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. 65 shows an ordinary symmetrical constant acceleration valve
lift diagram plotted against crankshaft degrees where 0° represents the inner centre and 180° the outward centre of the piston.

Fig. 65.—Valve Diagram. Constant Acceleration throughout

Fig. 66 shows the same diagram re-plotted in terms of piston displacement, while the dotted line represents the relative velocity of the piston throughout its stroke (assuming in both cases that the ratio of connecting-rod length to crankthrow is 3.6 : 1).

Fig. 66.—Valve Opening Diagram in relation to Piston Displacement

Fig. 67.—Effect of displacing Valve Opening Diagram in relation to Crankshaft

In fig. 67, 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. 68 shows the change of velocity through the valve throughout the stroke in the case of diagram fig. 66, 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 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.
VALVES AND VALVE GEAR

(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°, as shown in figs. 69 and 70, 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. 69. Valve Opening Diagram with Period extended 20° of Crank Angle](image)

![Fig. 70. Valve Opening Diagram in relation to Piston Displacement](image)

although the opening period of the valve is 20° 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 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. 71; 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. 72. In such a case it will

Fig. 72.—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. 73, 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;

![Diagram](image)

Fig. 73.—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° and consecutive firing.
3. Four cylinders with cranks at 180°.
4. Six cylinders with cranks at 120° fed by two carburettors, each distributing to a group of three cylinders.
5. Six cylinders with cranks at 120° 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 cylinder: 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. 74.—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. 74, 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 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...
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 confirmed 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.

![Indicator Diagram at 1500 R.P.M. shows Pressure Variations during Exhaust Stroke](image)

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. 74, 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
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. 75 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.
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 2⅛ inches diameter, and is operated more 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 1800 ft. per second per second, corresponding to a spring pressure of 56 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. 76, 77 and 78. 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 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, ⅝ in. wide and a tappet diameter of, say, 1¾ 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 weak film, 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. 79 shows the acceleration on tangent flanks (dotted curve) and round noses with circular followers, also harmonic cams with flat or mushroom followers.

Fig. 80 gives the figures for hollow flanks with circular followers.

Fig. 81 deals with round (convex) flanks with circular followers.

Fig. 82 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. 82),

\( C = \) 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 engine speed).

A preliminary layout 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
the lift to the minimum, at the same time providing ample surface on the cam and follower to take the heavy loading.

![Diagram of cam and follower mechanisms]

Fig. 82

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 (1\(^{1/3}\) 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. 83). Whenever possible the proportions of the nose of the cam should be such that \(R/L\) 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 velocities in the first instance and plot the cam accordingly.*

A convenient method of construction is appended (fig. 84).

**Construction of Harmonic Cams**

(see fig. 84)

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° to 16° total (crankshaft degrees), i.e. 3° to 4° a side actually on the cam.)

On BO produced mark off OD, making OD : OA :: acceleration on flank of cam : 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.)

With D as centre, DA as radius, describe arc AB.

With O as centre, OB as radius, describe arc BF.

Then as AF : the required lift :: OB : 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 1/8 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 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 at 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 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. 79, 80, and 81, and the key diagrams in fig. 82, are worked out for all reasonable proportions, but the formulae 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

\[
\text{Acc.} = W^2 R \left( \frac{1 + 2 \tan^2 \theta}{\cos \theta} \right) \quad \ldots \quad \ldots \quad (1)
\]
THE INTERNAL-COMBUSTION ENGINE

Round nose, and round or hollow flank

\[
\text{Acc.} = W^2 R \frac{(\cos \theta + n^2 \cos 2\theta + \sin^4 \theta)}{(n^2 - \sin^2 \theta)^{3/2}}. \quad \ldots \quad \ldots \quad (2)
\]

Simple harmonic cam with flat follower

\[
\text{Acc.} = W^2 R \cos \theta, \quad \ldots \quad \ldots \quad \ldots \quad (3)
\]

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 \( \frac{1}{3} \)th to \( \frac{1}{4} \)th of the lift is suitable.

**Valve Springs.**—These must be considered from four points:
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.

(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 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. 85).
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 20 to 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 to 15 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:

$$N = 513 \sqrt{\frac{R}{W}}$$

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. 86.—Spring Graph—Stresses

Fig. 87.—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. 86 for light gauges and fig. 87 for heavier gauges) when the force and approximate diameter of spring are known. The deflection per coil \( (d) \) may be read off from the deflection graph (fig. 88). 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 \( \frac{D}{d} + 2 \).

The maximum stress \( = \frac{8FD}{\pi d^3} \) lb. sq. inch,

where \( F' \) = maximum force in lbs.,

\( D \) = mean dia. of coils in inches,

\( d \) = dia. of wire in inches;

also the maximum deflection \( = \frac{8FND^3}{Cd^4} \),
where \( N \) = effective number of coils (total minus 2), and \( C \) = the transverse modulus of elasticity = 13,000,000.

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.
CHAPTER IX

PISTON DESIGN

In the first volume of this book the question of piston design was dealt with at considerable length, and though written some eight years earlier, the principles laid down and the views expressed therein hold good, generally speaking, at the present day. 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.

As has been shown in the first volume, 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° to 110°. 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. In the case of aluminium alloy pistons the position is somewhat different, because the softer metal of the piston permits of particles of grit embedding in its surface, and so grinding or lapping the cylinder walls; also there is evidence to show that some at least of these alloys are liable to form a highly abrasive surface due to segregation of a hard component of the alloy.

It has already been shown in Volume I that, 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. 89 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 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. 90, 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 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

![Diagram](image)

Fig. 90.—Curve showing Drop in Piston Friction alone from fig. 89

reduced piston friction more than compensates for the reduction in indicated power. Fig. 91 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.

![Diagram](image)

Fig. 01.—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 should be made as close a fit as possible in their grooves.

(2) Ample venting should be provided below the lowest 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. 92 shows an arrangement of rings, &c., which has been found very effective in preventing the passage of oil into the combustion space.

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. 93 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 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, 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 directly 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.

Fig. 94.—Test Readings at three different speeds.
PISTON DESIGN

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. 94, 95, and 96 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. 95.—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. 97 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 it of great advantage. The example shown in fig. 97 is the largest yet made, for this single piston transmits 135 B.H.P. at 1400 R.P.M.

Fig. 98 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. 99 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. 98; 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

![Diagram of piston design]

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

![Diagram of aluminium slipper type piston for large high-speed engine]
and prevent seizures; others, again, have even gone so far as to introduce springs between the connecting-rod and piston, in order 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 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
PISTON DESIGN

illustrated in fig. 100, which shows the position taken up by the piston at various points in the cycle.

Fig. 100.—Diagram of Forces on Piston with offset Gudgeon Pin, showing effect on clearance

Fig. 101 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.
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 there shall be as little clearance as possible in the ring grooves, since this determines largely the amount of oil they will pump. 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.

Radial Thickness.—The radial thickness determines the
PISTON DESIGN

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 some 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 something less than the mean pressure of the cycle. This theory is strengthened by the observed fact that when an engine with new rings is motored for several hours without gas pressure in the cylinder the rings do not bed in, but that if run under load they bed very rapidly, particularly the top ring; this may be due to pressure behind the rings, or it may be due to the fact that when run under load the lubrication is less effective. Experience indicates that in a well-made ring a spring pressure of from 5 to 6 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.

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
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.

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. 102, 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.
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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. 103, and the photographs reproduced in figs. 104 and 105. 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
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. 102 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 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. 106. 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 petroIs, 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, 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. 107, 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
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Fig. 108—General Arrangement (Cross Section), 3½ × 9½ Single Cylinder. Experimental Engine
which the four cylinders and the upper half of the crankcase are cast in one piece.

(a) 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. 108, 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 question 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 attain a high torque at low speeds, and to do so without detonation or "pinging."

(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. 109 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.

We will consider both cases and assume that the engines are of

![Graph](image)

Fig. 109.—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. 110, 111, and 112 show the brake horse-power and general
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. 109, 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 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
curve of the two engines against fig. 109 in the above ratio, as shown in fig. 113, 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. 114 and 115 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 the case of the larger engine, but the difference is small and hardly worth taking into consideration. Figs. 114 and 115 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

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 12.5 per cent, and on its bottom speed about 31 per cent. It will be noted

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. 116 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

![Graph showing acceleration on various gears, two-litre engine](image)

Fig. 116.—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
casting, 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. 117 and 118 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. 117.—Load Factor and Fuel Consumption in Pints per B.H.P. Hour when running on level at average speeds of from 10 to 40 Miles per Hour. Three-litre Engine

Fig. 118.—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. 119 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

![Figure 119](image)

**Fig. 119.—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

![Image](image_url)

**Fig. 120.—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. 120, 121, 122, and 123 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 con-

Fig. 121.—14-H.P. Vauxhall Engine

...ustion 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. 122 and 123, 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. 122.

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.

The crankshaft has a diameter of 1.75 inches throughout, the widths of the several bearings being:

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<td>Flywheel</td>
<td>2.65</td>
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<td>Connecting-rod</td>
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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.

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. 124, from which it will be seen that a brake mean pressure of 108 lb. per square inch is obtained at a
Fig. 124.—14-H.P. Vauxhall Engine. Gas Velocity, Brake Horse-power, Brake Mean Pressure, and Fuel Consumption Curves

Fig. 125.—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. 125 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.

![Diagram](image)

Fig. 126.—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. 126, 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

![Diagram of a sleeve-valve engine](image)

Fig. 127

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. 127 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. 128-131 show various alternative methods adopted by Burt for operating a single-sleeve valve. The method shown in fig. 128...
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 crankshafts are employed and the sleeve is operated from the centre of 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. 129 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. 128.

Another very attractive form is that shown in fig. 130, in which a ball-and-socket joint is used. This form has the advantage of being considerably lighter and more compact:

also it is probably the least expensive and the most accessible. Fig. 131 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 128, 129, and 131, 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. 132 shows typical ports $a$ being the ideal shape, $b$ the same port 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.

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 settings are given in fig. 133, while fig. 134 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 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.
ENGINES FOR ROAD VEHICLES

The sleeve valve is usually centrifugally cast of good quality grey iron, and for engines up to 2½-in. diameter bore is made 1 in. thick, while for engines of 4½-in. diameter bore 1½ 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 \( \frac{1}{575} D \) 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.

\[
T = \frac{\pi D \times 0.575D}{D(2 \text{ No. of ports}) - 1} (\text{? No. of ports}) - 1
\]

![Diagram of sleeve valve settings](image)

Fig. 134.—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. 135.

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

\[
A = 2 \text{ inlet} \times 2 \text{ exhaust, for it will be seen from fig. 133 that this setting gives the required area.}
\]

\[
D = 68 \text{ mm.} + 2\text{ in.} = \text{say 2·9-in. diameter.}
\]

\[
C = 0.575 \times 2.9 = 1.66 \text{ in., say 1·66 in.}
\]
T = \frac{1.8 \times 2.9}{(2 \times 4) - 1} \approx 0.74 \text{ in.}, \text{ say } 0.7 \text{ in.},

a = \text{Angular travel of sleeve valve (see fig. 132).}

\sin \frac{1}{2}a = \frac{T}{C} \text{ when } T \text{ is central} = \frac{7}{1.65} = 4.2424 \text{ or } 25^\circ 6';

\therefore 25^\circ 6' \times 2 = 50^\circ 12'.

When T is offset \sin a = \frac{bT}{C}.

Referring to fig. 136:

H = \text{Horizontal travel} = \frac{\alpha \pi D}{360} \times \frac{50^\circ 12' \times 3.1416 \times 2.5}{360} = 1.275''.

L = \text{Length of port} = H - \text{cover} = 1.275'' - 0.05'' = 1.225''.

Fig. 135.—Maximum Sleeve-valve Crankthrow. Various Port Settings

Minimum cover = 0.04''. 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. 134) = H + \(\text{cover} = 1.275'' + 0.05'' = 1.325''.

\(l = \text{Height of ports} = T + (T \sin \beta) = 7 + (7 \sin 221/2^\circ) = 9.7''.

4 \text{ ports at 1.225'' long } = 4.9''
2 \text{ spaces at 1.325'' } = 2.65''
2 \text{ spaces at } 0.775'' = 1.55''

Circumference of sleeve valve = 9.1''.

\(\bar{l} = \text{Inlet port tail} = \frac{H}{2} - \left(T \sin \frac{5D}{C}\right),

= 0.6375 - \left(7 \sin 71\frac{1}{2}^\circ \times \frac{1.45}{1.65}\right) = 6375 - 0.08 = 6575''.
\[ le = \text{Exh. port tail} = \frac{H}{2} \left( 1 \sin \frac{5D}{C} \right) \]

\[ = 6375 + \left( 0.7 \sin 15° \times \frac{1.45}{1.65} \right) = 6375 + 1.595 = 6535". \]

With straight-sided ports the flank angle can be solved as follows (see fig. 136):

\[ AU = h - 2r, \quad \tan \theta = \frac{BC}{AC}. \]

\[ r = \text{corner radius usually } 1/8", \quad AB = \frac{BC}{\sin \theta}. \]

\[ AD \text{ and } EB = r, \quad \sin \phi = \frac{r}{AB}. \]

\[ BC = L - (l + 2r), \quad z = 90° - \phi. \]

Flank angle = \( X = z - \phi \).

In fig. 137 the crankshaft timing diagram for the example worked

**CRANKSHAFT TIMING**

**VALVESHIFT TIMING**

UNBALANCED

BALANCED + 7° CRANK PIN ADVANCE

Fig. 137.—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 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

![Diagram](image)

Fig. 138. Valve opening Diagram. 68-mm. Bone Engine; $C = 1.657$, $T = 7^\circ$

![Engine](image)

Fig. 139. -Four-cylinder Bone 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. 137 at e; the corresponding port opening diagram being as shown in fig. 138.

Fig. 139 shows an example of a small four-cylinder motor-car engine of 68-mm. bore and 103 mm. stroke, in which single-sleeve valves actuated by the ball-and-socket mechanism are employed; while

figs. 140 and 141 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. 142-149 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,
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:

(1) To obtain the maximum possible thermal efficiency, with a view to getting the utmost possible power output from the available air.

Fig. 145.—Vauxhall Three litre Racing Engine
(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

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 and water-jackets were cast in one piece and rigidly attached to form an additional 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-

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 

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 crank-pins 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 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. 150, 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.

![Diagram](image_url)

Fig. 150 — 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 inches
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 ... ... ... ... = 3-41 : 1.

Fig. 161.—Vauxhall Three-litre Racing Engine

The chart, fig. 161, 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 wire-drawing or insufficient valve area. With a combustion chamber of the form used giving the maxiumum of turbulence, and with central ignition, the efficiency reckoned in 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

\[
\frac{69 \times 50·5}{100} = 34·8 \text{ per cent at the highest speeds, when turbulence is at a maximum and direct heat loss at a minimum.} \]

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. 151. 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—

<table>
<thead>
<tr>
<th>R.P.M.</th>
<th>Volumetric Efficiency per cent.</th>
<th>Combustion Efficiency per cent.</th>
<th>Indicated mean Pressure lb. per sq. in.</th>
<th>Observed I.M.E.P. lb. per sq. in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>75·0</td>
<td>31·0</td>
<td>133·4</td>
<td>138·5</td>
</tr>
<tr>
<td>2000</td>
<td>77·6</td>
<td>32·6</td>
<td>145·1</td>
<td>148·0</td>
</tr>
<tr>
<td>2500</td>
<td>79·7</td>
<td>33·9</td>
<td>155·0</td>
<td>156·0</td>
</tr>
<tr>
<td>3000</td>
<td>80·7</td>
<td>34·3</td>
<td>159·0</td>
<td>161·0</td>
</tr>
<tr>
<td>3500</td>
<td>80·7</td>
<td>34·6</td>
<td>160·4</td>
<td>162·5</td>
</tr>
<tr>
<td>4000</td>
<td>79·9</td>
<td>34·7</td>
<td>159·2</td>
<td>162·2</td>
</tr>
<tr>
<td>4500</td>
<td>78·5</td>
<td>34·8</td>
<td>156·9</td>
<td>159·0</td>
</tr>
<tr>
<td>5000</td>
<td>77·0</td>
<td>34·8</td>
<td>153·7</td>
<td>...</td>
</tr>
</tbody>
</table>
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.

In the first volume of this book certain empirical formulae, based on an accumulation of experimental results, were given for determining the mechanical efficiency of an engine. These formulae were arrived at from a large number of experiments mostly carried out on quite slow-running engines, but later experiments indicate that they are applicable also to high-speed engines with, perhaps, certain small reservations.

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—

\[
\text{Piston Friction} = \frac{\text{Mean fluid pressure including compression}}{4} + \frac{2 \times \text{Mean inertia pressure}}{10} + \frac{3}{20}.
\]

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—

<table>
<thead>
<tr>
<th>Speed R.P.M.</th>
<th>Piston Friction (lb. per sq. in. M.E.P.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>... 7-7</td>
</tr>
<tr>
<td>2000</td>
<td>... 8-6</td>
</tr>
<tr>
<td>2500</td>
<td>... 10-0</td>
</tr>
<tr>
<td>3000</td>
<td>... 11-4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Speed R.P.M.</th>
<th>Piston Friction (lb. per sq. in. M.E.P.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3500</td>
<td>... 13-1</td>
</tr>
<tr>
<td>4000</td>
<td>... 15-1</td>
</tr>
<tr>
<td>4500</td>
<td>... 17-3</td>
</tr>
</tbody>
</table>
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. In Volume I, fig. 23, a curve is given showing the observed fluid pumping losses at different gas velocities: from this curve, and that of gas velocity given in fig. 151, we find that the fluid pumping losses in this particular engine again expressed in terms of mean piston pressure should amount to:

<table>
<thead>
<tr>
<th>Speed R.P.M.</th>
<th>Fluid Pumping Losses (lb. per sq. m.)</th>
<th>Speed R.P.M.</th>
<th>Fluid Pumping Losses (lb. per sq. m.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>2.0</td>
<td>3500</td>
<td>5.4</td>
</tr>
<tr>
<td>2000</td>
<td>2.3</td>
<td>4000</td>
<td>6.9</td>
</tr>
<tr>
<td>2500</td>
<td>2.9</td>
<td>4500</td>
<td>8.6</td>
</tr>
<tr>
<td>3000</td>
<td>4.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

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:

<table>
<thead>
<tr>
<th>Speed R.P.M.</th>
<th>Total Losses (lb. per sq. m. M.E.P.) estimated</th>
<th>Observed Losses</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>12.7</td>
<td>10.9</td>
</tr>
<tr>
<td>2000</td>
<td>14.2</td>
<td>12.6</td>
</tr>
<tr>
<td>2500</td>
<td>16.5</td>
<td>15.2</td>
</tr>
<tr>
<td>3000</td>
<td>19.3</td>
<td>18.6</td>
</tr>
<tr>
<td>3500</td>
<td>22.7</td>
<td>22.7</td>
</tr>
<tr>
<td>4000</td>
<td>26.6</td>
<td>28.8</td>
</tr>
<tr>
<td>4500</td>
<td>30.9</td>
<td>35.0</td>
</tr>
</tbody>
</table>

Fig. 152 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

![Graph showing estimated and observed fluid and friction losses vs RPM.](image)

Fig. 152.—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. 46 and 47; 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

![Graph showing fuel consumption](image)

**Fig. 153.**—Three-litre Vauxhall Racing Engine

![Graph showing fuel consumption and velocity](image)

**Fig. 154.**—Three-litre Vauxhall Racing Engine
Curve showing Variation in Fuel Consumption with Speed. Most Economical Mixture in all cases. Fuel, Petrol. Oih, “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:

<table>
<thead>
<tr>
<th>B.H.P.</th>
<th>Fuel Consumption (Pints per B.H.P. hour.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>0.455</td>
</tr>
<tr>
<td>70</td>
<td>0.465</td>
</tr>
<tr>
<td>60</td>
<td>0.480</td>
</tr>
<tr>
<td>50</td>
<td>0.505</td>
</tr>
<tr>
<td>40</td>
<td>0.535</td>
</tr>
<tr>
<td>30</td>
<td>0.565</td>
</tr>
<tr>
<td>20</td>
<td>0.610</td>
</tr>
</tbody>
</table>

At 1950 R.P.M. the fuel consumption was:

<table>
<thead>
<tr>
<th>B.H.P.</th>
<th>Fuel Consumption (Pints per B.H.P. hour.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>0.170</td>
</tr>
<tr>
<td>50</td>
<td>0.180</td>
</tr>
<tr>
<td>40</td>
<td>0.195</td>
</tr>
<tr>
<td>30</td>
<td>0.225</td>
</tr>
<tr>
<td>20</td>
<td>0.270</td>
</tr>
<tr>
<td>15</td>
<td>0.305</td>
</tr>
</tbody>
</table>

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 \(48.5 \times \frac{8}{11.4} = 33.9\) 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
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. 155 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 4800 R.P.M., the actual engine speed being probably about 4800 R.P.M.
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 aéroplanes 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 of 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 aéroplanes 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 aéroplane, 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 overall 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. 156, 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 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 at one 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 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. 157 and 158 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

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 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, since the individual cylinders are larger and the losses therefore less in proportion, while, 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

Fig. 158.—Napier Lion Engine, End View
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.
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 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, since the individual cylinders are larger and the losses therefore less in proportion, while, 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

Fig. 158.—Napier Lion Engine, End View
that under normal working conditions local over-heating can be avoided and results obtained comparable with those of a water-cooled cylinder.

It has, however, been found very difficult to ensure a sound job when casting the head on to a steel barrel, while the alternatives of bolting, screwing, or shrinking have none of them proved sufficiently reliable. Probably casting on to the barrel is the most hopeful, for, it is not impossible, and the difficulties which have been encountered are mostly questions of foundry technique, and, as such, will probably be surmounted as more experience is gained. Given that, as in a radial, or rotary engine, the position of the cylinder is such that it has the best of facilities for cooling, it has been demonstrated that the air-cooled cylinder will give from 90 to 95 per cent as high power and efficiency as a corresponding water-cooled cylinder, so long as there is no distortion, leakage, or detonation; but while the water-cooled cylinder can generally survive such ailments due to the automatic intensification of heat transference from the seat of the trouble, the air-cooled cylinder has no such advantage, and must quickly give way to local overheating followed by pre-ignition, and perhaps also by distortion and seizure of the piston or burning out of the valves.

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. 165 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 opinion prevails as to the relative merits and demerits of the different forms in use.

Fig. 159 shows the cylinder construction used in the 290 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.

Fig. 159.—Cylinder Construction used in the 290 H.P. Mercedes Engine

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. 160 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. 161 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 scarless steel tube also screwed to the cylinder head in the same manner as the liner.

The construction shown in fig. 162 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. 163 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. 161.—Zeppelin Airship Engine Cylinder Fig. 162.—Benz Cylinder

Fig. 164 shows the cylinder construction used in the earlier Sun-beam engine, 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 are 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. 165 shows the construction of the Hispano Suiza cylinder block already referred to, in which a complete steel thimble, forming both the cylinder liner and valve seats, is screwed directly into an aluminium cylinder block.

In the Siddeley Puma engine shown later in fig. 175, 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 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. 166, which is the form employed in the Clerget rotating engine, and fig. 167, which shows a section of the Gnome single valve engine.

Fig. 168 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 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. 169, 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. 170 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.

![Diagram of a cylinder](image)

Fig. 169.—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. 171 the whole of the cylinder barrel and head are of steel, but a cast aluminium poultoce 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 poultries 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,

however, two such engines have been developed, namely, the Bristol Jupiter and the Siddeley Jaguar, figs. 170-174.

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.
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Fig. 170 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.

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In the Bristol Jupiter engine shown in Fig. 171 the whole of the cylinder barrel and head are of steel, but a cast aluminium poultece containing the inlet and exhaust valve elbows and heavily
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

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
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 Armstrong, though in both cases the earlier

The internal-combustion engine down, and so to retain the same tapet clearance at all cylinder temperatures. These control rods can be seen on the forward end of the engine (see fig. 172). The Siddeley Jaguar shown in figs. 173 and 174 is a four-tapet
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 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. 175 and 176 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
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 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. 157 and 158, also in the sectional drawings figs. 177 and 178, 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 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. 179 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. 179.—Rolls-Royce Eagle](image)

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
cooling system, &c., is 1177 lb. or 3.37 lb. per horse-power. The fuel and oil consumption are given as 6.50 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. 180 is shown the 600 B.H.P. Fiat twelve-cylinder, Vee engine, which also may be taken as a fairly typical example of the type of engine developed by the Allies during the latter phases of the war. In figs. 181 and 182 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. 183 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 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 cater's 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 36,000 R.P.M., and this, in itself, introduces very serious mechanical difficulties.

Thirdly, it is of course absolutely necessary to balance the pressure in the carburettor, float chamber, &c., and to deliver the fuel against the increased pressure; also, it is important to guard against leakages anywhere in the induction system or around the valve stems. This condition of affairs is of course not impossible of achievement, but it involves a good deal of added complication in the first instance, and is very difficult to maintain in service.

Fourthly, in addition to the added weight of the blower, its driving mechanism and attendant pipe-work, there is also the increased weight of radiator to be taken into account; again it is necessary to provide an additional radiator to cool the air after compression in the turbo-blower; and last, but not least, the increased stresses in the engine, both heat stresses and mechanical, must reduce greatly the reliability of the engine itself; for a normal aero-engine is not expected to develop its ground-level horse-power continuously, and although it may do so under favourable conditions on the test bed, its margin of safety is usually very small, and its reliability generally varies about inversely as the square of the power output.

While the earlier experiments with supercharging were for the most part carried out with mechanically-driven blowers, opinion in this country appears to be veering in favour of exhaust-driven blowers such as those shown in figs. 184, 185, and 186, 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 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 lbs. per square inch above the air pressure). Also the very serious problem of cogling the air after compression is still further aggravated by the addition, by 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 affected 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 possibly 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.

There is another system of supercharging, of which the author is in favour, in that it involves very little additional complication and is inherently automatic. This system was originally devised by Sir Dugald Clerk for large gas-engines. It consists in admitting above the piston, through ports in the cylinder wall, an additional charge of pure air, or air and cooled exhaust products, after the completion of the normal suction stroke, this supplementary charge being maintained as far as possible in a stratified layer over the piston. It may be utilized either as an inert diluent in order to lower the flame temperature, and so both increase the efficiency of the engine and reduce the heat stresses, or it may be used as an addition to the active working fluid, depending upon whether the initial charge is normal or rich. If rich, then there is sufficient fuel available to saturate the supplementary air; if normal, then the supplementary air acts merely as an inert diluent.

Using an ordinary standard carburettor, this gives automatic compensation for altitude, for if the mixture is adjusted to be about normal at ground level, then the supplementary air acts merely as a diluent, while, as the machine rises, and the mixture from the carburettor grows richer, more and more of the supplementary charge becomes active working fluid, so that until a height is attained at which the whole of it is consumed, the torque falls only as the square root of the density and not directly as the density of the atmosphere, while the air speed with a propeller of fixed pitch would remain nearly constant. By employing a cross-head type of piston and making use of the annular displacement of the piston, it becomes possible to add a supplementary charge corresponding to from 30 per cent to 35 per cent of the initial charge, and so to obtain a net increase in torque of about 35 per cent. The general arrangement of an experimental unit built on this principle is shown diagrammatically in fig. 187.

The efficiency of the whole system hinges on the possibility
or otherwise of working with a stratified charge and of obtaining a smooth transition from a stratified to a homogeneous charge.

Fig. 188 shows the calculated indicator card (a) when running normally without any supplementary charge and with an economical mixture strength, i.e. 16:1 air/petrol ratio, and (b) when running with the same mixture but admitting 35 per cent supplementary air as a diluent. Fig. 189 shows the indicator diagrams actually obtained from an experimental engine, under just such conditions, and indicates very fair agreement, the gross fuel consumption being exactly the same in both cases while the torque is increased by approximately 8 per cent. Figs. 190 and 191 are light spring indicator diagrams taken above and below the piston, which show the compression and introduction of the air supercharge. The absence of the "peak" when supercharging is due to the slower burning of the charge when a large proportion of diluent is present. Actually the total net gain in efficiency, after making allowance for the losses due to pumping in the supplementary air, was found to be
approximately 8 per cent when the whole of the supplementary air was used as a diluent, while the over-all efficiency was almost exactly the same when the maximum torque was obtained, whether with or without the admission of the supplementary air. In other words, applied to aircraft this method of supercharging by means of a
stratified charge, would give a net increase in fuel economy when the engine is operating at a normal torque of about 8 per cent, or alternatively it would give an increase of about 35 per cent in torque without gain or loss in economy. It would appear to have other advantages also, for, in the first place, it probably eliminates losses due to irregular distribution, since, if any one cylinder receives an over-rich mixture, the result is merely that more of the supplementary air is carburetted, and that particular cylinder develops a greater torque, so that, until a stage is reached when the whole of the supplementary air is used as active working fluid, irregularities in distribution are automatically compensated. Again, this system affords automatic compensation for mixture strength at different altitudes, for, if a normal type of carburettor is used, the torque

![Fig. 191 — Light Spring Diagram Supercharge Chamber](image)

will fall only as the petrol flow, or as the square root of the density, instead of directly as the density as in a normal engine, since, between wide limits, the torque depends not on the density of the surrounding air, but rather upon the flow of fuel.

Unless a variable pitch propeller is used, it is very doubtful whether it is desirable to maintain the torque much higher than that which this system provides.

The curves, fig. 192, give a summary of the performance of a single-cylinder experimental engine operating in the manner described above.

The objections to this system are (1) the comparatively small increase in torque available, when the differential area of the piston is utilized for supercharging, namely about 35 per cent. This, however, can readily be increased by increasing the density of the air supplied to the underside of the piston, in which case, since it
is only the supplementary and not the main air charge which
requires boosting, a comparatively small pump or blower will suffice;
(2) the additional weight incurred, which amounts to from 10 per
cent to 15 per cent.

Fig. 192.—Performance Curves, Experimental Supercharging Engine, bore 41 in., stroke 51 in.

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 the 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. The gain in power is not of the same magnitude because, for some at present inexplicable reason, the volumetric efficiency of an engine falls as the compression is increased. When the compression ratio is raised from 5:1 to 7:1, the indicated mean pressure was found (in experiments on the variable compression engine) to rise from 141 lb. per square inch to 157 lb. per square inch, a gain of only 12 per cent as compared with the 16.5 per cent gain in economy. Careful measurements of air consumption have proved that the whole of this large discrepancy is to be accounted for by reduced volumetric efficiency, the air consumption per hour for this particular engine at 1500 R.P.M. being 209.5 lb. at a compression ratio of 5:1, and 190.0 lb. per hour when the compression ratio is raised to 7:1. Though less than might be expected, this gain in torque is by no means to be despised, all the more so since it is obtained without any added complication. 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 may be dismissed at once as impracticable—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. 193, show the maximum indicated mean pressure obtainable with, in this case, a somewhat inferior aviation petrol detonating normally at a compression ratio of 4.85:1. The compression ratio was gradually raised and the throttle closed just sufficiently 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 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. Fig. 12, Chapter II, shows the results of experiments on the same variable compression engine, when the same fuel was used and the compression gradually raised, just sufficient cooled exhaust products being admitted at each stage to check detonation. It will be seen that by these means the ground-level power output with a compression ratio of 7:1 was equal to that
obtained at a ratio of 4·2 : 1, i.e. 125 lb. per square inch or 110 lb. per square inch brake mean pressure. Of the three, this is probably the most hopeful method.

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 withstand 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 very considerably. The use, however, of a fuel of lower flame temperature and higher latent heat, so that neither the temperature nor pressure is increased appreciably, such, for example, as alcohol, would appear very hopeful.
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 1500 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° to the horizontal, as shown in the photographs, figs. 194 and 195, which show a tank climbing out of a deep trench, while fig. 196 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

pistons and induction pipes, while the use of high tensile steel was entirely banned.
The standard 150-H.P. type is shown in the photos, figs. 197-200, and 215, and in the drawings, fig. 201. Six separate cylinders are employed, each of 5½ in. bore and 7½ 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 brught 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 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 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-

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 8 in. The general design is clearly shown in the general arrangement drawings, fig. 201, 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
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. 202).

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 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. 203 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. 204. The centre pump circulates the oil through
the various bearings, and the two scavenging pumps collect at the used oil to the external oil tank. Each of the scavenging 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. 206, which also illustrates the oil relief valve at the flywheel end of the main oil lead. In the original design, the scavenging 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 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. 206 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. 205.—Oil Pipe Arrangement

Fig. 206.—Auxiliary Drive Diagram

Fig. 207 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. 208.

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. 207.—Water Pump Drive Arrangement

Fig. 209 shows the general arrangement of the governor; it is a miniature of that used by Messrs. Mirrlees, Bickerton & Day, Ltd., for their large Diesel engines.

![Performance Curves of Water circulating Pump](image)

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 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 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 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° first in one direction and then in the
other. When tilted at this angle the engine was required to be run 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 be kept up firing regularly on all six cylinders without showing any smoke and without any oil leaking out of the base chamber (see fig. 229).
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 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. 210-213) give in full the average performance of these engines. Fig. 210 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. 214 shows the results obtained from a particularly good example after the conclusion of its 50-hour full-power list. Fig. 212 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 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. 213 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-

![Throttle Curve, 150 H.P. Engine](image)

Fig. 216.—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. 214 shows the power and consumption at full throttle and varying engine speeds.

Fig. 216 shows the fuel consumption at varying loads when running on the governor at speeds ranging between 1200 and 1300 R.P.M.

Fig. 217 shows in detail the cam formation and valve timing.
The following table taken from a sample test-sheet gives the heat distribution —
THE INTERNAL-COMBUSTION ENGINE

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.T.U.s per lb.
Air standard efficiency.—44·4 %.
Mechanical efficiency.—87 %.

Mean Results of Last Eight Hours of Test.

<p>| | |</p>
<table>
<thead>
<tr>
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<tbody>
<tr>
<td>Brake horse-power</td>
<td>162·9</td>
</tr>
<tr>
<td>Fuel (lb./b.h.p.-hour)</td>
<td>0·554</td>
</tr>
<tr>
<td>Brake thermal efficiency</td>
<td>24·7 %</td>
</tr>
<tr>
<td>Indicated horse-power</td>
<td>187·0</td>
</tr>
<tr>
<td>Indicated thermal efficiency</td>
<td>28·4 %</td>
</tr>
<tr>
<td>Relative efficiency (per cent of air standard)</td>
<td>64·0 %</td>
</tr>
<tr>
<td>Heat loss to jackets (B.T.U.s per hour)</td>
<td>418,000</td>
</tr>
<tr>
<td>Heat to indicated work</td>
<td>28·4 %</td>
</tr>
<tr>
<td>Heat to cooling water</td>
<td>24·9 %</td>
</tr>
<tr>
<td>Heat to exhaust, radiation, etc.</td>
<td>46·7 %</td>
</tr>
</tbody>
</table>

Fig. 218.—Interior of Mark V Two star Tank

Fig. 218 shows the installation of the engine in the Mark V two-star tank, while fig. 219 gives an exterior view. (In this model the larger 225 H.P. engine was used.)
The leading dimensions of these engines and general data are given in the following tables:

Number and arrangement of cylinders: 6, vertical, separate.

Bores: 5.625 in.

Stroke: 7.500 in.


Area of one piston: 24.85 sq. in.

Total piston area of engine: 119.10 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. (250 ft./sec.).

Indicated mean pressure: 110.6 lb. sq. in. (157.7 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. 0.75),
0.580 lb./b.h.p.hr. (18,600 B.T.U./lb).

Brake thermal efficiency: 23.6 %.

Indicated thermal efficiency: 26.8 %.

Irr 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.

Vol. II.
<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carburator body</td>
<td>168.8</td>
</tr>
<tr>
<td>Vertical induction pipe</td>
<td>169.5</td>
</tr>
<tr>
<td>Induction manifold</td>
<td>156.3</td>
</tr>
<tr>
<td>Inlet port</td>
<td>109.6</td>
</tr>
<tr>
<td>Inlet valve</td>
<td>168.3</td>
</tr>
<tr>
<td>Exhaust valve</td>
<td>148.25</td>
</tr>
<tr>
<td>Exhaust port</td>
<td>140.3</td>
</tr>
<tr>
<td>Exhaust branch pipes</td>
<td>140.3</td>
</tr>
<tr>
<td>Exhaust manifold</td>
<td>126.6</td>
</tr>
</tbody>
</table>

**Cross Sectional Area (sq. in.)—**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Choke tube</td>
<td>1.760</td>
</tr>
<tr>
<td>Carburator body</td>
<td>3.680</td>
</tr>
<tr>
<td>Vertical induction pipe</td>
<td>3.760</td>
</tr>
<tr>
<td>Induction manifold</td>
<td>3.976</td>
</tr>
<tr>
<td>Inlet port</td>
<td>5.672</td>
</tr>
<tr>
<td>Inlet valve</td>
<td>3.610</td>
</tr>
<tr>
<td>Exhaust valve</td>
<td>4.130</td>
</tr>
<tr>
<td>Exhaust port</td>
<td>4.430</td>
</tr>
<tr>
<td>Exhaust branch pipes</td>
<td>4.430</td>
</tr>
<tr>
<td>Exhaust manifold</td>
<td>4.910</td>
</tr>
</tbody>
</table>

**Diameter (in.)—**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Choke tube</td>
<td>1.496 in. (38 mm.)</td>
</tr>
<tr>
<td>Carburator body</td>
<td>2.165 in. (55 mm.)</td>
</tr>
<tr>
<td>Vertical induction pipe</td>
<td>2.1875 in.</td>
</tr>
<tr>
<td>Induction manifold</td>
<td>2.250 in.</td>
</tr>
<tr>
<td>Inlet port</td>
<td>2.6875 m.</td>
</tr>
<tr>
<td>Exhaust port</td>
<td>2.375 in.</td>
</tr>
<tr>
<td>Exhaust branch pipes</td>
<td>2.375 in.</td>
</tr>
<tr>
<td>Exhaust manifold</td>
<td>2.500 in.</td>
</tr>
</tbody>
</table>

**Weight of piston, complete with rings and gudgeon-pin, etc.**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight of piston, complete with</td>
<td>7.25 lb.</td>
</tr>
<tr>
<td>rings and gudgeon-pin, etc.</td>
<td></td>
</tr>
<tr>
<td>Weight per sq. in. piston area</td>
<td>0.292 lb.</td>
</tr>
<tr>
<td>Weight of connecting-rod, complete with</td>
<td>10.70 lb.</td>
</tr>
<tr>
<td>bearings, etc.</td>
<td></td>
</tr>
<tr>
<td>Total reciprocating weight per cylinder</td>
<td>10.82 lb.</td>
</tr>
<tr>
<td>Weight per sq. in. piston area</td>
<td>0.435 lb.</td>
</tr>
<tr>
<td>Length of connecting-rod</td>
<td>16.00 in.</td>
</tr>
<tr>
<td>Ratio connecting-rod/crank throw</td>
<td>4.27 : 1</td>
</tr>
<tr>
<td>Inertia pressure, top centre</td>
<td>82.2 lb./sq. in. piston area</td>
</tr>
<tr>
<td>Inertia pressure, bottom centre</td>
<td>51.0 lb./sq. in. piston area</td>
</tr>
<tr>
<td>Inertia pressure, mean</td>
<td>33.3 lb./sq. in. piston area</td>
</tr>
<tr>
<td>Weight of rotating mass of connecting-rod</td>
<td>7.13 lb.</td>
</tr>
<tr>
<td>Total centrifugal pressure</td>
<td>1094 lb.</td>
</tr>
<tr>
<td>Centrifugal pressure, lb. per sq. in. piston area</td>
<td>44.4 lb./sq. in.</td>
</tr>
<tr>
<td>Mean average fluid pressure, including compression</td>
<td>43.0 lb./sq. in.</td>
</tr>
<tr>
<td>Total loading from all sources, lb./sq. in. piston area</td>
<td>109.5 lb./sq. in.</td>
</tr>
</tbody>
</table>
HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

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.

Inlet Valve (one per cylinder)——
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) 6.8125 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) 1.720 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 bucket clearance, cold .................. 0.002 in.
Weight of valve, complete with spring, etc. .................. 1.6234 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) ................. 
Inlet valve closes (deg. on crank) ................. 
Material for valve .................. Valve leaves seat 29° early.
Exhaust Valve (one per cylinder)——
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.
THE INTERNAL-COMBUSTION ENGINE

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over-all length of valve</td>
<td>12.8125 in.</td>
</tr>
<tr>
<td>No. of springs per valve</td>
<td>Twq.</td>
</tr>
<tr>
<td>Free length of spring</td>
<td>(a) 5.875 in. (b) 8.8125 in.</td>
</tr>
<tr>
<td>Length of spring in position, no lift</td>
<td>(a) 4.000 in. (b) 3.8125 ip.</td>
</tr>
<tr>
<td>Mean diameter of coils</td>
<td>(a) 1.5703 in. (b) 1.120 in.</td>
</tr>
<tr>
<td>Gauge of wire</td>
<td>(a) No. 6 B.W.G., 0.203 in. diam. (b) No. 11 B.W.G., 0.120 in. diam.</td>
</tr>
<tr>
<td>Valve tappet clearance, cold</td>
<td>0.010 in.</td>
</tr>
<tr>
<td>Weight of valve, complete with spring, etc.</td>
<td>2.0934 lb. Two springs, 0.6718 lb. Spring cap, etc., 0.1406 lb.</td>
</tr>
<tr>
<td>Weight of spring, bare</td>
<td>0.61818 per pair.</td>
</tr>
<tr>
<td>Exhaust valve opens, degrees on crank</td>
<td>53° early.</td>
</tr>
<tr>
<td>Exhaust valve closes, degrees on crank</td>
<td>9° late.</td>
</tr>
<tr>
<td>Material for valves</td>
<td>3% nickel steel stamping, case-hardened.</td>
</tr>
</tbody>
</table>

Connecting-rod:
- 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. |

Crankshaft:
- 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 | Six |
HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

<table>
<thead>
<tr>
<th>Specification</th>
<th>Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, flywheel-end bearing</td>
<td>1.000 in.</td>
</tr>
<tr>
<td>Length, forward-end bearing</td>
<td>2.875 in.</td>
</tr>
<tr>
<td>Length, centre bearing</td>
<td>2.875 in.</td>
</tr>
<tr>
<td>Length, intermediate bearing</td>
<td>2.125 in.</td>
</tr>
<tr>
<td>Width of crank webs</td>
<td>3.500 in.</td>
</tr>
<tr>
<td>Thickness of crank webs</td>
<td>3.500 in.</td>
</tr>
<tr>
<td>Radius at ends of journal and crank-pins</td>
<td>0.250 in.</td>
</tr>
<tr>
<td>Diameter of drilled oil-paths in shaft</td>
<td>0.250 in.</td>
</tr>
<tr>
<td>Weight of complete shaft</td>
<td>182 lb., includes thrust race pinion, starting clutch, nuts, etc. (no balance weights).</td>
</tr>
<tr>
<td>Material</td>
<td>Mild-steel forging</td>
</tr>
</tbody>
</table>

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,
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 86 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 crankcase, 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 of its being drawn over the taper wedges, a steel wire ring was embodied in the pressing, as shown in Fig. 220. 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.
HIGH-SPEED HEAVY-DUTY ENGINES FOR TANKS

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. 221, 222, 223, 224, 225. 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 on the test-bed and subsequently in tanks to prove that the 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 63" and 74" 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. 225. 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

Fig. 221.—225 H.P. Engine, Carburettor Side
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 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
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 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. 225.—425 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.

The author desires to thank the proprietors of "The Automobile Engineer" and Messrs. J. Gardner & Sons for permission to use several of the illustrations in this volume.