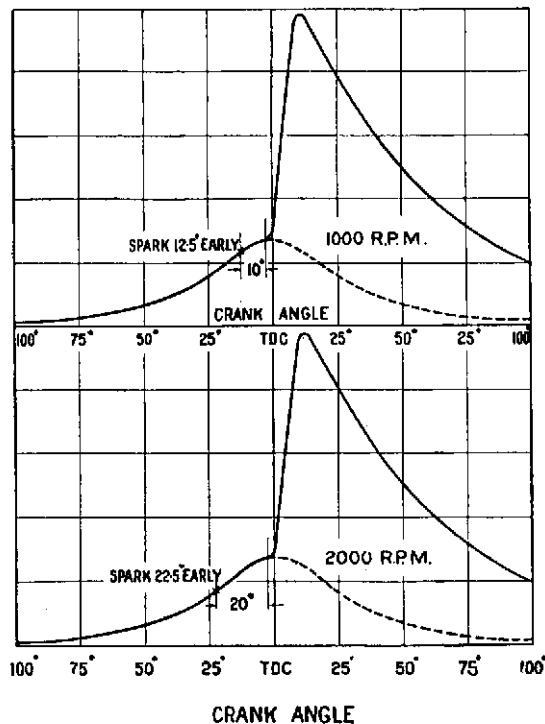


Fig. 266.—Diagrams showing increase of "Delay" Period with increase of Speed

localized nucleus. It seems probable that ignition first starts from some nuclear point so soon as only a minute portion of the injected fuel has had time to vaporize and to mix with sufficient oxygen to form a readily combustible mixture, and that from this nucleus the flame spreads at first in much the same manner as in a spark-ignited petrol/air mixture during the first phase. As in the case of the petrol engine the delay period appears to be constant in time and to depend upon:—

- (1) The nature of the fuel.
- (2) The temperature and pressure of the highly compressed air.

(3) The fineness of pulverization, not so much of the fuel stream as a whole, but rather of the finest particles, for clearly an infinitesimal proportion of the total fuel charge will suffice to provide for this initial nucleus of flame.

Except for (3) the conditions as regards this first delay period are substantially the same for the petrol as for the Diesel engine, and in both cases the period of delay is constant in terms of time but not of crankangle. The building up of the nucleus is, however, much more rapid owing to the higher pressure and temperature of the cylinder contents.

The second phase consists, as in the petrol engine, of the mechanical spread of flame from the initial nucleus to the main body of the combustion chamber. Here again the conditions are similar to that of the petrol engine in that the rate of spread of flame and therefore the rate of pressure rise is dependent on turbulence and is constant in terms of crankangle rather than of time. As in the case of the petrol engine this phase can be speeded up to any desired extent by varying the turbulence. There is, however, this essential difference that whereas in the petrol engine the whole of the fuel necessary to complete combustion is already present in the cylinder, in the Diesel engine only a proportion and in some cases a small proportion is present during phases (1) and (2), hence the pressure though its rate of rise is of the same order as that of a petrol engine with similar conditions as to turbulence, does not reach to anything like the high maximum which it would do were sufficient fuel present in the cylinder to combine with the whole of the oxygen. Clearly also phase (2) is influenced strongly by the amount of fuel present in the cylinder and the manner of its distribution throughout the air, for the mixture is no longer homogeneous as in the petrol engine, and the spread of flame during this period can take place only throughout those zones where a correctly proportioned fuel/air mixture exists at the time. The actual pressure attained at the completion of phase (2) depends upon:

(1) The extent of the delay period, which, in turn, depends upon the engine speed.

(2) Upon the rate of injection of the fuel which determines the amount of fuel available in the cylinder at this period.

(3) Upon both the temperature and pressure of the highly compressed air, and

(4) Upon the fineness of pulverization and the uniformity of distribution at this early stage in the proceedings.

Once the flame has spread through the combustion chamber, the increase in pressure and temperature is so great and the rate of combustion so accelerated in consequence, that the remainder of the fuel burns as it enters from the injector nozzle, thus bringing about a further rise of pressure or the maintenance of constant pressure depending upon the rate at which fuel is being delivered by the supply pump. During this, the third phase, the burning is under direct mechanical control, and the pressure can be varied at will by varying the rate of injection.

Fig. 267 shows a diagrammatic indicator diagram in which the three stages are shown as quite distinct, while fig. 268 shows an actual

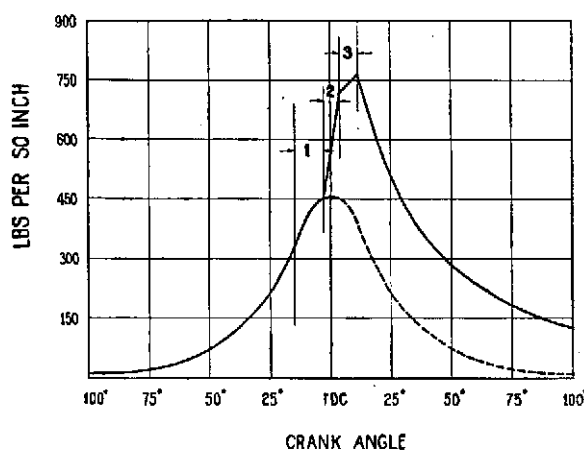


Fig. 267.—Diagram showing three phases of combustion process in Compression Ignition Engine

card taken from a small high-speed Diesel engine of  $5\frac{1}{2}$ -in. bore by 7-in. stroke running at 1500 R.P.M., from which the three phases above referred to can be identified though they are of course merged together to some extent.

Consideration of the above suggests that since:

(1) The delay period is constant in time while the rate of injection is proportional to crankangle; the higher the engine speed, the greater the proportion of fuel that will be admitted during this period, hence the greater the quantity available for phase (2) and the higher the maximum pressure. From this it would appear, as experience indeed indicates, that under any given set of conditions as to rate of injection, temperature, &c., the higher the speed the higher the maximum pressure involved if the same efficiency is to be maintained. It would appear also that in order to keep down the



maximum pressure it is desirable to inject the fuel not at a constant rate, but rather gradually at first, so that not too much shall enter during the delay period.

It used to be, and to a large extent still is argued, that the combustion process in a compression ignition engine, including the vaporization of the liquid particles, the subsequent admixture of this vapour with sufficient air for combustion, and finally the combustion of the air/vapour mixture thus formed must necessarily take so long a period of time as to render the high-speed compression ignition engine either entirely impracticable or at least so inefficient as to be of very little service. Such arguments appear to be entirely

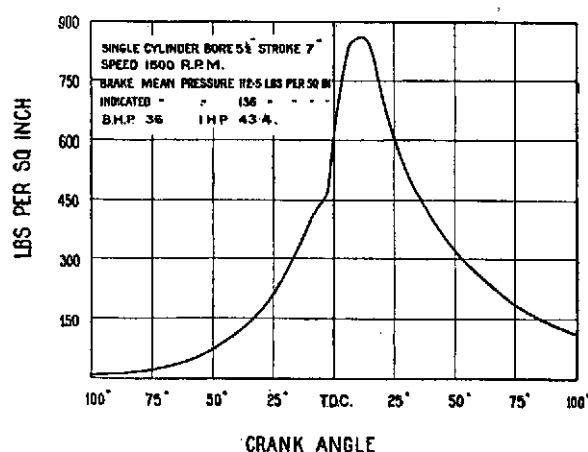


Fig. 268

unfounded; not only have they been refuted by practical results, but they appear also to be based on entirely false premises. Except that a very minute quantity of vapour must be present to provide the initial nucleus of flame, there does not appear the slightest ground for supposing that the rest of the fuel must be vaporized completely before combustion can take place; on the contrary, it appears far more probable that combustion actually takes place from the surface of the liquid droplet and that it is fed with fresh oxygen and the burnt products scoured away by the rapid passage of this droplet through the air. The really essential condition for high speed being sufficient relative motion between the drops of fuel and the air—a condition which can be met either by rapid movement of the fuel or of the air, or both (provided, of course, their direction is not the same).

It does appear, however, that for any given fuel the relative

proportion of the delay period must increase as the engine speed is raised, that the extent of this period in relation to the rate of injection determines the maximum pressure and that therefore the higher the speed, the higher the maximum pressure. In other words, the higher the speed, the greater the departure from the constant pressure and the nearer the approach to the constant volume cycle.

A large number of experiments have been carried out at the author's laboratory during the last eight years with a view to investigating these several stages in the combustion process. For this

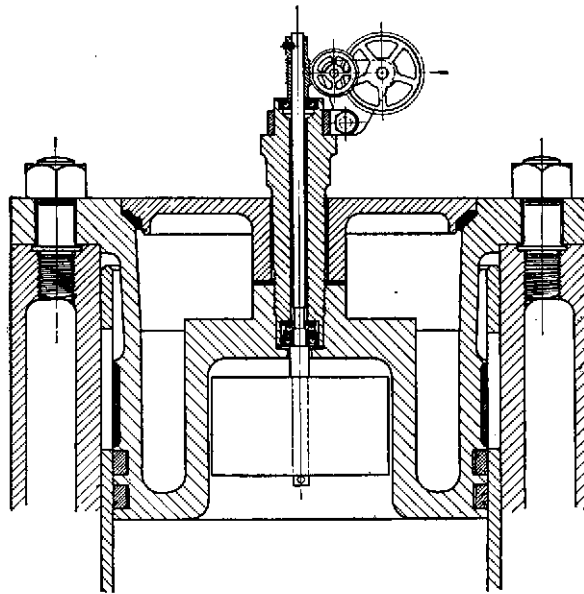


Fig. 269

purpose a number of sleeve-valve engines have been used because with this type of valve gear

(1) Both the degree and the nature of turbulence can be varied at will by means of guide vanes in the entry ports. Thus by admitting the air tangentially an almost perfectly organized rotary swirl can be obtained—by admitting the air at various angles and at varying velocities, almost any degree of general rough and tumble turbulence can be produced at will; further by the use of external guide vanes the conditions as to turbulence can be varied while running and their effect observed by merely altering the position or angle of the vanes.

(2) The use of a sleeve valve allows complete freedom of manoeuvre as regards the shape and proportion of the combustion chamber.

(3) An anemometer can be fitted within the combustion chamber as shown in fig. 269 whereby the speed of rotation of the air can be measured and recorded while the engine is running, if not under its own power, at least when motored at normal speed.

(4) For measurements of pressure rise, the latest type of Farnboro electrical indicator shown in fig. 270 has been used throughout

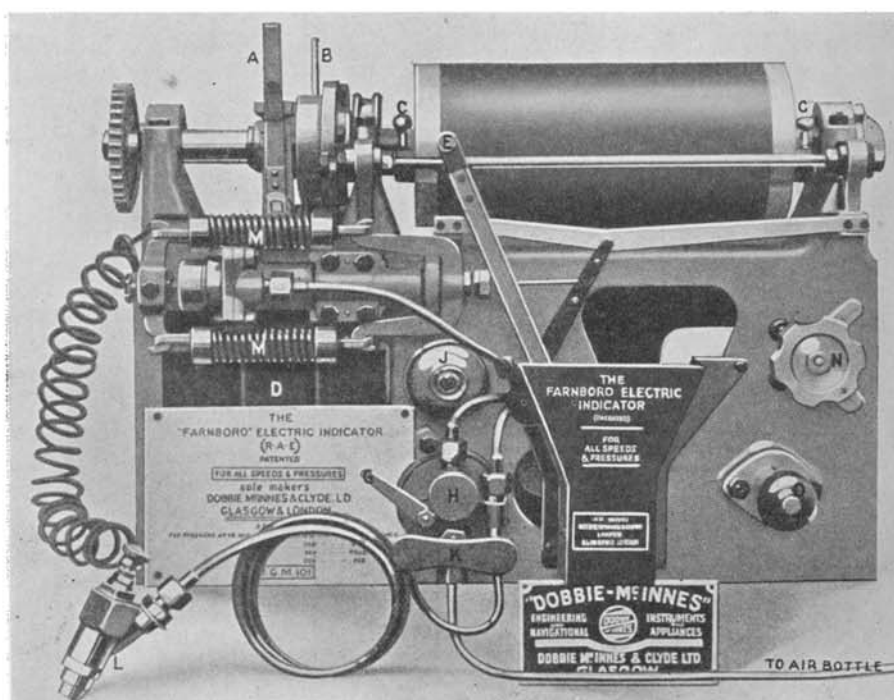


Fig. 270.—Dobbie-McInnes Indicator

all the experiments and a special adaptation of this indicator has been made whereby the timing and pressure in the fuel injection system can also be observed and recorded at the same time. A sectional drawing of the special indicator valve evolved for this purpose is shown in fig. 271, and a typical diagram obtained therefrom in fig. 272.

(5) For measurements of indiscriminate turbulence, means have been provided for blowing through the inlet ports at certain predetermined air velocities, and the direction of the air currents explored by means of a pitot tube and air-speed indicator.

The general gist of all the experiments has shown that:

(1) As regards Phase 1—In a Diesel engine the delay period depends, as would be expected, primarily upon the nature of the fuel, and what in a petrol engine would be considered as its tendency to detonate, the lower the ignition temperature of the fuel, and therefore the greater the difference between its self-ignition temperature and that of the surrounding air the shorter the delay period. This period can be further shortened by the addition of small quantities of pro-detonating dopes; such, for example, as amyl nitrate, which when added to petrol will increase greatly the tendency to detonate. In this respect the requirements are the exact converse of those for petrol.

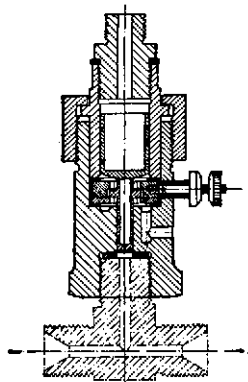


Fig. 271

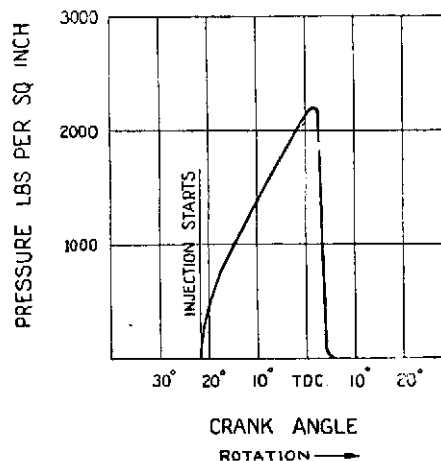


Fig. 272

(2) The delay period can be shortened by employing a higher compression ratio, thus both increasing the temperature difference between the highly compressed air and the self-ignition temperature of the fuel, and also the density and therefore the intimacy of the two. Increase of compression, however, is not altogether a satisfactory method, for while it reduces the delay and therefore also the rise of pressure above that of compression, yet this is almost completely compensated for by the higher initial compression pressure, while the slightly higher efficiency due to the increased expansion ratio is almost, if not completely, counterbalanced by increased friction and heat losses.

(3) The delay period can also be shortened a trifle by pre-heating the air or by running at a higher cylinder temperature, or both, but only, of course, at some expense in volumetric efficiency.

(4) The delay period appears—but here the evidence is much less definite—to depend also to some extent upon the degree of pulverization of the fuel. It is probable that it depends only on the degree of pulverization of some infinitesimal proportion of the fuel, and since almost every form of fuel spray contains at least a small fringe of finely atomized particles, it is exceedingly difficult to determine this point by experiment.

As regards Stage 2, namely, the spread of inflammation from the initial flame nucleus through the air, and such of the fuel as has been admitted during the delay period, this depends solely on the nature and degree of turbulence—the more violent and the more indiscriminate the turbulence, the more rapid the spread of inflammation, and, therefore, of the rate of pressure rise. In this respect, the conditions are exactly on all fours with those which obtain in a petrol engine. Just as in a petrol engine, it has been found that the rate of pressure rise follows the degree of indiscriminate turbulence as this is increased or diminished by means of the guide vanes referred to above. Also, as in a petrol engine, organized turbulence has, during this stage, little or no influence, since it merely carries the flame along with the procession instead of disseminating it through the mob. In order to spread the flame throughout the mass, and so raise the temperature in preparation for Phase 3, a high degree of indiscriminate turbulence is essential, and it is unfortunate that this requirement involves a rate of pressure rise considerably beyond that consistent with smooth running. Since this high turbulence and consequent rapid rate of pressure rise appears to be essential, the ensuing roughness of running can probably best be minimized, either by reducing as far as possible the delay period, or by employing at first a relatively slow rate of injection, or both, in order that the minimum quantity of fuel shall be present in the cylinder during this period of rapid pressure rise. Here certainly there is scope for much further investigation.

As regards the third phase, namely that in which the fuel burns from the surface of the liquid droplets as they enter from the fuel injector. This is the only one in which one has direct mechanical control over the rate of burning, and therefore over the pressure. It should be the aim of every designer so to arrange his conditions that Stage 3 shall be the main theme and shall form the bulk of the combustion process; Stages 1 and 2 being merely the prologue. This stage becomes possible only when the temperature and pressure have been raised by Stage 2 to beyond a certain minimum.

For Stage 3 the all-essential condition is that there shall be rapid relative motion between the droplets of liquid fuel and the air for combustion, so that each individual droplet shall travel like a rocket, leaving behind it its trail of combustion products, and shall proceed onwards to fresh fields of adventure, thus to search out and combine with as large a proportion as possible of the oxygen in the combustion space. This condition can be met only by securing rapid relative motion between the fuel and the air—such rapid motion can be obtained in various ways:

(1) By the use of very high injection pressures, thus giving to the fuel itself a high initial velocity. This method is, however, somewhat handicapped, since the speed at which a liquid jet can issue from an orifice is limited; for beyond a certain point increase of pressure results merely in breaking up the droplets rather than in increasing their velocity; moreover, the initial velocity gained by high initial pressure is soon dissipated. It is open also to the practical objection that very high pressures in conjunction with a somewhat elastic medium such as fuel oil add greatly to the problem of timing and metering. The general principle of this system is illustrated in fig. 273.

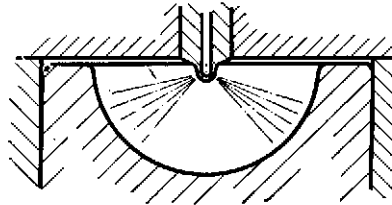


Fig. 273

(2) By the use of a so-called pre-combustion chamber which consists of a small chamber separated from the main combustion space and communicating through a number of small holes. Fuel is injected into this small antechamber where Stages 1 and 2 develop; during Stage 2 the rapid rise of pressure projects the remaining unburnt fuel along with highly heated air, and some products of combustion, into the main combustion chamber at a velocity far in excess of that which can be obtained by any purely mechanical injection of the liquid fuel. This method has certain obvious advantages, unlike (1) the engine speed is not limited by the speed of the injected liquid, nor is there the same difficulty in distributing the liquid particles throughout the air, since any number of holes can be provided in the pepper castor interposed between the pre-combustion chamber and the main chamber, whereas the number of jets for liquid fuel is strictly limited by mechanical considerations. On the other hand, the loss of heat entailed by passing the products through a number of small holes at so high a tem-

perature and velocity is very great, and curtails very seriously the efficiency and power output. This principle is shown diagrammatically in fig. 274.

(3) By producing organized unidirectional flow of the air itself and then projecting the fuel across the air stream. In this manner the highly heated air sweeps past each drop of fuel as it enters, carrying away the products of combustion as they are formed, and continually bringing up fresh or only partially contaminated air to the attack. Since the velocity of the air can in any case be enormously greater than that of the liquid fuel, and is at all times proportional to the engine speed, further, since it can be maintained almost constant throughout the combustion process, this system

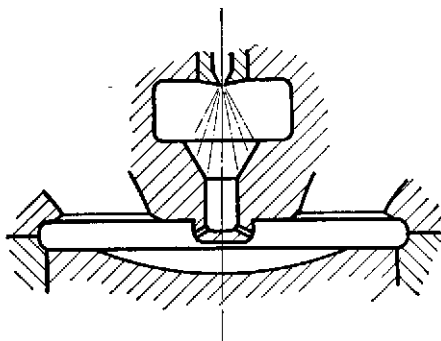


Fig. 274

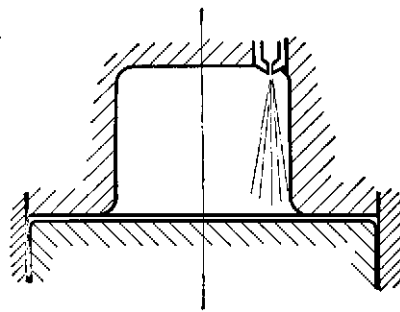


Fig. 275

is, in the author's opinion, much the more promising of the three; moreover, it has some very important practical advantages which will be dealt with later. This principle is shown diagrammatically in fig. 275.

In the case of Stage 3 the problem is very closely related to that of carburation in the average petrol engine; in both cases the aim is to distribute the particles of liquid fuel uniformly throughout the air, the chief difference being that in one case they evaporate and in the other they burn from the surface; but this difference does not affect the mechanical problem, which remains much the same in either case.

The three systems referred to above correspond in effect with three systems of carburation, namely:

(1) By projecting a spray of liquid petrol under pressure into the air—a method which has been tried from time to time, but never adopted on any considerable scale.

(2) By the use of a "diffuser" jet as in the Claudel type of car-

burettor—this corresponds to the pre-combustion chamber system and

(3) By concentrating all the air at a high velocity past the fuel jet by means of a choke tube as in most ordinary carburettors, this corresponds with the last system described, namely, that of air swirl or unidirectional flow.

Yet a fourth principle, and one which does not fall strictly into any of the above categories, is that known as the Acro system. In this system practically the whole of the air for combustion is compressed into a chamber, usually in the piston, and communicating with the rest of the cylinder through a narrow restricted neck. The fuel injector is so placed that the jet of liquid is aimed at the centre of this narrow neck as shown in fig. 276.

Fuel injection starts slightly before top dead-centre and is continued relatively slowly. Stages 1 and 2 of the combustion process are enacted more or less inside the chamber, where the indiscriminate turbulence is very intense, but since the fuel injection is timed rather later and is spread over a fairly long period, very little fuel has actually been injected before the piston starts to descend, when the highly compressed air within the

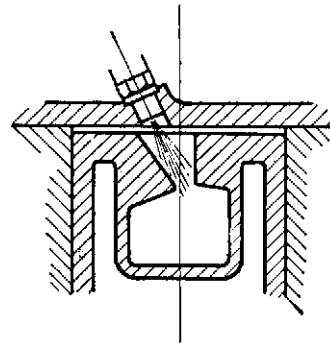


Fig. 276

chamber commences to rush out through the neck. Stage 3 is therefore enacted outside the chamber under conditions of intense air flow, for during this stage air is rushing helter skelter out of the chamber, and in so doing meets the injected fuel, which is approaching to meet it. The combustion process therefore, during Stage 3 resembles that of the last system described, but with this essential difference, that in the Acro system Stage 3 can occur only while the piston is descending fairly rapidly, that is, relatively very late, hence the expansion ratio is curtailed, and both the power output and efficiency somewhat limited in consequence. In the Acro system it is difficult to determine just where combustion takes place, but it probably oscillates rapidly about the open neck of the bulb; the bulb itself is quite uncooled, and since it does not attain a high temperature, it is quite clear that the bulk of the combustion process must take place outside it and beyond the neck; on the other hand, as the pressure rises, even though the piston be descending, there must be reversals, probably rapidly alternating



reversals, of flow between the bulb and the cylinder, so that some combustion even during Stage 3 must take place within the bulb.

And now to compare again the conditions as between a petrol engine and a high-speed solid-injection Diesel engine. Stages 1 and 2 are common to both, with the exception that in the case of the petrol engine the whole combustion process is consummated in Stage 2, while the equivalent of Stage 3 is anti-dated, and is enacted not in the cylinder, but in the carburettor.

In the petrol engine we need indiscriminate turbulence in the combustion chamber and processional flow in the carburettor, in the Diesel engine we need both within the combustion chamber itself, and since the conditions are conflicting, we have to make the best compromise we can.

In the petrol engine, under all ordinary circumstances, the whole of the oxygen is consumed, hence the flame temperature is almost constant at any mixture strength or throttle opening. Since the flame temperature governs the efficiency it follows that in a petrol engine the true indicated thermal efficiency reckoned on the air consumption, as described in Chapter IV, is almost constant, irrespective of mixture strength or throttle opening, provided, of course, that the degree of turbulence and time of ignition are appropriate in all cases. In the Diesel engine under no circumstances is the whole of the oxygen consumed, since it is, in practice, impossible to bring the whole of it into contact with the injected fuel. As the load is reduced, less and less of the available oxygen is consumed, hence the mean flame temperature is diminished, and the efficiency rises. Even at full load, it is extremely difficult to consume more than about 75 per cent of the oxygen present in the combustion chamber, so that at all loads the flame temperature is lower and the efficiency on that account somewhat higher than in a petrol engine of equal expansion ratio.

In a petrol engine we aim to work at the highest possible compression ratio consistent with freedom from detonation or auto-ignition under any circumstances. In the Diesel engine we aim to work at the lowest compression ratio which will ensure auto-ignition under all circumstances. In the case of the petrol engine and with the average fuel available on the market we are compelled, except in the case of very small cylinders, to limit the compression ratio to about 6.0 : 1. In the case of the Diesel engine, to ensure auto-ignition without undue delay under the most adverse circumstances we are compelled to work with a ratio of about 12.0 : 1 despite the

fact that the heavier petroleum products have a lower self-ignition temperature than petrol. Between these two limits lies a wide neutral zone unattainable by either type of engine except by the use of anti-detonating dopes on the one hand or pro-detonating dopes or hot plates, &c., on the other hand; all of which expedients bring in their trail evils greater than those which they are designed to overcome.

For light high-speed Diesel engines, the use of air-blast injection may be ruled out as almost impracticable on the grounds of cost, complication and weight, and one has to fall back on solid injection, of which three general systems are in vogue.

(1) In which the fuel pump delivers to a small elastic reservoir in which a more or less constant pressure is maintained and the timing and to some extent the metering of the fuel is accomplished by mechanical operation of the fuel-injection valve.

(2) In which the fuel valve is spring loaded, but "automatic" i.e. without any direct mechanical control. In this case we have to rely on the fuel pump for both timing and metering. In certain circumstances this system can be employed without any fuel valve at all, but merely an open jet as in a carburettor.

(3) So-called spring injection, which is really only a variant of (1). In this system the plunger of the fuel pump is itself spring loaded, and in readiness to deliver the fuel at the moment when the fuel valve is lifted mechanically. System 3 in effect, differs only from (1) in that the pump body itself is utilized as the elastic reservoir.

In the case of System 2, which is by far the simplest and most generally used, we have to depend entirely on the pump both for metering and for timing to within very narrow limits, in practice to within about 5 per cent as to metering and as to timing to within about 1 crankshaft degree or  $\frac{1}{2}^\circ$  of pump shaft. Such close limits can be obtained either by simple design coupled with superlative workmanship or by more complex design and indifferent workmanship, and the choice must lie with the resources available. In either case the desired result can be achieved, and the above degree of accuracy as to timing and metering can be maintained.

In all fuel-injection systems the essential features are correct metering and an abrupt cut off at the completion of delivery in order to avoid dribbling at the fuel orifice, and therefore the formation of a carbon crater around the orifice. In the case of slow-speed engines it is common practice to employ a very abrupt cam acting direct on the plunger of the pump itself, and timed to give injection at the

correct moment, the stroke of the plunger being varied by suitable mechanical means. For obvious mechanical reasons such a system is not practicable in the case of a high-speed engine, nor is it a desirable one in any case. Almost all the successful fuel pumps for high-speed engines operate on the following general principle. The plunger is given a relatively long stroke either by means of an eccentric or of a cam with nearly constant acceleration, i.e. a harmonic motion. Of this long stroke a portion only is utilized, namely that during which the plunger is travelling at its highest velocity, this portion being picked out either by the use of a "flash" or piston valve, or by poppet valves. In all cases the time of commencement of injection is maintained constant, and the metering and period of delivery is controlled by the time of release, which is varied according to the load.

In the original experiments at the Royal Aircraft Establishment, referred to at the beginning of this chapter, the fuel was supplied under a pressure of some 6000 lb. per square inch by means of an ordinary plunger pump, and was stored under this pressure in a small steel receiver. The fuel valve, consisting of a heavily spring loaded needle valve, was timed and opened mechanically by means of a cam and the fuel delivered to the cylinder through a number of fine jets each directed to various portions of the combustion chamber. This system worked admirably from a functional point of view, but introduced some rather grave mechanical difficulties, chief of which were the operation of the fuel valve in so short a period of time, and under so intense a spring loading, and the accommodation on the cylinder head of the somewhat massive mechanism involved. The experiments, however, served admirably to demonstrate that the operation of a high-speed engine was not only possible but remarkably efficient. The results obtained at the R.A.E. in 1922, namely a brake mean pressure of 120 lb. per square inch at a piston speed of 2000 ft. per minute and a minimum consumption of 0.36 lb. per B.H.P. hour at this high speed, have never been surpassed. The pioneer work of the Air Ministry was quickly followed up by Messrs. Beardmore under the guidance of Mr. A. E. L. Chorlton, who decided to abandon the mechanically operated fuel valve in favour of an automatic spring-loaded valve operated by the fuel pressure as is customary in the case of solid injection engines. In the Beardmore engine, one fuel pump running at engine speed (instead of half engine speed as is more usual) serves alternately two cylinders. Each fuel pump is driven by an eccentric and

has a long stroke, only the centre portion of which is made use of. The timing and duration of delivery is determined by means of a "flash" valve—a quick-moving piston valve which passes over a port serving alternately as suction or release; thus pressure is created and delivery takes place only during the period while the flash valve is passing and so obstructing this port. The fuel valve itself is of the ordinary differential-area spring-loaded type, and the fuel finally issues through a number of directed jets as in the R.A.E. experimental engine.

In Germany the Robert Bosch Co., of magneto fame, have developed and have undertaken the commercial manufacture of an exceedingly neat fuel pump of the same general type as the Beardmore; though functionally exactly the same, it differs from the latter in that the pump runs at half engine speed (for four cycle engines) hence a separate pump is needed for each individual cylinder, also the flash valve is made integral with and forms the top portion of each plunger.

Since the pump operates at half engine speed, and since the period of delivery is necessarily very short—of the order of  $20^{\circ}$  to  $25^{\circ}$  crankshaft degrees only, on full load, it follows that operation by a crank or eccentric would necessitate a very long stroke, hence a cam is used in place of an eccentric giving a total plunger movement of rather more than double that actually utilized for fuel delivery.

Fig. 277 illustrates the general design and details of the Bosch pump. It will be seen that the flash valve is flat on its upper surface, but the lower edge forms a spiral; the period of delivery is varied by rotating the whole plunger, flash valve included, by means of a rack and pinion; since the top of the flash valve is flat, its rotation does not affect the time of start of delivery, but the spiral lower edge determines the period and quantity delivered.

The whole pump is extremely neat, simple, and compact, and in the author's experience behaves admirably, but it will readily be appreciated that its accuracy both of metering, and of timing, and its consistency, all depend on a standard of workmanship and a degree of hardness requiring very special manufacturing facilities and highly skilled craftsmanship. With such a design of pump there is practically no scope for adjustment either for wear or even for the slightest manufacturing error. That, in fact, it behaves so well is due to the superlative workmanship and material embodied in it. The Bosch pump may, in fact, be taken as an almost classic instance where superb workmanship and material permit the use of a simple

design such as would be impracticable unless made to limits far closer than those adopted or attainable by most of the highest class makers.

The fuel pump designed by the author, and illustrated in fig. 278, is an example of more complex design for which indifferent workmanship will suffice. In this pump the plunger has a long stroke, and may be operated either by a cam or, in the case of sleeve-valve

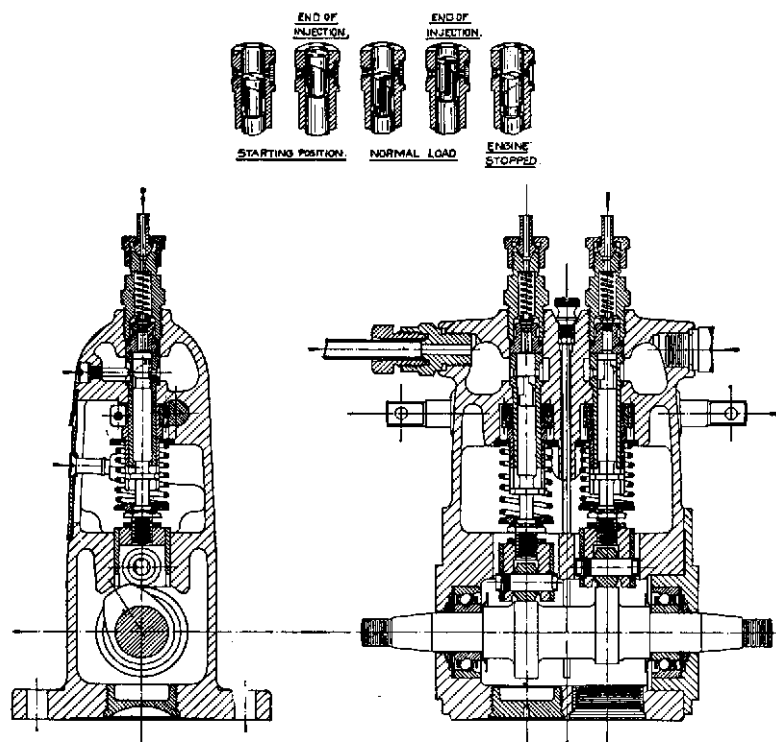


Fig. 277.—Bosch Fuel Pump

engines, directly from the beam operating the sleeve drive. Two cam-operated poppet valves are provided, one serving as the suction valve only, and the other serving both as a "spill" or release valve and, if desired, as an additional suction valve. The operation of the pump is as follows. During the outward stroke of the plunger one or both valves are open and functioning as inlet valves. Towards the latter part of the outward stroke the spill valve closes, but the suction valve remains open throughout, and until nearly half way up the delivery stroke, when it is closed mechanically; since the second valve has already closed, the only outlet remaining is through the fuel valve and into the cylinder. Thus the closing of

the suction valve determines the start of injection, and from then on injection continues until the second or "spill" valve is opened by means of a second cam; thereafter and during the remainder of the delivery stroke of the plunger, fuel is returned to the suction side of the pump. The closing of the suction valve determines the time of commencement of injection and is set by adjusting the tappet clearance as in any ordinary poppet valve gear; in the event of any

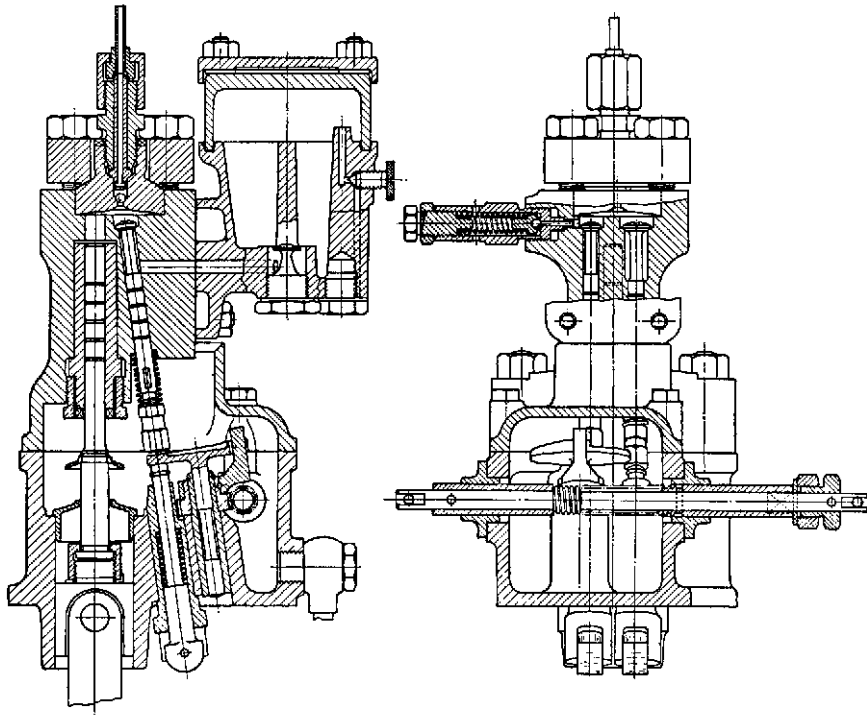


Fig. 278

derangement or wear, it can be readjusted with the same ease. The period of injection and therefore the metering is controlled by interposing between the valve and its tappet either a plain sliding or a rotating scroll wedge, movement of which varies the clearance and so forms the sole control of the engine; means are provided on this valve also, whereby the tappet clearance can readily be adjusted to ensure equality as between any number of pumps or cylinders. Fig. 279 and 279a show the valve-timing diagram for this pump. Since the pump plunger does not over-run any ports and since the fuel is under pressure only during the short delivery period, it follows that quite a slack fit will suffice, since the scaling length is very long. Again, in the event of leakage, the valves can readily be ground in,

( R 246 )

28a

and in the event of wear, the timing, &c., can be restored by means of tappet adjustment, hence the successful operation of the pump does not depend upon great accuracy of workmanship or fitting, or upon very accurate cam timing, &c. From a functional point of view, the three types of pump referred to are all practically identical, except

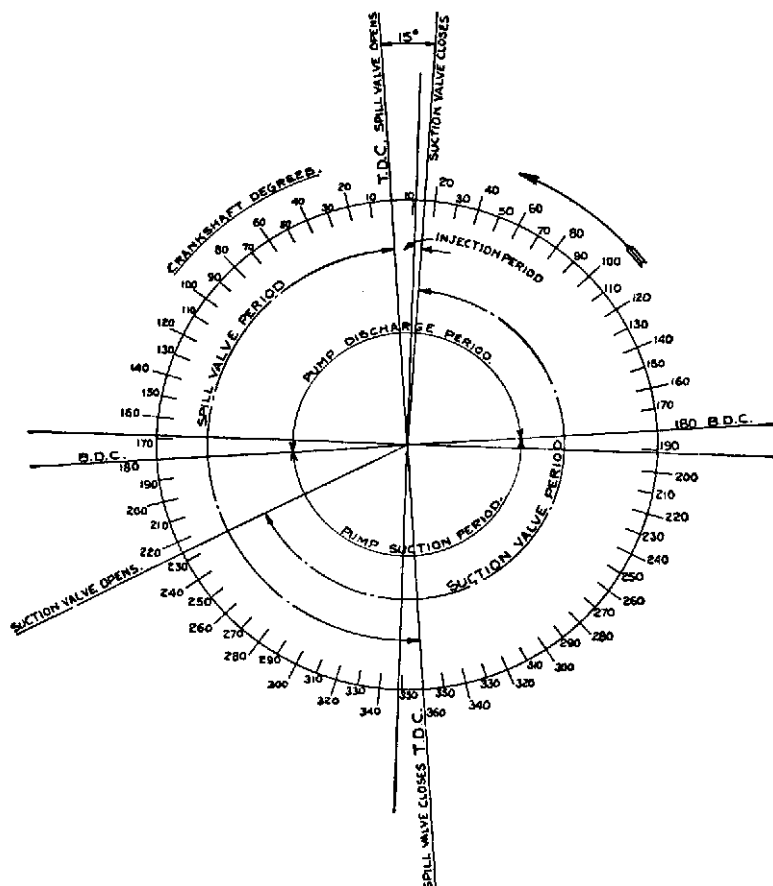


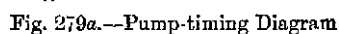
Fig. 279.—Pump-timing Diagram

in so far that the release of fuel at the end of the delivery stroke is much less abrupt in the Bosch pump (fig. 280) than in either of the others. Fig. 272 shows a typical indicator diagram taken from the author's pump. It will be noted that the characteristics of the two pumps are very similar, but that the quick-opening poppet type spill valve of large area gives a much more rapid "cut off" than does the flash valve of the Bosch pump.

Of the various methods of distributing the fuel within the com-

It has been found that the best results are obtained when the rotational speed of the air within the combustion chamber, as measured by means of an internal anemometer, is from 9 to 10 times the crankshaft

( F 24A )





speed, i.e. when the whole mass of the air within the chamber makes one revolution in from  $36^\circ$  to  $40^\circ$  of the crankshaft, which corresponds with the full-load combustion period.

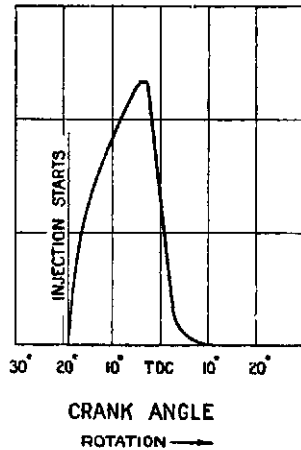


Fig. 280

With the use of a sleeve valve it is a very easy matter to adjust the speed of the air swirl by deflectors placed just outside the air inlet ports, and once these are set at the requisite angle the ratio of air rotation to crankshaft rotation remains practically constant at all engine speeds and under all conditions, since, of course, the entering air velocity increases *pro rata* with the crankshaft speed. Once the desired angle of entry has been determined by experiment on the first sample cylinder, the air inlet passages are thereafter cored in the casting to the requisite angle and no further adjustment is required.

Fig. 282 shows the results of a series of experiments on a cylinder of  $5\frac{1}{2}$ -in. bore 7-in. stroke, running at a constant crankshaft speed of 1300

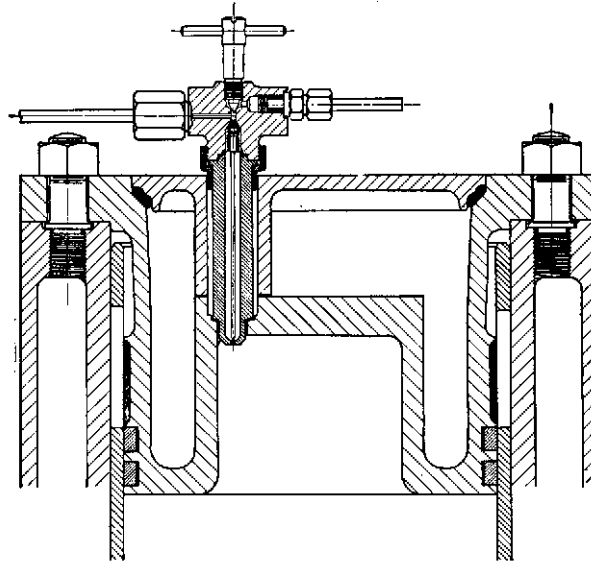


Fig. 281

R.P.M., in which by means of external deflectors the rotational speed of the air was varied from four times to twelve times the

crankshaft speed. It will be seen that as the rotational speed of the air is increased with no other change whatsoever, so the power output and efficiency increase, until a maximum is attained at, in this case, between nine and ten times crankshaft speed.

On the other hand, as the rotational speed is increased, so also is the loss of heat during compression owing to the more violent scrubbing of the air against the cylinder walls; the direct result of this is that the delay period Stage 1 is increased, and in consequence both the maximum pressure and the general roughness of running

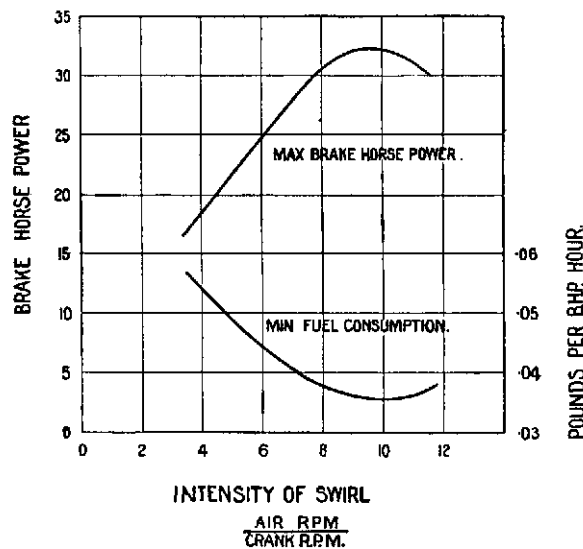


Fig. 282

are increased also. To some extent this can be counteracted by the use of a higher ratio of compression or a later injection timing, but to a limited extent only. It is necessary, therefore, to effect a compromise between smooth running and efficiency in exactly the same manner as one has to do in a petrol engine by adjusting the degree of turbulence. When working on this system the fuel is injected through a plain single-hole orifice of relatively large diameter, and without any valve or moving part other than the provision of a light non-return valve somewhere between the pump and the injector orifice merely to prevent air being forced back into the fuel system. In practice it is found preferable, for additional security, to employ two ball valves in series and as far apart as possible, one close up to the injection orifice and the other close to the fuel pump, leaving the whole oil capacity of the pipe line between the two. In this case,

should the first valve leak or fail to seat properly, the whole of the fuel in the pipe line would have to be driven back past the second valve before any air could reach the pump. The fuel stream is projected vertically downwards near the extreme outside edge of the combustion chamber, and no attempt whatever is made to pulverize it; a plain parallel hole of between 1 and 2 diameters long being used. A comparatively low oil pressure is required ranging from 1500 to 3000 lb. per square inch, according to speed, sufficient merely to propel the oil stream to nearly the bottom of the chamber; since there is no loaded valve, it follows that the pressure is controlled entirely by the resistance of the actual nozzle the size of which is selected merely to give the necessary penetration. With such a nozzle the fuel jet consists for some distance of an apparently solid core of liquid surrounded by an envelope of fine particles.

With such a system the progress of events during the combustion period is intended to be and probably is substantially as follows. Since, however, no means are known whereby the actual movements within the combustion chamber can be observed while running, one is compelled to fall back on speculation, but aided by a large amount of circumstantial evidence.

At the time of injection, the air within the combustion chamber is rotating at a very high speed. Into the outer edge of this whirlpool of dense air there is injected at right angles to the air flow a jet of liquid of which a portion, probably only a small portion, is in a finely divided state. Since the entering fuel has no tangential velocity and since, owing to centrifugal force, the pressure and velocity of the air are greatest at the outside edge, it follows that the main fuel stream will be deflected towards the centre while the finer particles will make almost straight for the centre, which is also the zone of highest temperature, and is relatively quiescent so far as mechanical movement is concerned. Here then the initial nucleus of flame is formed so soon as a few of the finer particles have reached this hot and peaceful zone, and have found the correct proportion of air for combustion. It is in this zone therefore that Stage 1 of the combustion process is enacted. Meanwhile, more fuel has been entering from the injector, some of which has found its way towards the centre, and an envelope of combustible mixture is being formed round the initial nucleus; to this envelope the flame spreads with great rapidity, thus raising the temperature and pressure of the central core and so reversing temporarily the pressure difference between the centre and outside. The flaming mass, now well estab-

lished and spreading rapidly, due to the indiscriminate turbulence which has reached its maximum intensity at about this time, extends radially outwards until it meets the entering fuel. This completes Stage 2. By this time the general temperature and pressure

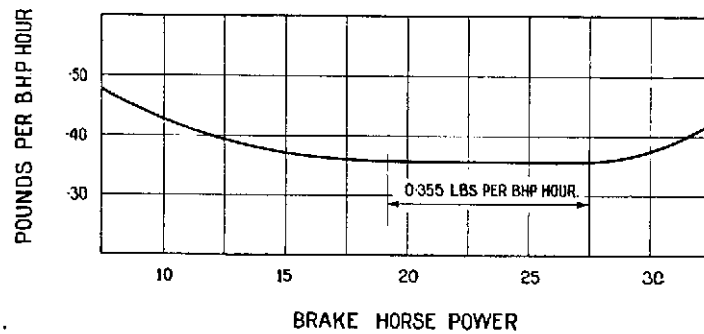


Fig. 283

conditions have risen so high that the fuel probably ignites as it issues from the jet. It then burns from the surface of the main core of liquid as it enters, fed by the high velocity rush of air past it, which carries off the products of combustion and continually replenishes with fresh oxygen. This process continues until the end of the

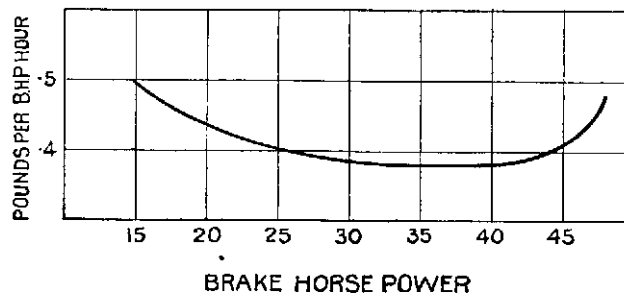


Fig. 284

injection. During this period, Stage 3, the rate of pressure rise can be controlled by the rate of fuel injection. By careful analysis of indicator diagrams and by other indirect means it would appear that from 20 to 30 per cent of the total available oxygen has been consumed before the fuel starts burning direct from the jet; there is still therefore some 70 to 80 per cent of oxygen available of which nearly the whole is swept past the entering fuel by the rapid rotation of the air. This, then, is the progress of events, so

far as it is possible to determine, from analysis of indicator diagrams, from markings on the piston, and in the combustion chamber,

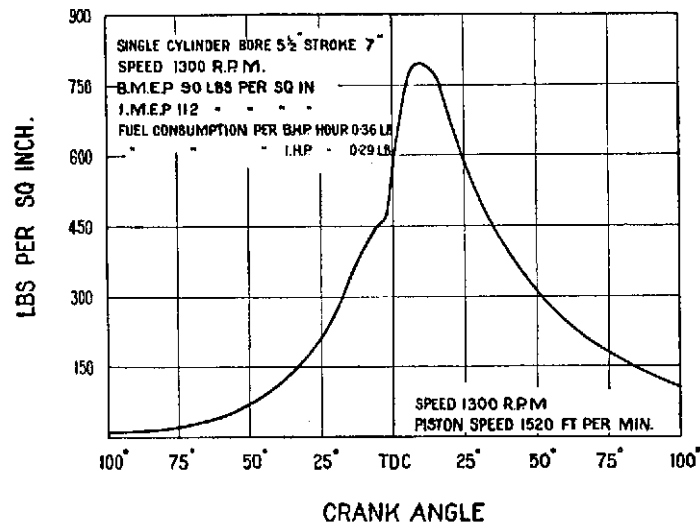


Fig. 285

and from its logical reaction to changes in the rate of swirl, rate of fuel injection, position of fuel injector, &c.

With such a method it is possible to utilize nearly 80 per cent of

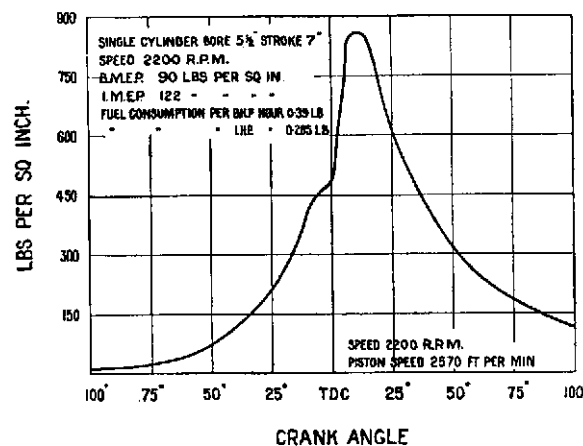


Fig. 286

the total oxygen available in the cylinder, thus realizing an indicated mean pressure of from 135 to 145 lb. per square inch, and that at any speed at which the engine is mechanically capable of running. Since the whole process with the exception of Stage 1 speeds itself

up automatically as the engine speed is increased, it follows that mechanical limitations and breathing capacity alone determine the speed. Experiments, employing this system, have been carried out on a number of experimental high-speed engines of from one to six cylinders, ranging from a small four-cylinder engine of  $3\frac{1}{2}$ -in. bore and stroke, running at a normal speed of 2500 R.P.M., to a large single-cylinder of 12-in. bore by 16-in. stroke, running at a normal

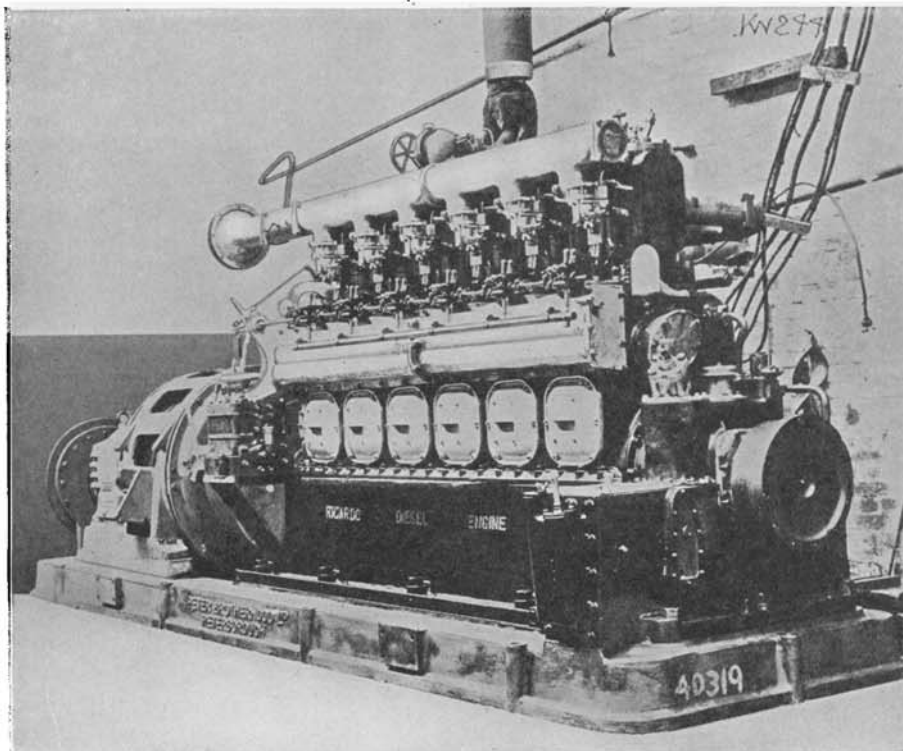


Fig. 287.—Brotherhood-Ricardo Six-Cylinder Engine

speed of 750 R.P.M. and developing 175 B.H.P. In fig. 283 is shown a graph of the performance of an experimental single-cylinder engine of  $5\frac{1}{2}$ -in. bore and 7-in. stroke, running at a normal speed of 1300 R.P.M., and developing up to 33 B.H.P. at this speed. This engine is completely self-contained, and drives all its own auxiliary gear, so that the performance is absolutely net. The maximum B.M.E.P. at 1300 R.P.M. is 119 lb. per square inch, corresponding to an indicated mean pressure of 141 lb. per square inch. The minimum consumption per brake horse-power hour is 0.354 lb. of commercial Shell Diesel oil of specific gravity 0.875. This particular engine is

fitted with adjustable deflectors to control the rate of air swirl, and the figures given are taken with a rotational swirl of 14,000 R.P.M., or about eleven times that of the crankshaft speed.

Fig. 284 shows a similar graph of performance of another single-cylinder experimental engine of the same dimensions but designed for higher speeds, the normal crankshaft speed being 2200 R.P.M., corresponding to a piston speed of 2570 ft. per minute, at which

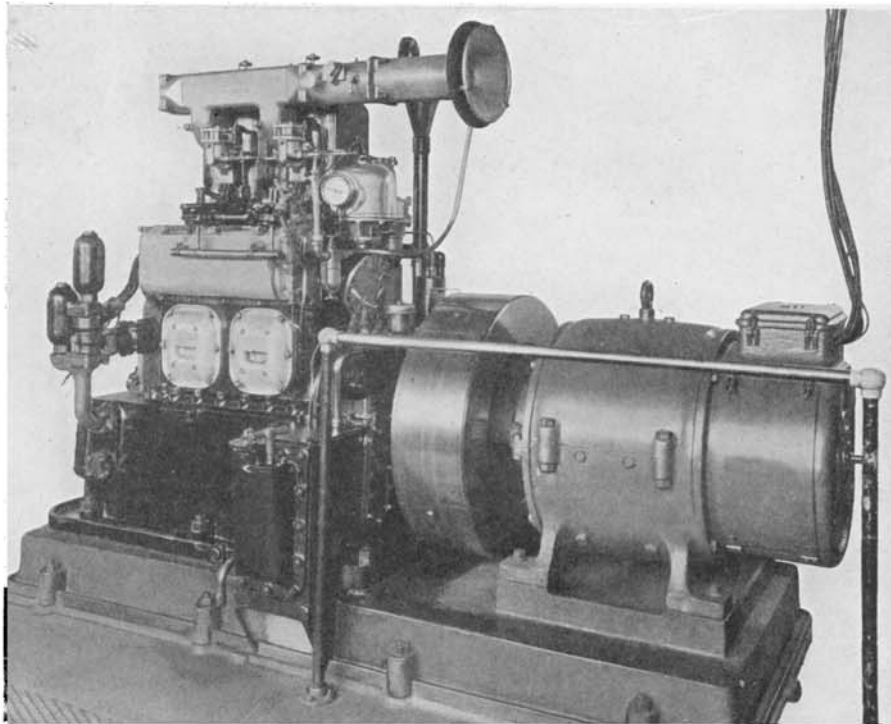


Fig. 288.—R 100-2 Engine and Dynamo on Combination Bedplate

speed the engine develops a maximum of 50 brake horse-power. Owing in part to the higher speed, and in part to other factors, the mechanical efficiency of this engine is not so high as the former, and though the brake mean pressure and fuel consumption are not so good, viz. a maximum of 108 lb. per square inch, and a minimum consumption of 0.385 lb. per B.H.P. hour at 2200 R.P.M., yet the indicated figures for both engines are almost exactly the same.

Figs. 285 and 286 show comparative indicator diagrams taken from the former engine at 1300 R.P.M., and from the latter at 2200 R.P.M., from which it will be seen that the influence of speed has little apparent effect on the indicator card. Figs. 287, 288, and 289

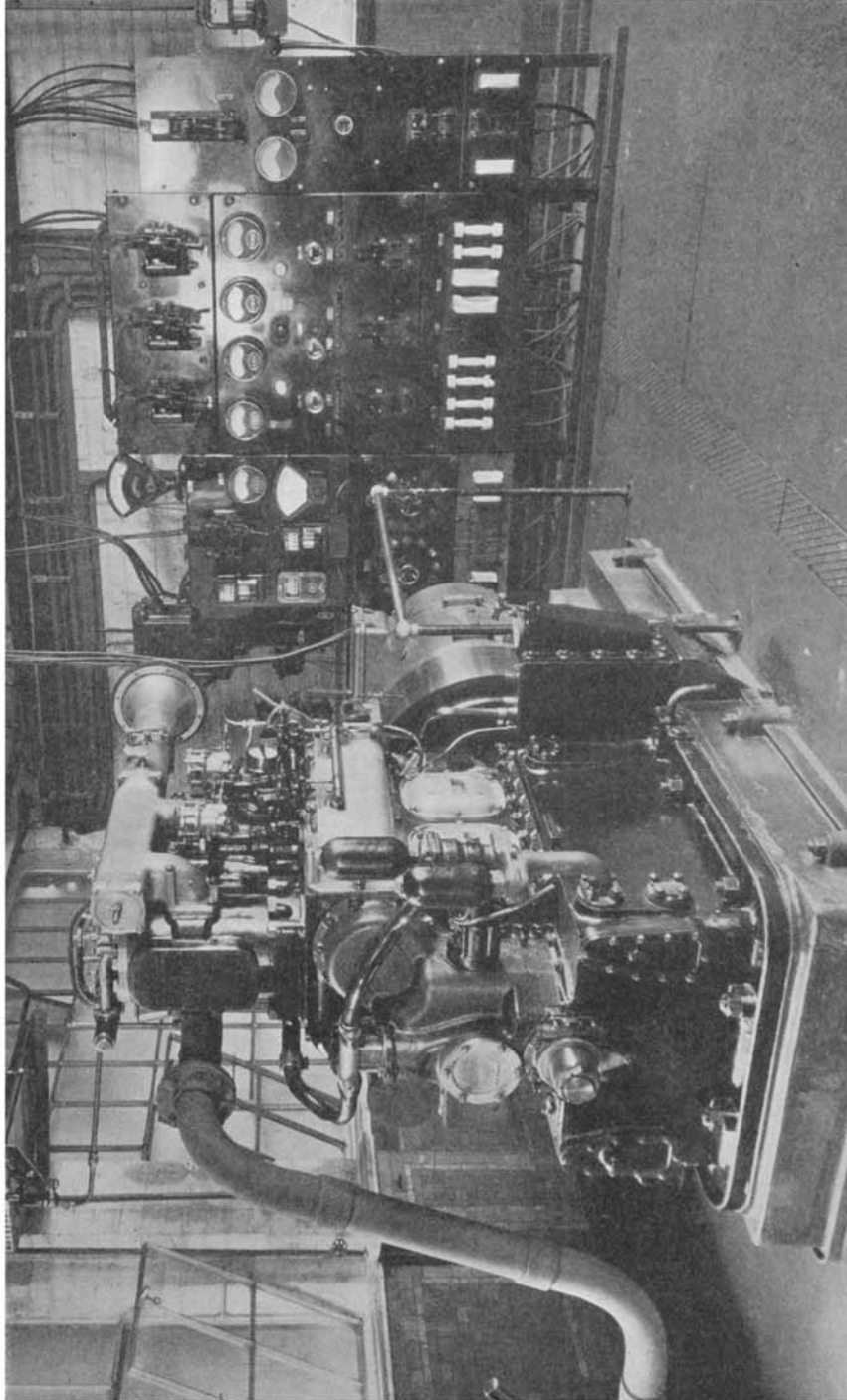


Fig. 289.—R 101-2 Engine Dynamo Unit and Power Station Switchboard



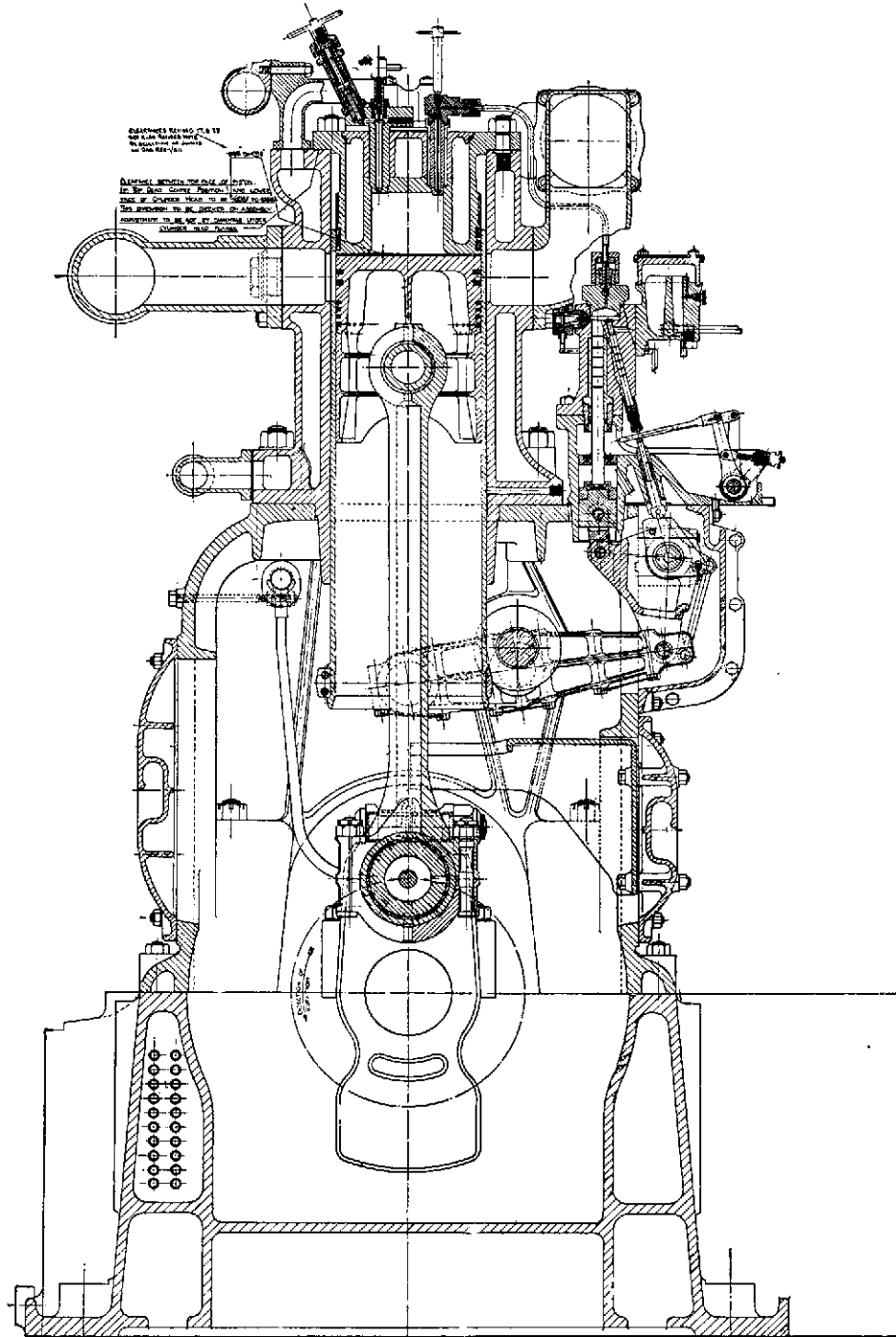


Fig. 290.—Arrangement of 7½" × 12" Brotherhood-Ricardo High Speed Diesel Engine  
Cross Section

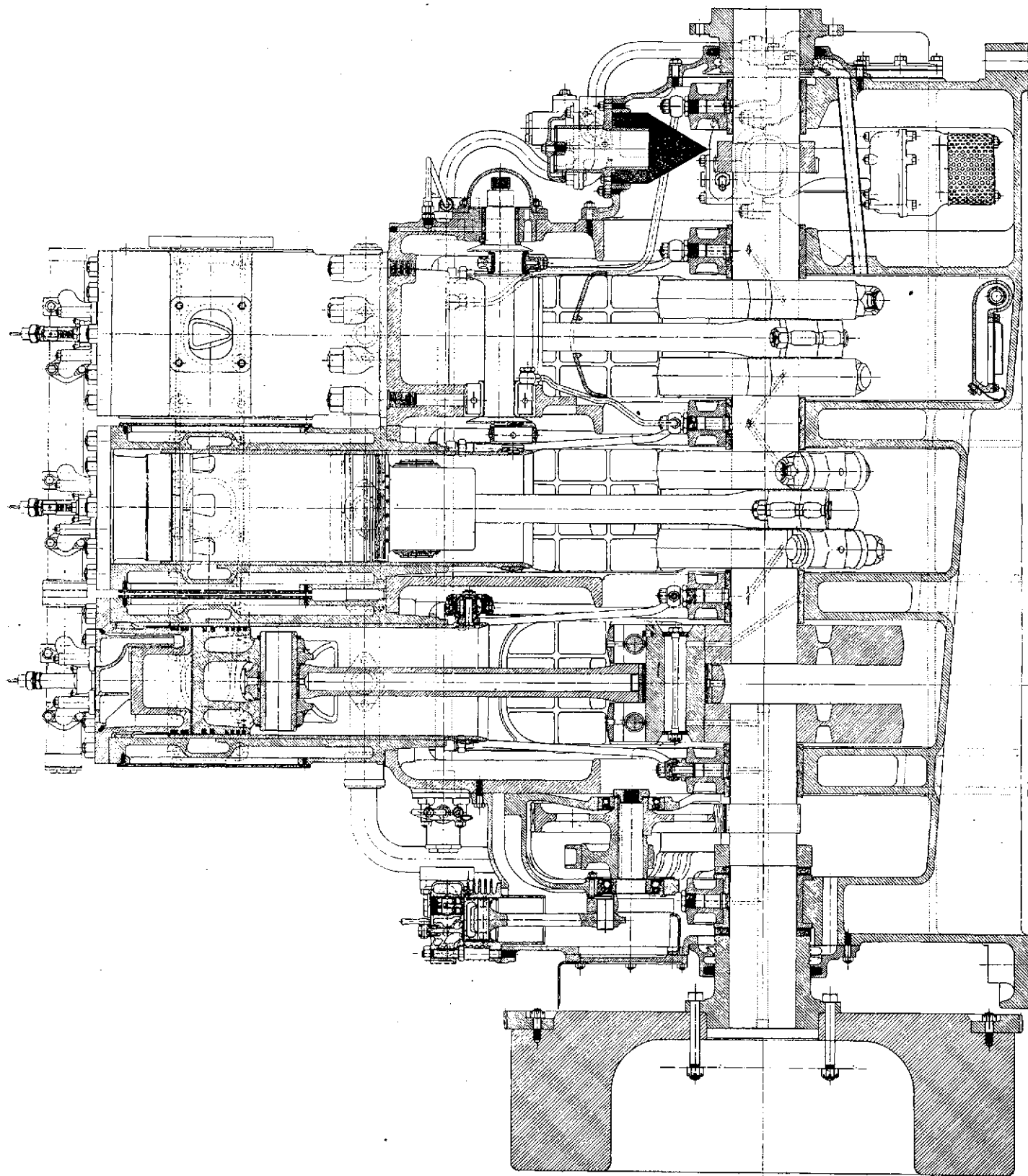


Fig. 291.—Longitudinal Arrangement of  $7\frac{1}{2} \times 12$ " "Brotherhood-Ricardo" High Speed Diesel Engine

show photos and details of a moderately high-speed engine designed for rough commercial work, and manufactured in this country by Messrs. Peter Brotherhood, Ltd., and in Holland by Messrs. Thomassen. It is made with any number of cylinders from two to eight, and is intended to run at a normal speed of 900 R.P.M. The dimensions of the cylinders are  $7\frac{1}{2}$ -in. bore 12-in. stroke, and the power rating at maximum economy is 50 B.H.P. per cylinder corresponding to an indicated mean pressure of 110 lb. per square inch. The fuel consumption at full load and normal speed and with a maximum pressure of 730–750 lb. per square inch ranges from 0.37 to 0.385 lb. per B.H.P. hour depending on the number of cylinders, the larger the number the higher the mechanical efficiency and the lower the fuel consumption. The normal piston speed is 1800 ft. per minute, but several engines are in regular service running at 1000 R.P.M., corresponding to a piston speed of 2000 ft. The sectional drawings, figs. 290 and 291 show clearly all the mechanical details. The single sleeve is operated by means of an overhanging beam driven by a lay shaft running at half engine speed. This gives to the sleeve an elliptical motion with the major axis vertical. By employing an elliptical motion of this nature, it becomes possible to use a considerable number of relatively high and narrow inlet ports—actually five in number—a form and arrangement of ports which is best suited to provide a tangential entry for the air. The fulcrum end of the operating beam is carried by a swinging bell crank, from the second arm of which the fuel pump plunger is operated directly. This provides a long stroke such as is needed when the plunger is not cam operated, it also has the advantage that the shock due to the sudden application of pressure at the time of injection is taken by the heavy mass of the sleeve and its driving mechanism and so is not transmitted to the gearing. The fuel pump is of the type already described with two mechanically operated valves. In the earlier engines these valves were operated from a rotating camshaft, which formed also the fulcrum shaft for the bell cranks and sleeve driving mechanism. This camshaft is now, however, being displaced in favour of rocking cams attached directly to the bell cranks, an alteration which has tended towards increased rigidity and simplicity.

The fuel injector is shown in fig. 292. It will be seen to consist of a hollow steel stem in one end of which is the actual fuel orifice—a plain  $\frac{1}{32}$  in. hole—and in the other end one of the ball check valves. In order to keep down the clearance and at the same time

to avoid drilling a long hole of small diameter, the intervening space between the ball valve and the actual orifice is filled in with a fluted steel plug. The top portion of the injector together with the fuel pipe union and a priming cock is formed as a separate piece.

The crankshaft is of the built-up type, with case-hardened pins and journals. The webs are shrunk on to the journals, but the pins are detachable, and held in place by means of clamp bolts, in the manner used in several aero engines. This allows of the use of float-

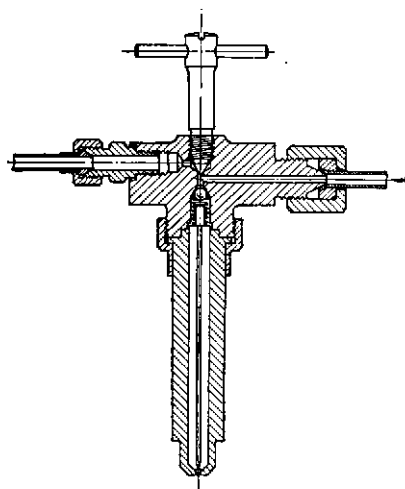


Fig. 292.

ing bearings for the connecting-rods, and also renders dismantling of the crank quite easy, for any crankpin can be withdrawn and replaced through the inspection doors in a comparatively few minutes merely by removing the adjacent main bearing cap, and slacking off the two clamping bolts.

The bedplate is water-cooled along each side and also under the centre crankshaft journal, and a hairpin tube oil cooler is inserted the full length of one of the bedplate water jackets. One end of the bedplate is divided off into a separate compartment to form the

main oil tank containing some fifteen gallons of lubricating oil. In this compartment are fitted two rotary oil pumps, one of which draws oil from the crank pits and delivers it through an Auto-Klean filter into the main tank, the second pump draws from the tank and delivers the oil under a pressure of about 12 to 15 lb. per square inch through the oil cooler and thence to all the important bearings.

The control of the engine is by means of a large centrifugal governor of orthodox type, which operates directly all the spill valve wedges. An over-riding hand control is, of course, fitted for use at starting or in any emergency. The remainder of the engine follows quite orthodox high-speed engine practice and calls for no particular comment.

*The author desires to thank the proprietors of "The Automobile Engineer" and Messrs. L. Gardner & Sons for permission to use several of the illustrations in this volume.*

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