PART TWO

DESIGN

CHAPTER X

GENERAL ENGINE LAYOUT

1. Basic comparative values

In designing an engine of a given power output, the requirements of light weight, compact overall dimensions and economy of operation must be borne in mind. By the latter not only a low specific fuel consumption is understood, but also simple maintenance, fatigue resistance and long engine life. If all these requirements are to be met, they must be first of all thoroughly studied by the designer in order to enable him to select the fundamental engine parts and general layout best suited for given conditions.

In determining the most suitable engine size, use is made of the familiar formula:

$$\text{b.h.p.} = \frac{A \,.\, L \,.\, i \,.\, n \,.\, p_e \,.\, z}{75 \times 60} = \frac{V n\, p_e}{C} \tag{41}$$

where $C = 450$ for two-stroke engines;

$C = 900$ for four-stroke engines;

$A = \dfrac{\pi D^2}{4}$, i.e. the piston area (cm^2)

L denotes the stroke (m)

i ,, ,, number of cylinders

n ,, ,, revolutions per minute

p_e ,, ,, mean effective pressure (kg/cm^2)

D ,, ,, bore (cm)

V ,, ,, total displacement (dm^3)

z ,, ,, number of expansion strokes per revolution.

By using the mean piston velocity formula

$$c_m = \frac{L \,.\, n}{30}$$

engine performance may be expressed as

$$b.h.p. = \frac{A \,.\, c_m \,.\, i \,.\, p_e \,.\, z}{150} = D^2 \,.\, c_m \,.\, i \,.\, p_e \,.\, z \,.\, \frac{\pi}{4 \times 150}$$

This equation makes it clear which values affect the engine performance. By expressing the stroke in terms of stroke: bore ratio, $\xi = \dfrac{L}{D}$ the following formulae are obtained:

$$\text{b.h.p.} = D^3 \,.\, \xi \,.\, n \,.\, i \,.\, p_e \,.\, \frac{\pi}{36\,000} \quad \text{for four-stroke engines and}$$

$$\text{b.h.p.} = D^3 \,.\, \xi \,.\, n \,.\, i \,.\, p_e \,.\, \frac{\pi}{18\,000} \quad \text{for two-stroke engines} \tag{42}$$

Before deciding on the size of cylinder to be used, the values of p_e, n, ξ, i must be chosen. Before this is done, relationships derived from the principle of similarity of engines must be considered.

Any two engines with the same basic layout and number of cylinders of differing dimensions may be considered as geometrically similar. For complete similarity some further conditions would have to be met: together with the changes of cylinder bore, proportional changes would have to be made not only in stroke, but also in connecting rod length, the size of the crankshaft etc., and even in the thickness of castings and the dimensions of electrical accessories, carburettors, fuel injection equipment, bolts and so on. These latter conditions cannot be met in practice, but this may be assumed to have been done if the dimensional differences in question are not considerable. In-line engines cannot be compared with V engines etc. The weight: bore ratios of six cylinder in-line engines are shown in Fig. 139, where weights of various

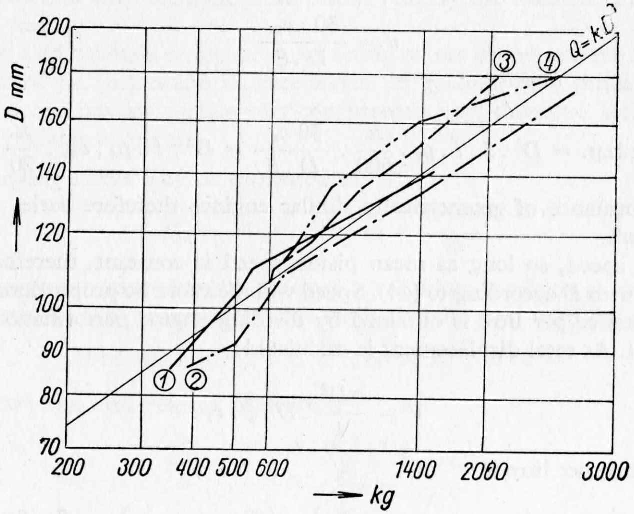

FIG. 139. Weights of six-cylinder in-line engines plotted against bores.

engines of one make are plotted against their bores and connected by one line in the graph. It is apparent that the weight: bore lines of all engines very closely follow the line $Q = k\ D^3$ which expresses the relationship between engine dimensions and weight.

In comparing two geometrically similar engines a means of comparison is provided by the ratio

$$m = \frac{D_0}{D} \tag{43}$$

where D_0 denotes the bore of the derived engine
D „ „ bore of the original engine.

For geometrically similar engines the following relationships hold:

all length dimensions are proportional to m,
all areas (of pistons, valves etc.) are proportional to m^2,
engine displacement is proportional to m^3,
weight of engine and parts is proportional to m^3.

Supposing that the mean effective pressure and mean piston speed of both engines under consideration are equal, further relationships may be expressed:

engine performance	$\propto m^2$,	performance per litre	$\propto m^{-1}$
speed	$\propto m^{-1}$	weight/performance ratio	$\propto m$.

Engine performance should be proportional to D^3 according to (42). Since, however, mean piston velocity is limited and remains constant, n may be expressed in terms of piston velocity as

$$n = \frac{30\ .\ c_m}{D\ .\ \xi} \tag{44}$$

Therefore

$$\text{b.h.p.} = D^3\ .\ \xi\ .\ i\ .\ p_e\ .\ \frac{\pi}{600}\ .\ \frac{30\ c_m}{D\ .\ \xi} = D^2\ .\ i\ .\ p_e\ .\ c_m\ .\ \frac{\pi}{20} \tag{45}$$

The performance of geometrically similar engines therefore varies with D^2 i.e. with m^2.

Engine speed, so long as mean piston speed is constant, therefore varies inversely with D according to (44). Speed will therefore be proportional to m^{-1}.

Performance per litre is obtained by dividing engine performance by displacement. As total displacement is calculated:

$$V = \frac{\pi D^2}{4}\ .\ D\ .\ \xi\ .\ i,$$

performance per litre is

$$\text{b.h.p.}/l = \left(D^2\ .\ i\ .\ p_e\ .\ c_m\ \frac{\pi}{20}\right) : \left(\frac{\pi}{4}\ .\ D^3\ .\ \xi\ .\ i\right) = \frac{p_e\ .\ c_m}{5\ .\ D\ .\ \xi} \tag{46}$$

The performance per litre varies therefore inversely with D and with m^{-1}. The smaller the bore, the larger will be the performance per litre.

The weight of an engine is proportional to the cube of any of its longitudinal dimensions and a given constant C_1. The weight/b.h.p. ratio can be expressed as

$$G_{hp} = \frac{D^3\ .\ C_1}{D^2\ .\ C_2} = D\ .\ C_3 \tag{47}$$

where C_1 C_2 and C_3 are constants.

The weight/b.h.p. ratio of geometrically similar engines therefore varies in direct proportion to their bore D and is directly proportional to the ratio m.

The volumetric efficiency which is affected by various conditions is also directly proportional to the velocity of the mixture (or air) at the inlet valve. Other considerations in respect of volumetric efficiency may be omitted with geometrically similar engines showing only small differences in dimensions. Gas velocity in the inlet valve is proportional to the port area (m^2) and piston displacement (m^3). It follows:

$$v_1 = \frac{m^3\ .\ C_1}{m^2\ .\ C_2} = m\ .\ C_3$$

But keeping the mean piston velocity constant, the amount of gas admitted is not proportional to m^3 but to m^2. Therefore

$$v_1 = \frac{m^2\ .\ C_1}{m^2\ .\ C_2} = C_3 \tag{48}$$

It follows that with constant mean piston velocity the volumetric efficiency does not vary.

Bearing load depends on gas pressure acting on the piston and the accelerating forces of the piston and its accessories. In geometrically similar engines piston area and bearing surface vary concurrently and, therefore, unit bearing pressure remains constant.

Accelerating forces may be expressed as

$$P_a = M\ .\ r\ .\ \omega^2; \quad \omega = \frac{\pi\ .\ n}{30}$$

Unit bearing pressure may be obtained by dividing the force P_a by bearing surface area A_1. The respective values vary with the ratio m as follows:

$$M \propto m^3,\ r \propto m,\ n \propto m^{-1},\ \omega^2 \propto m^{-2},\ A \propto m^2$$

It follows that unit bearing pressure is

$$\frac{P_a}{A} = \frac{m^3\ .\ m\ .\ m^{-2}\ .\ C_1}{m^2\ .\ C_2} = C_3 \tag{49}$$

Unit bearing pressures, therefore, remain constant in geometrically similar engines in which the mean piston velocity is constant.

With greater dimensional differences between geometrically similar engines it has to be remembered that some factors are not governed by engine size; these are, oil and air viscosity, the rate of combustion, pressure oscillations in the fuel system, the induction and exhaust pipes, etc. For this reason the compression ratio is generally reduced with increased cylinder dimensions. This connection between displacement and compression ratio is considerable as is shown by the results obtained from an experimental single cylinder engine with two stroke dimensions of 98·4 mm and 127 mm and bore varying from 70 mm to 95 mm. The experimental engine was water-cooled, its cylinder head was of cast iron, compression ratio variable and its temperature was kept at 100 °C. Spark advance was set for maximum b.h.p. and the fuel air mixture for maximum engine knock. The fuel used was of 80 octane rating. The test was conducted at the engine speed of 600 rev/min and the results are shown in Fig. 140.

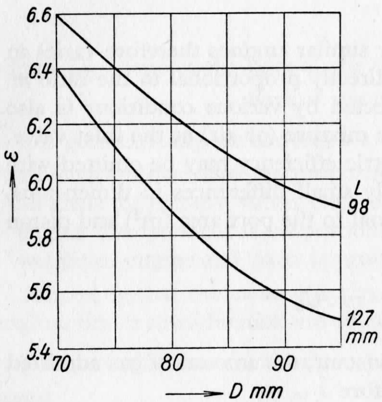

FIG. 140. Effect of bore upon maximum admissible compression ratio.

Another interesting fact is apparent from Fig. 140. Short - stroke engines can work with a higher compression ratio. This is in keeping with the fact that smaller cylinders will operate with higher compression ratios, since the mixture is better cooled during compression, and compression temperatures and pressures are, therefore, lower.

An engine with a stroke of 127 mm and bore 80 mm has a displacement of 638 cm^3. In order to have the same displacement, an engine with a 98 mm stroke must have a 91 mm bore. The curves in Fig. 140 show that with cylinders of equal displacement, a higher compression ratio may be used in the engine with the shorter stroke (in this particular case 5·96 instead of 5·9).

By lowering the compression ratio, both thermal efficiency and engine performance are reduced. The considerable influence of the compression ratio upon thermal efficiency is shown in Fig. 141.

This diagram refers to the thermal efficiency of an ideal cycle following the formula

$$\eta_t = 1 - \left(\frac{1}{\varepsilon}\right)^{\nu-1}$$

The thermal efficiency rises rapidly with increasing compression ratio up to 8 : 1, after which the increase becomes comparatively lower. The interests of high thermal efficiency therefore dictate the choice of the highest possible compression ratio.

To complete the picture, the influence of some major factors upon engine performance are tabulated in Table 25. This shows the influence of altering one single factor in order to raise engine performance on the displacement of a single cylinder V_1, total engine displacement V, performance per litre, bore

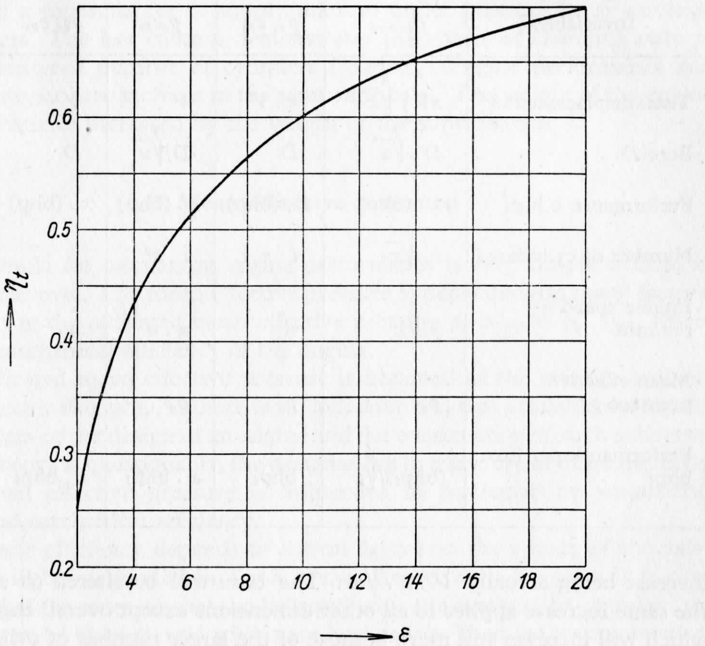

FIG. 141. Effect of compression ratio upon thermal efficiency.

D, number of cylinders i and n. The engines in all cases are geometrically similar.

The upper line of the table contains the fraction expressing in its numerator the factor to be increased x times and in its denominator the factor which remains unchanged.

The first column refers to an engine, the performance of which was increased by increasing cylinder dimensions, and thereby its displacement per cylinder, without altering the mean effective pressure p_e and mean piston velocity c_m. In order to adhere to the same piston velocity with the derived engine, speed must be reduced by $n/\sqrt[3]{x}$. As a result the performance per litre would be reduced and to counteract this a larger number of cylinders $i\,.\,\sqrt[3]{x}$ must be employed. The engine capacity will therefore rise more than x times, the

Table 25

Geometrical Similarity of Engines

Values increased x times	V_1	i	n	z	p_e
Invariables	p_e, c_m	p_e, c_m	$p_e c_m$	p_e, c_m	i, c_m
Total displacement V	$xV . \sqrt[3]{x}$	$x . V$	V	V	V
Bore D	$D . \sqrt[3]{x}$	D	$D/\sqrt[3]{x}$	D	D
Performance b.h.p.	x . (bhp)	x . (bhp)	x . (bhp)	x . (bhp)	x . (bhp)
Number of cylinders i	$i . \sqrt[3]{x}$	$i . x$	$i . x^3$		i
Engine speed n rev/min	$n/\sqrt[3]{x}$	n	$x . n$	n	n
Mean effective pressure p_e	p_e	p_e	p_e	p_e	$x . p_e$
Performance per litre bhp_l	$(\text{bhp}_l)/\sqrt[3]{x}$	bhp_l	$x . \text{bhp}_l$	$x . \text{bhp}_l$	$x . \text{bhp}_l$

increase being actually $V . x . \sqrt[3]{x}$. The bore will be altered to $D . \sqrt[3]{x}$ and the same increase applies to all other dimensions except overall engine length, which will increase still more because of the larger number of cylinders. The increase in engine weight is approx. $x . \sqrt[3]{g}$. An increase of performance is, in this instance, accompanied by a considerable increase in the size and weight of the engine.

The second column refers to an increased performance solely by employing a larger number of cylinders, while p_e and c_m remain unchanged. This is a simple case and only the number of cyliders increases x times. The size and weight of the engine are also increased approximately x times.

The third column refers to an increase of performance gained by an increase in speed. In order to keep the mean piston velocity down at its original value, cylinder dimensions have to be decreased $1/\sqrt[3]{x}$ times. Performance per litre rises x times, engine displacement remains unchanged and the number of cylinders will rise to $i . x^3$. Such a solution is advantageous considering engine dimensions and weight only, for the increase in the number of cylinders is impracticable. In automobile prime movers the number of cylinders is limited practically to eight or twelve.

The last two columns are of particular interest. In the fourth column the change concerns the number of expansion strokes per crankshaft revolution, i.e. the comparative values of a four-stroke and two-stroke engine are considered. The performance per litre of a two-stroke engine is double that of the four-stroke assuming that the p_e and c_m remain constant. Other values also remain unchanged. The engine weight however is increased by the addition of a supercharger to supply mixture under pressure or to scavenge the cylinders. The last column demonstrates the effects of changing only p_e with an unaltered number of cylinders i and c_m. Engine performance and performance per litre increase in the same ratio as p_e. The weight of the engine is in this instance increased by the weight of the supercharger.

2. Mean effective pressure

The formula for calculating engine performance is very simple with p_e as the only unknown. The mean effective pressure is dependent on many factors. Its relation to the indicated mean effective pressure p_i is $p_e = p_i . \eta_m$, where η_m is the mechanical efficiency of the engine.

The indicated mean effective pressure is obtained as the average ordinate of the indicator diagram. As there is no indicator diagram available during the various stages of the design of an engine and the construction of such a diagram based on theory is questionable, the designer has to make use of other methods.

The mean effective pressure is influenced in particular by volumetric, thermal and mechanical efficiency.

Volumetric efficiency depends to a great degree on the system of the valve gear and will be mentioned in detail in the chapter on valve gear. It is clear however, that the more air that can be supplied to the combustion chamber, the more fuel can be burned; and with more heat energy liberated by combustion, engine output will be higher for a given thermal efficiency.

Thermal efficiency expresses the percentage of the heat contained in the fuel converted into useful work done by the engine. On any engine, the thermal efficiency may be calculated from

$$\eta_t = \frac{632}{V_p . b}$$

where V_p denotes the calorific value (kcal/kg)

b „ „ specific fuel consumption (kg/b.h.p.-h.)

The theoretical thermal efficiency depends chiefly on the compression ratio and for a petrol engine it is

$$\eta_t = 1 - \left(\frac{1}{\varepsilon}\right)^{\nu-1}$$

This relationship is plotted in Fig. 141, but it must be borne in mind that thermal efficiency is also influenced by engine design. Two engines of equal compression ratio but unequal distances of flame travel (the one having a compact combustion chamber, the other an elongated one) will have different thermal efficiencies.

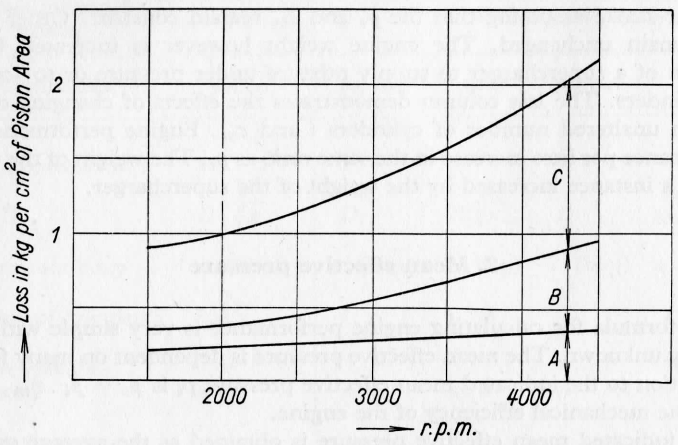

FIG. 142. Distribution of mechanical losses plotted against engine speed.
A — Bearing friction and auxiliaries, B — Induction losses, C — Losses by piston friction.

Mechanical efficiency is a measure of the losses during the transference of work from the power stroke to the flywheel. With lower frictional losses in the engine, the mechanical efficiency rises and so does the engine performance. The principal mechanical losses in the engine are caused by piston friction. Fig. 142 shows the various losses in relation to the engine speed. It will be seen that frictional losses caused by pistons and by charging show a sharp increase with rising speed thus reducing mechanical efficiency. The accelerating forces acting on the moving mass of the engine increase with higher engine speed, thereby increasing the frictional losses caused by pistons as the square of the speed value.

The use of roller bearings makes for higher mechanical efficiency as compared with plain bearings. Engine accessories, such as the dynamo, supercharger, fan or ventilator, water

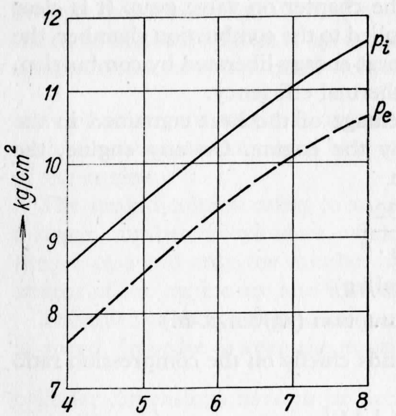

FIG. 143. Influence of the compression ratio upon mean indicated pressure p_i and mean effective pressure p_e.

pump, etc., all have an adverse effect on mechanical efficiency and lower mean effective pressure.

Fig. 143 illustrates the variation of p_i and p_e with the compression ratio. By increasing the compression ratio of an automobile engine from 6 : 1 to 11½ : 1 the results shown on Fig. 144 were obtained. With high compression ratios, iso-octane with 0·8 cm³ of tetra-ethyl-lead added per litre was used for fuel. The improvement on b.h.p. at 3000 rev/min with high compression ratios is relatively small.

During the initial stages of designing an engine the m.e.p. must be approximated, taking into account

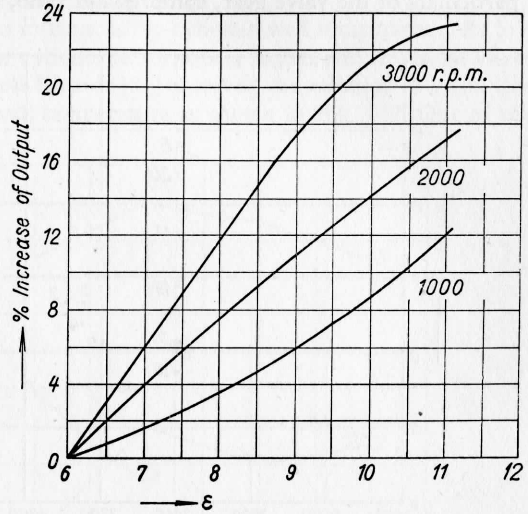

FIG. 144. Effect of compression ratio upon engine performance at various speeds. Note the small increase in output at 3,000 rev/min and $\varepsilon > 10$.

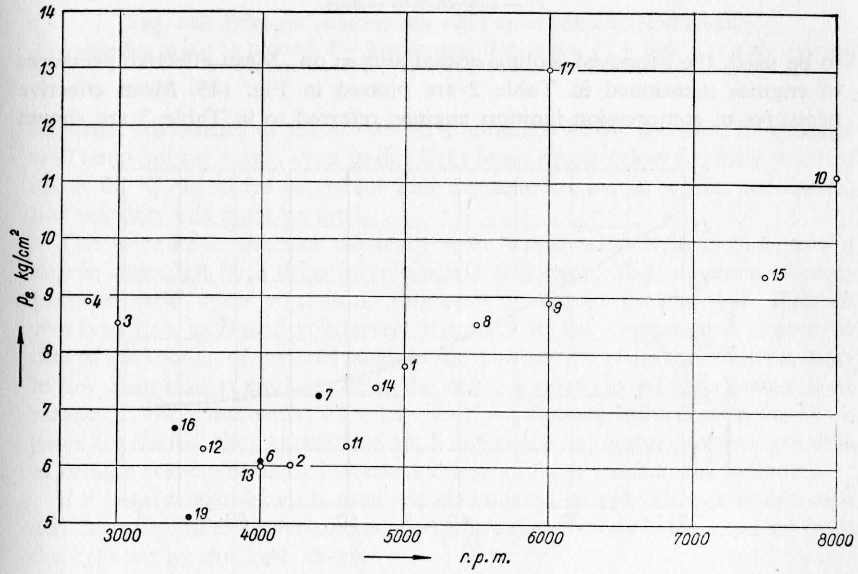

FIG. 145. Mean effective pressures in air-cooled petrol engines.
○ — hemispherical combustion chamber, ● — wedge-shaped combustion chamber.

experience with other engines and allowing for differences caused by the particulars of the valve gear, compression ratio, the octane rating of the fuel

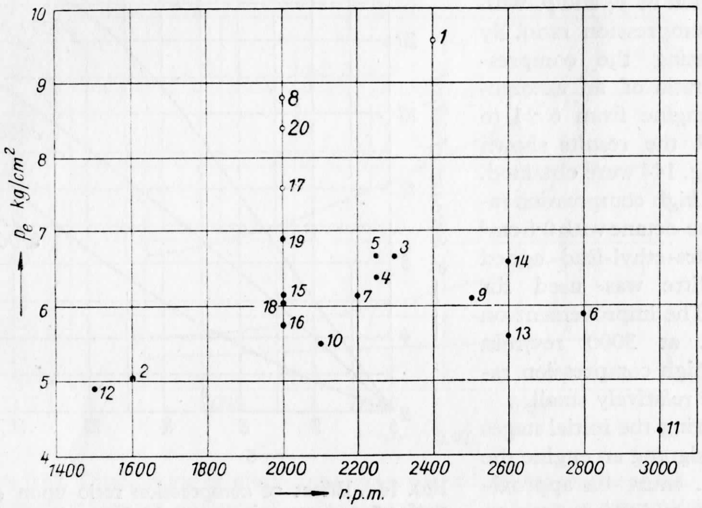

FIG. 146. Mean effective pressures in air-cooled oil engines.
○ — supercharged engines.

to be used, the proposed cooling system and so on. Mean effective pressures of engines mentioned in Table 2 are plotted in Fig. 145. Mean effective pressures of compression-ignition engines referred to in Table 3 are shown

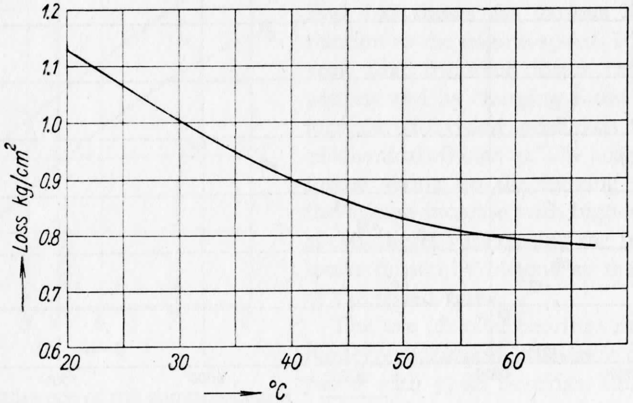

FIG. 147. Effect of cylinder temperature upon friction losses in a four-cylinder lorry engine at 900 rev/min.

in Fig. 146. These values are suitable for reference when estimating the mean effective pressure of a proposed engine design.

Mechanical losses are also influenced by cylinder wall temperature. At low temperatures the viscosity of cylinder lubricants is higher and frictional losses are correspondingly increased. Ricardo [51] observed the variation of mean loss on pressure with cylinder-wall temperature as shown in Fig. 147. One of the

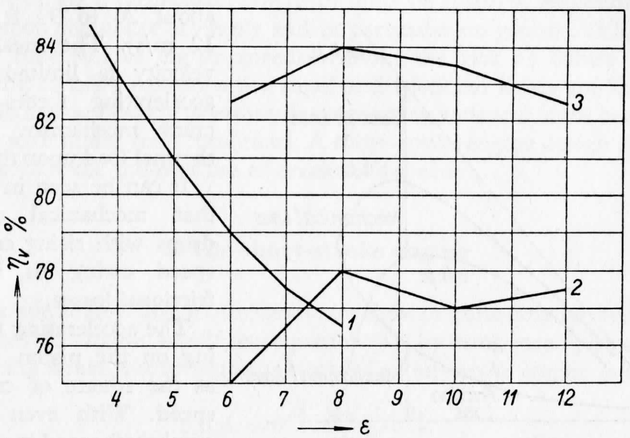

FIG. 148. Effect of compression ratio upon volumetric efficiency.
1 — according to test by Ricardo, *2* — experimental GM engine for a high ε at 3,200 rev/min, *3* — experimental GM engine for high ε at 2,000 rev/min.

apparent advantages of the air-cooled engine lies in the fact that its cylinder wall temperature never, even under light load, drops below a critical point of about 80 °C. In order to reduce loss through friction in winter conditions, low viscosity oils must be used.

The increase in thermal efficiency with compression ratio is to a certain degree cancelled by a drop in volumetric efficiency. The influence of compression ratio upon volumetric efficiency is shown in Fig. 148. Ricardo observed that increased volumetric efficiency in low compression engines is due to the cooling of residual gases in the combustion chamber, which is large in low compression engines. After the exhaust valve closes, hot exhaust gases remain in the combustion chamber. In the following induction stroke these gases are cooled, they contract and the difference in volume makes it possible to bring a correspondingly increased charge of fresh gas into the cylinder.

If a large valve overlap is used the above is no longer valid, as under such conditions cylinder scavenging is good, the exhaust gases being expelled from the cylinder by the fresh charge.

3. Engine speed

In order to keep the engine small in size and light in weight, the best possible use has to be made of crankshaft speed up to the highest permissible mean piston velocity c_m. The mean piston velocity of the modern automobile petrol engine is about 32 to 49 ft/s (10 to 15 m/s) and in the case of the compression-ignition engine about 26 to 39 ft/s (8 to 12 m/s). The mean piston velocity is limited by the accelerating forces of the crank mechanism and the thermal load upon the piston.

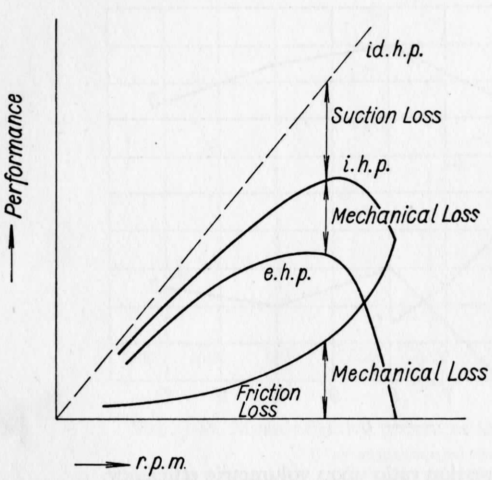

FIG. 149. Dependence of ideal, indicated and effective performance, and losses on engine speed.

It can be seen in Fig. 142 that mechanical efficiency drops with rising crankshaft speed owing to increased frictional losses.

The accelerating force acting on the piston increases as the square of crankshaft speed. With even a small crankshaft speed increase, the drop in efficiency is therefore rapid.

The influence of crankshaft speed on b.h.p. is shown in Fig. 149. Assuming that ideal mechanical and volumetric efficiencies of 100% could be obtained, engine performance would rise in direct proportion to crankshaft speed as indicated by the dashed straight line of ideal engine performance. Under real conditions however, losses in the induction system must occur and therefore, at higher crankshaft speeds, a charge lower than the displacement enters the cylinder, the volumetric efficiency not being 100%, but rapidly decreasing with growing crankshaft speed. That is why inside the cylinder only an i.h.p. reduced by suction losses is produced, as shown by the full curve. This output is then further reduced by mechanical losses to the resulting b.h.p. curve. The aggregate performance loss rises rapidly with rising crankshaft speed and at a certain speed equals i.h.p. This marks the maximum possible crankshaft speed of an engine, reached without load and on a fully open throttle. Under such conditions all i.h.p. is used up to overcome frictional losses and none is left to do useful work. The mechanical efficiency in such a case, expressed as a percentage, equals 0%.

Crankshaft speed may be increased only to about 60% of mechanical efficiency as below this percentage running economy is very low and both

specific consumption and engine wear are high. A large proportion of the i.h.p. is used up by the mutual abrasion of various engine parts. Bearing only this point in mind, the lowest possible crankshaft speed commensurate with high mechanical efficiency and low specific consumption would be of great advantage. This would however produce large and heavy low speed engines.

The modern automobile prime mover must be small and light and hence the apparent effort to increase crankshaft speed. For this trend of development to be successful, good mechanical efficiency must be ensured. Attention must be focused upon the crank assembly and in particular on pistons. A light crank assembly may run at higher speeds without the risk of failure since the accelerating forces in question are small and frictional losses are kept within reasonable limits. Bearing pressures in connecting rods and main bearings can be coped with under such conditions. A short-stroke engine design also makes it possible to make the best use of crankshaft speed.

4. The short-stroke engine

Automobile engine design of recent years shows a quite definite trend toward the short-stroke engine. Modifying equation (41) by using mean piston velocity c_m in place of stroke, the formula for b.h.p. of a four-stroke engine is obtained:

$$\text{b.h.p.} = \frac{A \cdot c_m \cdot i \cdot p_e}{300}$$

Stroke is omitted in this formula and it is thus evident that output depends primarily on piston area (the bore D) and mean piston velocity c_m. Long stroke and lower crankshaft speed or short stroke with high crankshaft speed may be used with equal results.

The use of high crankshaft speed with a short stroke, however, makes it possible to design a smaller and lighter engine, which with its smaller displacement, will give a performance equal to that of an engine with larger displacement.

In order to exploit this advantage stroke is being reduced so radically today that in many cases it is smaller than bore and such engines are said to be "over-square". Mean piston velocity

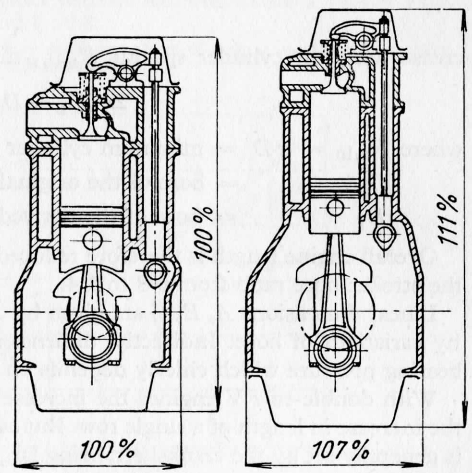

FIG. 150. Comparison of dimensions of engines with stroke: bore ratios of 0.8 and 1.4. The 0.8 ratio is taken as 100 %.

generally does not rise, indeed in some cases it is even reduced. As a result of a reduced stroke, connecting rod weight is reduced and often pistons are lighter, thus increasing mechanical efficiency.

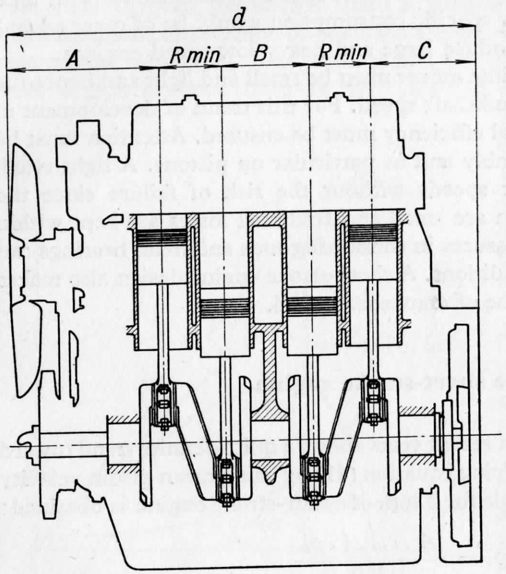

FIG. 151. Longitudinal section of a four-cylinder engine with a three-bearing crankshaft.

Volumetric efficiency of short-stroke engines is good, as with a given c_m it varies with the piston area: port area ratio, which is favourable in this case, since a large diameter cylinder head can house large valves.

Fig. 150 shows the cross sections of two 4-cylinder in-line engines, each having a displacement of 1·4 litres, one with a stroke: bore ratio of $\xi = 1{\cdot}4$, the other 0·8. Taking the latter as a basis for comparison, the overall height of the former is 111% and the width 107%.

Fig. 151 shows a simplified drawing of a four-cylinder engine with a three-bearing crankshaft. It is apparent that an increase in bore D increases only the cylinder spacing $R_{\min}$, the overall increase in length being

$$2x\,(D_1 - D_0),$$

where $R_{\min} = x\,D$ = minimum cylinder spacing

D_0 = bore of the original engine

D_1 = bore of the derived engine.

Overall engine length is therefore reduced from 100% to 94% by increasing the stroke: bore ratio from 0·8 to 1·4.

Linear dimensions A, B, C are given by design and are not directly affected by variations of bore. Indirectly, dimensions A, B, C may be influenced by bearing pressure which chiefly depends on mean piston velocity.

With double-row V engines the increase of the engine length is limited to the increase in length of a single row. But as the cylinder spacing of a V engine is generally set by the crankshaft, being larger than $R_{\min}$, the variation of bore usually does not affect overall length.

Fig. 152 illustrates the influence of the stroke: bore ratio on crankshaft design. In order to reduce engine weight, the removal of metal (bore A) in the

crankpins is worthwhile. Crankshaft weight is reduced by the amount bored out and the counterweight needed to balance the crankpin is correspondingly reduced. Assuming that crankpin diameter varies with bore and not with stroke, then in engines having a shorter stroke than bore (over-square

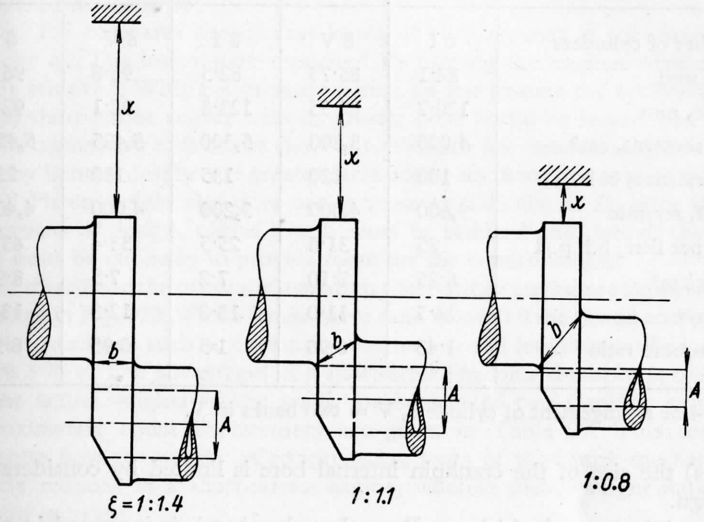

FIG. 152. Shape of crankshaft for engines with stroke: bore ratios 1 : 1.4, 1 : 1.1, and 1 : 0.8.

A — diameter of the crankpin relief bore, b — minimum width of web, x — distance of piston bottom from crankshaft.

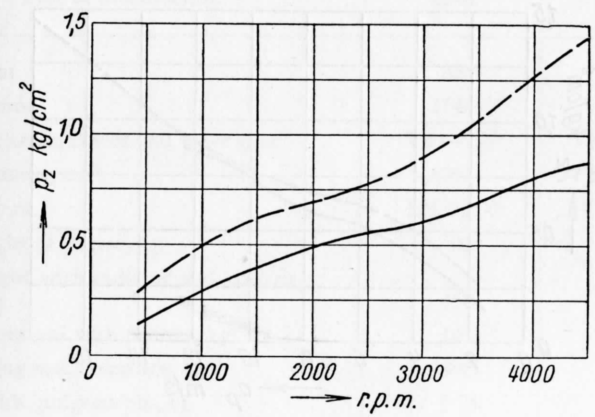

FIG. 153. Mean pressure loss in kg per sq. cm of piston area plotted against engine speed. Dash line — Studebaker engine 1949, full line — 1951 engine.

Table 26

Year of production	Studebaker		Chrysler		
	1949	1951	1950	1951	1954
Number of cylinders	6 I	8 V	8 I	8 V	8 V
Bore, mm	84·1	85·73	82·5	96·8	96·8
Stroke, mm	120·7	82·55	123·8	92·1	92·1
Displacement, cm³	4,020	3,800	5,300	5,425	5,425
Performance, b. h. p.	100	120	135	180	238
Speed, rev/min	3,600	4,000	3,200	4,000	4,400
Perf. per litre, b.h.p./1	25	31·6	25·5	33·4	43·8
p_e, kg/cm²	6·25	7·10	7·2	7·5	8·95
c_m, m/s	14·5	11·0	13·2	12·3	13·5
Stroke/bore ratio	1·43	0·96	1·5	0·95	0·95

I = in-line arrangement of cylinders, V = two banks in V.

engines) the size of the crankpin internal bore is limited by considerations of design.

The minimum web thickness B on the other hand, is increased in short-stroke engines by crankpin overlap. The crankshaft of a short-stroke engine is therefore extremely rigid and thus highly suited for modern high-compression engines.

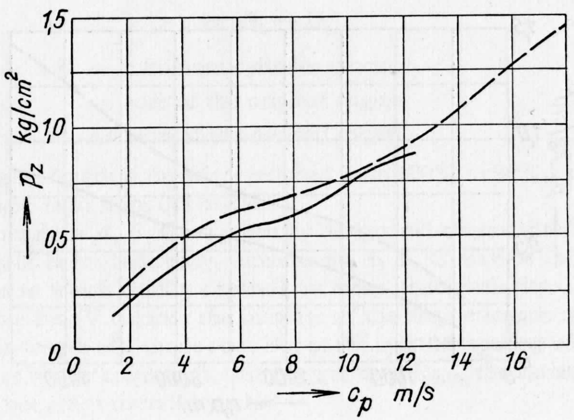

Fig. 154. Mean pressure loss in kg per sq. cm plotted against mean piston velocity. Dash line — Studebaker engine 1949, full line — 1951 engine.

Fig. 153 illustrates the readings obtained on two Studebaker engines [70]. The older, a six-cylinder of 4 litres displacement was replaced by a 3·8 litre eight-cylinder engine. The new short-stroke engine with smaller cylinders has a lower mean piston velocity despite higher crankshaft speed and losses through friction are therefore considerably reduced. Both these engines are compared in Table 26.

Fig. 154 compares the friction losses of both engines at the same piston velocity c_p. The losses were measured by turning the engines with cylinder heads removed. With gas pressure acting on the pistons the frictional losses of the short-stroke engine with its greater bore would be larger than shown.

The connecting rods of the short-stroke engine are considerably shorter and thereby lighter, despite the greater force acting on them (because of the larger bore). Piston weight also does not increase as the cube of D, since for considerations of design, piston length must be reduced and part of the piston skirt must be cut away to provide room for the counterweight.

For this reason the pistons of a modern short-stroke engine are slipper shaped, as shown in Fig. 322. With a stroke: bore ratio below 0·9 the piston and counterweight overlap to such an extent that connecting rod length must be increased above 3·75 r. The advantages of a short-stroke become doubtful at this stage.

The actual weights of the crank mechanism of two Cadillac engines of approximately equal displacement are given in Table 27. This serves to illustrate how the weight of reciprocating parts of the crank mechanism is greatly reduced in a short-stroke engine, whereas piston weight shows only a slight increase.

Table 27

Weights of Sub-Assemblies of two Cadillac V8 Engines

Year of manufacture	1948	1949
Bore, mm	88·9	96.8
Stroke, mm	114·3	92.1
Cylinder arrangement and valve gear	V8 — SV	V8 — OHV
Displacement, cm³	5,675	5,425
b.h.p./r.p.m.	124/3,200	133/3,500
Dry weight of engine, kg	402	318
Wet weight with radiator and cooling water, kg	450	350
Connecting rod with pistons, kg	14·2	12·2
Connecting rod, complete, kg	8·45	6·05
Piston with gudgeon pin, kg	5·75	6·15
Crankshaft, kg	41·9	27·9

The example given in Fig. 155 is still more instructive. It concerns the mechanical efficiency of two eight cylinder Chrysler engines [15]. The older engine was fitted with side valves, the other with overhead valves. Notwithstanding the fact that both these engines have eight cylinders and that the

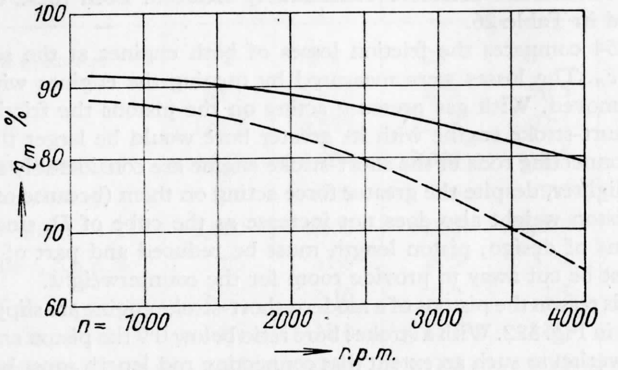

FIG. 155. Mechanical efficiency plotted against speed. Dash line — Chrysler engine made in 1950, full line — made in 1951. Dimensions of both engines are given in Table 26.

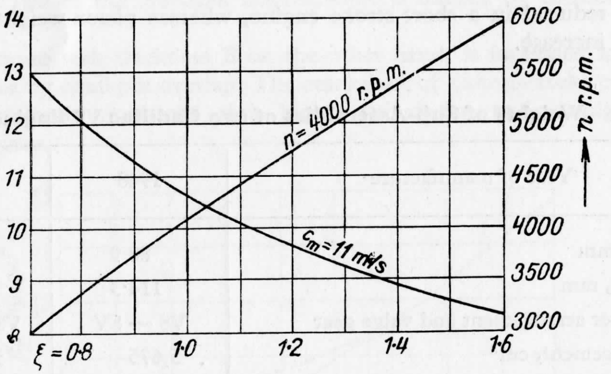

FIG. 156. Dependence of mean piston velocity c_m on the stroke: bore ratio at a constant speed of 4,000 rev/min and dependence of engine speed on the stroke: bore ratio at a constant mean piston velocity of $c_m = 11$ m/s.

second has a somewhat increased displacement, the mechanical efficiency of the new engine is above that of the old despite the higher crankshaft speed employed. The deciding factors in this case are the increased thermal and volumetric efficiencies and the choice of a short-stroke engine layout, keeping the mean piston velocity below the original value despite the higher speed.

Under the heading of mechanical efficiency, pumping losses are included; they are lower in the case of the overhead valve engine with hemispherical combustion chambers than in the case of the side-valve engine.

The short-stroke engine attains a higher crankshaft speed with reasonable mechanical efficiency. High crankshaft speed makes for a high performance per litre and for a good weight: power ratio.

Fig. 156 shows the variation of speed n with the stroke: bore ratio ξ with $c_m = 11$ m/s constant and conversely the variation of c_m with ξ and constant $n = 4000$ rev/min and the same displacement (in this case 350 cm³). It follows from the diagram that, for instance, with the change of the stroke: bore ratio of a 1·4 litre four-cylinder engine from 1·4 to 0·9 crankshaft speed may be increased from 3450 to 4650 rev/min without affecting the mean piston velocity. But by raising crankshaft speed only to 4200 rev/min c_m is reduced to 10 m/s and an improvement of mechanical efficiency can therefore be expected.

The influence of the stroke: bore ratio on cylinder surface area and heat loss is of some interest. For the sake of simplicity the combustion chamber will be considered to be cylindrical in shape and of the same diameter as the cylinder bore. Calculation will be based on a cylinder having a displacement of 350 cm³ and the variations in surface area of the combustion chamber itself and of the whole inside surface area will be observed. The results are given in Table 28. The basis for comparison (100%) is provided by a cylinder with a stroke: bore ratio of $\xi = 0{\cdot}8$. It is apparent that with an increase in stroke the surface area of the combustion chamber is reduced, whereas cylinder surface area is increased.

Table 28

Effect of Stroke/Bore Ratio ξ Upon Change of Dimensions and Internal Surface of a 350 cm³ Cylinder

ξ	*D* mm	*L* mm	Surface area of comb. chamber cm²	%	Surface area of cylinder cm²	%	Total surface area cm²	%
0.7	86·025	60·218	90·736	104	162·743	95·1	253·479	98·5
0.8	82·280	65·824	87·290	100	170·150	100	257·440	100·0
0.9	79·112	71·201	84·453	97·0	176·962	104	261·415	101·5
1.0	76·382	76·382	82·300	94·3	183·289	107·8	265·589	103·2
1.1	73·994	81·393	80·669	93·0	189·205	111·0	269·874	104·8
1.2	71·878	86·255	79·530	91·1	194·773	114·7	274·303	106·9
1.3	69·986	90·982	78·476	89·9	200·040	117·8	278·525	108·2
1.4	68·278	95·590	77·581	89·0	205·049	120·5	282·624	110·1
1.5	66·726	100·09	76·934	88·1	209·813	123·0	286·747	111·6
1.6	65·306	104·49	76·371	87·5	214·376	126·0	290·747	113·0

Heat losses to combustion chamber walls will therefore vary inversely as the stroke: bore ratio.

Heat loss to cylinder walls, on the other hand, becomes greater with an increased stroke: bore ratio, as this also increases the cylinder wall area. The cylinder wall however is screened by the piston skirt during piston travel and the uncovered part is contacted by exhaust gas which has already lost some heat. Experiments with a supercharged water-cooled aircraft engine have shown the coefficient of heat transfer to combustion chamber walls to be 400 000 kcal/m^2 h and to the cylinder wall surface only 125 000 kcal/m^2 h.

With an automobile engine (as opposed to an aircraft engine), working at a lower load, the coefficient of heat transfer to the cylinder head may be taken to be 300 000 kcal/m^2 h and that of heat transfer to the cylinder wall 100 000 kcal/m^2 h [22] [42]. Total heat losses are compared in Table 29. It will be noted that they are approximately equal in both instances, but since the short-stroke engine gives a higher performance per litre owing to higher speed, its b.h.p. is higher. The heat carried off by the cooling medium expressed as a percentage of total heat loss is therefore smaller for a fast running short-stroke engine, assuming the cooling area to be equal. This would seem to endow the short-stroke engine with the double advantage of a smaller radiator and a smaller portion of engine power used for operating the cooling system. In practice, conditions may be even more favourable than assumed as shown by actual engine measurement in Table 30.

Table 29

Stroke: Bore ratio ξ =	0.8	1.4
Surface area of combustion chamber, cm^2	87·29	77·58
Surface area of cylinder, cm^2	170·15	205·04
Heat from comb. chamber, kcal/h	2620	2320
Heat removed by cylinder, kcal/h	1701	2050
Total heat removed, kcal/h	4321	4370

Table 30

Chrysler engine made in	1951	1950
Number of cylinders	8V	8 in-line
Displacement, cm^3	5425	5300
Heat removed by cooling, kcal/min	1385	1510

5. The number and arrangement of cylinders

Multi-cylinder engines are desirable because of small engine weight; their torque being more even, they may be designed with smaller and lighter flywheels. An excessively large number of cylinders, however, makes the engine too costly and complicates operation. The driver of a vehicle fitted with an engine of more than eight cylinders needs considerable experience to be able to detect and locate a cylinder which is not firing. Engines of more than eight cylinders are therefore seldom used for passenger cars.

When forced by circumstances to produce a high-output engine having a low weight, as is the case with aircraft or armoured vehicle engines, designers resort to using engines with a large number of cylinders. 16, 18 and even 24 cylinder engines have proved reliable under operating conditions and engines having as many as 28 cylinders are used in aircraft, while others with a still greater number are undergoing tests. A sixteen-cylinder engine is not unusual for tanks and twelve-cylinder prime movers are used in railway practice.

Long development has shown a 160 mm bore to be the maximum for aircraft engines, if cooling is to remain adequate. The overall engine output therefore depends on the number of cylinders which can be grouped into a serviceable unit.

With the reduction of engine weight in view, the arrangement with a common crankpin for the largest possible number of cylinders is most desirable. This is best met by radial engines with their short and rigid build and small size of crankcase per cylinder. With air-cooled engines this arrangement has the further advantage of individual well separated cylinders making possible the use of extensive finning.

The radial arrangement offers fewer advantages for automobile use. Such an engine is rather high, cannot easily be accommodated under the bonnet and accessibility of some cylinders is impaired. The inlet manifold of radial engines presents difficulties, which are absent in supercharged aircraft engines. But owing to the small weight of the radial engine and the ease of cooling, the possibility of developing a good automobile engine of this type should not be ruled out. In aircraft engine practice several radial rows (up to four) of cylinders are arranged in line making a compact and powerful engine.

The cylinders or banks of multi-cylinder engines (e.g. 16-cylinder tank engines) are often arranged in X form or with one set of horizontally opposed banks superimposed on another. These types are low and easily accessible. The latter have two crankshafts. Twin V (or W) type engines are rare, although they are low and accessible.

For use in vehicles the largest V engines are eight- or twelve-cylinder types. This layout is convenient as all engine parts are accessible and the engine is light and short. The crankshaft is simple and well balanced. Eight cylinder V engines are progressively gaining more and more adherents.

The V arrangement is very suitable for air-cooled engines as, owing to considerations of crankshaft housing, the cylinder spacing is large enough for the required size of cooling fins to be used. There is therefore no difference in the lengths of water- or air-cooled V engines if plain bearings are used and if pairs of connecting rods share common crank pins. The induction manifold presents no problems and cooling air may be directed where it is needed without difficulty.

Opposed cylinder engines retain the first mentioned advantage but the induction manifold, if only one carburettor is to be used, is long and it is more difficult to direct cooling air flow. Despite this difficulty horizontally opposed engines are better suited for vehicles than in-line types and are very popular in conjunction with air-cooling.

In-line petrol engines are the most numerous because they are the simplest from the production point of view. Accessibility is good and the inlet manifold simple. The in-line arrangement is used in four- or six-cylinder engines, eight-cylinder in-line engines being too long and, with present day high running speeds, presenting great difficulties caused by torsional vibration of the crankshaft. There are no recent types of in-line eight-cylinder engines in production.

An air-cooled in-line four-cylinder engine is too long, the same applying, but to a greater degree, in the case of a six-cylinder engine.

Motor cycle and stationary engines have a smaller number of cylinders. Twin cylinder engines are arranged as in-line parallel twins, V engines or horizontally opposed (flat twins). Air cooling for twin cylinder engines offers every advantage, whereas water cooled twins and flat-twins in particular, are too complicated.

The opposed twin is unusually well balanced, for which reason it is also used to propel small passenger cars. The size of cooling fins is not limited by any considerations, but the induction manifold is too long and two carburettors may therefore be used with advantage. A short stroke is desirable.

The parallel twin has been popular with motor cycle designers in recent years. Its balance is poor, but firing intervals are regular. The V engine strikes a compromise between these two types. The smallest type of unit uses only a single cylinder, such an engine being simple and cheap to produce. Most engines under 250 cm^3 displacement are singles.

As the positioning of the cylinders of an air-cooled engine is influenced in no way by water jackets, unions or radiators, engine design may be adapted to suit the vehicle layout. Some interesting examples of this are shown in Fig. 157.

6. Balancing of engines

Engine balance of the most common multi-cylinder four-stroke air-cooled engines is dealt with in this section.

Twin cylinder engines are usually designed as opposed twins. (Tatra 12 Fig. 158, VW Fig. 212, Citroen 2 CV Fig. 372, Panhard Dyna Fig. 372, etc.)

The revolving masses are balanced by crankshaft counterweights. Primary and secondary couples remain unbalanced, but owing to the minute offset of opposing cylinder axes (comprising the centre web width and the width of one connecting rod), they are small.

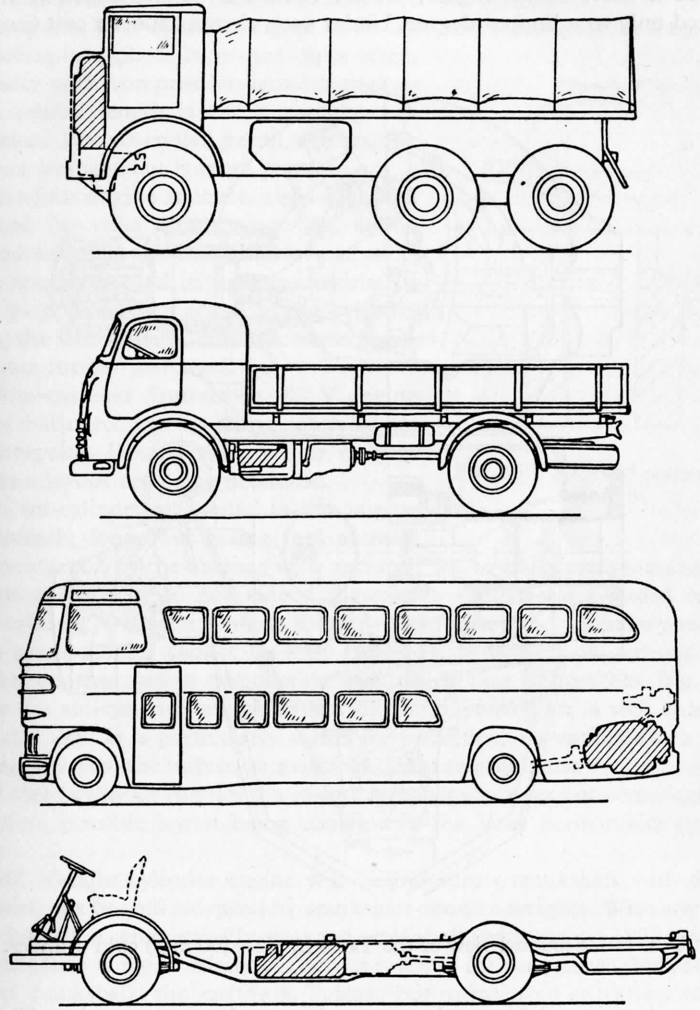

FIG. 157. Characteristic vehicle layouts.

The GM-T 51 truck for off-the-road operation with a vertically mounted Continental AO-536-1 engine. Second – The Tatra 116 lorry with a flat six-cylinder engine mounted underneath the floor. Third — Greyhound bus with two six-cylinder engines installed in the rear. Bottom — The Chassis of a Movag bus with a flat SLM eight-cylinder mounted underneath the floor.

In designing the crankshaft it has to be borne in mind that any saving in crankpin weight will also reduce the weight of the counterweights by the same amount, and, therefore, will be doubly valuable in terms of overall weight reduction.

With a short-stroke engine, weight reduction of the crankpin may be effected only to a limited degree. Under such circumstances a cast crankshaft

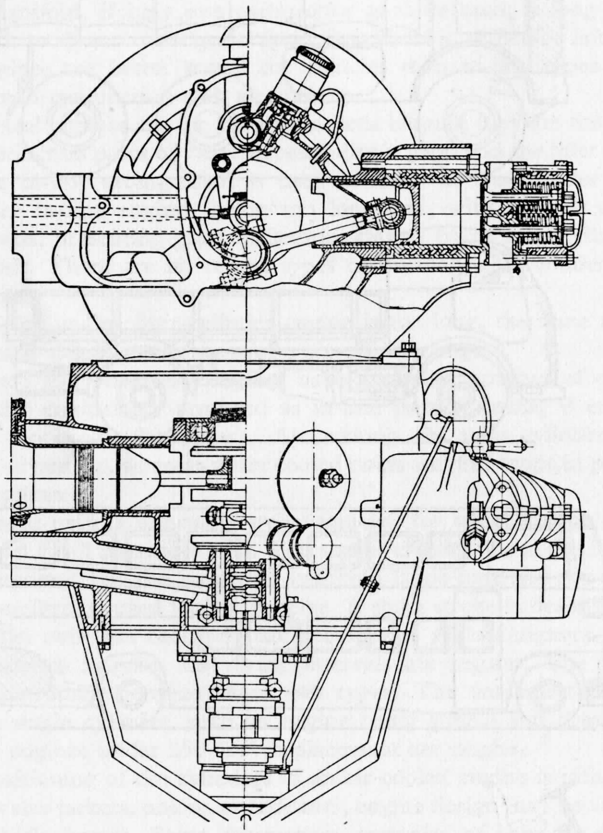

FIG. 158. Sectional view of the Tatra 12 (82 × 100 mm) twin cylinder.

is convenient as weight may be removed in the best manner (see Fig. 159). The parallel twin cylinder engine is as imperfectly balanced as a single cylinder engine since both pistons travel up and down simultaneously. The total revolving masses are balanced together with about 50 to 70 % of the reciprocating masses (depending on the cylinder position and engine mount-

ing). An example of a parallel twin Sunbeam motor-cycle engine is given in Fig. 382. With larger high speed engines there is a risk of transverse vibrations of the crankshaft and a third bearing is therefore desirable. The parallel twin arrangement permits the design of a compact engine with no difficulties with the inlet manifold and lubrication system.

The in-line four-cylinder engine in its accepted form is rather long when air cooling is applied. It is used quite often, especially with compression-ignition engines where overall length is not a primary consideration. For air-cooled petrol engines, the flat-four arrangement is more popular.

A flat-four engine is short, rigid and well balanced. Its chief disadvantage lies in the long induction manifold, if only one carburettor is employed, and in its great width. In order to provide easy access to the cylinder heads, the demands on engine compartment width are further increased.

A four-cylinder four-stroke 90° V engine can be balanced, but its firing intervals are very irregular. Despite this, V-four engines of such a layout are being produced.

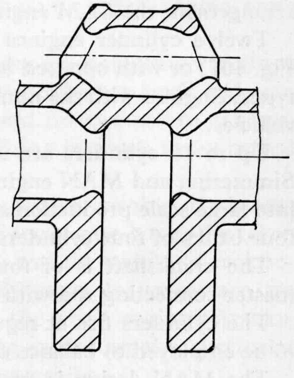

FIG. 159. Relieved casting of a crankpin.

The six-cylinder air-cooled in-line engine is extremely long, but taking into account the space saved by the absence of a radiator, its overall space requirements in a motor vehicle do not exceed those of a similar water-cooled engine. A six-cylinder V engine is much shorter, but primary and secondary reciprocating mass couples cannot be fully balanced. It is used on account of its compact construction in conjunction with air cooling. (Tatra 926, Fig. 414.)

The flat six-cylinder engine with a six-throw crankshaft is well balanced, short and low. It is particularly suited for underfloor mounting. On a petrol engine, however, the induction manifold presents problems.

For eight-cylinder engines the in-line arrangement does not come into consideration, possible layout being confined to the V or horizontally opposed types.

A 90° V-eight cylinder engine with a four-throw crankshaft with throws 90° apart, can be well balanced by crankshaft counter-weights. With any other angle than 90° a symmetrically arranged crankshaft (with throws 180° apart) in a single plane is more suitable. There is no need for counterweights with this type of crankshaft, the engine is lighter, but unbalanced secondary couples remain, although they are relatively small and cause no disturbance (Tatra 928, Fig. 413, etc.).

The case of the opposed eight cylinder engine proves interesting. With the choice of a symmetric four-throw single plane crankshaft the engine is well balanced and cylinder spacings are small, all this at the expense of two cylinders always firing concurrently. In order to arrange the firing intervals regularly and 90° apart, an eight-throw crankshaft must be used with a resulting increase of cylinder spacings (the minimum possible increase being that of crank web thickness).

In another variation of the opposed eight-cylinder engine, the crankshaft has four throws at 90° intervals. In this case firing intervals are regular but reciprocating primary couples remain unbalanced. As an example of this arrangement the SLM engine may be quoted.

Twelve cylinder engines may be designed either as V types (Tatra 111, Fig. 407) or with opposed banks of cylinders. Balancing is satisfactory in both types. Engines with the number of cylinders exceeding 12 are not used in road vehicles.

Up to 16 cylinders are used in prime movers for armoured vehicles, the Simmering and MAN engines being two examples. The first was about to go into large scale production at the end of World War II. It is an X engine with four banks of four cylinders each disposed at 45° and 135°.

The crankshaft is of four-throw design with throws 90° apart and one master connecting rod with three connecting rods acting on each crankpin.

The cylinders fire at regular intervals but crankshaft counterweights have to be employed to balance couples. Primary couples cannot be fully balanced.

The MAN design is an H type with two crankshafts. From the point of view of balancing this engine may be regarded as two opposed eight-cylinder engines with a common crankcase.

Both these sixteen-cylinder engines combine short, low and light build with relatively good accessibility. Beside the arrangements described, a great number of others exist, such as in-line engines with an odd number of cylinders, radial engines etc. In order to be of the greatest possible use, an engine must be fully adapted for the type of vehicle in which it is to be installed. Some examples of air-cooled engines designed for a particular vehicle type are shown in Fig. 157.

Common types of automobile engines have been treated so far. Aero-engines, from which considerably larger performances are required, are built with a greater number of cylinders.

Air-cooled aircraft power units are in the majority of cases arranged as radial engines. It is feasible to balance this type fully even when the number of cylinders is small. As all cylinders are in one plane, primary and secondary couples are absent and primary and secondary forces can be balanced by crankshaft counterweights. Balancing is calculated on the assumption that all connecting rods revolve round the crankpin centre. With one master rod and other connecting rods acting upon it the conditions of balancing are somewhat different.

By grouping balanced radial engines in tandem, an engine with a multiple number of cylinders is obtained, the balancing of which is also perfect. 14, 18, 28 and even 36-cylinder engines (4 × 9 cylinders) may thus be arranged, whereas a grouping of in-line engines produce 12, 24 and even 48 cylinder

units. These high-output engines are, however, today being replaced by the gas turbine.

In order to complete this section, mention must be made of the balancing of in-line engines with an odd number of cylinders. Although such an arrangement is not common in automobile practice, several five-cylinder engines exist (Lancia, Gardner etc.). Resort to such an arrangement is usually only made when necessity dictates the production of an engine to bridge the gap between a four-and a six-cylinder type and standardized engine parts must be used.

It was found, that primary and secondary couples are unbalanced in these engines. With a larger number of cylinders, these couples are small and may be disregarded. A similar arrangement can be used for two-stroke engines, balancing problems remaining the same, but the firing intervals being halved and the order of firing changed.

So far it has always been assumed that the respective weights of pistons, connecting rods etc. are equal for all cylinders. Special care has to be taken during manufacture to ensure that this is really so and the weight of reciprocating parts must be kept within close limits. If, for example, we replace one piston in a six-cylinder engine by a heavier piston, the whole engine balancing is upset. When replacing only some pistons during engine repairs, particular care has to be taken to fit only pistons of matching weight.

For engine balancing to be perfect, all moving masses must be taken into account. The flywheel, which is the heaviest moving part, must be balanced and attention must be paid to all gear wheel teeth, etc.. In very highly stressed engines, even camshafts must be balanced, as the cams are not symmetrically disposed and produce unbalanced couples.

7. The influence of compression ratio on engine economy

Economy of operation is one of the foremost requirements with modern engines. With a petrol engine, running economy is chiefly dependent on the fuel: air ratio and the compression ratio.

The influence of the fuel: air ratio upon m.e.p. and upon thermal efficiency is shown by Fig. 160 which makes it clear that the use of a higher percentage of air than in the ideal fuel-air mixture leads to an improving thermal efficiency

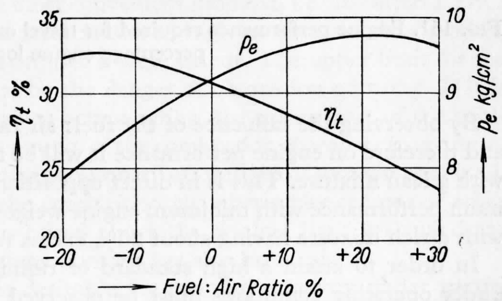

FIG. 160. Influence of fuel: air ratio upon thermal efficiency and mean effective pressure.

of the engine. But as a leaner mixture burns more slowly, spark advance has to be increased in order to provide optimum conditions. With correct spark advance, the highest thermal efficiency of an engine is attained with about 15% of air surplus. With thermal efficiency at its peak, specific consumption is lowest and running economy will be at its best.

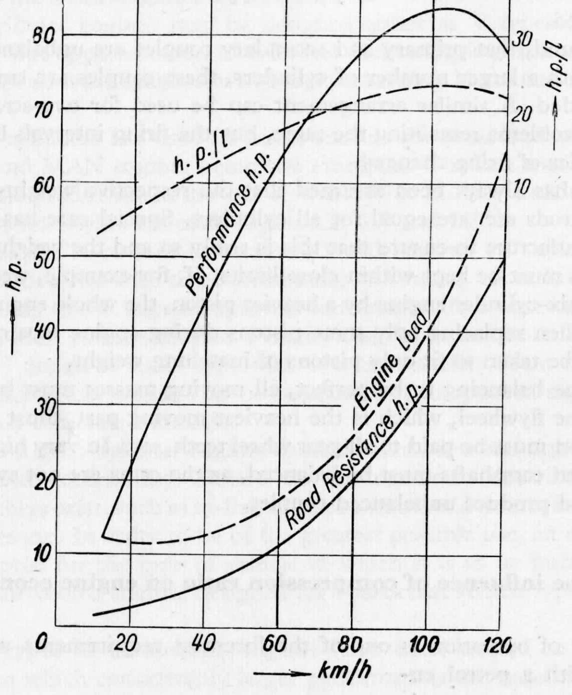

FIG. 161. Engine performance required for travel on level roads, road resistance and percentage engine load.

By observing the influence of the fuel: air ratio on mean effective pressure and therefore on engine performance it will be noted that the latter is reduced with a lean mixture. This is in direct opposition to the requirements of maximum performance with minimum engine weight. Maximum output is reached with a rich mixture having about 20% excess fuel.

In order to attain a high standard of running economy, the engine load under operating conditions must be observed in detail. Fig. 161 shows the maximum engine performance at various road speeds and the output required to propel the vehicle on a level road. It is apparent that with the vehicle in

uniform motion on a level road and at speeds which are usual today, the engine load is relatively low. Extra performance is required for overcoming gradients and for acceleration, but for most of the time the engine is operating under only partial load. This condition is of great interest to the engineer whose aim it is to produce an economical engine. The need for a lean mixture supply to the engine under part-load conditions is evident.

The carburettor of a modern petrol engine must be set to supply a lean mixture on small throttle openings and a rich mixture for a full throttle opening when maximum power is to be provided by the engine. This is effected in two ways in modern carburettors, either by thottling the flow of fuel to the main jet (Solex) or by bleeding air into the fuel jet by a separate valve (Zenith). The fact that an engine may overheat on full throttle with a lean mixture is recognised as is the possibility of a breakdown caused by excessive heat. With part throttle running the amount of heat produced by combustion is proportional to engine output and therefore small. Engine cooling is then efficient despite the lean mixture used and there is no danger of overheating.

In order to provide optimum operating conditions for the engine with both rich and lean mixtures, it is necessary to vary ignition timing with mixture strength, independently of engine speed. This cannot be done by using the common type of centrifugal ignition advance control which accounts only for engine speed.

With the throttle fully open a rich mixture is required, and for a partial throttle opening the mixture must be lean. The vacuum in the inlet manifold being dependent on throttle opening (at constant engine speed), it may be utilized to modify the setting of the centrifugal governor. A vacuum controlled ignition advance regulator unit is shown in Fig. 162. With the throttle fully open, pressure in the inlet manifold rises and the ignition setting is correct for maximum output. With the throttle partly closed, pressure in the inlet manifold is reduced and draws in the diaphragm which overcomes the force of a spring and rotates the whole distributor head against the direction of distributor arm rotation, thereby increasing ignition advance. A greater degree of advance compensates for the slower rate of burning and maximum gas pressure inside the cylinder is again attained at the most convenient moment, i.e. 10° after T.D.C.

The compression ratio should be chosen as high as circumstances permit, for thermal efficiency depends on it to a large extent. The upper limit for the compression ratio used is given by the danger of detonation occuring. If it is intended to use a fuel with a given octane rating, the upper limit of the compression ratio is given by the geometry of the combustion chamber. In conjunction with this, the "mechanical octane number" of an engine is sometimes quoted.

If, by altering the shape of the combustion chamber or by the use of other means, the compression ratio is raised above the original limit without detonation occuring, the engine octane number has also been raised.

A high compression ratio is desirable for running economy under partial load conditions. The shape of the combustion chamber should therefore be chosen with the object of making the highest compression ratio possible. As an automobile engine is mostly working only under partial load, the compression ratio is also chosen for partial load conditions. With partial load operation the entry of air into cylinders is throttled, whereby the real compression ratio, pressure or temperature at the end of the compression stroke are reduced. With part throttle conditions prevailing, the compression ratio

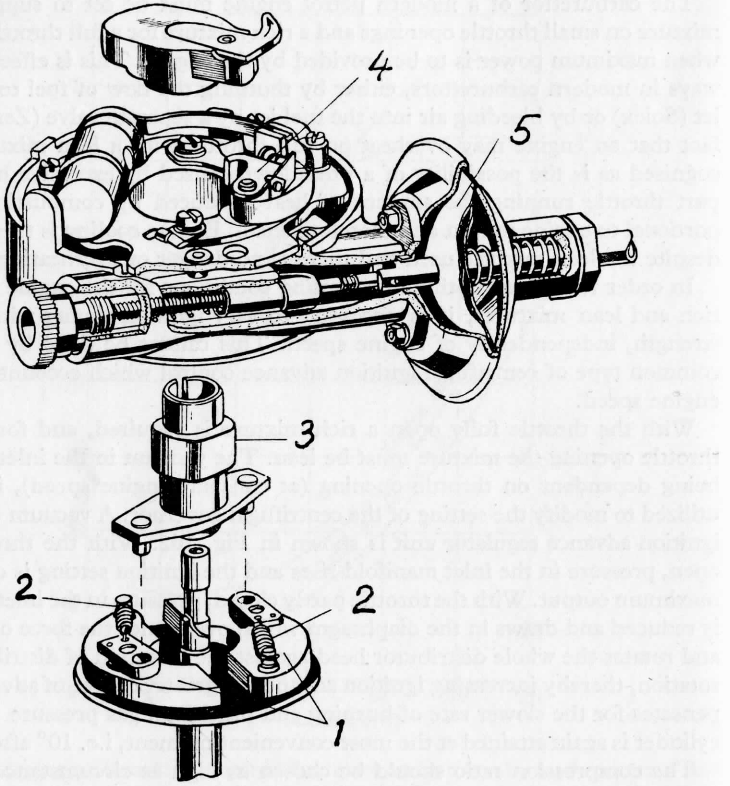

FIG. 162. Vacuum control of ignition advance.

1 — centrifugal weights, *2* — governor springs, *3* — distributor cam, *4* — moving contact, *5* — vacuum control.

may therefore be raised without causing detonation. At full throttle, measures have to be taken to prevent detonation. This may be done in the following ways:

1. enriching the mixture,
2. internal cooling,
3. raising the fuel octane number,
4. altering ignition advance.

The tendency to detonate may be radically curbed by enriching the mixture. Fig. 163 shows the dependence of the maximum useful compression ratio on the point of detonation in a laboratory engine upon the fuel: air mixture ratio. It will be seen that the resistance to detonation of a fuel rises rapidly with a richer mixture. With a lean mixture the resistance to detonation is increased

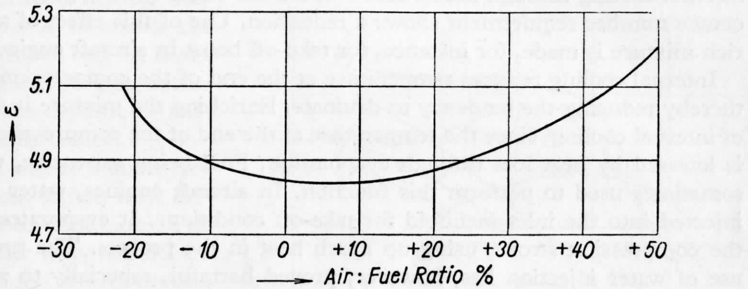

FIG. 163. Effect of air: fuel ratio upon the maximum admissible compression ratio.

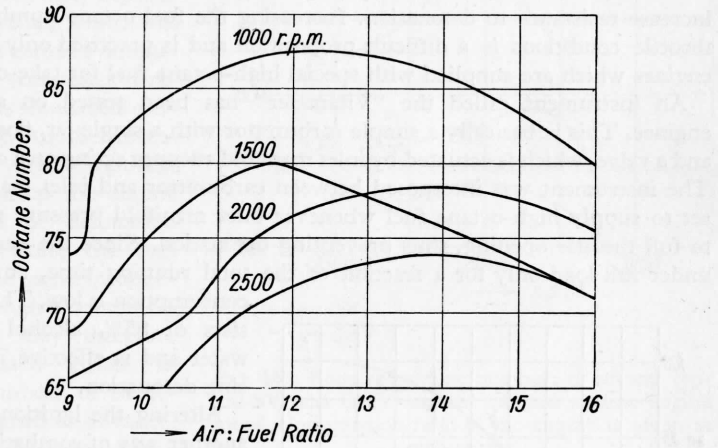

FIG. 164. Effect of air: fuel ratio upon desired octane number in a single-cylinder experimental OHV engine.

by the slower rate of combustion, but this is of no use when maximum power is aimed at.

Efforts have been made to exploit this phenomenon by supplying low grade fuel to the engine at uniform road speed and high grade fuel whenever the need of acceleration or of overcoming gradients called for high performance. These experiments did, however, not meet with success.

The fuel: air mixture ratio influences the velocity of flame travel and combustion, internal cooling etc. and therefore it also influences the octane requirement of an engine. Fig. 164 shows the variation of the octane requirement of a single cylinder overhead valve engine with engine speed. By setting the mixture extremely rich at more than 1 : 11, the beneficial effects of internal cooling through excess fuel evaporation begin to be apparent and the octane number requirement shows a reduction. Use of this effect of an over-rich mixture is made, for instance, for take-off boost in aircraft engines.

Internal cooling reduces temperature at the end of the compression stroke, thereby reducing the tendency to detonate. Enriching the mixture is a means of internal cooling, since the temperature at the end of the compression stroke is lowered by heat loss through evaporation. Fuel being expensive, water is sometimes used to perform this function. In aircraft engines, water may be injected into the inlet manifold for take-off conditions. It evaporates during the compression stroke, using up much heat in the process. The prolonged use of water injection has, however, proved harmful, especially to sparking plugs.

For the maximum engine output to be reached, a mixture about 20 % rich has been found most satisfactory (Fig. 160), and further enriching does not increase resistance to detonation. Increasing the fuel octane number for full throttle conditions is a difficult proposition and is practised only on aircraft engines which are supplied with special high-octane fuel for take-off.

An instrument called the "Vitameter" has been tested on automobile engines. This is basically a simple carburettor with a single jet, float chamber and a valve, which is actuated by inlet manifold vacuum acting on a diaphragm. The instrument was interposed between carburettor and inlet manifold and set to supply high-octane fuel whenever inlet manifold pressure rose owing to full throttle opening, thus preventing detonation. Since the engine works under full load only for a fraction of the total running time, auxiliary fuel consumption is low. The fuel consists of 85% alcohol and 15% water and is effective in suppressing detonation.

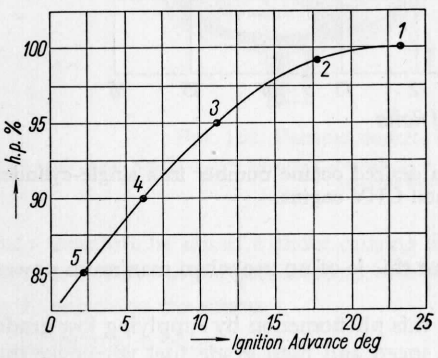

FIG. 165. Influence of octane number and ignition advance upon the detonation point.

Altering the ignition setting is another way of combating detonation. Not even with fuel of a low octane number will detonation occur if the ignition setting is retarded. The results obtained by laboratory test on a single-cylinder engine with a compression ratio of 7·25 : 1, running at 1000 rev/min on full load are shown in Fig. 165. For 100% output to be reached, the engine must be run on 98 octane fuel and the ignition setting must be 23° advance. If fuel of only 93 octane rating is used, 11° is the optimum ignition advance and the output will be 95% (point 3). At point 5 only 86 octane rating is required. Low grade fuel may therefore be used at the expense of output. This requirement can be met when plotting the ignition advance curve for the governor. Failing this, the basic ignition setting may be modified to enable the engine to be run on lower grade fuel, though naturally at the expense of reduced performance.

The engine octane requirement differs with differing load conditions. Put in another way, this means that under all conditions the engine should ideally be run on such a mixture of the basic fuels, heptane and iso-octane, which will keep it just short of detonating. The compression ratio remains unaltered during the test. Another method uses a variable ignition setting to keep the engine just short of detonation occuring, while the fuel used is the same throughout. The engine octane requirement may therefore be expressed either in octane numbers or in degrees of ignition advance.

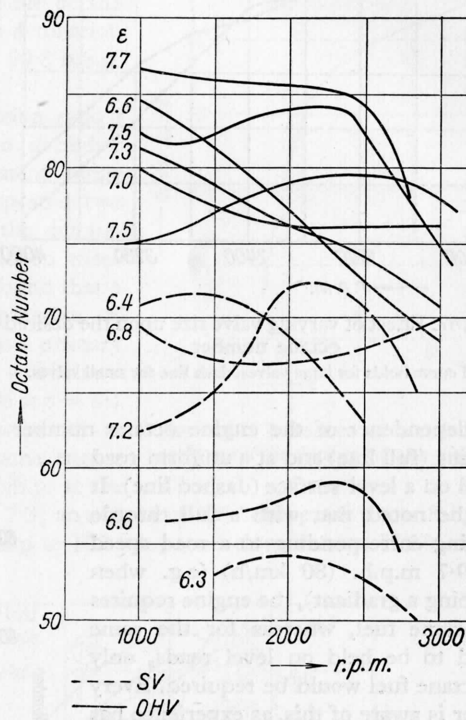

FIG. 166. Required octane numbers of several typical SV and OHV engines plotted against engine speed. Compression ratio of the engine is given at each curve.

Fig. 166 shows the octane number requirement variation with engine speed [68]. The typical curves for overhead valve and side valve engines are of interest. Overhead valve engines require a high-octane number fuel at low engine speeds, whereas at higher speeds the octane number requirement decreases. Side-valve engines do not require high-octane number fuels at low speeds, their octane requirement growing with rising engine speed up to a point below maximum engine speed, after which it drops. The probable cause of the higher octane number requirement of overhead valve engines is their higher volumetric efficiency and their resultant higher mean effective pressure.

Octane requirement values are influenced to a great extent by combustion chamber geometry and by volumetric efficiency. Their curve is also influenced by variation in valve size (Fig. 167).

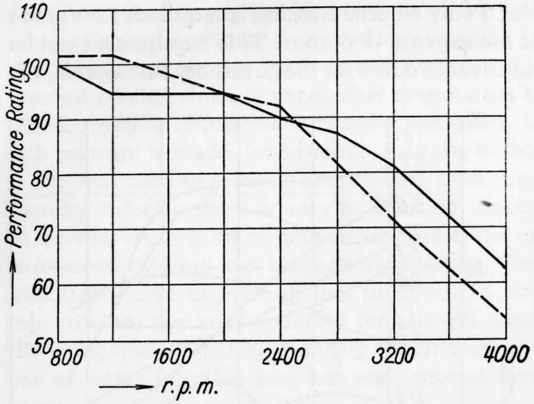

FIG. 167. Effect of varying valve size upon the desired octane number.

Full curve holds for large valves, dash line for small valves.

The effect of throttle opening on the octane requirement of an engine is also considerable. With the throttle only partly open, less air enters the cylinder and compression is low. The effect is the same as when the compression ratio of the engine is reduced.

Fuel with considerably lower detonation resistance is therefore satisfactory under partial-load conditions, as is apparent in Fig. 168, which shows the dependence of the engine octane number requirement at full throttle running (full line) and at a uniform road speed on a level surface (dashed line). It will be noted that with a full throttle opening corresponding to a road speed of 49·7 m.p.h. (80 km/h) (e.g. when climbing a gradient), the engine requires 72 octane fuel, whereas for the same speed to be held on level roads, only 14 octane fuel would be required. Every driver is aware of this, as experience has taught him to close the throttle slightly whenever the engine begins to knock, upon which detonations cease.

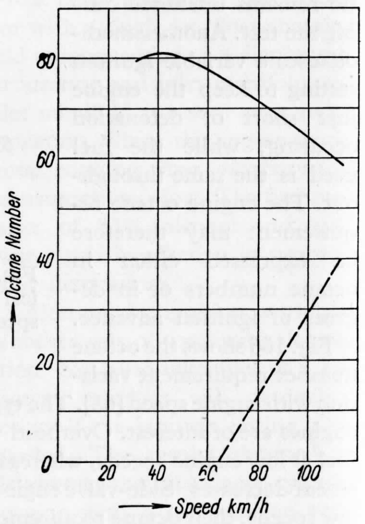

FIG. 168. Influence of engine load upon the desired octane number.

Full line holds good for full load (acceleration, up-hill travel, open throttle), dash, line for partial loads (uniform travel on level roads).

The engine octane number requirement is naturally influenced to a major degree by the compression ratio (Fig. 169). As already stated, the highest possible compression ratio with a given fuel should be used in order to maintain high thermal efficiency and fuel economy.

The compression ratio best suited is determined in the following manner. First of all the octane number requirement of the engine in dependence on

power output and ignition advance is ascertained. The result, found by varying the ignition setting during a dynamometer test, is then entered in a graph. For each octane number, a corresponding performance and ignition setting exists for the detonation point. To quote actual figures, in this instance the corresponding ignition advance for octane number 85 is 20° (in circle) at detonation point and the engine output 99·8 b.h.p. (See Fig. 170.)

By increasing the compression ratio a similar curve, transferred to a higher output and octane requirement range will be obtained. This is repeated two or three times. By marking the detonation points of the fuel tested on these curves (crosses), it will be found that a fuel behaving under compression of $\varepsilon = 7{\cdot}3$ in the same manner as a primary fuel with an 85 octane rating, will behave as if its octane number were 86 at $\varepsilon = 8{\cdot}1$ compression, and as if it were 88 at $\varepsilon = 8{\cdot}7$ compression. The same engine will have an output of 99·8 b.h.p. at the point of detonation with $\varepsilon = 7{\cdot}3$; an output of 105·5 b.h.p. with $\varepsilon = 8{\cdot}1$; and at $\varepsilon = 8{\cdot}7$ the output will drop to 103 b.h.p., which represents 99·8% of the

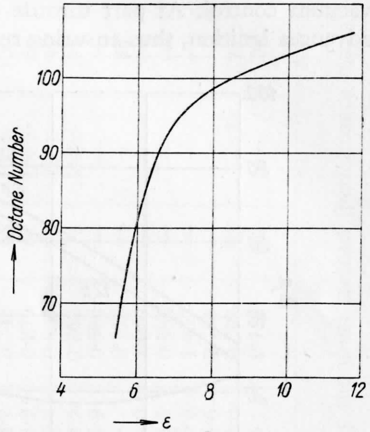

FIG. 169. Influence of compression ratio upon the desired octane number.

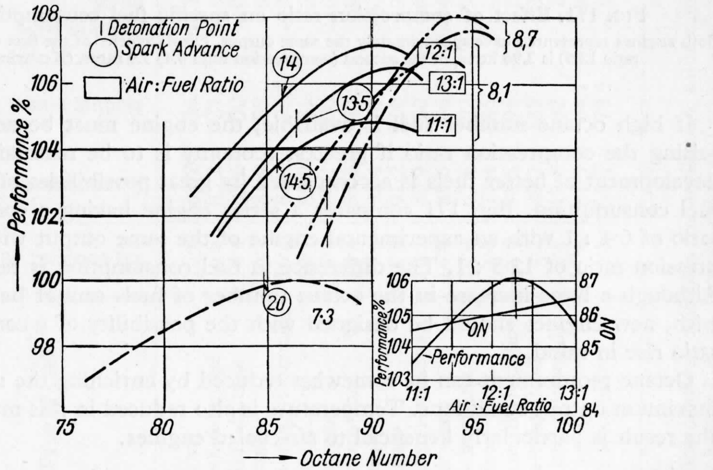

FIG. 170. Dependence of engine performance on octane number at varying compression ratio, ignition advance and fuel: air ratio.

maximum output at $\varepsilon = 7{\cdot}3$; 98% b.h.p. max. at $\varepsilon = 8{\cdot}1$ and only 94% b.h.p. max. at $\varepsilon = 8{\cdot}7$.

It will therefore be found convenient to choose $\varepsilon = 8{\cdot}1$ and to prevent detonation at maximum output by reducing ignition advance to 14° by the vacuum control. At part throttle openings the vacuum controlled regulator advances ignition, thus ensuring running economy.

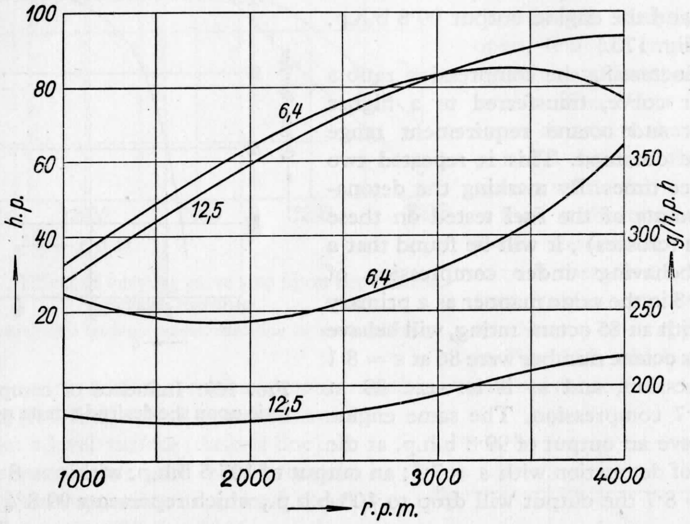

Fig. 171. Effect of compression ratio on specific fuel consumption.
Both engines represented have approximately the same output. Displacement of the first (compression ratio 12.5) is 2.96 litres, of the second (compression ratio 6.4) 3.9 litres. (Kettering).

If high octane number fuel is available, the engine must be adapted by raising the compression ratio if greatest economy is to be maintained. The development of better fuels is accompanied by great possibilities of reducing fuel consumption. Fig. 171 compares a series engine having a compression ratio of 6·4 : 1 with an experimental engine of the same output with a compression ratio of 12·5 : 1. The difference in fuel consumption is remarkable. Although a large increase in the octane number of fuels cannot be reckoned with, new engines should be designed with the possibility of a compression ratio rise in mind.

Octane requirement can be somewhat reduced by enriching the mixture at maximum output conditions. Temperature is also reduced in this manner and the result is particularly beneficial to air-cooled engines.

Properties of Engine Fuels Table 31

Type of fuel	H_2 content, %	C content, %	Boiling point, °C	Solidification point, °C	Calorific value, kcal/kg	Heat of evaporation, kcal/kg	Density kg/litre	Anti-knock value performance rating: without T.E.L. lean mix.	without T.E.L. rich mix.	with 1.06 cm³ T.E.L. per litre lean mix.	with 1.06 cm³ T.E.L. per litre rich. mix.
Pentane	16·8	83·2	36	−130	10 720	85·6	0·626	41	41	63	63
2-methyl-butane	16·8	83·2	27.8	−160	10 720	81·1	0·620	76	—	120	130
2,2-dim.-butane	16·4	83·6	50	−100	10 670	73·9	0·649	78	—	130	130
Heptane	16·1	83·9	98	−90	10 670	76·7	0·684	22	22	36	36
2,4-dimethyl-pentane	16·1	83·9	80.5	−120	10 620	71·1	0·673	62	62	95	95
Triptane	16·1	83·9	81	−25	10 620	69·5	0·690	140	200	200	300
Iso-octane	15·9	84·1	100	−108	10 620	65·0	0·692	100	100	153	153
Cyclopentane	14·4	85·6	49.5	−93	10 440	88·9	0·745	65	100>	100	160>
Benzene	7·6	92·4	80	+5.4	9 560	93·9	0·879	68	160>	68	160>
Toluene	8·8	91·2	110.5	−95	9 670	86·7	0·867	93	160>	95	160>
Isopropyl-benzene	10·1	89·9	150	−96	9 840	74·5	0·862	78	160>	93	160>
Isobutylene	14·4	85·6	−6.8	−140	10 780	93·9	0·594	—	—	—	—
Di-isobutylene	14·4	85·6	102	−92	10 500	—	0·715	64	—	85	160>
Aniline	7·6	77·4	185	−6	8 340	104·0	1·022	—	—	—	—
Monomethyl-aniline	8·5	78·5	196	+22	8 670	95·5	0·986	—	—	—	—
Tetra-ethyl lead	6·2	29·7	182	−137	—	40·6	1·653	—	—	—	—
Ethylene dibromide	2·1	12·8	132	+10	—	45·6	2·181	—	—	—	—
Methanol	12·6	37·5	64.7	−94	4 660	263·0	0·793	75	— (3)	75>	— (3)
Ethyl alcohol	13·1	52·1	78.3	−113	6 460	206	0·789	75	—	75>	—

In rich alcoholic mixtures the anti-knock value cannot be measured by the usual method (performance rating). However, they have a high anti-knock value and great resistance to self-ignition.

8. Fuels

The quality of fuel determines to a large extent the output which can be obtained from an engine. The suitability of fuels is judged neither by their calorific value nor by their specific gravity, but chiefly by their resistance to detonation.

The properties of some fuels are given in Table 31. If a fuel of high calorific value is used, the vehicle will travel a long distance per unit of fuel. The calorific value has only a small effect on engine performance, as the engine burns a fixed mixture of air and petrol and it is the calorific value of this mixture which influences engine output. For complete combustion to take place, various fuels need different amounts of air. Heptane, for example, needs 15·2 units of air for the complete combustion of 1 unit of fuel, benzole 13·2 units of air and methyl alcohol only 6·44 units. The calorific values of these fuels on the other hand, are: heptane – 10 720 kcal/kg, benzole – 9630 kcal/kg, methyl alcohol only 5340 kcal/kg.

The energy liberated by the combustion of respective fuel/air mixtures will be: heptane – 410·7 mkg/litre, benzole – 401·5 mkg/litre, methyl alcohol – 406·5 mkg/litre. The energy content per litre of chemically correct fuel/air mixture is therefore very nearly the same for all hydrocarbon fuels.

The resistance of fuels to detonation is generally expressed to-day by their octane number, giving the percentage of iso-octane in a fuel used as a basis for comparison, composed of heptane and iso-octane. The octane number is found by mixing the two fuels in such proportions as to obtain equal detonation or "anti-knock" resistance as that of the tested fuel in a laboratory test engine. Nowadays the so called "motor method" is employed, by which the fuel is tested in a single cylinder CFR test engine with a variable compression ratio. The other method used, referred to as the "research method", varies from the above in engine speed and inlet air temperature (Table 32). The research method gives a higher octane number and approximates real running conditions more closely. If various brands of fuel are tested by both methods, the octane

Table 32

Comparison of the "Motor" and "Research" Test Methods

Test Method	ASTM Motor	CFR Research
Engine speed, rev/min	900	600
Temperature of cooling water, °C	100	100
Temperature of intake mixture, °C	149	52
Spark advance	variable with compression ratio	13 deg

numbers obtained will differ widely in some cases, in others they will be practically the same. This proves that some fuels are highly sensitive to compression temperatures and engine speed, while others remain unaffected by variation of these conditions. If the difference in octane numbers gained by the two methods is less than 4, the fuel is said to be "non-sensitive" and if it exceeds 8, the fuel is said to be "sensitive". By adding the same proportion of knock inhibitor or "dope" to both fuels, the octane number of the "sensitive" fuel may be raised considerably, whereas the octane number of the "non-sensitive" fuel will be improved only slightly.

It follows that the octane number as determined by the "motor method" does not classify fuels in a very reliable manner. This led to a search for other means of classification.

Dopes

The octane number of a fuel can be increased and its qualities improved by the addition of a small amount of chemical compounds known as dopes, which raise the resistance to detonation of the said fuel.

The best known and most extensively used dope to-day is tetra-ethyl lead (TEL). The lead deposits produced by combustion of the mixture cover the surface of the combustion chamber and also tend to soil spark plugs. By the addition of ethylene dibromide the lead oxides are converted into gaseous bromides which leave the combustion chamber with the exhaust gases. A mixture of these chemical compounds is marketed under the name of "ethyl-fluid". Even when this is used, considerable deposit forms and for this reason the maximum tetra-ethyl lead addition is 0·8 cm³ per litre of fuel. Even in such small quantity, tetra-ethyl lead is a very efficient dope. It is also strongly poisonous and fuel containing it must, therefore, be handled with care.

A less known and also less extensively used dope is iron carbonyl, which deposits a reddish powder (iron oxide) inside the cylinder. This, mixed with oil, forms a grinding paste, for which reason its use is not recommended. Of the other substances raising the resistance to detonation, monomethyl aniline may be named as an example. It must be added to fuel in much larger quantities.

The most effective dope is tetra-ethyl lead, the addition of a diminutive

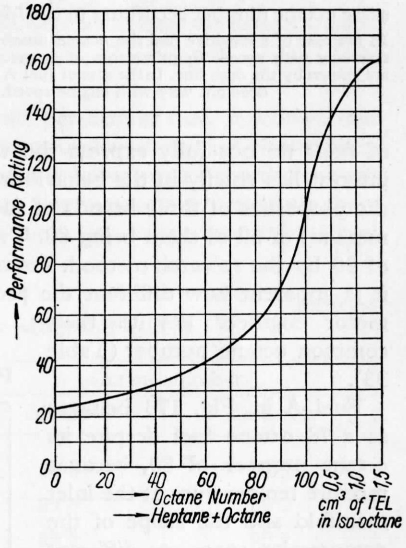

FIG. 172. Relationship between octane performance rating and contents of tetra-ethyl lead in one litre of iso-octane.

percentage of which considerably raises the octane number of a fuel. By adding tetra-ethyl lead to high grade fuel or to iso-octane, fuels with octane numbers exceeding 100 are produced. In such cases the resulting fuel cannot be compared with a mixture of iso-octane and heptane; a mixture of iso-octane and tetra-ethyl lead is used as a basis for comparison. For such comparisons, the octane number has been replaced by the "performance rating". This scale was set as a result of experiments in order to provide a means of classifying fuels of over 100 octane rating by numbers which are directly proportional to the gain in output obtained by using the fuel concerned. The "performance rating" is sometimes used also for fuels of octane numbers below 100. The relative values of octane and performance numbers are given in Fig. 172.

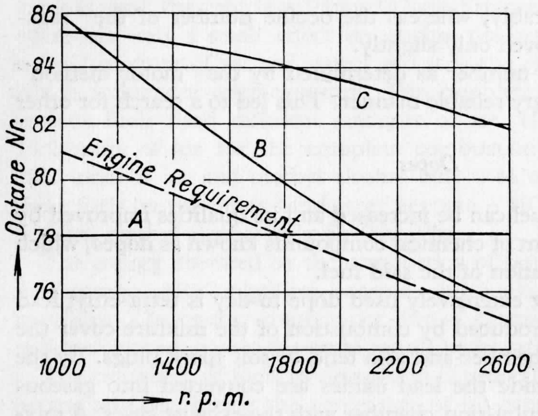

FIG. 173. Comparison of three types of fuel of the same octane number according to the Motor method.
In the case of a sensitive fuel the octane numbers varies substantially with speed. Requirements of a typical OHV engine are shown by the dash line. In the case of fuel A octane number does not vary with engine speed.

The anti-detonation value of a fuel

As already stated, the methods used today to determine the octane ratings of fuels do not fully express the value of the fuel. The motor engineer's interest lies chiefly in the behaviour of the fuel in an engine. Fig. 173 shows the properties of three brands of fuel – *A, B, C*, the octane number (motor method) of all of them being 80. Two of them, *B, C*, have the same number of 90 by the research method. From their behaviour at various engine speeds it is apparent how different the properties of these fuels are as seen by the motor engineer despite their common octane number (Table 33).

Table 33

Properties of Fuels According to Fig. 173

Octane number according to the method	Motor	Research
Petrol A	80·0	80·0
Petrol B	80.0	90·0
Petrol C	80·0	90·0

Fuel A in Fig. 173 behaves as a 78 octane fuel despite its octane number of 80, because mixture temperature at the inlet manifold and the shape of the compression space are different from those in the CFR test engine.

A so called "road method" has been developed for fuel testing and this classifies fuels according to their real value. By this method the curve of detonation points can be determined at different ignition settings and varying engine speeds while driving the vehicle on the road (the automatic ignition control having been put out of action).

Fig. 174 illustrates the results obtained by such a test with a typically saturated fuel bearing a chemical resemblance to straight chain hydrocarbon paraffins. Fig. 174 together with Table 34 illustrates the variation of the octane number of this fuel with the addition of tetra-ethyl lead. The dashed line marks the engine octane requirement with normal ignition advance and correct carburettor settings. Fuel *A* would detonate up to 2900 rev/min and only at higher speeds would knock be absent. After the addition of 0·4 cm³ per litre of tetra-ethyl lead, the fuel is nearly satisfactory, the desired properties being fully gained by the addition of only 0·8 cm³ tetra-ethyl lead per litre of fuel. It follows that with the addition of tetra-ethyl lead the octane number of the fuel rises rapidly at high engine speeds, whereas improvement at lower engine speeds is much slower. This fuel is "non-sensitive" and an improvement in octane numbers with both methods does not neccessarily mean an improvement in actual road performance.

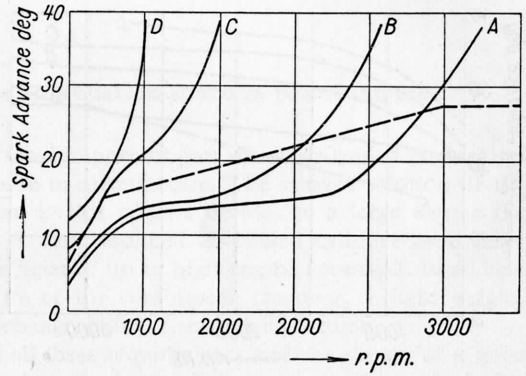

FIG. 174. Relationship between the maximum permissible spark advance at the detonation point and engine speed for a non-sensitive paraffinic fuel *A*.
By the addition of tetraethyl lead according to Table 34 the required advance varies in accordance with curves *B*, *C* and *D*. The dash line shows the requirement on ignition advance of a normal engine, according to which ignition advance was set.

Table 34

Properties of Fuels According to Fig. 174

Fuel	T.E.L. cm³/litre	Octane number	
		Motor	Research
A	0·0	80·0	80·5
B	0·13	85·0	86·0
C	0·4	89·5	90·5
D	0·8	93·5	98·0

A similar diagram, obtained with a non-saturated fuel is shown in Fig. 175. Bearing in mind the difference in octane numbers obtained by the two methods this fuel is obviously "sensitive". It is almost satisfactory without doping except in a small range between 2800 and 3000 rev/min. After a small modification to the automatic ignition control graph this fuel would be perfectly satisfactory. The addition of tetra-ethyl lead raises the octane number of the fuel almost consistently over the entire engine speed range. According to Table 35, the addition of 0·8 cm³ tetra-ethyl lead will raise the octane number (motor method) by 3 as compared with the increase of 13·5 with the fuel considered before. This small rise means a great improvement in terms of road performance as the compression ratio of the engine can be raised (Fig. 175).

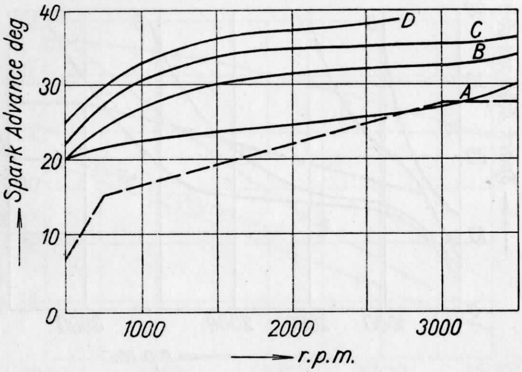

FIG. 175. Relationship between the maximum permissible spark advance at the detonation point and engine speed for a sensitive fuel.

By the addition of tetra-ethyl lead according to Table 35 the requirement varies in accordance with curves *B*, *C*, and *D*. The dash line represent the ignition advance given by the disributor.

These two examples illustrate how the octane number does not classify fuels according to their real value in the engine and how modern 80 octane fuel, produced by catalytic cracking is of much higher quality than fuel with the same octane number produced 20 years ago and composed mainly of paraffin hydrocarbons.

Table 35

Properties of Fuels According to Fig. 175.

Fuel	T.E.L. cm³/litre	Octane number	
		Motor	Research
A	0·0	82·0	94·5
B	0·13	82·5	96·5
C	0·4	84·0	98·0
D	0·8	85·0	99·0