AUTOMOBILE ENGINEERING

VOLUME V

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DIESEL ENGINES

BY

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ENGINE LAY-OUT, SUSPENSION AND MOUNTING

BY

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FRAMES, SPRINGS AND SUSPENSION

BY

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AUTOMOBILE ENGINEERING

A PRACTICAL AND AUTHORITATIVE WORK FOR AUTOMOBILE ENGINEERS, DESIGNERS, AND STUDENTS

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PREFACE

DIESEL engines continue to gain in popularity, especially among owners of heavy commercial and passenger carrying vehicles, and Mr. Judge has dealt with all the latest improvements in Section XVIII. The text has been thoroughly revised and many new illustrations have been included. Cylinder head and injection nozzle design have received special attention and particulars of successful types of engines are given.

Section XIX, dealing with Engine Lay-out, Suspension and Mounting, has now been illustrated, and the text has been revised where necessary in accordance with the latest practice. Section XX, dealing with Frames, Springs and Suspension has practically been rewritten. Reference is made to the latest mass production methods and there are six useful tables and three inset plates. With the present leaning of the motoring public towards comfort, problems of suspension and springing are extremely important and the revised section covers the various factors involved in design.

PREFACE
TO THE FIRST EDITION

IN this, the fifth volume of the series, we deal in Section XVIII with what is probably the most important modern development in Automobile Engineering. For many years the Diesel Heavy Oil Engine was employed exclusively for stationary purposes; then it passed into the marine field, and finally, by constant refining of the design, it became suitable for use in motor vehicles and even aircraft. Mr. Judge deals with the main problems of Diesel engine design at considerable length, and the section forms a complete introductory textbook on the subject.

Section XIX is devoted to the general consideration of engine lay-out, and the student will find much helpful suggestion in reducing the various elements of an engine to an harmonious whole. It should be remembered that when all the problems have been dealt with, the development of a neat and pleasing design, which at the same time leaves all the vital parts easily accessible, is far from being a simple matter, and much actual practice is necessary before proficiency is attained. The important matter of engine suspension has not been overlooked.

In Section XX the chassis frame and its spring suspension are treated together. The frame and its component fittings are fully described, together with the necessary information regarding determination of stresses, and adequate space is devoted to the study of laminated springs.

H. K. T.

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SECTION XVIII

DIESEL ENGINES

INTRODUCTION

One of the theoretical cycles of operation dealt with in the section on "Thermodynamics," is the Constant Pressure one, in which, after the charge is compressed in the cylinder, heat is added under constant pressure conditions. The pressure does not actually rise above the maximum compression pressure, but during the period of heat addition the pressure line remains parallel to the zero pressure line. This cycle, when interpreted into practice consists in first drawing a charge of pure air into the cylinder from the surrounding atmosphere, and then compressing it to a much higher pressure than in the case of the petrol engine; the usual compression pressures are from 450 to 550 lb. per sq. in. At, or near the end of the compression stroke the fuel is introduced into the combustion chamber of the cylinder under a much higher pressure than the compression pressure, so that it forms a finely-divided spray which penetrates the compressed air. Owing, however, to the high temperature attained (from 550° to 600° C.) during compression by the compressed charge of air, the fuel immediately ignites as it meets the compressed air, and combustion then proceeds for as long a period as the fuel is injected. After the latter has ceased the hot products of combustion expand and thus do useful work on the piston.

This cycle of operations is generally termed the Constant Pressure or Diesel cycle, but as we shall see later, the actual cycle employed by high-speed Diesel engines is not a true constant pressure one, but is, usually, something intermediate between the constant pressure and constant volume, or Otto, cycle.

Although the credit for the discovery of the heavy oil engine, using fuel injection into a compressed charge of air, the temperature of which is raised above the ignition temperature of the fuel, is usually given to the German inventor, Dr. Rudolf Diesel, it should not be overlooked that an Englishman, Herbert Ackroyd-Stuart, two years previously to the date of Diesel's patent (in 1892) took out patents for heavy oil engines using a separate combustion chamber connected by a narrow orifice to the cylinder. Air for combustion was compressed separately in the cylinder and the fuel for combustion was injected into the compressed air. In this case, however, the fuel was ignited by the heat of the combustion chamber walls, and not by the heat of compression. The primary object of the invention, as one reads the patent specification, was to get rid of the pre-ignition, or premature ignition, effects previously experienced. The compression pressures used were too low for ignition by the heat of compression alone, and the type of engine referred to in the Ackroyd-Stuart patent may therefore be classed as a *hot-bulb* engine, in which a subsidiary combustion chamber, or hot bulb, is kept sufficiently hot by the products of combustion of previous explosions to ignite the fuel when injected.

Diesel's patent of 1892 specifically relates to the introduction of fuel into highly-compressed air, in order to ignite the fuel, and thus to avoid all complications of the igniter and vaporizing chamber. He also aimed at a *high economy by cooling the cylinder gases after combustion through a large expansion ratio*; in other words, he sought to obtain a high thermal efficiency by using high compression ratios. It is this high compression ratio which, as we shall see later, is the secret of success of the *compression-ignition* engine, working upon the cycle outlined by Dr. Diesel.

In view of the fact that the idea is prevalent in some circles of the Diesel engine being associated only with high pressure *air blast injection* of the fuel, it should be pointed out that *airless* injection of the fuel was undoubtedly conceived by Diesel in his original patent ideas, so that it is not in accordance with the facts to associate direct, airless injection solely with Ackroyd-Stuart's name.

THE HIGH-SPEED DIESEL

Before proceeding with the theoretical aspects of this subject it is proposed to give a brief outline of some of the principal features of the high-speed Diesel engine, in order that the reader may be in a better position to appreciate the attractions of this type for motor vehicle purposes.

Owing to its high compression ratio, the Diesel engine gives a much higher thermal efficiency than any petrol engine. It is thus able to generate a greater amount of power for a given weight of fuel consumed, or, in other words, to show a much lower fuel consumption per b.h.p. developed.

The thermal efficiency* of the modern Diesel engine, as reckoned on the b.h.p., lies between about 33 and 39 per cent, whereas in the case of the petrol engine it ranges from about 22 to 27 per cent. The fuel consumptions of high-speed Diesel engines are generally of the order 0·38 to 0·45 lb. per b.h.p. hour. In the case of the petrol engine, the usual values are 0·5 to 0·6 lb. per b.h.p. hour.

The graph reproduced in Fig. 1 shows this advantage clearly. Prior to 1935 the fuels employed for

* Maximum values under most favourable conditions of fuel-air ratio, speed throttle-opening, etc.

commercial Diesel engines were only subject to a tax of 1d. per gallon, the prices of commercial petrol and Diesel oil being, respectively, 1s. and about 4½d. per gallon.

Under these conditions the average fuel costs of a well-designed motor vehicle Diesel engine worked out

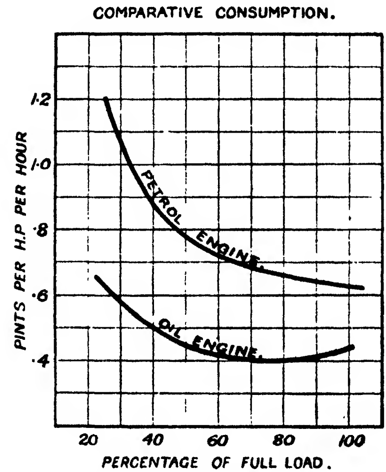


FIG. 1. RELATIVE FUEL CONSUMPTION OF DIESEL AND PETROL ENGINES

at about 0·18 to 0·24d. per b.h.p. per hour. The corresponding costs for a commercial vehicle petrol engine were 1·8d. at 20 per cent full load, falling to 0·9d. at full-load; the marked economy of the Diesel engine is clearly shown by these figures.

In 1935, however, a revenue tax of 8d. per gallon was imposed upon Diesel fuels, so that the differences in the fuel costs per B.H.P. hour were reduced appreciably. Nevertheless, under these conditions the Diesel engine has continued to exhibit its superiority, in the matter of low fuel costs, for these are still from

30 to 50 per cent lower than for the equivalent petrol engine.

Fig. 2* illustrates the running costs of two identical heavy commercial vehicles fitted with Diesel and petrol engines, respectively. The costs shown include other items besides that of fuel, namely, annual licence duty,

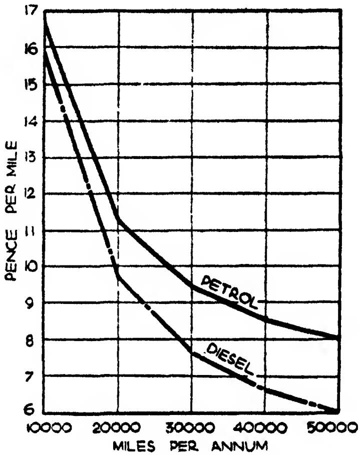


FIG. 2

depreciation, lubricant, and maintenance. The costs per gallon of the fuels are approximately equal so that the appreciably lower running costs of the Diesel engine may be regarded as being due largely to its higher operating efficiency.

Apart from its lower fuel costs for a given power output, the Diesel engine requires no carburettor—with its complicated mixture regulation devices—and no electrical ignition apparatus.

* "Diesel Influence on Road Transport Costs." P. M. Sanders and E. K. Wenlock. *Proc. Diesel Engine Users Assoc.*, 8th December, 1937.

In the case of the Diesel engine, the fuel pump and injection valve take the place of the carburettor, but with the important difference that there is no loss of volumetric efficiency as in the case of the carburettor, with its venturi tube and its various ports and passages. With the Diesel engine the fuel required per working stroke is accurately measured, or metered, out by the fuel pump, so that the air-fuel proportions remain practically constant at each fuel pump setting.

As the fuel ignites spontaneously on its entry into the combustion chamber, no subsidiary electric ignition apparatus is necessary, so that no sparking plugs, magneto, high-tension cables, etc., are required. It is true, however, that in certain designs of Diesel engine electric glow plugs are fitted for starting purposes. In these cases plugs having small spirals of high resistance wire are heated from a low-tension battery (4 or 6 volts) until the wires glow. As soon as the engine has commenced to "fire," the current to these glow plugs is switched off and the engine operates in the usual compression-ignition manner.

The use of a high flash-point fuel, and the absence of a carburettor and high-tension electrical ignition apparatus render the Diesel engine much safer from the point of view of fire risks. This is an important advantage for both aircraft and automobile purposes.

Another advantage of the high-speed Diesel engine is that it gives a more uniform torque over a wider range of speeds than the petrol engine, so that a motor vehicle fitted with the former type of engine will have a better top-gear performance; a certain amount of gear-changing will therefore be avoided.

In connection with the fuel consumption of the Diesel engine, it is interesting to observe that the low values previously mentioned are well maintained over a wider range of engine speeds than in the case of the petrol engine. There is no appreciable increase in fuel consumption between about one-half and full-load in the more recent designs of high-speed Diesel engines.

WEIGHT DISADVANTAGES

The earlier types of Diesel engine were designed on the lines of the steam engines with which they had to compete, viz. the vertical reciprocating types, with piston rods, crossheads, and connecting rods; as a result these engines were of massive construction. Further, they were run at low-engine speeds, viz. 150 to 250 r.p.m., so that when the weights of these engines were reckoned in connection with their power outputs, the figures obtained were excessive from the automobile engineer's point of view.

Thus, whereas most commercial vehicle type petrol engines weigh from 15 to 30 lb. per b.h.p., the slow-speed marine type of Diesel engine usually lies between 200 lb. and 300 lb. per b.h.p.; the former weight refers to two-cycle and the latter to four-cycle engines.

It was not until recent years that any serious attempts were made to reduce these heavy weights, except in the case of certain Diesel-generator sets used for fast marine craft, when the weight was brought down to about 60 lb. per b.h.p.

Realizing the advantages of the Diesel engine from the low fuel consumption viewpoint, its design was considered from the point of view of aircraft requirements, and it was shown that high power outputs could be obtained from much smaller engines by increasing the engine speed and by applying the experience in engineering materials and methods derived from automobile practice.

Engine speeds of 1,700 to 4,000 r.p.m., and even higher, are now employed, whilst mean effective pressures over 100 lb. per sq. in. are obtained in recent

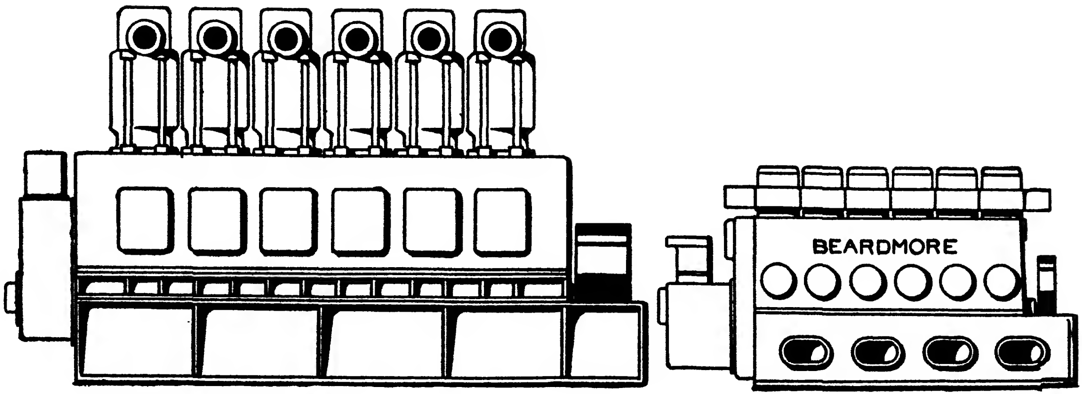


FIG. 3. SHOWING RELATIVE SIZES OF A LARGE SLOW-SPEED DIESEL ENGINE AND A HIGH SPEED ONE OF THE SAME POWER OUTPUT

engines, so that a considerable improvement has therefore resulted in the amount of power output from a given size of cylinder. The modern commercial Diesel engine's weight has been brought down to 7 to 15 lb. per b.h.p., and there are indications of a still further reduction in this figure.

Incidentally, it may be mentioned that the Packard, air-cooled, nine-cylinder radial engine, designed for aircraft purposes, had a weight of only 2¼ lb. per b.h.p., thus showing what can be done by careful attention to design.

The high-speed Diesel engine, unless it can be operated at much higher speeds than the petrol engine, will always be the heavier type however, for it employs appreciably higher maximum pressures in its cylinders. It is the maximum pressure that determines the dimensions of the parts to which it is applied, i.e. the cylinder, piston, connecting rod, crankshaft, and crankcase. Obviously, if the loads on these parts are of greater magnitude, then the dimensions must be increased in order to obtain the same stress values—or factors of safety.

The modern automobile Diesel engine possesses all the advantages previously referred to, in connection with low fuel costs, freedom from fire risks, more uniform torque, etc.

At present it is mainly the commercial vehicle type engine which has been developed, since the fuel costs per mile of this type are a big proportion of the total running costs. The experiences of numerous transport companies and of several municipal authorities running Diesel-engine buses, show very conclusively that big savings in fuel costs are regularly obtained, in comparison with similar vehicles fitted with petrol engines.

In regard to starting, the modern Diesel engine is no more difficult than the petrol engine; in two or three

British designs starting from cold in all weathers is actually easier than with many petrol engines.

A big advantage of the automobile Diesel engine is that as soon as it has been started, it requires no preliminary warming-up period of a few minutes as is the case with the petrol engine. It can be driven off, and will give practically full output almost immediately; there are no initial fuel condensation troubles as when petrol-air mixtures are used in ordinary petrol engines of the multi-cylinder type.

The excellent fuel economy of these automobile engines is well maintained under road service conditions. On full-load the fuel consumption is from 30 to 40 per cent lower per b.h.p. hour, or per mile of running of the petrol engine under similar conditions. As the load is reduced, this figure either remains about the same, or actually decreases, whereas in the case of the petrol engine the fuel consumption increases with load reduction. Under normal road service conditions the engine works at about one-third full power, and at this part-load the high-speed Diesel engine uses only about one-half the fuel of the petrol engine.

A further advantage claimed for the high-speed Diesel engine by many of its commercial vehicle owners is the smaller amount of maintenance attention that is required, and hence lower maintenance costs. The principal reason for this is that whereas with the petrol engine the carburettor, ignition unit and sparking plugs require periodical attention, in the case of the Diesel engine only the injection nozzles and fuel filters need this. The fuel pump, beyond lubrication, requires no attention over extremely long periods, if the fuel filters are kept clean.

The automobile Diesel engine in its present designs is not without its disadvantages, however; but as this type of engine is of fairly recent origin, and has had only a fraction of the experimental work devoted to it, these drawbacks are being, and will no doubt be, overcome in due course.

Apart from its somewhat greater weight per horse-power, certain designs of this engine have been associated with a heavy thumping noise, when running, known as *Diesel knock*. This has now been shown to be due to too-rapid ignition, and has been cured in subsequent designs by alterations in the shape of combustion chamber, and in the type and location of the fuel injection valve.

Another drawback, namely, that of *smoky and pungent exhausts*, has also been shown to result from unsuitable combustion methods, faulty injection timing, incorrect periods of injection and defective injectors. These troubles have definitely been overcome in the latest designs.

WEAR

In regard to the life of a modern high-speed Diesel engine designed for road transport purposes, the cylinder pressures are greater and of longer duration at about their maximum values than in the case of the petrol engine, so that the bearing surfaces must be of ample proportions to withstand these pressure effects. Hitherto, the life of the automobile Diesel engine, from the point of view of the wear of the cylinder and bearings, has been shorter than that of petrol engines of corresponding output, operating under the same conditions. Thus it has been found that the usual cast-iron cylinders required re-boring and new pistons fitted after about 40,000 miles of road work, whereas the petrol engine would give from 60,000 to 70,000 miles of useful running under similar working conditions. On the other hand, the inlet and exhaust valves appear to last appreciably longer, owing no doubt to the lower cylinder temperatures.

The fuel pump, which is the only other important component subject to wear, has been shown to have a useful life of at least 100,000 miles, before the wear of the plungers and barrels has any appreciable effect upon the metered amounts of fuel injected into the cylinder.

The latest high-speed Diesel engines are fitted with *separate cylinder liners* of the "wet" or "dry" types, so that when wear occurs it is only necessary to draw out the liner and to insert a new one; it is not necessary to fit new pistons in this case.

As these cylinder liners* are made from a high grade alloy cast iron containing nickel, chromium, etc., and they are cast by a centrifugal process ensuring maximum density of the metal and freedom from air bubbles and other defects, the wearing qualities are much better than those of ordinary cylinder irons; the liners, therefore, have much longer lives.

It should here be mentioned that the *wet* type of liner forms part of the cylinder, so that its outer surface is in contact with the cooling water, whereas the *dry* type fits into a cylinder barrel, and is therefore not in direct contact with the cooling water.

THE FOUR-CYCLE DIESEL

The four-cycle Diesel engine differs from the four-cycle petrol engine in respect of the method of ignition of the fuel-air explosive charge; in other respects, viz. the induction, compression and exhaust strokes, it bears a close resemblance to the latter. The important distinction, however, is that whereas in the case of the petrol-type engine it is a mixture of fuel and air which is drawn into the cylinder and afterwards compressed, in the Diesel engine it is a charge of air only that is drawn in and compressed.

* See page 125.

Further, as we have previously stated, combustion is caused by the spontaneous ignition of the fuel as it is injected, owing to the heat of the compressed air, whereas an electric spark is employed for the ignition in the case of the petrol engine.

Let us now consider the actual cycle of operations in the modern Diesel engine, by reference to Fig. 4. The four diagrams A, B, C, and D show respectively the suction, compression, injection (or ignition), and exhaust processes. During the suction stroke (Diagram A) the piston is moving downwards, and the inlet valve is kept open, mechanically, as in the case of the petrol engine. A charge of air is thus drawn into the cylinder; the exhaust valve is closed during this stroke. At the end of the downward stroke, or soon after, the inlet valve closes, and as the piston ascends on the next stroke (Diagram B), the air is compressed. At the end of the compression stroke the pressure may be anything between 450 and 550 lb. per sq. in., whilst the temperature, due to the heat of the compression is something of the order of 550° C. to 600° C.—a temperature well above the self-ignition temperature of the oil fuel used.

In the true constant pressure cycle, fuel is injected in the form of a fine spray, at the end of the compression stroke, i.e. when the piston is on its top dead centre. In order to inject this atomized fuel it is, of course, necessary to employ pressures "behind the fuel"—or injection pressures, considerably higher than the maximum compression pressure. Unless these high injection pressures are employed the fuel particles will not be projected far enough into the compressed air charge, and burning of the fuel will be sluggish; it may even be incomplete, so that burning still occurs when the exhaust valve opens—in this case the exhaust gases will be smoky.

In large slow-running Diesel installations it has been

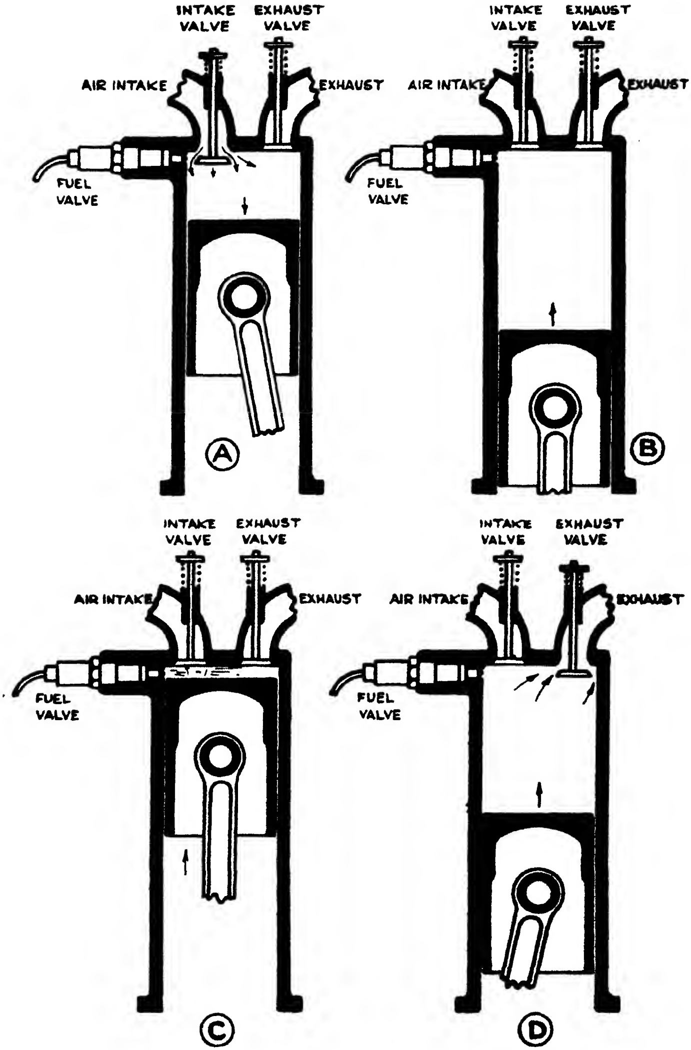


FIG. 4. THE DIESEL FOUR-CYCLE OPERATIONS

the practice, in the past, to use compressed air, at from 800 to 1,200 lb. per sq. in. pressure, for the purpose of injecting the fuel. Whilst this method gives most satisfactory results so far as atomization and penetration of the fuel spray are concerned, it necessitates the use of heavy and bulky air compressors; the latter are inadmissible for automobile purposes if only on account of their weight.

The system of fuel injection, known as the *airless* or *mechanical* one, is now universally employed on high-speed Diesel engines for aircraft and automobile purposes. In this case a fuel pump forces the fuel under an injection pressure of 1,500 to 3,000 lb. per sq. in. through the injection valve; the injection pressure need only be maintained for the relatively short injection period, viz. from $\frac{1}{12}$ to $\frac{1}{20}$ of a revolution, and on alternate revolutions. We shall return to this subject of fuel injection later.

Reverting to Fig. 4, the piston, in Diagram C, is shown at the top of the compression stroke, and fuel is being injected through the injection valve, or sprayer, shown on the left. Both inlet and exhaust valves are closed. The fuel, as it is injected, burns in the compressed air, the latter, having a certain amount of turbulent motion, assists in this combustion process.

In the true *constant pressure* cycle, injection commences at top dead centre and continues for a certain period after; this period corresponds, theoretically, to the complete utilization of all of the oxygen in the air charge.

The pressure in the cylinder is thereby maintained at a constant value, viz. that of the maximum compression pressure, so that the actual indicator diagram obtained in practice is of approximately the shape shown in Fig. 5. In this case injection commences at A and ends at B, so that the line AB is more or less straight and parallel with the pressure datum or zero line.

Expansion of the products of combustion within the cylinder then proceeds, the piston being forced downwards by the pressure of the gases until, when near the

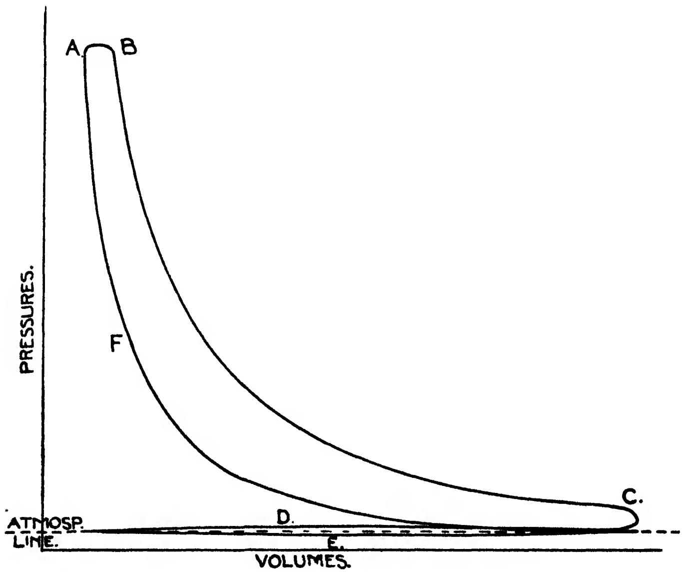


Fig. 5. Diesel Engine Indicator Diagram

lower end of its stroke, the pressure is released by the exhaust valve opening.

On the indicator diagram (Fig. 5) the expansion curve is shown by the line BC, the exhaust valve opening at C. Thereafter the pressure falls to practically atmospheric value along the line D; this corresponds to the exhaust stroke of the piston, as shown in Diagram D (Fig. 4). The line marked E, on Fig. 5, corresponds to the suction stroke shown in Diagram A

(Fig. 4). In practice the line D (Fig. 5) lies above, and the line E below, the atmospheric pressure line; the area between these lines represents work which has been done by the piston in overcoming the exhaust pressure and inlet suction resistances. This work must therefore be subtracted from that represented by the area of the main indicator diagram $ABCF$ in order to obtain the resultant work, or indicated horse-power.

THE DIESEL CYCLE IN PRACTICE

Hitherto, we have dealt with the constant pressure type of cycle, in which fuel injection commenced at top dead centre and continued for a definite period afterwards. In the majority of high-speed Diesel engines, however, the injection is timed to commence before the piston has reached its top centre, and it may either cease at top centre or at some period after this. The cycle of operations then no longer follows the constant pressure one shown in Fig. 5.

If high injection pressures are employed and the injection period is advanced so that combustion is complete when top centre is reached, it is possible to obtain the *constant volume cycle* conditions; in certain commercial engines using heavy oil fuel, these conditions are actually realized. *By varying the instant at which the injection of the fuel is commenced, it is possible to pass from constant volume to constant pressure conditions.*

In the former case the indicator diagram is characterized by its sharp pressure peak, following the compression line; the maximum cylinder pressures in this case (Fig. 6) are very much higher than when the injection commencement point is made later, i.e. nearer the top dead centre of the piston.

The disadvantage of the constant volume cycle, in this case, is that of the high cylinder pressures, for these necessitate the strengthening of all the parts under stresses due to the piston load; the engine therefore tends to become appreciably heavier than when the constant pressure cycle is approximated to.

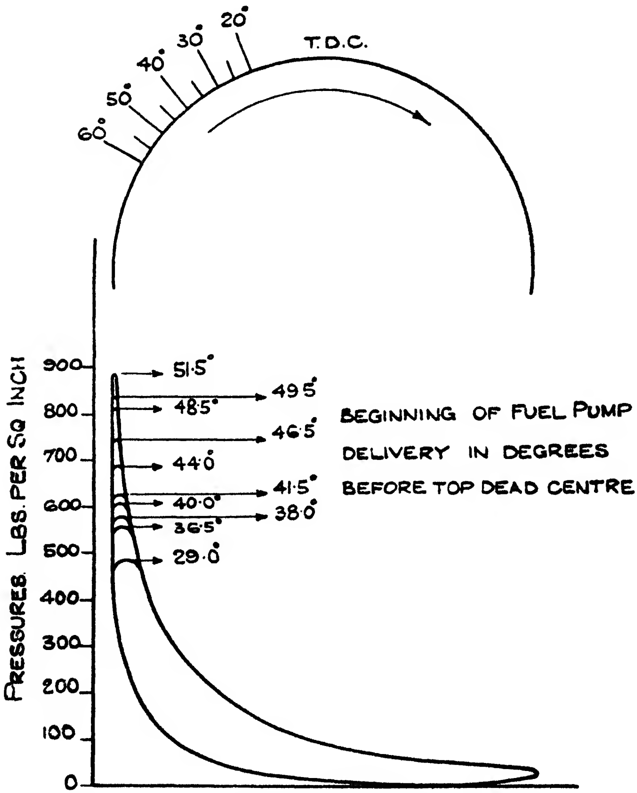


Fig. 6. Illustrating the Effect of Advancing the Injection Timing on Nature of Indicator Diagram

In practice, the point of commencement of the injection is so arranged that the resulting cycle is intermediate between the constant volume and the constant pressure ones; the maximum cylinder pressures are therefore kept within reasonable limits.

The majority of high-speed Diesel engines used for motor vehicle purposes in this country employ compression pressures up to 600 lb. per sq. in., and attain maximum cylinder pressures of 700 to 850 lb. per sq. in.; the injection pressures usually range from about 1,200 to 3,000 lb. per sq. in., but may be even higher.

In the case of the Packard aircraft Diesel engine, the fuel injection was arranged so as to give practically constant volume combustion; in this case the maximum cylinder pressures were over 1,200 lb. per sq. in. The injection timing, in this case, was such that fuel commenced to enter the combustion chamber some 45° before top centre; the compression ratio employed was 16 : 1.

The constant volume type of combustion in high-speed Diesel engines has the advantage of giving better fuel economy, for a greater expansion ratio is obtained. This will readily be apparent from the fact that the expansion commences at top centre, whereas with the constant pressure cycle it is from 20° to about 30° past this position.

There is a tendency to "Diesel knock," however, if the injection is advanced too much, in a somewhat similar manner to ignition knock in petrol engines; the design of the combustion chamber and certain other factors are also contributory items.

As will be shown later, it is necessary to arrange for a certain amount of fuel injection advance, in order to allow for the delay period in the combustion process; the ignition advance in the case of the petrol engine is arranged for a somewhat similar reason.

THE TWO-CYCLE DIESEL ENGINE

Just as in the case of the petrol engine, the two-cycle method is applicable to compression-ignition engines;

indeed, this cycle can be made more efficient in the latter types. The two-cycle operation is now so well-known to automobile engineers that it is hardly necessary to deal with the petrol engine sequence of operations.

In the two-cycle Diesel engine, after the compressed air charge has received and burnt its injected fuel. expansion occurs in the usual manner, until the piston, in its downward path, uncovers the exhaust port, or until an exhaust valve is opened by the camshaft mechanism. Shortly afterwards a charge of fresh air is introduced through another valve, or port system, and as the piston ascends this charge is compressed in exactly the same manner as the four-cycle case. At or before the end of the compression stroke fuel injection commences and then proceeds for a definite period.

The principle of the two-cycle Diesel engine is illustrated in Fig. 7 for the case of a three-port type. The three ports, shown in Diagram A, are as follows: *B* is the exhaust port, *C* the air inlet port, and *D* the transfer port.

When the piston is in the position shown in Diagram A, both the exhaust and transfer ports are uncovered by the piston, the former one having been uncovered by the upper edge of the piston a little before the latter port.

In this position the exhaust products resulting from the previous combustion will have escaped through *B*, towards the end of the expansion stroke. A charge of air previously drawn into the crankcase through the port *C* (Diagram A), when the piston was in the position shown in Diagram C, and compressed by the descending piston, when in the position shown in Diagram B, escapes into the cylinder when the transfer port *D* (Diagram A) is uncovered at the bottom of the firing stroke.

When the piston next ascends, as shown in Diagram

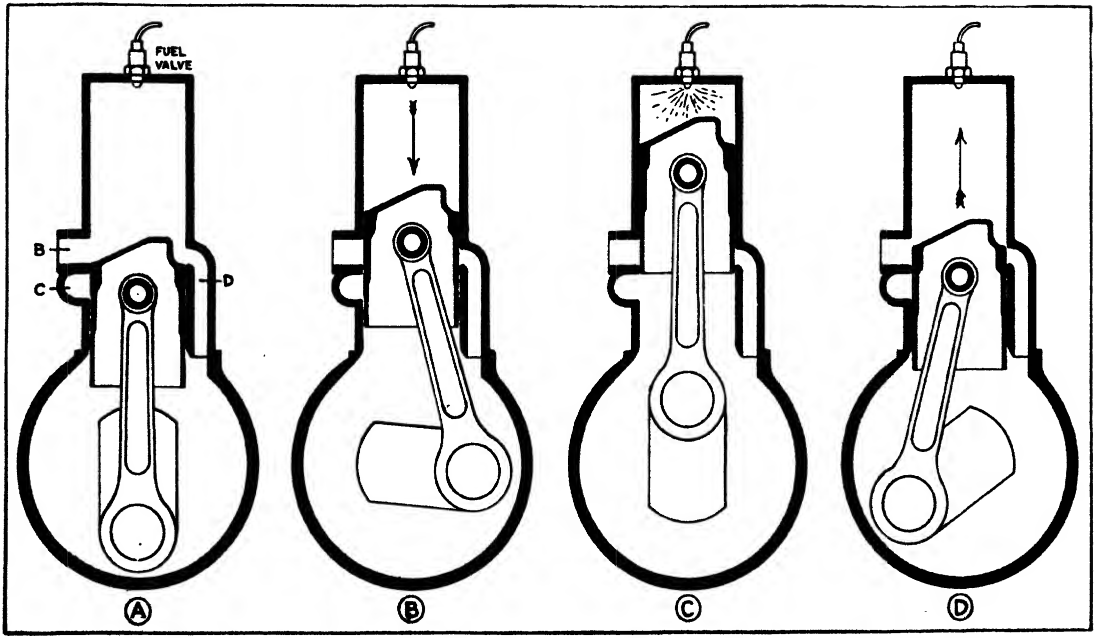


Fig. 7. Illustrating the Sequence of Operations for the Two-cycle Diesel Engine

D, this air charge is compressed progressively until, when the piston has very nearly reached its top dead centre position, fuel injection occurs, and the fuel burns in the highly-heated compressed air just as in the case of the four-cycle engine. Thereafter expansion occurs until the piston uncovers the exhaust port *B* (Diagram A) when the burnt gases escape. It should be noted that during the entry of the fresh air charge through the port *D*, some of this air escapes into the exhaust, thus tending to expel the residual burnt products and giving a partial scavenging action.

It will be observed that there is one important difference between the petrol and the two-cycle Diesel engine, namely, that in the former case the charge of air and petrol is introduced at, or near, the bottom of the piston's stroke, whereas air only is compressed in the latter case.

In the two- and three-port two-cycle petrol engines, as both the inlet and exhaust ports are open together for a certain period, part of the fresh charge escapes into the exhaust; with the two-cycle Diesel engine air only escapes. By arranging for a surplus charge of air to be admitted it is thus possible to scavenge the cylinder of its exhaust gases. The petrol engine, on the other hand, loses part of its fuel when the charge escapes; this accounts for the relatively poor fuel economy of the two-cycle petrol engines of normal design.

The two-cycle Diesel engine, therefore, has a better fuel consumption than the petrol engine on this account and quite independently of its higher compression ratio.

Although theoretically the two-cycle Diesel engine should give double the amount of power, at any given speed, of the four-cycle engine, in practice it is not possible to attain this, for it is impossible to fill the whole of the cylinder efficiently. The necessity for exhaust scavenging and the presence of exhaust and inlet ports are among the principal reasons for this lower efficiency. Another reason is that the full expansion stroke does not occur owing to the presence of the ports. Further, where scavenging air is introduced there is a loss of power of from 5 to 10 per cent in driving the scavenging pump.

The two-cycle Diesel engine is undoubtedly lighter per horse-power than the four-cycle type, and it is fairly widely employed in the larger slow running Diesel engines used for marine and stationary purposes.

Several high speed two-cycle engines have, however, been built for automobile and aircraft purposes. Typical examples of these are the Junkers (made in both the aircraft and automobile types), Petter, and G.M. engines.

FUTURE DEVELOPMENTS OF TWO-CYCLE ENGINES

In view of its important advantages in the matter of lightness, more even torque and, in most cases, greater simplicity, the two-cycle engine offers an attractive proposition as a high-speed Diesel type, provided its principal drawbacks can successfully be overcome.

The chief difficulty experienced with two-cycle engines is that of filling the cylinder with its air charge, in the very small interval of time the transfer, or air inlet, port is open; this period is considerably less than for the four-cycle engine; apart from the smaller port opening period, it must also be remembered that there is an air admission operation *twice as frequently* as in the four-cycle engine. Further, as the period of uncovering of the exhaust port is also much shorter than for the four-cycle type, it is not possible to get rid of the exhaust products so efficiently.

The only satisfactory method, at present, of overcoming these two difficulties appears to be that of air-scavenging. For this purpose a separate air compressor of the rotary type—on the lines of the supercharger pump—is often employed. The compressor delivers air under a pressure of from 4 to 8 lb. per sq. in. above atmospheric, so that an excess of air is provided for charging and scavenging over that obtainable from crankcase compression alone.

In some designs, the exhaust port has a mechanically-operated poppet valve, which is closed when the air port is opened. In other cases, there is an ordinary exhaust valve, or two such valves, in the cylinder head, so that no exhaust port is required in the cylinder barrel. The sleeve-valve engine appears to offer several advantages from the point of view of the two-cycle engine, for adequate port areas can be arranged around the sleeve, or sleeves.

In regard to the use of air compressors for two-cycle engines, it may be of interest to the automobile student to note that the power required to supply air to the cylinders at from 4 to 6 lb. per sq. in., and at a speed of 1,200 to 1,500 r.p.m., is from 7 to 10 per cent of the total b.h.p. developed. The weight of the compressor and its drive is usually about 0·2 to 0·3 lb. per b.h.p. It has been shown* that a well-designed two-cycle engine should have an output of 5 gross h.p. per lb. per sq. in. of air per minute, allowing an excess of 50 per cent for combustion and 10 per cent for scavenging and wastage. These results correspond to a brake m.e.p. of about 97 lb. per sq. in.

EFFICIENCY OF THE FOUR-CYCLE DIESEL

We have already stated that, owing to its high compression, the Diesel engine has a much higher thermal

* "High Speed Two-stroke Oil Engines." P. E. Biggar, *The Automobile Engineer*, April, 1930.

efficiency than the petrol engine; the effect of increasing the compression ratio upon the theoretical efficiency has also been demonstrated in the section on "Thermodynamics." In practice the efficiency will depend upon the amount of fuel injected into the compressed air charge; it will be lower for small fuel amounts and higher for larger quantities, within certain limits. *The best efficiencies* are obtained in modern engines *when the quantity of fuel injected is sufficient to combust from* 75 *to* 80 *per cent of the air present.* If a greater quantity of fuel is injected the efficiency will fall, although it is a fact that more power will be obtained.

There is thus a definite parallel between the compression ignition and the petrol engine, for in the latter case the highest thermal efficiencies are obtained from weaker mixtures, whilst the greatest power is given by richer ones.

In the case of high-speed Diesel engines, indicated thermal efficiencies up to 53 per cent have been obtained by Ricardo, from a sleeve-valve engine of 15 : 1 compression ratio, running at 1,300 r.p.m. In practice the brake thermal efficiencies obtained are from 33 to 39 per cent.

In connection with the calculation of thermal efficiencies, a useful formula, based upon experimental results obtained from actual engines of the direct-injection poppet-valve type is as follows—

$$\text{Efficiency} = 1 - \left(\frac{1}{r}\right)^{0.236}$$

It will be observed that this expression is similar in form to the one for the air standard efficiency, but with a different index.

In regard to the effect upon the efficiency *as the combustion conditions are varied* from constant pressure to constant volume ones, Fig. 8 shows the theoretical indicator diagrams for a compression-ignition engine in which the same amount of fuel is burnt at different maximum pressures, by suitably arranging the point of commencement and the period of the fuel injection.

The lower plain diagram corresponds to constant pressure combustion at a maximum pressure of 450 lb.

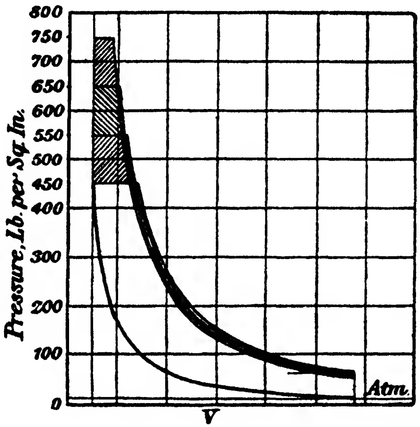


Fig. 8. Showing Effects of Different Compression Pressures in Diesel Engines (Ricardo)

per sq. in., whilst the uppermost diagram is for an intermediate type of combustion approaching that of the Otto cycle, with a maximum pressure of 750 lb. per sq. in. The gains in efficiency as the maximum is increased are shown by the shaded areas.

From 450 to 550 lb. per sq. in., there is a gain of efficiency of about $4\frac{1}{2}$ per cent; by allowing a further rise to 650 lb. per sq. in., we gain another 3 per cent, whilst a still further increase to 750 lb. per sq. in. gives another 2 per cent gain. It will thus be obvious that there is a limit to the increase of the maximum pressure where the small efficiency gain is offset by practical considerations of design of the stress-bearing members.

DIESEL ENGINE OUTPUTS

Owing chiefly to its greater expansion ratio, and in part to the advance of its injection, the high-speed Diesel engine does not attain such high mean effective pressures as the petrol engine at similar piston speeds, so that the power output requires rather larger engine dimensions in the former case. In the case of many commercial high-speed Diesel engines the brake m.e.p. is of the order 80 to 110 lb. per sq. in., whereas modern high-speed petrol engines give brake m.e.p.'s of 130 lb. and above per sq. in. at speeds of 2,000 r.p.m., corresponding to piston speeds of 2,300 ft. per min. The mean pressure values given for the Diesel engines correspond to piston speeds of 1,500 to 1,800 ft. per min.

At higher speeds the brake m.e.p. falls off progressively. Higher mean pressures than those given for ordinary commercial Diesel engines have actually been attained in the laboratory. Thus, in the case of an experimental single cylinder engine tested at the R.A.E., Farnborough, the highest brake m.e.p. measured was 122 lb. per sq. in. This value corresponded to the relatively low speed of 800 r.p.m., and to a maximum cylinder pressure of 800 lb. per sq. in. On a power-to-weight basis this result is therefore not quite so favourable as it at first appears.

An interesting comparison, taken under somewhat similar conditions, has been made of the performances of a petrol and a Diesel engine.

The petrol engine had a compression ratio of 6 : 1 and was run at 1,500 r.p.m. The time from the passage of the spark to the attainment of maximum pressure was 40° of crank-angle, i.e. the ignition occurred 28° before and ceased at 12° after top centre. The Diesel engine had a compression ratio of 13·5 : 1. In this case the time from the start of injection to that of the maximum pressure (800 lb. per sq. in.) was 28° before and 12° after dead centre.

The brake m.e.p.'s of the petrol and Diesel engines were 139 and 112 lb. per sq. in, respectively.

A similar comparison made on a single-cylinder engine at the R.A.E., Farnborough, showed that at 1,000 r.p.m., in each case the brake m.e.p.'s for petrol engine and Diesel cycles were, respectively, 134 and 121 lb. per sq. in.

At present the petrol engine scores from the point of view of power for weight, since it can run at much higher speeds, and can maintain a higher m.e.p. at all speeds; moreover, it has to withstand lower cylinder pressures and so can be made lighter.

FUEL CONSUMPTIONS

We have already referred to the low fuel consumption of the Diesel engine, and now propose to give some further information on this point. The lowest recorded fuel consumptions, according to Ricardo, are 0·35 lb. per b.h.p.-hour for a 1,000 h.p. per cylinder engine running at 80 r.p.m., and 0·347 for a small engine of only 5½ in. bore running at 1,500 r.p.m.

Fig. 21 shows the fuel consumption graph for a single cylinder Diesel engine running at 1,300 r.p.m. The bore and stroke were 5½ in. and 7 in., respectively, and the maximum output 35 b.h.p. The fuel consumptions are plotted against power outputs, so as to show how the minimum values, viz. 0·355 lb. per b.h.p.-hour, are maintained over a relatively wide range of outputs. Normally, this low fuel consumption would occur from *about one-half to nearly full-load.*

In the case of the petrol engine, the fuel consumption at reduced loads falls off very appreciably, so that in this connection the Diesel engine scores from the motor vehicle user's point of view.

For practical purposes it is usual to take the average fuel consumptions, under full load conditions, of the petrol and high-speed Diesel engine as 0·55–0·60 and 0·35–0·40 lb. per b.h.p.-hour, respectively. The usual commercial grade of Diesel oil employed in this country will give the average values mentioned; with inferior grades higher consumptions will be experienced.

THE COMBUSTION PROCESS

The combustion of the liquid fuel droplets forced into the hot compressed air may be regarded as an extreme case of oxidization of the oil drops, in which the carbon and hydrogen of the latter are oxidized at a very rapid rate.

It can readily be shown that this rate of oxidization depends chiefly upon the following factors, namely: (1) The nature of the fuel. (2) The degree of atomization, or breaking up into small particles. (3) The rate of the relative movement between the fuel particles and the oxygen "particles" of the compressed air, and (4) the pressure and temperature of the compressed air. Speaking generally, the smaller the fuel particles and the more rapid their movement in the compressed air the quicker will this combustion occur. Further, the higher the temperature of the compressed air the greater will be the rate of oxidization.

In the Diesel engine, as we have seen, the temperature of the air charge at the end of the compression stroke is arranged to be above the self-ignition temperature of the fuel in air, and we then obtain this rapid combustion process that is in reality a very quick oxidization of the fuel particles.

The actual temperature of the air charge in the cylinder, necessary for the self-ignition of Diesel engine fuels, is of the order of 500° to 600° C.; this is the maximum temperature attained at the end of the compression stroke. If the fuel is injected—as is usually the practice—some 10° to 20° before the end of the compression stroke, the temperature will be somewhat lower than this.

The temperature, corresponding to any compression ratio, at the end of the compression stroke can readily be calculated from the pressure-volume relation that has been shown, experimentally, to exist in the case of high-speed Diesel engines,* viz.—

$$P\,V^{1\cdot32} = \text{constant}.$$

The corresponding formula for the temperature is obtained by substituting the value of P from the well-known formula combining the laws of Charles and Boyle, viz.—

$$\frac{P\,V}{T} = \text{constant}$$

where T is the absolute temperature.

We then arrive at the temperature-volume relation—

$$T_c = T_s \times r^{0\cdot32}$$

where T_c and T_s are the compression and suction temperatures (expressed on the absolute scale), respectively, and r is the compression ratio.

The index 0·32 is obtained from the general expression r^{n-1} where $n = 1{\cdot}32$.

Reverting to the subject of the combustion process in the Diesel engine, let us next consider what occurs when fuel is injected into the compressed air charge, in the cylinder, remembering that the air is very hot and above the self-ignition temperature of the fuel.

When a small drop of fuel, from the fuel atomizer, enters this air, the surface layer of the fuel at once commences to evaporate, so that an envelope of vapour

* D. R. Pye. *The Internal Combustion Engine*, p. 129.

is formed; there will be a certain small lowering of temperature of the air around due to the heat absorbed in the evaporation process.

The fuel vapour next ignites in the air, for the latter is well above the self-ignition temperature. This process of evaporation and burning continues from the surface inwards until all of the fuel is burnt.

The rate of burning will depend upon the initial size and number of the oil particles and upon their relative movement in the compressed air, as we have already mentioned.

COMBUSTION PROCESS FROM INDICATOR DIAGRAMS

The high-speed electric indicator, developed by Messrs. Dobbie McInnes and Clyde, Glasgow, has enabled the combustion process to be analysed, for it is possible to obtain pressure diagrams on a time basis, the diagrams being to quite a large scale.

An examination of a typical pressure-time (or crank-angle base) diagram obtained from a high-speed Diesel engine shows that the pressure-rise portion of the diagram, which commences at the point of fuel injection, can readily be divided into three separate portions. These correspond, respectively, with three distinct phases of the fuel injection process, as follows—

(1) **The Delay Period.** This is indicated by the portion marked "1" of the diagram in Fig. 9. During this period, which commences at the moment injection begins, there is an initial lag or delay, but no actual combustion occurs. It is possible, according to Ricardo, that ignition first proceeds from some nuclear point, as soon as only a minute portion of the injected fuel has vaporized and mixed with enough oxygen to form a combustible mixture; the flame spreads from this centre in a somewhat similar manner to that occurring

in the case of the petrol-air mixture ignited by a spark from the sparking plug. As in the latter case, the delay period appears to be constant in time and to depend upon the nature of the fuel, the temperature and pressure of the highly compressed air and upon the size of the fuel particles. i.e. the degree of atomization.

(2) **The Rapid Combustion Period.** This is shown at "2" in Fig. 9. This period corresponds to the mechan-

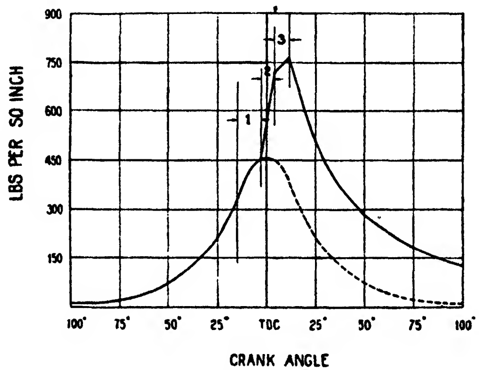


FIG. 9. ILLUSTRATING THE THREE-PHASES OF COMBUSTION IN DIESEL ENGINES

ical spread of the flame to the main body of the combustion chamber. As in the petrol engine, the rate of speed of the flame, and therefore the rate of pressure rise, is dependent on turbulence and is constant in terms of crank-angle rather than of time; incidentally, this phase can be altered by varying the turbulence. There is an important difference, however, from the state of affairs occurring in the petrol engine, for in the latter case the whole of the fuel for combustion is already present in the cylinder head, whereas in the Diesel engine only a portion of the fuel is present during the first and second (or present) phase. For this reason the pressure, although its rate of rise is of the same order as that of a petrol engine, with similar conditions of turbulence, does not attain anything like the same maximum which it would do if sufficient fuel were present to combine with the whole of the oxygen present.

The actual value of the pressure reached at the end of the second phase, now under consideration, depends upon the extent of the delay period, i.e. upon the engine speed, upon the rate of injection of the fuel, and finally upon both the temperature and pressure of the highly compressed air.

(3) **The Final Phase.** This consists of a period, as shown by the portion "3" of the pressure rise line in Fig. 9, during which the remaining fuel burns as it enters the combustion chamber. Once the flame has spread throughout the latter, the increase in pressure and temperature is so great that the rate of combustion is accelerated so that the fuel actually burns as it leaves the injector nozzle, thus bringing about a further rise of pressure, or in the case of constant pressure cycles, of a maintenance of constant pressure conditions.

During this final phase the combustion is under direct mechanical control, and it can be varied by altering the rate of injection; it can be stopped by shutting off the fuel supply.

In this final phase the limiting factor is the remaining oxygen of the air charge, for as the oxygen becomes used up there will obviously be less for the fuel to find and to combust with. One must therefore anticipate a slowing up of the rate of combustion during the latter phase; this is borne out in practice by the greater obliquity of the third portion of the pressure line of the indicator diagram (Fig. 9).

The latter diagram has intentionally been reproduced in linear phase form, in order to explain the three separate combustion conditions. These conditions, in practice, are not so abruptly defined, the phases passing more or less gradually from one to another.

The actual indicator diagram is therefore a continuous curve, as shown in Fig. 10 ; here, it will be observed, the three phases are still well defined, but are smoothed

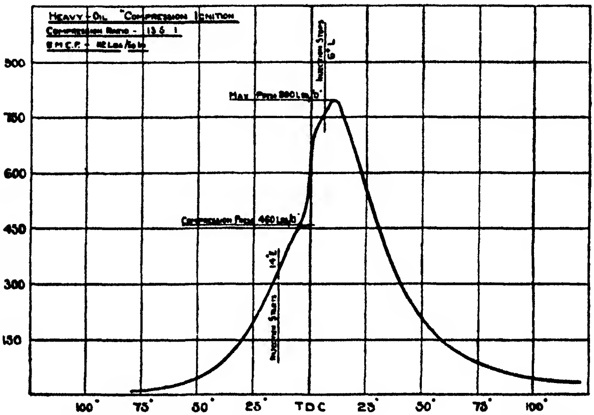


FIG. 10. AN ACTUAL INDICATOR DIAGRAM FROM A DIESEL ENGINE, RUNNING AT 1,500 R.P.M.

out to a certain extent. The diagram in question was obtained, by Ricardo, from a single cylinder engine of $5\frac{1}{2}$ in. bore and 7 in. stroke, running at 1,500 r.p.m. and giving a brake m.e.p. of 112 lb. per sq. in. The compression ratio was 13·5 : 1, the maximum compression pressure, 460 lb. per sq. in., and maximum ignition pressure 800 lb. per sq. in.

LIMITING COMBUSTION CONDITIONS

For a considerable period it was believed that the Diesel type of engine could only be operated at comparatively low speeds of the order of 200 to 300 r.p.m., due to the limiting combustion conditions, i.e. to the slowness of the process.

The experiments carried out by the Royal Aircraft establishment, Farnborough,* during 1925–7, showed, however, that there were no combustion limiting circumstances for speeds up to 1,200 r.p.m. More recently engines have been run at speeds of over 4,000 r.p.m. without experiencing any signs of a limit by the fuel-injection process of ignition.

Experimental work on the combustion side of high-speed Diesel engines, using electrical indicators and other apparatus, has shown that so far as is known at present, the Diesel engine should be able to run, not only as fast as, but even considerably faster than, the petrol engine ; the mechanical limitations are at present the only ones preventing such high speeds from realism.

FUEL INJECTION METHODS

In the high-speed Diesel engine the fuel injection system takes the place of the carburation, induction, and electrical ignition systems of the petrol engine. There is, however, one important difference between the two methods of fuel-air introduction, for whereas in the latter case the whole of the induction and compression strokes are utilized for mixing, intimately, the air and fuel, in the case of the Diesel engine the period of fuel injection and air mixing is only some 15° to 30° of crank angle—as against about 360° for the petrol engine.

Special methods have, therefore, to be adopted to ensure the intimate mixing of the fuel with the air during the considerably shorter period available. The latter is only about $\frac{1}{12}$ to $\frac{1}{24}$ that of the petrol engine, and amounts to a period of from ·002 to ·001 sec. only, in the case of an engine running at 2,400 r.p.m.

This problem of satisfactory atomization of the fuel

* "High Speed Compression-ignition Engine Research." H. B. Taylor, Proc. I.A.E., November, 1927.

and its thorough dispersal throughout the oxygen molecules of the air charge, in these extremely short periods of time, has been solved by several different methods in modern high-speed Diesel engines. If these methods be analysed carefully, it will always be found that the designer has aimed primarily at the rapid pulverization of the fuel and the intimate mixing of the fuel and air.

In some cases compressed air has been used to force the fuel through the injection nozzle and to atomize it, the minute particles of fuel being projected with sufficient velocity, and in various directions, into the compressed air to penetrate the whole mass of it rapidly. In most cases the shapes of the combustion chambers have been designed, in relation to the fuel nozzle positions, to facilitate this end.

In other cases smaller injection pressures are used and the combustion chamber is designed to give sufficient turbulence to the air, to bring practically all of its oxygen into the vicinity of the fuel stream from the nozzle. In yet another method a separate combustion chamber, known as an ante-chamber, or precombustion chamber, is employed, the fuel being injected into this chamber in which part of the air charge is in movement near the end of the compression stroke.

Speaking broadly, there are two chief systems of fuel injection, namely, the *Compressed Air* and the *Airless*, or *Mechanical* ones. The former has been used almost exclusively for large slower speed marine and stationary Diesel engines, although it has actually been employed on one or two high-speed types. Practically all of the successful automobile and aircraft Diesel engines of to-day use the airless injection system for reasons which will be mentioned later.

Compressed Air Injection. In this system, which is still widely employed on the larger slower speed engines, the fuel is forced through a special design of nozzle by means of compressed air at a much higher pressure than the maximum cylinder pressure; the compressions used for this purpose vary from 900 to 1,100 lb. per sq. in. When used with suitable nozzles a fine atomization of the fuel, combined with good penetration effect, is obtained. The fuel particles are thus rapidly mixed with the air in the combustion chamber, a smaller fuel injection advance being necessary than is the case with most of the airless injection systems.

It is possible to obtain combustion under practically constant pressure conditions and, therefore, to keep the maximum pressures comparatively low; these pressures need only be slightly higher than the maximum compression pressures.

With air injection systems it is necessary to employ mechanically-operated fuel valves to give the correct point of commencement and the correct period of duration of the fuel injection. It is possible in this way, and also by varying the pressure of the injection air, to control, effectively, the rate of admission of the fuel.

From the point of view of automobile engine requirements, however, the air injection method is ruled out by weight and bulk considerations, for the air compressors necessary to give such high pressures are both heavy and cumbersome, as a general rule. It is not proposed, therefore, further to describe this system.

Airless, or Mechanical Injection. This method, which was first successfully applied by Sir F. McKechnie, of Vickers, Ltd., in 1910, and subsequently developed by the Admiralty for submarine engines, employs a mechanical fuel pump to meter out the correct quantity of fuel required per working stroke and to inject it under high pressure into the cylinder. It is now almost universal for the lighter high-speed Diesel engines.

There are many variations of this method of injection, as we have already noted, but the different systems may broadly be grouped under three headings as follows: viz. (1) Direct-injection with plain combustion chambers. (2) Pre-combustion chambers into which the fuel is injected, and (3) auxiliary air chambers where the air is given its turbulent motion during the fuel injection period.

Each of these general systems has its own particular advantages in the matter of combustion efficiency,

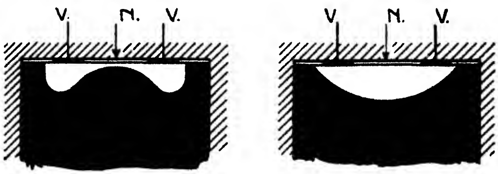


FIG. 11. TYPICAL DIRECT INJECTION CYLINDER HEADS

V denotes the inlet and exhaust valves and *N* the fuel injection nozzle position

power output, or ease of starting from the cold. It is now proposed to deal with the characteristics of each system, leaving out, for the time being, the question of the fuel pump and injection valve design.

The Direct Injection System. In this case the cylinder head is usually made of plain form, generally with a flat top, although it is the practice to make the shape of the top of the piston conform with the requirements of the fuel injection nozzle. The latter is of special design, and the spray is given the proper direction to penetrate all the air in the combustion space.

Fig. 11 illustrates, diagrammatically, two typical direct injection systems, in which the cylinder heads are flat, the fuel nozzle is placed centrally, and the piston crowns are shaped so as to give good penetration of the fuel particles injected. The pistons are shown in black in these diagrams.

Fig. 12 shows the direct injection system, illustrated in the left diagram in Fig. 11, but in more detail. In this case the piston is shown at *a*, the inlet and exhaust valves at *d* and *f*, respectively. The piston crown is of special shape, the edges *b* coming very close to the cylinder head. The fuel injection nozzle is shown at *e*. This nozzle belongs to the multiple hole type, in which the fuel is sprayed obliquely, as shown at *c*, *c*. The fuel therefore follows the direction of the contour of the

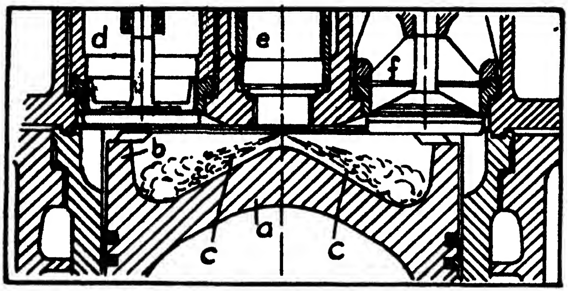


FIG. 12. SHOWING SECTIONAL VIEW OF A CYLINDER HEAD FOR DIRECT INJECTION ENGINE

piston, and as the compressed air is, itself, in motion, the admixture of the fuel and air is efficiently carried out.

In this case it is the fuel nozzle that is designed to give the desired penetration of the air charge; the turbulence of the air in the combustion chamber is more or less accidental.

The direct injection system has the advantage of a smaller loss of heat during the compression in the combustion chamber, owing to the absence of excessive turbulence. Moreover, the delay period of combustion is shorter and the general running smoother than in many other types.

Engines employing this system usually start up from the cold more readily, due to the smaller loss of compression heat, and they can run on a wider range of fuels.

On the other hand, the direct injection system requires higher fuel injection pressures in order to give the proper amount of atomization and penetration; the fuel nozzles are also of more complicated design. The fuel orifices are very small and are therefore more liable to partial or complete stoppage, and to the effects of erosion.

Generally speaking, direct injection system engines are not capable of such high speeds as some of the other systems described hereafter.

Pre-combustion Chamber Systems. In these, there is an auxiliary or ante-chamber which communicates with the cylinder head proper through a restricted passage or neck. This system was originated by Ackroyd-Stuart as long ago as 1890. It has since been used extensively by various engine builders, one of the best known designs being the Bosch-Acro type, described later.

In this case, when the piston moves up the cylinder on its compression stroke air is forced through the narrow neck into the smaller chamber and circulates therein. At the correct moment fuel, in a partially atomized condition, is injected into the pre-combustion chamber and immediately starts to burn. In burning, the ignited fuel, together with the unburnt portion, is projected through the narrow neck into the cylinder head proper. In some cases there is a perforated plate between the pre-combustion chamber and the cylinder proper; in other cases, a plug or wall with a hole of relatively small dimensions.

The admixture of the fuel with the air in the pre-combustion chamber and cylinder head is therefore effected by the projection of the earlier ignited fuel

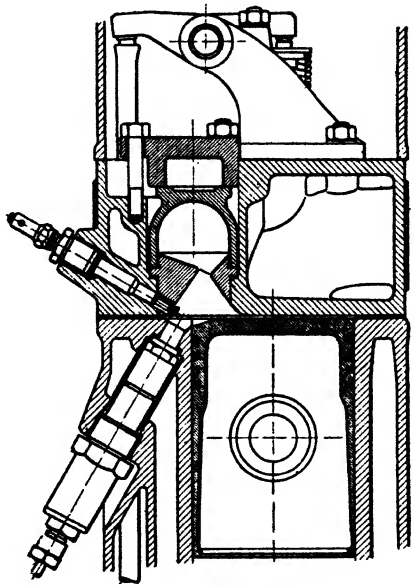


Fig. 13. The External Type Bosch-Acro Pre-combustion Chamber

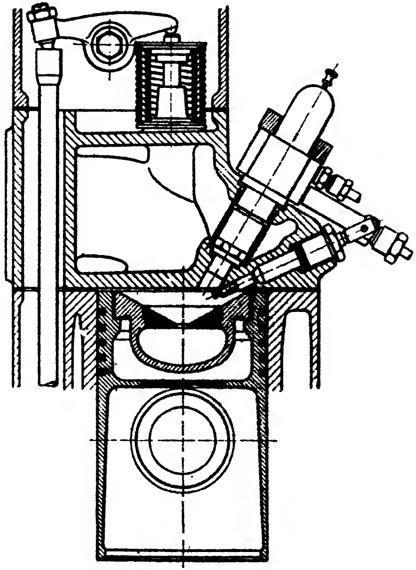


Fig. 14. The Internal Type Bosch-Acro Pre-combustion Chamber

particles. This method has the advantage of necessitating only a simple form of injector nozzle, having a relatively large fuel orifice that is not liable to choking up.

Further, a low injection pressure only is required, and there is no necessity for accurately locating the direction of the fuel spray.

The principal disadvantages of the pre-combustion chamber system are that (1) There is a loss of combustion efficiency due to the relatively large absorption of heat during the passage of the burning particles through the neck. (2) A higher compression pressure is required in order to obtain a sufficiently high temperature at the moment of fuel injection; this higher compression necessitates a somewhat heavier engine. (3) Starting is more difficult from the cold, so that special heating plugs or cartridges are necessary. Before proceeding to the third system we shall describe one or two typical pre-combustion chamber arrangements.

The Bosch-Acro Head. The Acro type of pre-combustion chamber, now licensed by Messrs. Robert Bosch, Ltd., is perhaps the most widely used of any.

It is made in two principal forms, viz. the *external*, or cylinder head, and the *internal*, or piston types. Fig. 13 shows a typical arrangement of the former, and Fig. 14 one of the latter type.

In the former case it will be seen that there is an ante-chamber, situated above the cylinder head and communicating with the latter by means of a funnel-shaped passage. The fuel injection nozzle is shown to the left of the piston, the fuel being sprayed upwards towards and through the neck of the communicating passage. The smaller plug shown above the fuel nozzle is an electric heating plug used for starting purposes.

The sectional view given in Fig. 15 shows in more detail this design of Acro head applied to the earlier commercial vehicle Diesel engines of the Associated

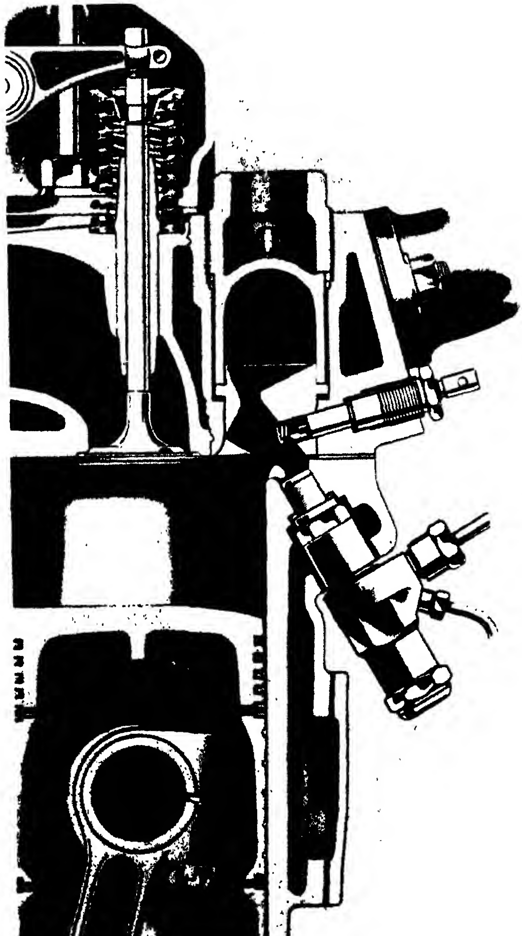


FIG. 15. THE BOSCH-ACRO PRE-COMBUSTION CHAMBER USED ON THE EARLIER A.E.C. DIESEL ENGINES

Equipment Co., Ltd. These engines were of the six-cylinder vertical type, developing 95 b.h.p.

The internal type of Acro pre-combustion chamber utilizes a space arranged in the crown of the piston as the ante-chamber, the fuel being sprayed through the neck of the communicating passage. In this case the air flows from the cylinder space into the auxiliary chamber in the piston during the compression stroke, and the fuel is injected shortly before the piston reaches the top dead centre. The fuel ignites in the communicating passage and combustion therefore occurs first in this place. Upon reversal of the piston motion, the space above the piston is enlarged and the pressure is decreased. A high pressure still exists in the air chamber, however, so that a strong blast of air is ejected through the orifice of the funnel into which the fuel is being injected until the piston has travelled about 15° past the top dead centre. This current of air comes into contact with the burning fuel in the funnel, and assists, therefore, in completing the combustion process. Further, the gases which are formed during combustion escape into the gradually increasing cylinder space outside the funnel and therefore do not interfere with the combustion.

Auxiliary Air Chambers. In this method of fuel injection and atomizing, the principal factor relied upon to attain these ends is the turbulent motion of the compressed air. In general, when the piston reaches the end of its compression stroke it approaches very close to the flat head of the cylinder, so that the air between the piston crown and cylinder head is forced out, more or less violently, into the auxiliary air chamber, or combustion chamber. The turbulent movement thus given to the air causes it to swirl past the fuel nozzle very rapidly so that in an extremely short interval the whole mass of air will have swept past the nozzle.

Since it is this air that causes the admixture with the fuel, in this case the fuel injection nozzle may be made comparatively large in diameter, so that it is of simple design and free from the possible choking defects of the multiple-hole type.

Moreover, it is not necessary to employ very high injection pressures, as with the direct injection method outlined under heading (1).

There are various methods of carrying out this air-turbulence method of fuel atomization, but in each case that is examined it will be noticed that there is a separate combustion or auxiliary air chamber above the cylinder, the volume of this being relatively small compared with that of the cylinder, or piston.

The piston head in most cases is flat, although in certain instances, where the auxiliary air chamber is offset from the centre line, the piston crown is cut away, or chamfered. One fuel nozzle is usually employed, but in certain cases multiple nozzles are used. Typical examples of engines using this method of fuel atomization are those of Junker, Hesselmann, Krupp, Ricardo, Omo, Leyland, Perkins, Victor and Fowler Sanders. Some of these representative types are described hereafter.

In regard to the disadvantages of this type of combustion head and fuel atomizer, it is usually more difficult to start from the cold, and there is, in some cases, a loss of efficiency due to the cooling of the compressed air charge and the igniting expanding gases owing to the turbulence action, for the loss of heat to the cylinder walls is greater for a rapidly moving gas than for a more or less quiescent one. Further, in some cases the delay period of ignition is greater than for the direct injection type.

The examples of turbulent ante-chambers, given in Fig. 16, show some alternative methods of carrying out the principles described. In the two examples shown at *A* and *B*, the inlet valve is arranged on the side of the smaller chamber, so that the entering air is given a swirling motion; this motion persists throughout the compression stroke. It is enhanced

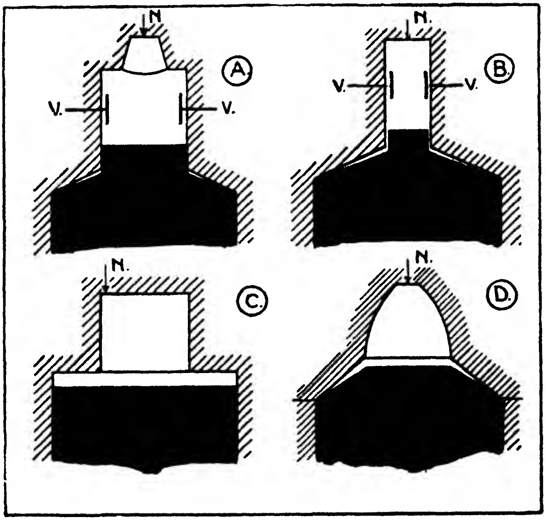


FIG. 16. SOME TYPICAL TURBULENT ANTE-CHAMBER DESIGNS

V denotes the valves, and *N* the fuel injection nozzle

A = Blackstone *C* = Ricardo

B = Wiseman *D* = Petters

further when the piston projection enters this smaller chamber, the compressed air in the main cylinder being forced violently into the smaller chamber. The position of the fuel injector is indicated by the arrow *N*.

In the case of the Ricardo engine, indicated at *C*, Fig. 16, no valves are shown since the engine is of the sleeve-valve type; this particular method of turbulent injection is described more fully, later. The example shown at *D*, Fig. 16, refers to one model of the

Petters two-stroke engine, with inlet and exhaust ports in the lower part of the cylinder barrel.

Certain other examples of turbulent cylinder heads now in demand utilize spherically-shaped ante-chambers

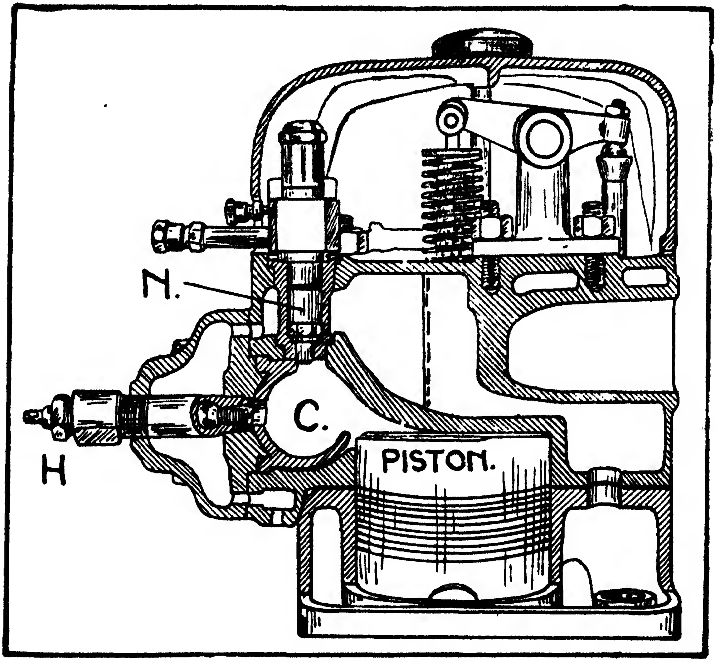


Fig. 17. The Omo Cylinder Head

into which the compressed air is forced tangentially, as it were, by means of a straight port connecting it to the cylinder. The Omo, Perkins, Leyland, and Comet cylinder heads, described later, are typical examples of this method.

The Omo Cylinder Head. This Continental engine, illustrated in Fig. 17, has a spherical type of ante-chamber arranged on one side of the cylinder; this chamber is of the detachable type, and it has an air space, or insulation, over part of its outer surface. The upward movement of the piston on the compression stroke causes the air to swirl rapidly in the ante-chamber

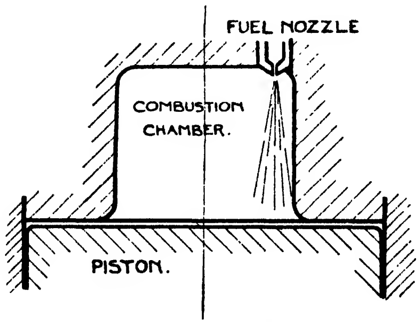


FIG. 18. PRINCIPLE OF RICARDO ROTATIONAL SWIRL SYSTEM

towards the end of the compression stroke. The fuel is injected through an "open" nozzle vertically downwards so that the swirling mass of air sweeps past it and mixes intimately with the fuel as it issues. An electric heating plug, consisting of a small spiral of resistance wire, which can be heated to incandescence by means of an electric current from a low-tension battery, is arranged horizontally in the ante-chamber; it is used for starting from the cold.

The Ricardo Turbulent Head. An original method of creating the necessary air turbulence during injection has been devised by Ricardo, and is used in single-sleeve valve engines, of which the Mirrless Diesel is a typical example.

The principle of the method—which is known more popularly as the " Rotational Swirl " system—is to give the compressed air a rotary movement, about the axis of the cylindrical ante-chamber, and to place the fuel injector parallel to the axis of the cylinder, but offset from the centre, as shown in Fig. 18, so that the fuel is injected across the air stream. The highly-heated air ignites each drop of fuel as it enters, carrying away the

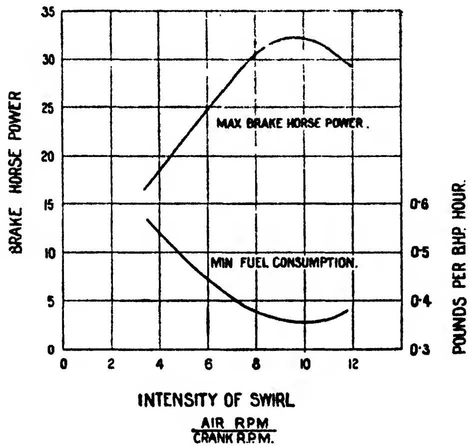


FIG. 19. EFFECT OF VARYING THE RATE OF SWIRL IN RICARDO SYSTEM

products of combustion and bringing a fresh supply of air for combustion of the next drop.

In this system the velocity of the air can be made considerably greater than that of the injected fuel. Moreover, it can be controlled, if necessary, by means of ports of variable inclination—in sleeve-valve engines. Further, the air velocity is always proportional to the engine speed, and is practically constant during the actual combustion process.

From tests carried out by Ricardo, it has been shown that maximum power and efficiency are obtained when the rotational speed of the air in the ante-chamber is from nine to ten times the crankshaft speed. (Fig. 19.)

In applying this method it has been found necessary to intensify still further the turbulence obtained from the inclined air inlet ports by compressing the whole of the cylinder contents into an ante-chamber cylinder

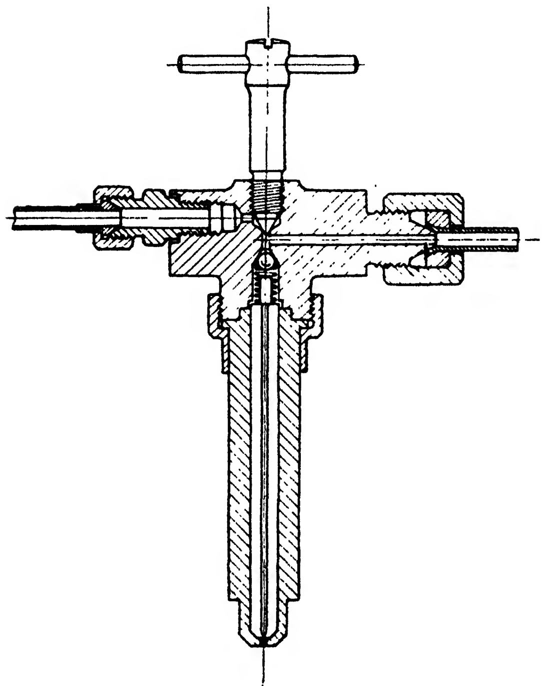


FIG. 20. THE ORIGINAL RICARDO INJECTION NOZZLE

having a diameter one-half that of the main cylinder; this results in a doubling of the rotational speed. Additional turbulence is given to the air at the last moment by the sudden injection of the air entrapped between the piston and the flat portion of the cylinder head, thus setting up a kind of vortex ring.

In carrying out this system the fuel was injected through a plain single-hole nozzle of relatively large diameter (Fig. 20); the oil pressure for injection was from 1,500 to 3,000 lb. per sq. in., according to the engine speed.

With the Ricardo turbulence system it is possible to utilize nearly 80 per cent of the total oxygen available in the cylinder, so that mean indicated pressures of from 135 to 145 lb. per sq. in. are obtainable at practically any speed at which the engine is capable, from mechanical considerations, of running.

In connection with the performances of high-speed Diesel engines utilizing this system, a number of tests have been made on experimental engines of the single and multiple cylinder types.

Fig. 21 shows the relation between the brake horse-power and the fuel consumption of a single cylinder sleeve-valve engine, of 5½ in. bore and 7 in. stroke, running at a speed of 1,300 r.p.m. and developing 32·5 b.h.p. at this speed. The maximum brake m.e.p. at 1,300 r.p.m. is 119 lb. per sq. in., this being equivalent to an indicated m.e.p. of 142 lb. per sq. in. One particularly interesting feature of the results shown is the relatively low fuel consumption over a fairly wide range of engine speeds. Thus, the fuel consumed from about 60 per cent full-load to full-load is about 0·355 lb. per b.h.p. hour.

An indicator diagram, taken with the Farnboro' electric type indicator, which gives the corresponding cylinder pressures on a crank-angle, or time base is reproduced in Fig. 22 for the same engine. In this case the engine was running at a speed of 2,200 r.p.m. and developing a brake m.e.p. of 90 lb. per sq. in. The fuel consumption was 0·39 lb. b.h.p. hour. Fig. 23 shows another indicator diagram taken at 1,300 r.p.m.

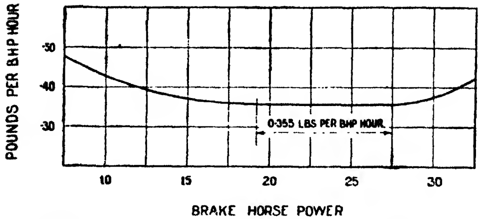


FIG. 21. POWER OUTPUT AND FUEL CONSUMPTION RELATION FOR RICARDO SYSTEM

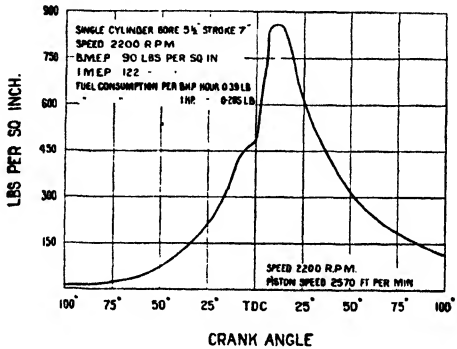


FIG. 22. INDICATOR DIAGRAM TAKEN AT 2,200 R.P.M.

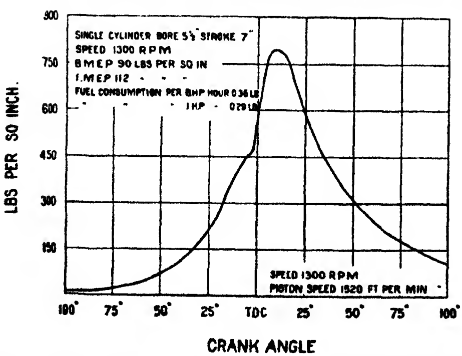


FIG. 23. INDICATOR DIAGRAM FROM SAME ENGINE AT 1,300 R.P.M.

Fig. 24 illustrates a cross-section through the cylinder head of an actual high-speed sleeve-valve Diesel engine employing the rotational swirl method of fuel injection.

The Comet Head. This design of cylinder head, due

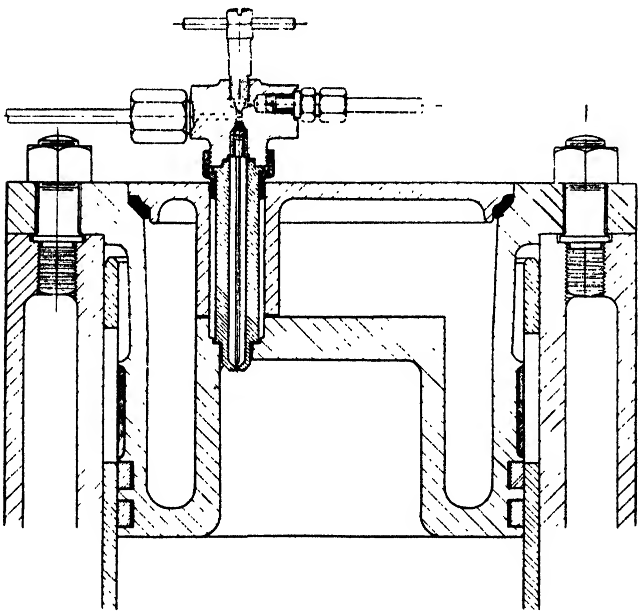


Fig. 24. Sectional View through Ricardo Sleeve Valve Engine Cylinder Head

to H. Ricardo, was introduced to the commercial vehicle engines made by Messrs. Associated Equipment, Ltd., London, for the London Passenger Transport Board, in place of those using the Acro type of cylinder head.

With the substitution of the Comet for the latter type of head it was found possible to increase the power output from the same capacity of engine by about 40 per cent, whilst improving the fuel consumption at the same time; moreover, the converted engines ran without any trace of Diesel knock.

The Comet head, shown in Fig. 25 consists of a spherical chamber communicating with the cylinder

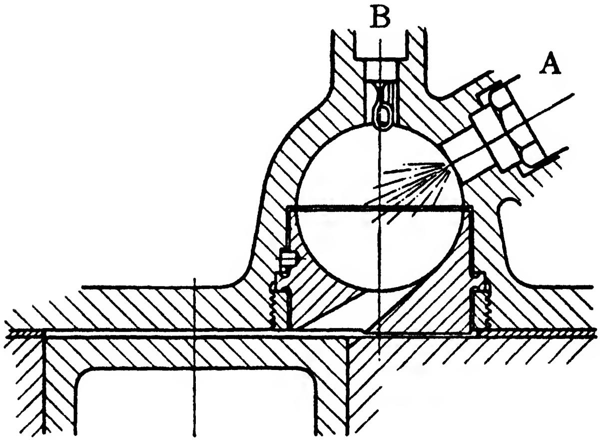


FIG. 25 THE COMET HEAD
A. Injector. *B.* Heater Plug.

by means of a tapered throat, the general direction of which is tangential to the spherical chamber. The lower half of the latter is made from a heat-resisting metal, insulated from the cylinder head by the provision of a small air gap. The object of this arrangement is to maintain the temperature of the lower half of the sphere, upon which the flame impinges, above about 350° C., for below this temperature it has been shown that the pungent smelling aldehydes—which account for the objectionable exhaust odour, previously noticed—are formed.

When the piston ascends on its compression stroke and approaches the top of same, the compressed air is forced at a high velocity through the throat and into the spherical chamber, where it is swirled around rapidly. In this connection it should be noted that there is very little clearance between the piston top and the cylinder head so that practically the whole of the air charge is forced into the auxiliary chamber.

Just before the end of the compression stroke, namely, about 5° to 10° of crank angle, the injection commences. The injector used in connection with the Comet head is of the plain single hole pattern operating at a relatively low fuel pressure. It is arranged in the side of the spherical chamber and sprays its fuel radially towards the centre, but not towards the opening of the throat.

The fuel particles are burnt by the swirling mass of air, as it passes the injector nozzle, in an efficient manner.

In this connection the heat-insulated lower half of the spherical chamber tends to reduce the heat losses from the swirling air charge, under normal working conditions, so that the delay period is reduced and a better control over the rate of burning obtained; in this way Diesel knock is avoided and the delay period is kept constant at all speeds.

It is on this account that fixed injection timing is employed on Comet head engines; for this reason, also, the maximum or peak pressures remain practically constant.

In regard to cold starting, it is necessary—in view of the heat losses to the throat and spherical chamber—to employ heater plugs for starting purposes.

The original Comet head has now been modified for the purpose of providing easier starting, better combustion conditions at low speeds and in order, also, to use inferior grades of fuel, without any appreciable falling off in power output or increase in the fuel consumption.

As it is not possible, with present space limitations, to describe the modified forms of Comet head the reader is referred to the Paper* given in the footnote reference for a full account of these types and their effects.

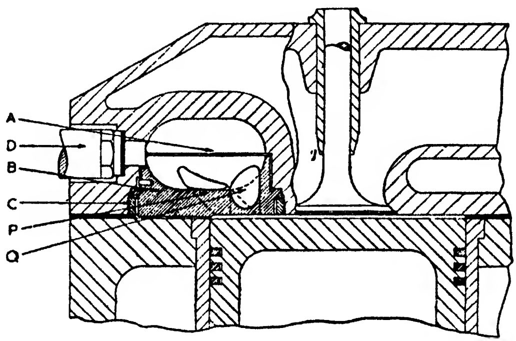


FIG. 26

The Ricardo Whirlpool Head. This design of head represents an improvement upon the Comet one, in that it dispenses with the use of heater plugs for starting from the cold and will operate satisfactorily over a wide speed range. Thus a single-cylinder experimental engine would run smoothly at 300 r.p.m. and would also work at speeds up to 4,000 r.p.m.; it also had good cold starting qualities.

Fig. 26 illustrates the arrangement of the Whirlpool head. It employs a vertical axis rotational motion instead of the horizontal one used in the Comet head.

The swirl chamber resembles in shape a kind of flattened sphere or Dutch cheese shape *A*, the upper half being machined in the cylinder head casting.

* "Compression Swirl Oil Engines." H. S. Glyde and E. N. Soar. *Proc. Inst. Autom. Engrs.*, December, 1936.

The lower half, as usual with the Comet heads, is made of a separate heat-resisting steel member *B*, which is insulated from the cylinder head and held in position by means of a dowel and retaining nut *C*. There are usually two communicating throats *P* and Q, consisting of plain drilled holes set at right angles to each

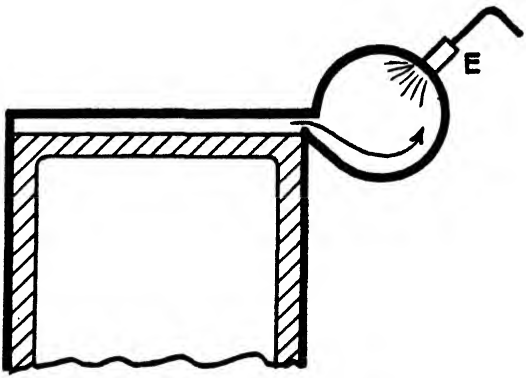


FIG. 27. THE LEYLAND ANTE-CHAMBER ENGINE
E—FUEL INJECTOR

other and at 45 degrees to the vertical. The injection nozzle *D* is located in the central horizontal plane.

The Leyland Ante-chamber Engine. The smallest of the three Leyland engines, namely, the 29·4 h.p. rated one, incorporates a different design of combustion head to that fitted on the other two models—which employ the direct-injection cavity piston combustion system.

The former engine is a six-cylinder one of 3½ in. bore and 5 in. stroke giving a cylinder capacity of 289 cub. in. (4,731 c.c.). The arrangement of the combustion head is shown in Fig. 27 from which it will be observed that it utilizes an offset spherical turbulence chamber, one-half of which is formed in the cylinder block casting and the other half in the detachable head; a small tangential throat places this ante-chamber in communication with the cylinder itself.

The injection nozzle is arranged on the left and sprays the fuel obliquely downwards towards the centre of the sphere, i.e. at right angles to the direction of the turbulent air flow caused by the displacement of

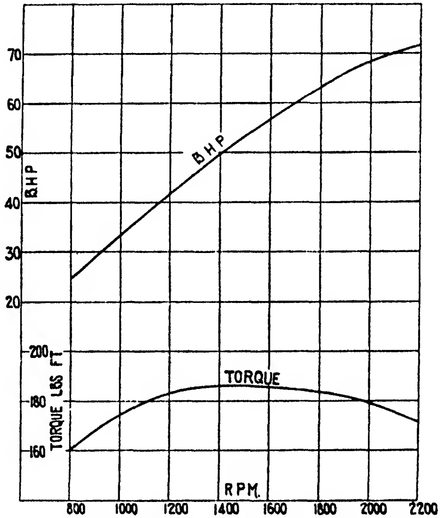


Fig. 28. Leyland Engine Performance Curves

the air by the ascending piston from the cylinder; this ensures efficient mixing of the fuel and air for combustion.

The power curve for this engine is reproduced in Fig. 28, from which it will be seen that the b.h.p. increases from about 25 at 800 r.p.m. to 72 at 2,200 r.p.m. The comparative flatness of the torque curve, which is a special feature of this engine, results in a relative high torque at the lower speeds, so that the engine pulls well on top gear at these lower speeds.

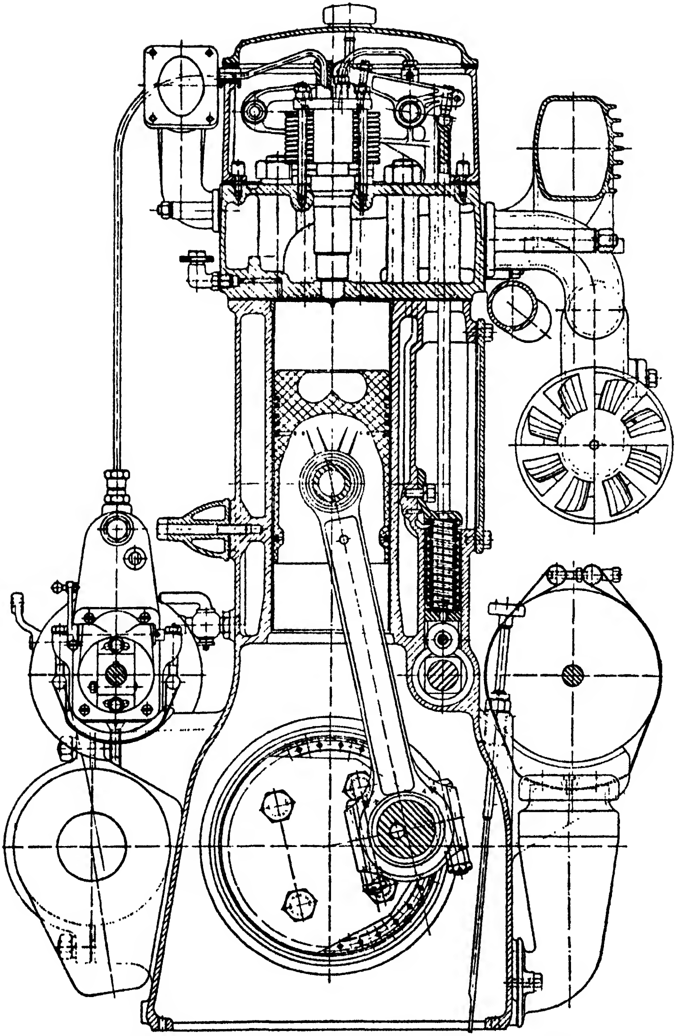


FIG. 29. THE SAURER DUAL TURBULENCE ENGINE

The Dual Turbulence System. An interesting combustion system which is employed on the Armstrong-Saurer engines is that known as the "dual turbulence" one. The engine has two air-inlet and two exhaust valves per cylinder, the inlet being masked on one side to give the incoming air a directional movement,

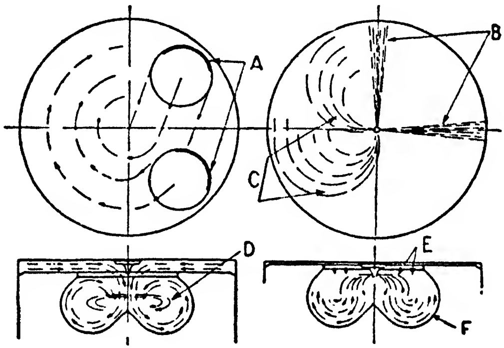


FIG. 30

namely, a rotation about the vertical axis. In addition the air charge, during the following compression stroke, is given a vertical movement which is superimposed upon the horizontal rotational one persisting from the suction stroke.

The engine in question has a flat cylinder head and a cavity-type piston. The cavity is in the form of a kind of circular section ring with a common centre portion as shown in Figs. 29 and 30.

Referring to the diagrammatic views, given in Fig. 30, showing the relative air movements and deflection of the fuel spray, from the centrally situated fuel injection nozzle, the upper left-hand view indicates the direction of the air charge as it flows into the cylinder past the masked inlet valves *A*. The path of the fuel particles sprayed from the nozzle in still air would be that shown at *B* in the upper right-hand illustration. Owing, however, to the rotary air movement in the cavity piston, the actual path of the fuel particles is that shown at *C*.

In the lower left-hand illustration the horizontal air stream is shown at *D* in combination with the vertical air swirl due to the air being trapped and then displaced into the cavity of the piston as it ascends on its compression stoke. In the lower right-hand diagram *E* shows the superimposed air swirl acting on the fuel spray *F* so as to give a very intimate mixing of the fuel particles with the air.

Engines employing this turbulent piston cavity system give increased b.h.p., torque and speed values, with lower fuel consumptions than the engines of similar dimensions previously used by the same firm and employing the Acro cylinder heads.

The "dual turbulence" engines use a compression ratio of 15 : 1 and yield maximum pressures of 600 to 700 lb. per sq. in. The maximum engine speeds are 1,800 to 2,000 r.p.m., and the fuel consumption (minimum), 0·385 lb. per b.h.p. per hour.

A somewhat similar combustion system with cavity piston, four valves per cylinder and central fuel injector, is employed on the more recent Dennis and one of the A.E.C. commercial vehicle engines.

The Perkins "Aeroflow" Head. The Perkins engines, which are made in sizes down to the four-cylinder type rated at 14·4 h p., employ a spherical turbulence chamber communicating with the cylinder proper by means of a passage of greater relative sectional area than in the case of the Comet head.

The injector is located in this passage or neck (Fig. 31) and is arranged with two fuel spraying orifices, one of which is directed towards the cylinder and the other towards the spherical chamber. The latter spray causes the fuel particles to become entrained in the turbulent air stream in the spherical chamber, whilst the former mixes with the air passing down to the cylinder when the piston commences to move downwards. In this manner, it is claimed, the system

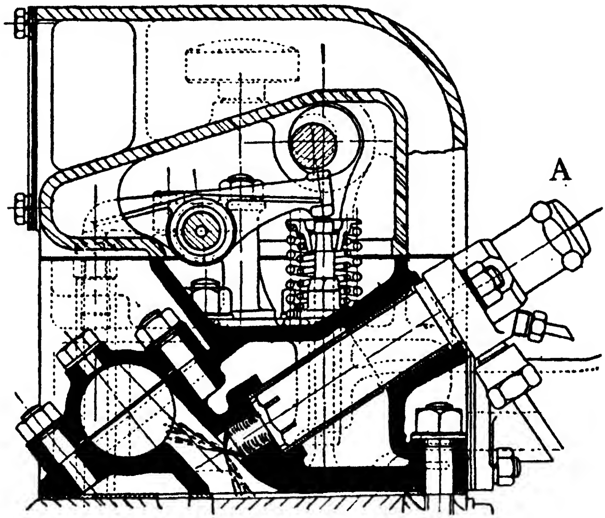


FIG. 31. THE PERKINS COMBUSTION CHAMBER

combines the advantages of both the turbulent air cell and direct injection combustion processes.

The engines are fairly easy to start and they run smoothly with no trace of Diesel knock. No heater plugs are necessary in this design, but two small priming holes are provided in the air passages, through which a small amount of paraffin may be injected in order to assist starting in cold weather. The usual compression ratio employed is 16 : 1.

The Lanova Combustion System. This Continental design of head has also been adopted by the American

Buda concern for motor vehicles; it has been used on British makes of Diesel engine as well.

The combustion chamber, situated above the top of the piston, consists of two lobed portions above the air inlet and exhaust valves, respectively.

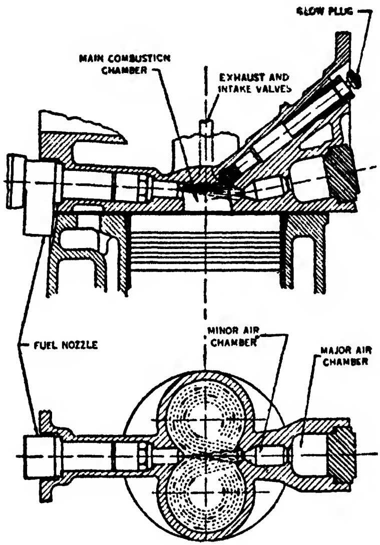


FIG. 32. THE LANOVA COMBUSTION SYSTEM

Referring to the sectional view in the upper diagram (Fig. 32) on the right is shown a double air chamber, the spaces of which are known as the major and minor air chambers. The fuel injection nozzle is located on the left-hand side in a position, such that the fuel spray is directed towards the mouth of the minor air chamber.

When the piston almost reaches the top of the compression stroke and fuel injection commences, the burning fuel particles are projected across to the mouth of and into the minor air chamber. As the piston commences to descend air from the air chambers at once rushes out, carrying the fuel particles with it and causing the burning mass to sweep round the two lobes of the combustion chamber, as shown in the lower illustration; this reversal of the air direction to that of the fuel spray provides a certain admixture of the fuel and the air. The swirling of the air around the lobes also provides two turbulent air streams which intimately mix the fuel particles with the air for combustion. The combustion process under these conditions is not of extreme rapidity, the rate of pressure rise being relatively low, since at first only a portion of the air is available for combustion. The maximum cylinder pressures are therefore lower than with many other combustion systems; the usual values lie between 550 and 650 lb. per sq. in.

The mean effective pressures, on the other hand, are relatively high, since the peak pressures tend to become maintained over a certain crank angle period. This is shown from indicator diagrams of the Lanova head engine, the corresponding mean effective pressures ranging from 100 to 125 lb. per sq. in. The compression ratios used are about 13 : 1. The original Lanova heads were fitted with a plug and screw device for shutting off the major from the minor air chamber in order to increase the compression ratio for starting purposes.

The Buda Lanova engine employs heater plugs, situated just above the air chambers, for starting purposes.

The injection pressures employed are relatively low, namely, about 1,200 lb. per sq. in. The maximum engine speeds in the case of the Buda engines are 2,000 to 2,200 r.p.m.

Other Typical Combustion Heads. There is a fairly wide range of combustion head designs used for high-speed Diesel engines. Whilst these differ considerably in the general locations of the combustion chambers and in detail design, it is the general purpose of all these systems to ensure the proper admixture of the injected fuel with the air charge, so that the combustion process will be an efficient one; in the more recent designs regulation of the rate of pressure rise is also aimed at, in order to prevent Diesel knock.

An examination of the various systems in present use indicates that the most popular types are the turbulent ante-chamber and direct-injection ones. On the Continental models the use of air cells and pre-combustion chambers is still prevalent, but modifications have been introduced to overcome the drawbacks of the earlier pre-combustion chambers, so that not only are the power outputs and maximum working speeds increased, but the fuel consumptions have been reduced; in some cases, also, the aim has been towards delayed combustion with a view to reducing the maximum cylinder pressures and extending the period of these pressures.

Although it is not possible to give a survey of all the different designs of combustion heads not hitherto dealt with in this section, some typical examples of the more important ones are shown in Figs. 33 to 36.

Fig. 33 illustrates the turbulent ante-chamber head known as the *Clerestory* one, in front and side sectional views. In this design the air inlet and exhaust valves are arranged horizontally in an ante-chamber of practically cylindrical shape. The piston is of special form, having a cylindrical projection on the top of the tapered crown, which almost closes the passage connecting the cylinder and ante-chamber; a somewhat wider space is left, however, between this projecting portion and the right-hand side of the passage—as shown in the left-hand diagram. The object of this is to direct the air, which is finally displaced by the piston, as it approaches the top of its compression stroke, tangentially into the cylindrical chamber, thus imparting a rotary movement to it. The injection nozzle *A* is arranged on top (Fig. 33) so as to spray the fuel vertically downwards, i.e. at right angles to the

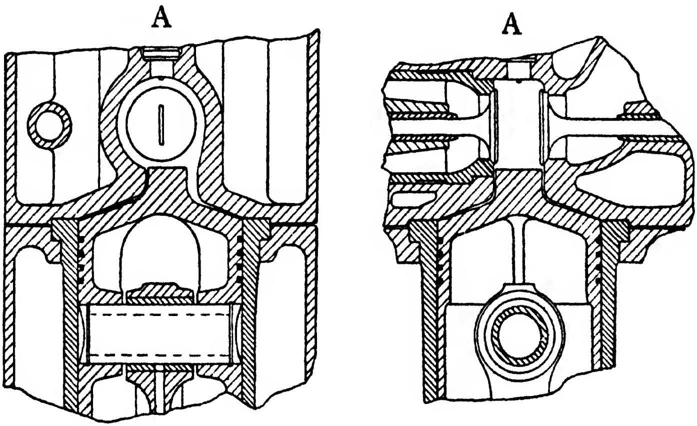


FIG. 33. THE CLERESTORY CYLINDER HEAD

direction of the air flow. This arrangement results in an efficient combustion process, but involves somewhat complicated valve operating mechanism.

Fig. 34 shows, diagrammatically, another combustion head of the direct injection type, which is used with various modifications on certain Continental engines, notably the Maybach and Renault ones. In this example the piston is cut away on one side in the form of a cavity. It is so arranged that the plain top of the piston, at the top of its stroke comes very close to the cylinder head so that the last movement of the piston suddenly forces the air between its plain crown and the cylinder head sideways into the cavity, whence it acquires much turbulence; the fuel injection occurs at this stage so that the conditions are appropriate for the efficient burning of the fuel. This type of combustion head gives excellent fuel economy and easy cold starting without the aid of heater plugs, but in general it is not an easy matter to control the delay

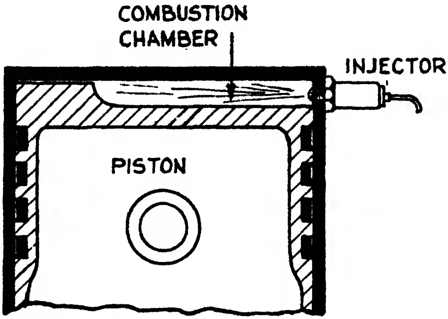


FIG. 34. OFFSET DIRECT INJECTION SYSTEM

period so that there is a tendency towards Diesel knock in some instances.

Fig. 35 (A) illustrates the Oberhänsli head used on certain German engines, of which the Vomag is a typical example. It employs a spherical pattern turbulence chamber *C* which is designed so as to be detachable; it is made of a highly refractory material in order to provide easier cold starting, more rapid ignition of the fuel and to avoid the formation of aldehydes—as in the Comet heads. The fuel is injected downwards from the injection nozzle *A* towards the centre of the spherical chamber, whilst the heater plug *B* is arranged horizontally. With this form of combustion head fairly high engine speeds are possible.

Fig. 35 (B) shows a design of combustion head that may be regarded as a modified version of the Acro one previously described. The auxiliary chamber is of practically spherical form and communicates with the space on the right above the piston, when the latter is on its top dead centre, by means of a perforated

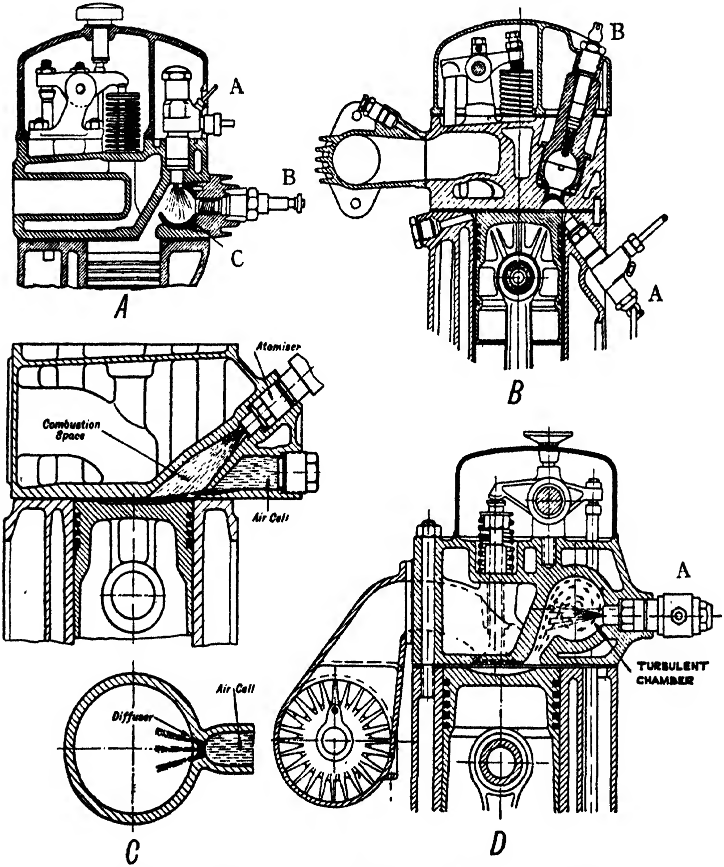


FIG. 35. TYPES OF CONTINENTAL COMBUSTION HEADS

plate; the latter is made detachable for replacement, when necessary.

The fuel injector *A* sprays its fuel into the smaller combustion space and on to the perforated plate, so that only a small quantity of fuel will find its way into the spherical chamber; the greater part of the fuel impinges on to the screen plate and thereby becomes vaporized. Owing to the turbulence existing in the air charge stored in the auxiliary chamber, the fuel that actually gets through the perforations is burnt and as the piston commences to descend the combustion of the bulk of the fuel then proceeds. The baffle or screen plate, according to the claims of the inventor of this system, not only prevents large quantities of fuel from reaching the air cell, thus confining the main combustion to the outer combustion chamber, but it ensures efficient atomization of the fuel spray and also heating of the fuel vapour. This form of cylinder head is more efficient than the Acro one and enables higher speeds to be attained.

The system described is employed in the German M.W.M. high-speed Diesel engines. These employ compression ratios of 17·5 : 1, with corresponding compression pressures of 500 lb. per sq. in. The fuel injection pressures are 1,350 lb. per sq. in. and maximum engine speeds 2,000 r.p.m.

The engines employ electric heater plugs (as shown at *B* in Fig. 35) for starting purposes. The baffle plate is made of a heat resisting nickel steel.

The combustion system employed in the M.A.N. Diesel engines used for automobile purposes is shown in Fig. 35 (c). The method in question utilizes the auxiliary air cell shown; the latter is independent of the combustion chamber and has for its object the storing up of air at the maximum compression pressure, until such time as the main combustion pressure begins to fall below this value, when the air commences to flow through the communicating passage into the combustion chamber. Therefore it not only provides a turbulence effect but also the necessary additional air required to burn the whole of the injected fuel. It will be evident that as the piston has commenced to descend whilst combustion is still proceeding, the rate of pressure rise will be less on this account than for the usual direct-injection type of engine. The net result is a general smoothing down of the usual high peak part of the pressure curve and, in consequence, better running characteristics; the combustion process is thus an efficient one.

The atomizer, or fuel injector, is situated at the top of the conical combustion chamber and sprays its fuel downwards at an average angle of about 45 degrees to the axis of the cylinder.

The arrangement of the Kamper cylinder head shown in Fig. 35 (D) belongs to the turbulent ante-chamber class, except that in this case the piston top is made concave in shape in order to provide a secondary combustion chamber, communicating with the spherical one by means of a venturi or throat. The fuel injection nozzle *A* is arranged horizontally and sprays its fuel across the direction of turbulent air flow.

Fig. 36 illustrates the form of cylinder head used on the Mercedes Benz commercial vehicle engines, employing the pre-combustion chamber method. The auxiliary chamber *B* into which fuel is sprayed from the injection nozzle *A*, supplied from the fuel pump through the connection shown at *E*, has its axis inclined to the cylinder axis. The pre-combustion chamber *B* communicates with the cylinder space, formed between the dished piston head *C* and cylinder head, by means of a nozzle having a relatively small orifice. The initial combustion of the fuel in *B* drives the main part of the unburnt fuel through the nozzle passage into the combustion space between the cylinder head and piston, where the final stage of the combustion occurs. The fuel spray emerges into this combustion space in the vaporized state owing to the high temperature of the nozzle passage between the two chambers. It is claimed that, on this account, com-

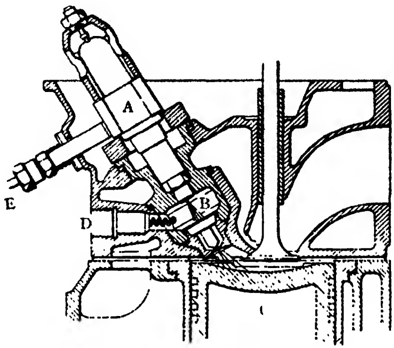


FIG. 36. THE MERCEDES BENZ COMBUSTION HEAD

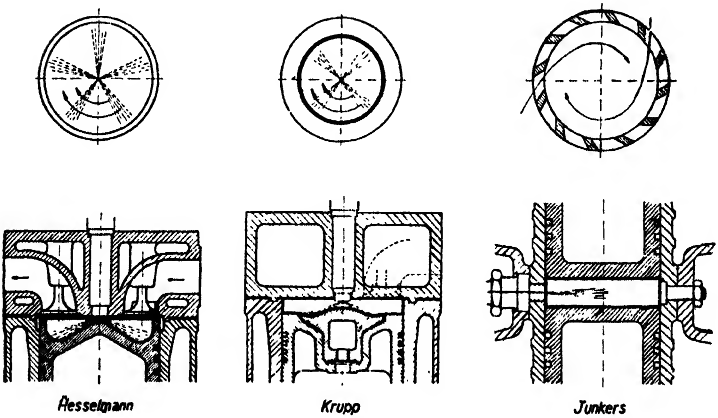
bustion of the fuel is perfect, even when the engine is idling.

The injection pressures employed are from about 1,000 to 1,300 lb. per sq. in.

Heater plugs, one of which is shown at *D*, are employed for starting purposes.

The Hesselmann Combustion Chamber. Some types of combustion chamber fall into one or other of the three principal classes described previously, or into a combination of these. The Hesselmann is a good example of the combination type.

The Hesselmann engine employs a plain cylinder head and a special form of piston, as shown in Fig. 37. The central portion, and also the periphery of the piston, are raised so as to obtain a contour to the lower part of the combustion chamber, that harmonizes with the form of the fuel jet. The peripheral part of the piston is provided for the purpose of preventing the fuel from striking the relatively cool cylinder walls; this would cause condensation and a smoky exhaust. The inlet valve is shaped in a special manner so as to impart a rotary movement to the incoming air. The air is thus in turbulent motion during the fuel injection, the fuel



SOME OTHER TYPICAL HIGH-SPEED DIESEL ENGINE CYLINDER HEADS

being sprayed at a similar angle, in radial directions, to the slope of the piston crown. In this case we have a combined direct-injection and turbulent combustion chamber.

FUEL INJECTION IN PRACTICE

Having described the fuel injection processes in their relation to combustion chamber design, it is now proposed to deal with the practical aspect of the fuel injection system as applied to motor vehicle type Diesel engines.

The fuel injection system in this case may be regarded as the parallel of the magneto and carburettor systems of the petrol engine, for in each case the final object is the ignition within the cylinder of the fuel supplied for combustion.

The system, in the Diesel engine, comprises the fuel pump, fuel filters, distributors, and injectors. The fuel pump draws its supply through an efficient filtering system from the main or an auxiliary fuel tank, measures out and delivers the correct quantity required for each cylinder of the engine, and, in virtue of the high hydraulic pressure produced, forces it to the injection nozzle in the combustion chamber at precisely the correct moment required. Moreover, the total charge of fuel metered out must be injected over a certain pre-determined period of time, at every working stroke of the piston.

The fuel pump and distributing system must therefore give the correct fuel amount, injection timing, and period.

Controlling the Power Output. In the case of commercial motor vehicle Diesel engines there must be some means of controlling the power output, similar to that of the carburettor throttle in the case of petrol engines. This is accomplished in modern Diesel engines by regulating the quantity of fuel injected per working stroke, by means of some device on the fuel pump, or—in the case of mechanically-operated injection valves—in the valve-operating mechanism.

In some cases a centrifugal governor is fitted to control the fuel pump's output so as to maintain a constant speed. In other cases, more particularly with automobile engines, the control is by means of the driver's accelerator pedal, and a hand control.

Another popular method of controlling the power output is to provide a governor to limit the maximum speed and also the minimum speed, when the engine is required to "idle"; otherwise, if there were no such control, the engine would generally stop. The accelerator pedal of the vehicle is then used to control the power at intermediate speeds, but it cannot override the governor at the maximum speed; the latter is the highest safe speed, as fixed by the manufacturers.

FUEL QUANTITIES

The fuel pump of a high-speed automobile Diesel engine has to deal with very small quantities of fuel at each injection stroke: and it must deliver these small amounts very rapidly at the higher speeds.

The whole injection period, in the case of an engine running at 3,000 r.p.m., occupies only about ·001 sec. The amount of fuel injected, per stroke of the pump, into the cylinder of a 60 h.p. high-speed engine running at full-load is of the order of 0·004 to 0·05 cub. in. At light loads it is about 0·001 to 0·015 cub. in.

FUEL PRESSURES

The injection pressures at which the fuel pump has to supply the fuel during the injection process, in practice, may vary from as low as 1,000 to as high as 10,000 lb. per sq. in. The actual injection pressure used depends upon the system of fuel injection and upon the combustion chamber design. In general, the direct injection systems require much higher pressures than the turbulent combustion head ones, for in the former case the high fuel pressure is necessary for breaking up the fuel and forcing it across the combustion chamber space.

The turbulent head systems generally employ injection pressures of 1,000 to 2,000 lb. per sq. in.; the direct injection systems use pressures of 1,500 to about 4,000 lb. per sq. in., for commercial engines.

FUEL INJECTION TIMES AND PERIODS

It is important, in Diesel engine operation, to be able to inject the fuel at precisely the correct time before the piston has reached the end of its compression stroke, in order to allow for the delay combustion effect; this is analogous to spark advance in the case of the petrol engine. Further, it is equally important to be able to vary this point of commencement of the injection to suit the engine speed. Thus, for light load running, a much smaller amount of fuel is injected per stroke, and the injection timing can then be advanced, i.e. made to occur earlier.

The usual period of fuel injection occupies from 20° to 30° of crank-angle before top dead centre, on the compression stroke, and from 10° to 15°, as a rule, after top centre; the total injection period is from 30° to 45°, according to the type of engine, speed, and other conditions.

The injection timing has an important bearing upon the power output, and cleanliness of the exhaust, so that it must carefully be chosen from experimental tests, as a rule. If the *injection is advanced too far*, i.e. made to commence too early, as measured on the compression stroke before the piston reaches the top dead centre, the *maximum, or "peak" pressures will be appreciably higher*. Such conditions are conducive to "Diesel knock." Thus, for every degree of advance of the injection the peak pressure generally increases by 35 to 40 lb. per sq. in.

It is equally important to ensure that the injection timings are the same for each of the cylinders in the case of multi-cylinder engines; otherwise the individual power strokes will be unequal and the crankshaft torques irregular in amounts. Further, each cylinder must receive exactly the same quantity of fuel as each of the others, in order to ensure equal power strokes.

In this respect it is generally specified that the fuel amounts per cylinder should not differ by more than 5 per cent at working loads and by 20 per cent when the engine is idling.

THE FUEL SYSTEM

The complete fuel system includes the main fuel tank; the fuel supply or feed pump for delivering fuel to the suction side of the fuel injection pump; the fuel filter or filters; the fuel-injection pumps for delivering metered quantities of fuel at the correct times to the fuel injection nozzles; the fuel injection nozzles and the necessary fuel piping.

The fuel feed pump may be a separate mechanical or electrical diaphragm one as used for petrol systems; a vacuum feed system may, however, be employed if desired.

In some installations the fuel feed pump is a separate unit away from the fuel injection pump, but in many instances the fuel feed pump is mounted on the fuel injection pump and operated by the latter's camshaft; this method is used for the C.A.V.-Bosch fuel feed pump.

The filtering of the fuel is a very important factor since the presence of any solid matter, even of extremely small dimensions might lead to the blocking of the fine holes in the fuel injector; moreover, it would also cause undue wear of the precision-finished fuel pump plungers and their barrels.

Apart, therefore, from careful filtering of the fuel before it enters the main fuel tank it should be made to pass through two filters before reaching the fuel injection pump.

The fuel filters employed for Diesel fuel are usually

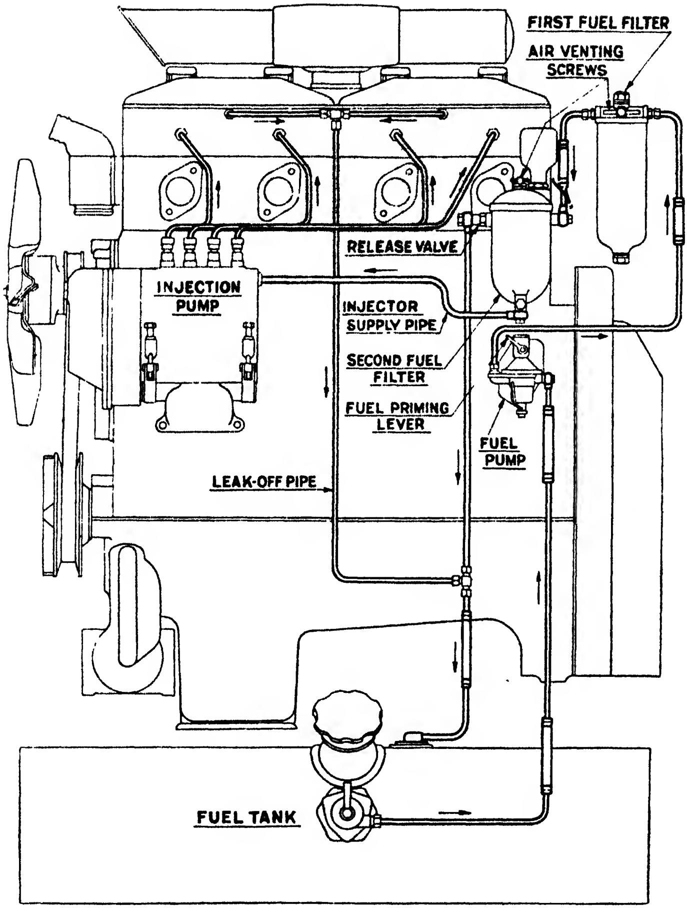


Fig. 38. The Dennis Fuel Feed System

of the felt or cloth filter variety, the fuel being forced under pressure from the outside of a cylindrical or star-sectioned felt or cloth element whence the delivery is taken from the centre of the element to the second filter or fuel injection pump. A pressure release valve is fitted to the filter to prevent excessive fuel pressure due to a choked filter element or other cause; under these conditions the fuel is then returned to the main fuel tank.

Fig. 38 illustrates a typical fuel system lay-out as used in the Dennis four-cylinder Diesel engine.

The fuel from the main tank below passes along and upwards—as shown by the arrows on the right—to the suction side of the diaphragm pattern fuel pump, from the delivery side of which it goes to the first fuel filter shown. Thence it passes to the second fuel filter on the immediate left of the former; this filter has a pressure release valve with fuel return pipe to the main tank.

From the second filter the fuel goes direct along the injector supply pipe shown to the suction side of the injection pump, whence it is delivered in metered quantities to the four injectors the supply pipes of which are denoted by the four arrows above.

As stated in the section on fuel injectors, the latter are provided with leakage or "leak-off" unions and pipes to direct any fuel leakage past the valve plungers of the injectors back to the main fuel tank or to any other convenient source of discharge. The leak-off pipe for the four injectors (Fig. 38) is shown by the vertical pipe, with downwardly directed arrow, in the centre of the diagram.

The fuel system is provided with air venting plugs in the tops of the two filters; these are used for priming the fuel system, i.e. eliminating all traces of air when the system is to be filled with fuel.

FUEL PUMP SYSTEMS

Disregarding the compressed air method of fuel injection, which is now seldom used on modern automobile engines, there are two principal methods, upon which the designs of fuel injection pumps are based, viz. as follows—

1. *The Storage or Timed Injection Valve Method, and*
2. *The Jerk Pump or Timed Pump Method.*

The Timed Injection Valve Method. In this case the fuel pump usually maintains a constant pre-determined pressure upon the fuel in the reservoir between the pump and the injection valve. At the proper moment for injection the injection valve is opened by mechanical means, kept open for the correct period, and then closed again. In order to be able to regulate the amount of the fuel injected, the valve lift must be made variable, by suitable mechanical means.

This method is usually referred to as the *timed value, storage* or *common rail system.*

In an alternative system that is often used, known as the *constant valve lift with variable fuel pressure,* the valve lift is always the same but the fuel pressure is varied to suit the load. The advantage of this method is that only one pump is required for a multi-cylinder engine so that all cylinders are supplied under the same conditions, if suitable piping for the fuel is provided to each cylinder. In these methods of fuel injection very accurate workmanship is necessary in the manufacture of the injector valves, or nozzles, for the lift of these small valves is of the order of $\frac{1}{30}$ to $\frac{1}{100}$ in. only. It is therefore essential for the valve mechanism to be absolutely rigid and free from wear, temperature, and vibration effects.

One important disadvantage of this "storage" system is that as the injection valve is exposed, always, to the full effects of the fuel pressure behind it, if any defect of the valve or its seating occurs, there will be a constant leakage of fuel past the valve during the whole of the cycle of operations; this will cause premature ignition and a smoky exhaust; the power output will be seriously diminished in the cylinder in question.

Leakage of fuel past an injection valve, after it is supposed to be closed, is usually referred to as *dribble.*

Fig. 39 illustrates a typical example of the timed injection method we have just described. The system

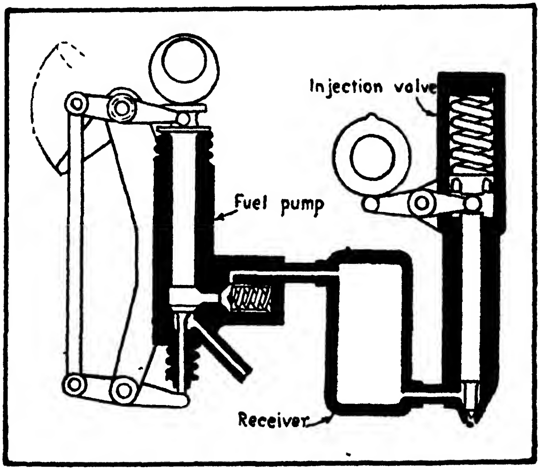


FIG. 39. THE R.A.E. TIMED INJECTION VALVE METHOD

shown was that employed at the R.A.E. in connection with the classical tests on a single-cylinder high-speed Diesel engine of the type known as the 20T. The results of these tests inspired Continental firms to produce the high-speed engines that were the forerunners of those designed in this country.

The system shown used a fuel storage reservoir, in which the fuel was stored under high pressure; the injection pressures used ranged from about 1,400 to 8,850 lb. per sq. in., the direct method of fuel injection being employed in this engine.

The fuel pump itself was used as the fuel measuring device; in this case the pump's output was controlled by the suction valve, the injection valve being opened, positively, by a cam and closed by a spring. No by-pass valve was fitted in the system since the pump was the only metering device used. The pressure in this system, it will be seen, becomes automatically that required to force into the cylinder through the injection valve, and in the period of time available, such quantity of fuel as the pump delivers into the system. It will be evident, therefore, that at a given engine speed the larger the quantity of fuel delivered by the pump the greater will the injection pressure become, if nothing else in the system is altered. In the system shown (Fig. 39) the time of injection of the fuel could be varied whilst the engine was running, but the actual period—which was fixed by the contour of the cam—could not be altered under running conditions.

The Timed Pump Method. The principle of this system of fuel injection is that the fuel pump itself controls the fuel injection timing, for it delivers fuel at the correct injection pressure, at the required moment and over the given injection period.

This system is by far the most widely-used one on high-speed automobile Diesel engines.

It is usually employed in conjunction with what is known as the "hydraulic" type of injection valve. The latter is a conical seat valve held on to its seating by means of a strong spring. When the fuel pump pressure is applied to it the pressure of the spring is overcome, the valve lifted, and the fuel is injected past the valve into the combustion chamber.

Fig. 40 shows the essentials of the timed, or jerk, pump system. The left-hand diagram illustrates the cam-operated fuel pump, with its suction and delivery valves, the direction of flow of the fuel being shown by means of the arrows. The right-hand diagram shows the principle of the fuel injection nozzle, the control spring being depicted above the valve for convenience of illustration.

The fuel enters, from the pump delivery, through a small-bore tube to the opening on the left, and thence

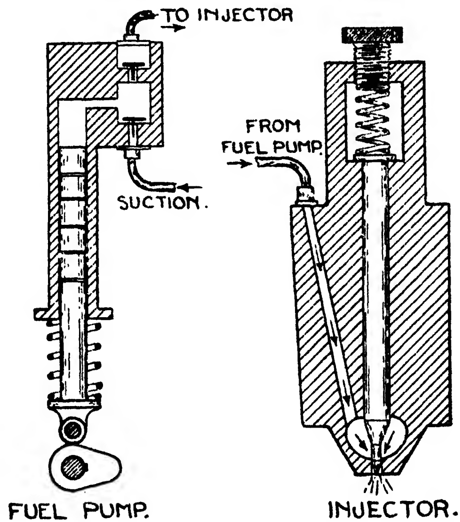


FIG. 40. ILLUSTRATING PRINCIPLE OF JERK PUMP METHOD

to the lower chamber near the valve seating. It will be observed that the valve is *a differential one*, spring pressure being applied to its central area and oil pressure to the annular area around the former; this gives good control of the beginning and end of the injection period.

FUEL PUMP DESIGNS

The design of the fuel pump is by no means a simple matter, since *it is necessary to have different rates of injection* at the beginning and over most of the period of injection, combined with a fairly sharp cut-off at the end of the injection period.

A typical arrangement is to aim at a fairly low initial rate of injection, with an increasing rate up to a maximum which is reached at the point where injection ceases. Space will not permit any account of the various methods employed by fuel-pump designers to attain this ideal result,* but it may be pointed out that the ordinary pump, having the usual relatively low start and stop rates, with its maximum rate at some intermediate point, is not satisfactory.

It is usual to arrange the pump operating mechanism to utilize only a portion of the stroke of the pump, so that the slowest periods are cut out and only the intermediate and fastest ones employed for the beginning and end of the delivery period; the central part of the stroke is used in such cases. The cam which operates the fuel pump is sometimes given a variable contour and a sliding movement in order to obtain the required control over the pump delivery for power regulation purposes. In other cases the *inlet valve* is controlled mechanically for a like purpose.

One common arrangement is to provide a suitable cam for giving the pump's plunger as uniform a movement as possible over the greater part of its stroke. The start of the injection is controlled by closing the pump's inlet valve or port; the end of the injection is obtained by opening a control port in the pump unit. When such ports are employed it is usual to employ a piston valve which over-runs a port cut in the pump barrel; this valve may either be the plunger itself or a separate piston valve. When the pump's valves are controlled mechanically it is usual to employ rocking levers operated from the camshaft of the pump.

* A full account of this subject is given in *High-speed Diesel Engines*, 3rd Edition, A. W. Judge. (Chapman & Hall, Ltd.)

In the timed pump system it is usual to employ a separate pump for each engine cylinder, although in one or two cases one pump is used to feed a pair of cylinders in turn.

The timed pump system has the advantage of freedom from full pressure of the fuel on the injection valve except at the period of injection. As the latter is only about $\frac{1}{20}$ to $\frac{1}{30}$ of the complete cycle in the case of a four-stroke engine, there is very little likelihood of fuel "dribble" occurring. Moreover, this jerk pump method has the additional advantage that its distributor system renders it less difficult to supply the correct amount of fuel to each cylinder, although it necessitates running the fuel pump at a higher speed.

The Importance of Rigidity. One important factor in fuel pump design is that of rigidity of the pump framing, for if there is any "whip" or "give" between the camshaft and the plunger, the amount of fuel injected will vary; further, the injection timing may also vary.

It is also essential to employ strong fuel piping between the pump and injector units and to make the individual cylinder pipes of equal length.

FUEL PUMPS

As we have previously emphasized, this is one of the most important items of the modern Diesel engine; it is comparable with the petrol engine's magneto in the matter of accuracy of design and workmanship, and in reliability.

As the amounts of fuel to be pumped during each injection stroke are so small the movements of the pump plungers are very small. The working members of the pump must therefore be made of the best materials and to high limits of accuracy.

The valves and their seatings must be very hard, and made very accurately to corresponding shapes, for they have to work under fuel pressures of the order 1,000 to 8,000 lb. per sq. in. Hardened steel and alloy cast iron are used for most of the working parts of the pump, ground fits being relied upon for freedom from leakage.

Typical Fuel Pump Regulation Devices. Fig. 41 illustrates four typical methods employed for regulating the fuel amounts delivered by fuel pumps. Diagram (1) shows a variable contour sliding cam used to vary the stroke of the plunger. Diagram (2) illustrates a pump having needle-valve regulation which allows variable leakage between the delivery and suction pipes. Diagram (3) shows a by-pass regulation in which a valve on the delivery side can be tripped at a variable point in the pump plunger's stroke. When this valve is lifted part of the fuel returns to the suction pipe side. Diagram (4) depicts an alternative method in which there is a sloping groove in the plunger that can be moved to give a by-passing effect.

We shall now proceed to describe one or two typical fuel pumps from the many different makes now on the market.

The Bosch Fuel Pump. This was one of the first successful small fuel pumps designed for high-speed Diesel engines, and owing to the excellence of its design, workmanship, and materials, it has attained a similar high standard of excellence to that of the Bosch magnetos. It has been adopted by a number of automobile Diesel engine manufacturers—usually in conjunction with the Bosch injection valve systems.

In this pump there is a separate plunger for each cylinder of the engine, but all of the plungers are housed in the same unit. The plungers have constant strokes and give accurate metering and timing of the fuel at all loads and speeds. A special feature is the anti-dribble device fitted to each pump plunger unit.

The pump has a novel output control actuated by

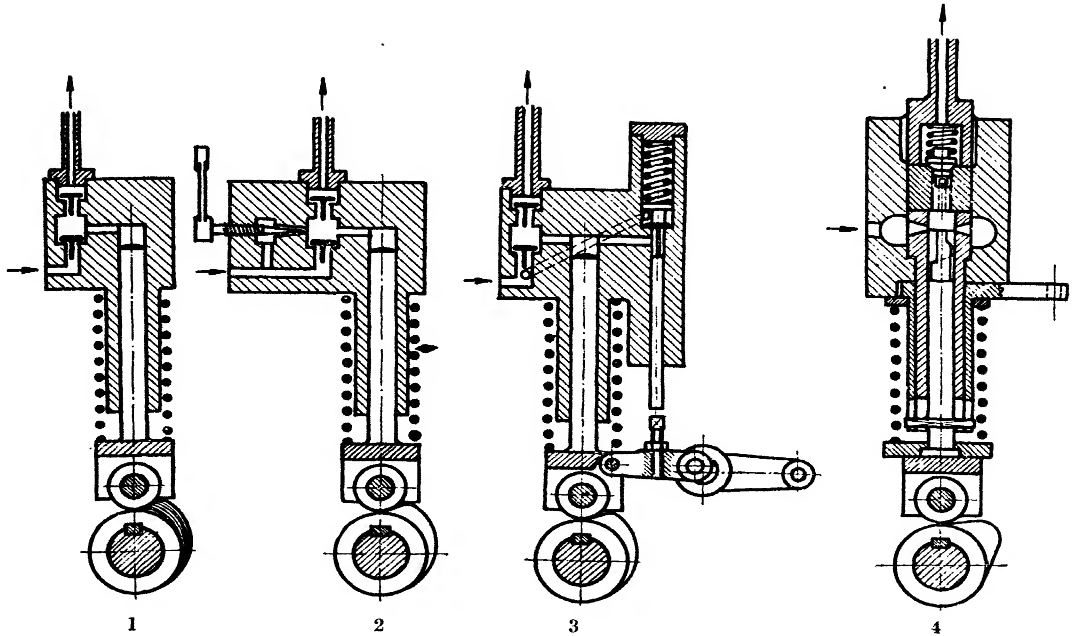


Fig. 41. Typical Fuel Pump Regulation Devices

1. Variable contour cam 2. Variable leakage by needle valve 3. By tripped delivery valve 4. By sloping groove in plunger

means of a sliding control rod (shown at a, b, c, in Fig. 42). This rod is used to shut off the supply for *stopping the engine*; it is the equivalent of the magneto switch in a

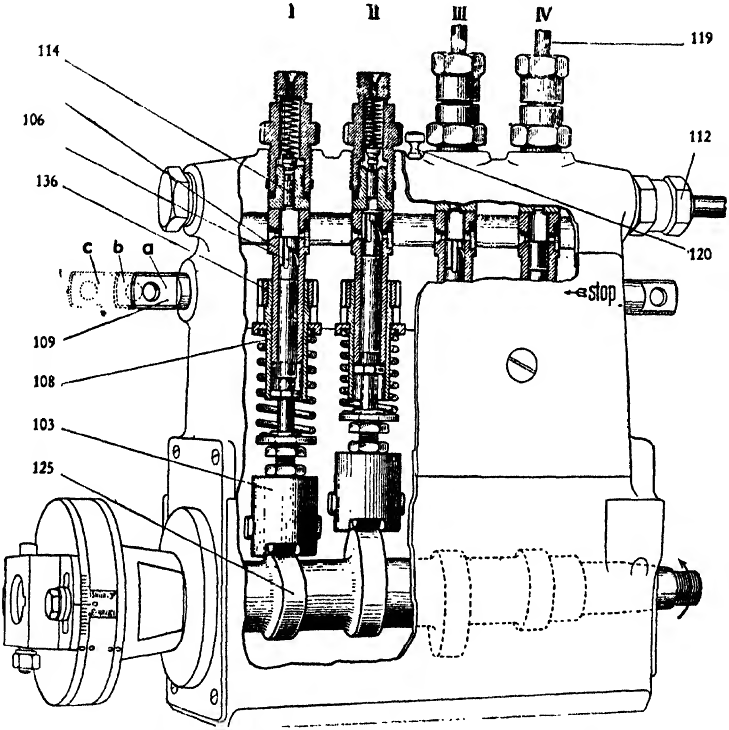


FIG. 42. THE BOSCH FOUR-CYLINDER TYPE FUEL PUMP

petrol engine. There is also an *injection advance device* to facilitate starting and to vary the point of fuel injection. A lever is provided on this unit for altering the pump drive shaft in its angular relation to the driven shaft. It is possible with this device to advance the moment of injection up to 8° (on the pump shaft) with the engine running.

Referring to Fig. 42, each pump element 106 consists of an accurately-ground steel pump barrel and a ground steel pump plunger which is a piston fit for the barrel.

The plunger is provided with a suitable guide and tappet roller 103 which bears on the cam 125. The discharge valve 114, fitted at the upper end of the barrel, completes the parts comprising the pump element.

Fuel is supplied from a tank (placed slightly higher than the pump), so that it flows easily through a suitable filter to the fuel inlet connection 112, and keeps the common suction chamber in the pump casing always flooded with clean fuel, whence it can be readily drawn into the pump barrels of the various elements through the two small ports provided. As already mentioned, the plunger 106 always moves vertically in the barrel with a constant stroke, which in this model is 10 mm. To enable the pump to vary the quantity of fuel delivered per stroke, the plunger is provided with a vertical channel (see Fig. 43) extending from its top edge *A* to an annular groove (the upper edge *B* of which is formed as a helix) a little way down the plunger length. External means (108 and 109, Fig. 42) are provided whereby the plunger can be rotated slightly in its barrel while working.

Referring to the four-cylinder engine model, shown in Fig. 42, the pump element I is shown at the bottom of its suction stroke when the two small ports are open through which fuel has already been drawn, thus filling the pump barrel. On the next up or delivery stroke (pump element II) the plunger displaces fuel back through the two small ports until its top edge *A* (Fig. 43) covers them, so that the remaining fuel is pressed out through the delivery valve 114, *via* the connecting piping to the nozzle in the engine cylinder. From this, it will readily be seen that, since the plunger is of constant stroke, its top edge will always cover the ports in the pump barrel at the same positions of the cam rotation, so that the commencement of fuel injection at the nozzle will always be the same relative to the

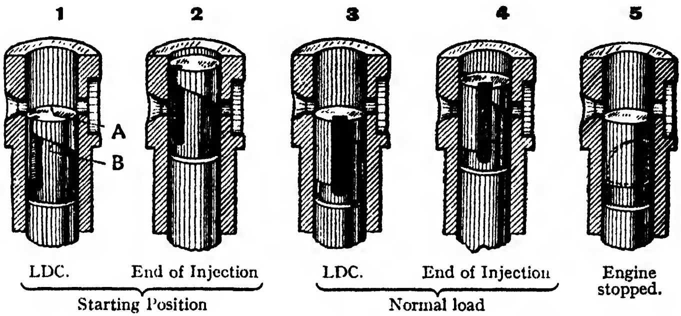


FIG. 43. SHOWING POSITIONS OF BOSCH FUEL PUMP PLUNGERS DURING A CYCLE OF OPERATIONS

position of the engine crank. As long as the ports are kept covered by the plunger, the pump will continue to inject fuel at the nozzle, but reference to 2 (Fig. 43) will show that before the plunger reaches the top of its stroke, the helical edge *B* of its annular groove (see Fig. 43) has uncovered the right-hand port, which enables the enclosed fuel to take the path of least resistance (*via* the vertical channel and annular groove in the plunger) back through the port in the barrel to the common suction chamber. The position of the plunger stroke at which the helical edge *B* will uncover the port, is adjustable by rotating the plunger through a certain angle axially by means of the toothed quadrant 107*b* (Fig. 44) which is clamped to a sleeve 107*a*, having slots engaging with the lugs of the plunger at its lower end.

The toothed quadrant 107*b* meshes with the teeth of a long rack provided on the control rod 107*d* which actuates the quadrants of all the pump elements simultaneously. This control rod—which is shown also at 109 (Fig. 42)—is connected externally either to

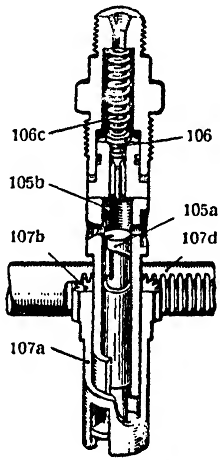


Fig. 44. Sectional View of Pump Element Showing Method of Controlling Fuel Quantity

105*a* = Pump plunger
105*b* = Pump barrel
106 = Delivery valve and seat
106*c* = Valve spring
107*a* = Control sleeve
107*b* = Toothed quadrant
107*d* = Control rod

the governor or to the ordinary accelerator pedal linkage.

The *anti-dribble device* operates as follows: When the helical edge *B* (Fig. 43) of the pump plunger uncovers the port in the pump barrel, near the end of the delivery stroke, the fuel pressure is immediately reduced so that the delivery valve at once drops on its seating, thus cutting off communication between the pump and the nozzle until the next delivery stroke. In coming to its seat to act as a non-return valve, the delivery valve is, however, made to act also as a *pressure pipe release*; this novel function is arranged by the special design of the delivery valve unit.

This valve, shown in the open and also closed positions in Fig. 45, is an ordinary conical faced valve

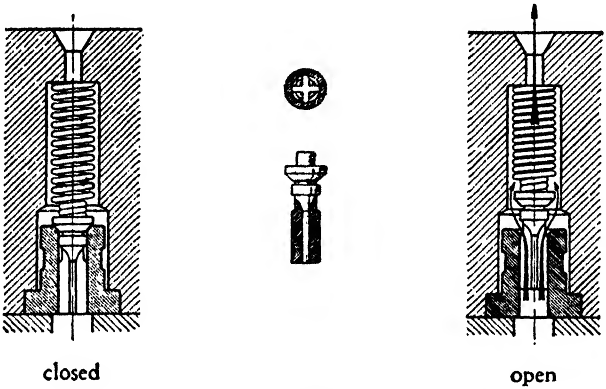


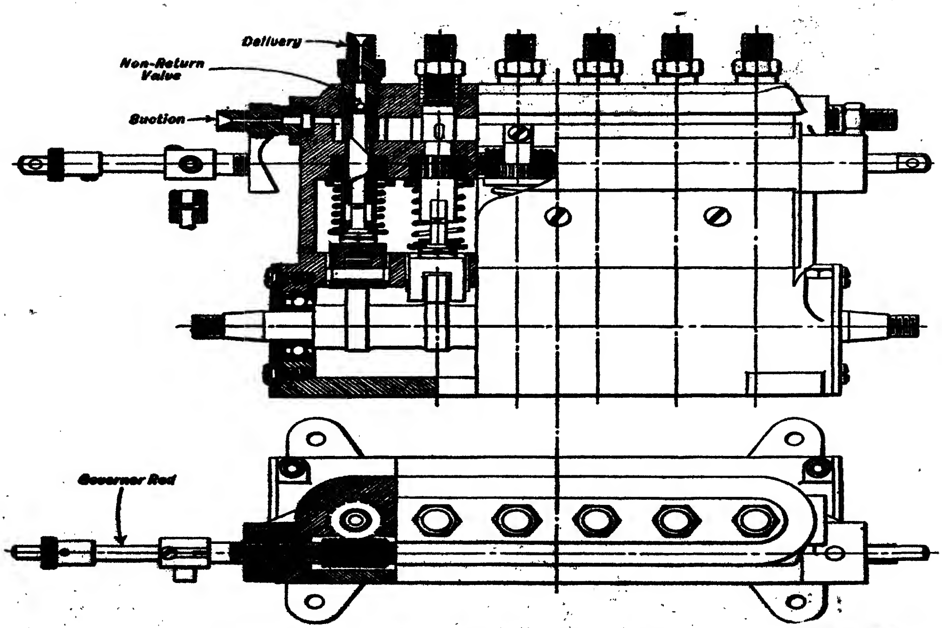
FIG. 45. THE FUEL PUMP DELIVERY VALVE

with fluted guide portion, a circular groove cut in it, dividing the guide into two parts. The lower fluted portion has four longitudinal grooves which communicate with the circular groove. The upper part of the guide acts as a small piston made a ground fit in the seating guide. When the pump is on its delivery stroke, as the pressure of the fuel increases the valve lifts until the fuel can escape through the longitudinal flutes over the valve face and thence to the injection nozzle. As soon as the pump plunger releases the pressure in its barrel, the delivery valve—under the influence of the spring above and the fuel pressure difference—becomes seated again, causing the small piston parts of the guide to return with a plunger action, thus increasing the space in the delivery pipe (by an amount equal to the volume of the small piston part of the valve guide) before the valve actually seats itself. The effect of this increase in volume in the delivery pipe system is to suddenly reduce the pressure of the fuel therein so that the nozzle valve in the injector can rapidly seat itself, thus giving a very quick cut-off of the fuel spray and eliminating "dribble."

Control of Power Output. The engine power output is controlled by varying the quantity of fuel injected per stroke of each fuel pump plunger. The word "Stop" and an arrow engraved on one end of the pump casing in line with the control rod 107*d* (Figs. 42 and 44) indicate which way the control rod should be moved to stop the engine. The corresponding position of the pump element is shown at (5) in Fig. 43. For engine starting purposes the control rod is moved to the right so that the pump elements are as shown at (1) and (2), Fig. 43; at this point the plungers are actually delivering more fuel than required by the engine at full load—in order to ensure easy starting. The positions of the pump elements for normal output are shown at (3) and (4), Fig. 43. When the engine starts, the control rod must be released to the normal or a lower speed position; to ensure this a trip collar is often provided. Unless the control rod is released the exhaust will be smoky and there will be excessive carbon formation in the combustion head.

The fuel pump described can be fitted with a hand-operated control for advancing and retarding the fuel injection to facilitate starting or to select the best instant of injection to suit the particular engine. Very few engine manufacturers encourage the use of such hand controls, however, the usual procedure being to fix the injection timing after having ascertained the best position for maximum power and a clear exhaust.

The Benes Fuel Pump. Fig. 46 illustrates this British design of fuel pump. It belongs to the "one plunger



(*The Engineer*)

FIG. 46. THE BENES FUEL PUMP, SIX-CYLINDER TYPE

(T.875A)

unit per cylinder" class, giving a sharp cut-off with freedom from "dribble." It has no complicated non-

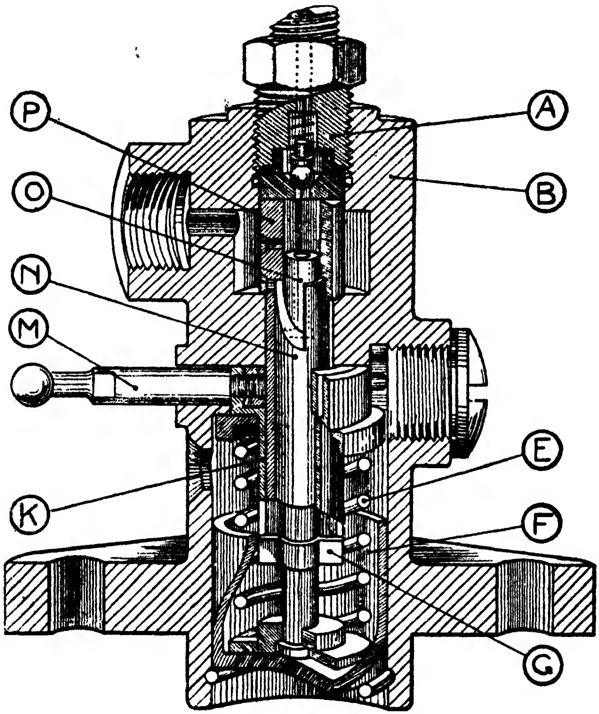


FIG. 47. THE BENES FUEL PUMP

return valves, and will give injection pressures up to 5,000 lb. per sq. in.

Referring to the sectional view of one plunger unit, given in Fig. 47, there is a plain cylindrical plunger *O* working in a sleeve *N* provided with inclined faces, and a barrel *P*; the sleeve is fixed relatively to this barrel. There is also a two-piece sleeve *K* mounted so that it

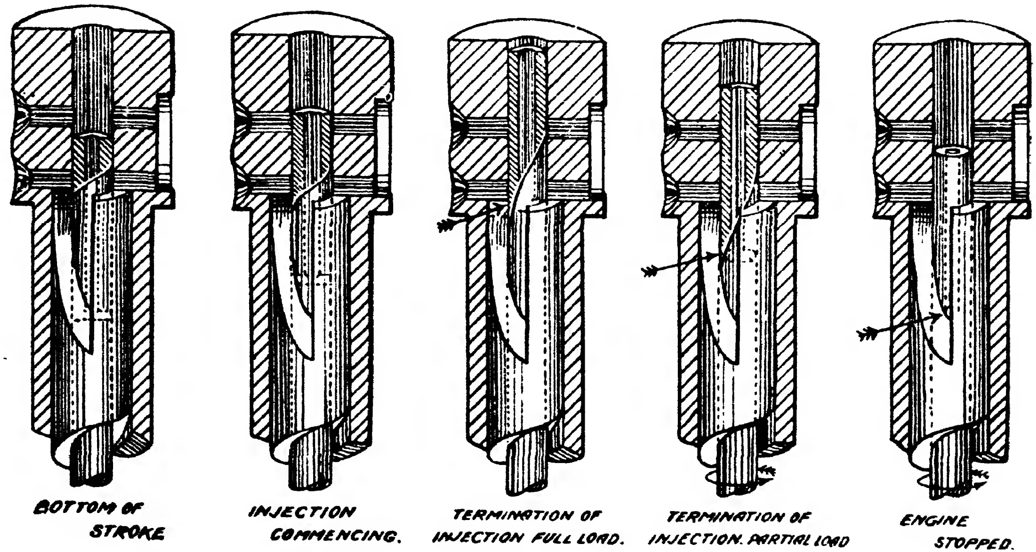


FIG. 48. ILLUSTRATING THE OPERATION OF THE BENES FUEL PUMP

can rotate about the barrel *P*. This sleeve is controlled by means of a rod *M*. The sleeve *K* is provided with slots at its lower end in which projections *G* on the plunger engage, so that rotation of the sleeve *K*, by means of the arm, also rotates the plunger, the slots permitting the plunger to move up and down whilst being rotated also. The plunger is returned, on its down stroke, by means of the spring shown at *E* and is guided accurately by the member *F*. The spring loaded non-return valve is shown at *A*. The complete unit is contained in an outer housing *B*.

The plunger is provided with an axial hole communicating with a radial hole forming relief ports, drilled through the plunger (as shown by the dotted lines in Fig. 48) and with the suction chamber formed by the space in the housing surrounding the top part of the barrel. The spaces formed by the inclined faces on the sleeve *N* are also in direct communication with the suction chamber.

The Bryce Fuel Pump. This fuel pump, which has been developed for stationary, marine, and motor vehicle engines, operates upon the constant stroke principle, with variable by-pass; it employs a cam-operated spring-returned plunger for each cylinder supplied.

A sectional view of the plunger is given on the left in Fig. 49; it will be seen that it is drilled along its axis at *a* and has two other holes *b* and *c* communicating with *a*; the lower hole *c* meets an annular groove *d* around the plunger. The pump barrel shown in the centre and right-hand diagrams has two triangular ports *e* situated upon opposite sides; these communicate with the bore of the barrel and the suction side of the pump, respectively.

The operation of the pump (Fig. 50) is as follows: When the plunger is at its bottom end of stroke *A*;

the ports are uncovered and fuel flows to fill the space above the plunger. When the delivery stroke occurs fuel is first displaced back through the ports until the plunger reaches the position *B*, when the fuel above

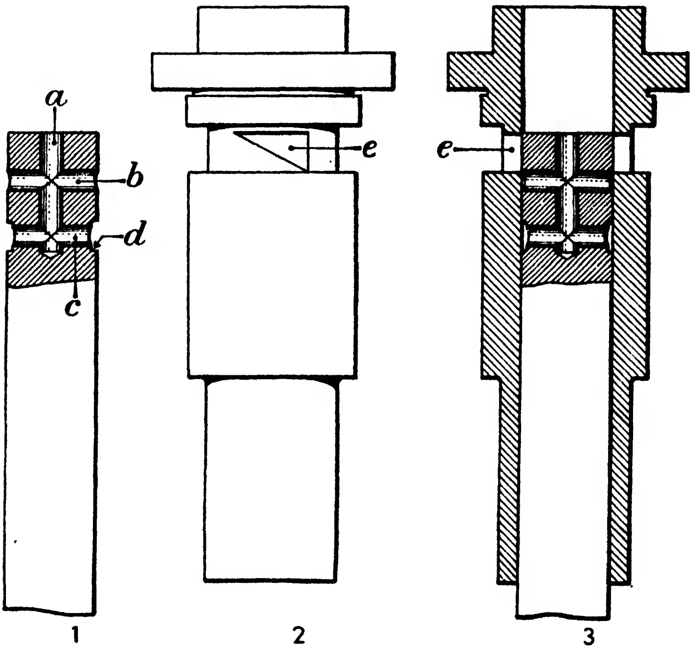


FIG. 49. THE BRYCE FUEL PUMP PLUNGER AND BARREL ELEMENTS

it is forced along the delivery pipe to the injector. The fuel is delivered continuously until the pressure is released by the holes *a* and *b*, when *b* rises above the sloping edge of the ports as in *C*. Before the hole *b* completely passes above the ports the annular groove *d* reaches the lower corners of the triangular ports (as in *D*) and by-passing of the fuel continues through the holes *a* and *c* until the plunger reaches the end of the delivery stroke *E*.

By rotating the plunger the hole *b* can be moved so that it will rise above the sloping edge of the port

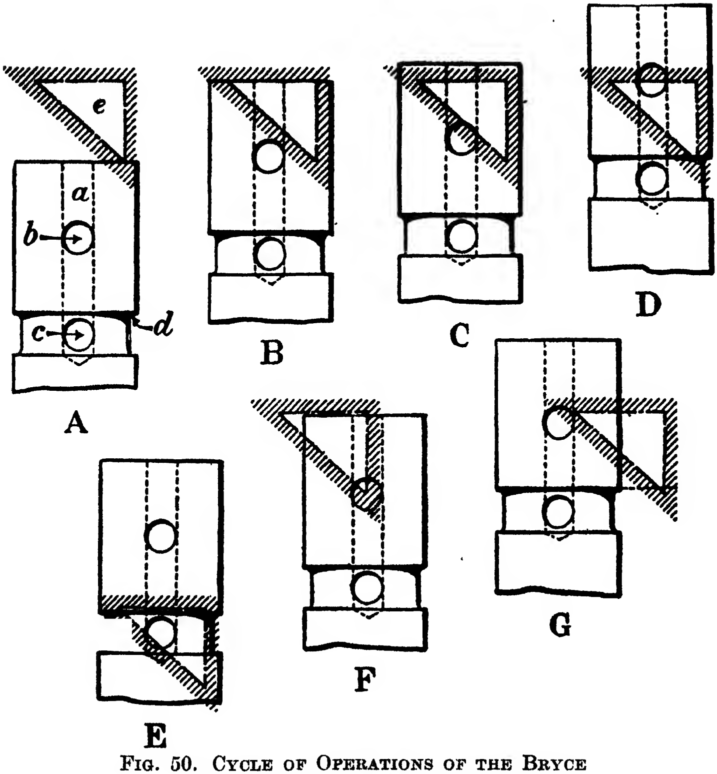


FIG. 50. CYCLE OF OPERATIONS OF THE BRYCE FUEL PUMP

earlier or later in the delivery stroke, and by this means the fuel delivery to the injection nozzles can be regulated. When the plunger is not required to deliver fuel, it takes the position shown at *F*; when delivering the maximum quantity of fuel, at *G*. The plungers, it should be mentioned, are rotated, for fuel regulation purposes, by means of a rack and pinion, and individual adjustment is provided on each pump element.

In connection with the delivery side of the pump a special design of delivery valve, having no spring, gives a practically instantaneous pressure release after each injection, so that no "dribbling" occurs at the injection nozzle.

The Simms Fuel Pump. This well-known fuel pump operates upon the jerk pump principle and has a separate plunger and delivery valve per engine cylinder. The plunger—as in many other fuel pumps of its class—is cam-operated and spring returned. The pump works upon the *uniflow* or constant directional flow principle.

The quantity of fuel delivered by the plunger is regulated by rotating the latter so as to give variable cut-off or spill to the fuel.

Fig. 51 shows the pump in end sectional view, with the end plunger at the highest part of its stroke—as shown by the cam position. The pump working chamber has been filled with fuel drawn into it during the previous suction stroke. The upward stroke of the plunger increases the fuel pressure and forces the fuel past a valve into the fuel injection nozzle system; a complete charge is thus delivered to the fuel injection nozzle.

In order to regulate the amount of fuel delivered, by foot accelerator or governor control, it is arranged for an earlier cut-off of the fuel by rotating the plunger with a rack and pinion movement. This causes a helix on the plunger to be brought into coincidence, at a certain point of the stroke, with the helical edge of a port cut in the upper end of the plunger's barrel. Thus, fuel trapped in the working chamber spills into the

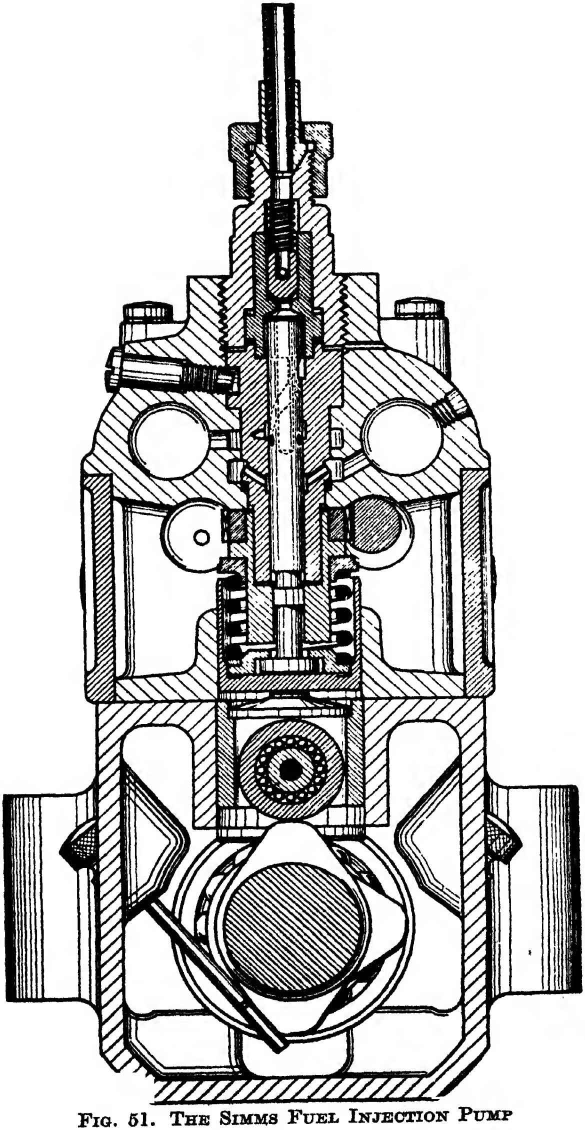


FIG. 51. THE SIMMS FUEL INJECTION PUMP

helical channel of the plunger whence it is ejected at high velocity into the spill annulus.

During the following down stroke of the plunger suction occurs and fuel enters the suction annulus by way of the ports provided and when the plunger is near its lowest position this fuel passes into the axial suction duct of the plunger through radial ports, whence it is delivered to the working chamber. The flow of fuel, it will be observed, is always in the same direction.

FUEL INJECTION NOZZLES

The purpose of the injection nozzle is to supply the measured amount of fuel to the air charge in the combustion chamber in such a state that it will be burnt as efficiently as possible, the fuel supply being promptly cut off at the end of the injection period without any dripping—or "dribble."

The design of the injection nozzle depends upon the size and shape of the combustion chamber and the method employed, e.g. air turbulence, air cell, precombustion or direct-injection systems; each combustion method usually requires its own special design of nozzle.

In certain cases, however, the injection nozzle unit apart from atomizing the fuel must also time the period of injection and measure the amount of fuel injected. This is known as the mechanically-operated type, a typical example being that used in the American Cummins engine.

The majority of injection nozzles used in British oil engines are of the hydraulic pattern, having a conically seated valve held in place by a calibrated compression spring. The fuel is delivered at a high pressure, namely from 1,000 to 3,000 lb. per sq. in. by the fuel pump, which "times" the moment of fuel injection and measures the quantity of fuel supplied to the injection nozzle. The hydraulic pressure of the fuel thus delivered is greater than the spring load pressure on the valve, so that the latter is lifted from its seating and the measured amount of fuel is sprayed into the combustion chamber under the fuel pump injection pressure. This form of injection nozzle, of which the C.A.V.-Bosch ones are typical examples, is known as the *Closed Type*.

The other kind of nozzle used on high speed Diesel engines is known as the *Open Type*, and has its supply of fuel controlled entirely by the fuel injection pump; there is no controlling valve—as in the closed type—to stop the flow of fuel from the nozzle.

SELECTION OF FUEL INJECTION NOZZLES

In regard to the selection of the most suitable type of nozzle, it has been shown that in the case of *auxiliary turbulence chamber* and *pre-combustion chamber* engines, a *plain single hole nozzle* is quite satisfactory; the fuel is broken up into very small particles and burnt efficiently in such engines. The principal advantages of such single hole nozzles are their reliability and comparative freedom from carbonizing; moreover, the fuel injection pressures are relatively low.

In the case of *direct-injection engines*, where little turbulence exists, the injection nozzle must not only atomize the fuel efficiently, but also provide the correct shape of spray and direct this spray to the proper parts of the combustion chamber in order to obtain intimate admixture with the air charge. For these reasons the injection nozzles are usually of the *multiple-hole* type, the holes being either symmetrical or offset to suit the combustion chamber shapes or characteristics.

Much higher fuel injection pressures are generally required for nozzles used in direct injection engines, since the atomizing process must be as complete as possible and the pressure employed sufficient to project the fuel particles to the farthest parts of the combustion chamber. Before proceeding to describe typical nozzles some experimental properties of injection nozzles will be considered.

Some General Test Results. In connection with the design of fuel nozzles, the results of experimental work indicate fairly conclusively that *the penetration* of the spray into the compressed air charge *increases with increasing ratio of the length of the fuel valve orifice to its diameter.*

The penetration also increases with the injection pressure for a given charge compression pressure.

Further, in regard to the *atomizing of the fuel*, the best results appear to be obtained by *decreasing the ratio of the orifice length to its diameter.*

The penetration effect also depends upon the *direction of the fuel spray*, and upon the *specific gravity of the fuel.* In the latter connection the heavy fuels are usually the more viscous, and are not so readily atomized; this makes the spray angle narrower but helps to produce greater penetration.

In connection with the relative merits of open and closed nozzles it has been shown, from tests made with fuel pumps and pipe lines in which these nozzles were included, that pressure variations exist in the pipe lines, such that after the pump delivery valve has closed these pressure fluctuations continue.

Under these circumstances the closed type of nozzle prevents delayed injection of the fuel whereas the open type is subject to such injection, sometimes for periods of 15° to 20° of crank angle after the pump supply ceases; this results in "dribble" at the mouth of the open nozzle.

TYPICAL FUEL INJECTION NOZZLES

As the quantity of fuel delivered per working stroke of the engine—in the case of automobile Diesel engines—is very small, namely, of the order of 0·005 to 0·015 cub. in., the sizes of the delivery holes in fuel injectors are also very small. The usual diameters of these holes lie between 0·010 and 0·004 in. and their lengths between 0·015 and 0·05 in.; under these circumstances absolute cleanliness of the fuel is essential.

The Open Type Nozzle. As this type is seldom used in modern Diesel engines it will suffice to mention a single example, namely, that employed in connection with the Ricardo rotational swirl system, shown in Fig. 20. This design of nozzle has a single hole which is left open and therefore uncontrolled by any valve, although a light spring-loaded ball valve is employed to prevent the cylinder pressure from acting upon the fuel in the pipe line system; this can be seen just below the shut-off screw down valve in the illustration.

The Closed Type Nozzle. On account of its advantages over the open and most other types, this form of nozzle is of greater importance in automobile engines and is now used almost exclusively on the latest designs of Diesel engines.

The closed nozzle is made in several different patterns to suit the various designs of engines in common use. All of these nozzles, however, have the same general construction, consisting of a nozzle body and valve (Fig. 52). In the C.A.V.-Bosch design the nozzle valve is a circumferentially grooved barrel which, after being case-hardened is ground to fit the nozzle body to a high degree of precision so that it will work freely but without leakage. At one end of the nozzle valve a stalk or "pen" is provided, whilst at the other end it is reduced in diameter to produce a stem upon which a valve face is turned. Fuel is fed into the mouth of the nozzle through small channels bored vertically in the nozzle body which terminate in an annular reservoir or "gallery" just above the valve seat. The effect of the fuel pump pressure is to cause the fuel in the gallery to act upon the lower

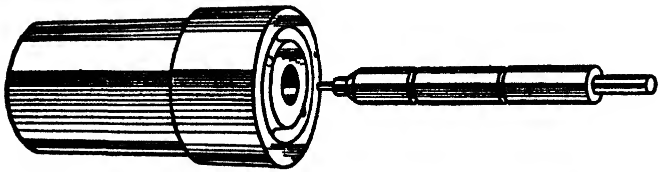


FIG. 52. INJECTOR NOZZLE BODY AND VALVE ELEMENTS

surfaces and to lift the valve; the valve pin above is subjected to spring pressure of predetermined amount.

The principal types of closed nozzle include (1) Pintle, (2) Single Hole, (3) Delay, (4) Multiple Hole and Long Stem Nozzles (Fig. 53).

Pintle Nozzles. In this design, which is used in certain "air cell" and pre-combustion chamber engines, the stem of the nozzle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle body. This type of nozzle can be made to give a spray of hollow conical form from an included angle of 4° upwards, to suit the requirements of the engine.

Single Hole Nozzles. In these the conical faced valve closes a single hole of cylindrical form bored centrally in the nozzle body. This hole can be made of suitable diameter, from 0·008 in. upwards to suit the size and design of the particular engine. A variant of the symmetrical single hole nozzle is the type in which the exit hole for the fuel is bored at an angle to the axis of the nozzle; this is known as the *Conical End Single Hole Nozzle*; it gives a fuel spray obliquely to the nozzle axes.

Delay Nozzles. These are designed to give modified

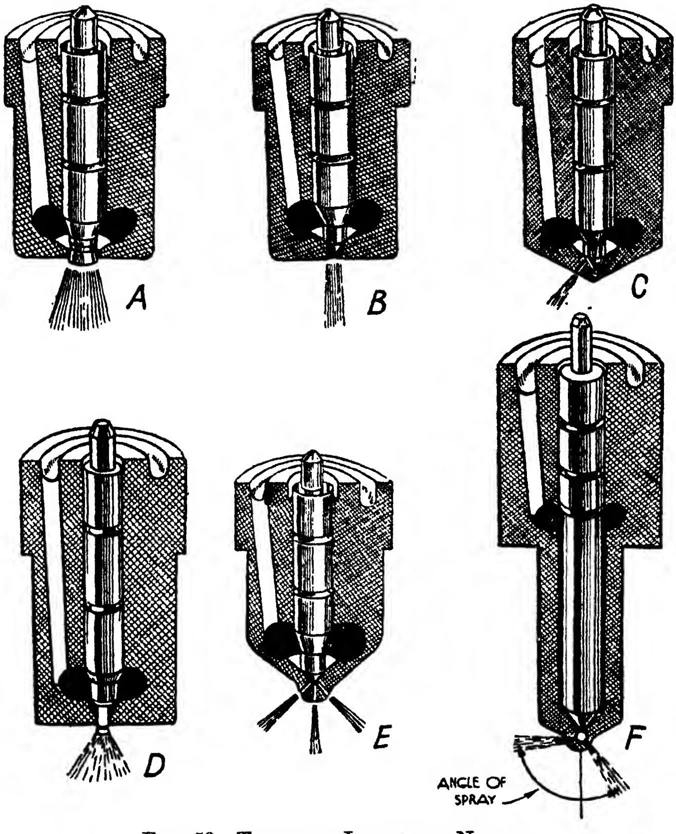


Fig. 53. Types of Injection Nozzle

A = Pintle
B = Single Hole
C = Conical End (Oblique Spray) Single Hole
D = Delay
E = Multi-hole
F = Long Stem (Oblique Spray) Multi-hole

spray characteristics for engines of the pre-combustion kind, in order to obtain a more stable performance when the engines are idling. To achieve this end, the pintle is modified so that the rate of injection increases towards the end of the delivery. The effect of this is to lengthen the periods of injection at idling speeds without affecting combustion at higher speeds.

Multiple Hole Nozzles. As the name indicates, these have more than one exit hole; any number of holes up to seven can be provided in the dome under the nozzle mouth. These holes are arranged radially in a circle with even pitch, about the nozzles' axis. The holes can be provided from 0·008 in. upwards by steps of 0·020 in., whilst the hole angle can be varied from 20° upwards to suit the design of engine.

Long Stem Nozzles. These are intended for direct injection engines where, owing to limited space between the valves and the cylinder head it is not possible to provide for cooling the nozzle, effectively.

Typical Timed Valve System. The method of fuel injection employed on the American Cummins Diesel engine is a typical example of the "timed valve" or "common rail" system. It utilizes a separate fuel pump and distributor device—consisting of a rotating valve unit—to supply fuel at a relatively low pressure, namely, from about 80 to 120 lb. per sq. in. to the supply side of a mechanically operated plunger device situated in the head of each cylinder. The single plunger pump supplying the fuel to the injection pump is given a variable stroke—operated either by hand or governor—so that in the maximum stroke position the pump can only deliver as much fuel as the air can burn in the cylinder. The direct-injection method of combustion is employed in this engine.

The cylinder head plunger (Fig. 54) injects the fuel through a wide-angle spraying nozzle, the shape of

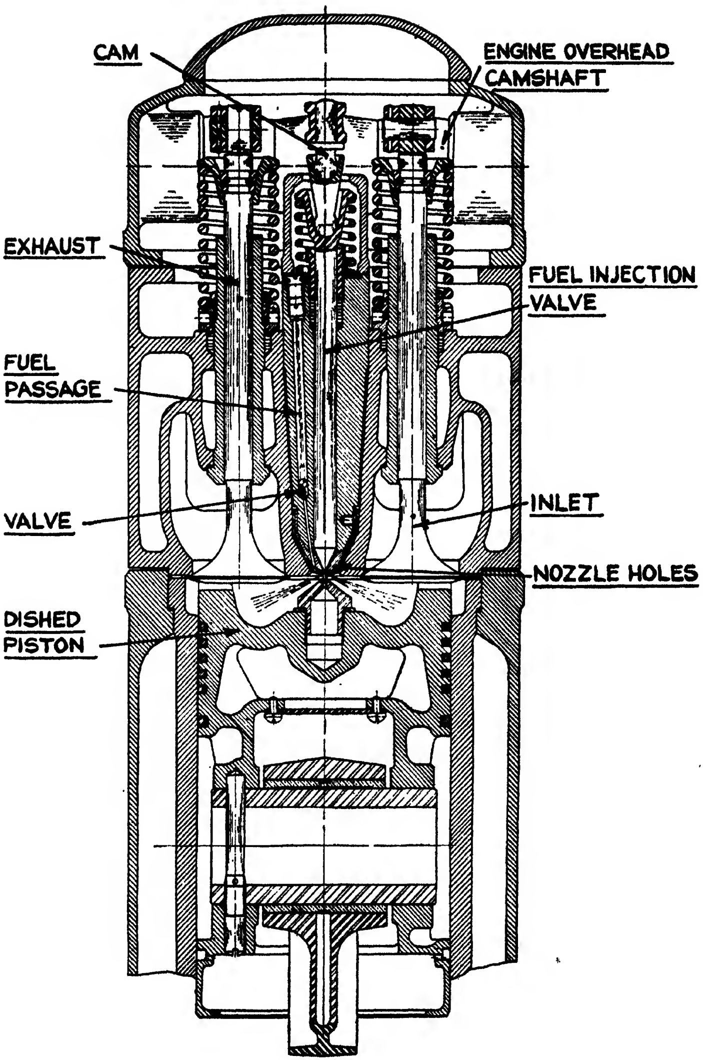


FIG. 54. THE CUMMINS CYLINDER HEAD

the spray being arranged to suit the shape of the air chamber formed in the piston head. In addition the latter has a plug in its centre to assist in deflecting the fuel spray. It should be mentioned that, apart from the rotary distributor unit there are no valves in the

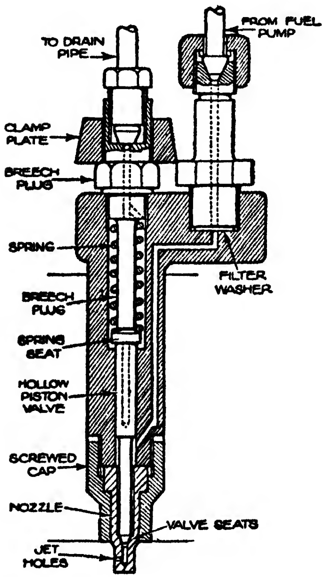


FIG. 55

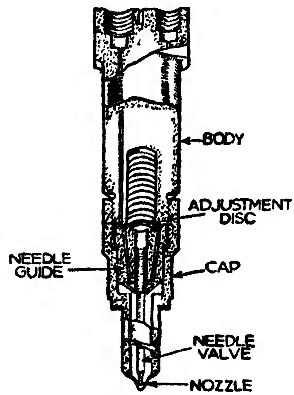


FIG. 56

fuel delivery and injection system, for the cylinder plunger merely has a conical end seating in the injection nozzle; the cone is lifted each time injection occurs.

Two typical examples of multi-hole injection nozzles are given in Figs. 55 and 56, for the Gardner and Armstrong-Saurer engines, respectively.

The Gardner nozzle which is used in conjunction with a direct-injection engine with shaped piston crown is of the hydraulic spring loaded type, the fuel under pressure being led from the fuel pump down the passages shown on the right to the conically seated valve and thence through a series of fine holes into the combustion chamber. Any fuel that may leak past the piston valve is led away by the drain pipe shown on the upper left-hand side. The nozzle for the LW type Gardner engines operates with an injection pressure of 2,000 lb. per sq. in.

The injection nozzle shown in Fig. 56 has four symmetrically disposed spraying holes, each of 2·5 mm. (i.e. about 0·01 in.) diameter. The nozzle is used in conjunction with the "dual turbulence" engine previously described in this section. With this nozzle fuel is sprayed at an oblique angle to the injector's axis. The opening pressure on the valve can be altered by adjusting the compression of the spring shown. The fuel supply to the valve is directed from the top left-hand screwed connection, down the small channels shown on the left.

Notes on Fuel Injection Nozzles. In connection with the fitting of injection nozzles to Diesel engines, two important points must be taken into consideration, namely, (1) the cooling of the nozzles, and (2) their accessibility for removal and replacement purposes.

Since the nozzles operate under high temperature conditions, they should be adequately jacketed for water-cooling or well-ribbed, if of the air-cooled pattern. The design of the cylinder head or combustion chamber in which the fuel injector is located should make the necessary provision for such cooling.

Normally, fuel injectors require removal for inspection or cleaning purposes about every 2,000 to 3,000 miles, so that they should be mounted on the cylinder or head in such a manner that they can readily be removed with a spanner or other suitable tool, without any difficulty.

The complete nozzle unit should have a fuel leak-off pipe leading away from the union near the top of the nozzle holder to drain away any small leakage that may occur past the valve guide part of the nozzle.

Most fuel injectors of the spring-loaded hydraulic pattern are provided with an extension of the valve spindle, consisting of a rod passing through the centre of the compression spring and its adjusting screw to the outside of the injector unit cover. This extension is provided with a small milled screw and is termed the "feeling pin." When the injector is working correctly the feeler pin pulsates, its action being readily felt by placing the finger on the pin; if these pulsations are erratic in nature, or absent altogether, it is a sign of a faulty injection nozzle. The feeling pin is therefore a useful indicator of the injection nozzle's behaviour.

COMPLETE INJECTOR UNIT

The complete Bosch fuel injector unit is shown in Fig. 57. In this case the end of the holder has a highly ground face which forms a joint with the flange of the nozzle body 150*a* by means of the nozzle cap nut 151.

Fuel is fed by the piping connected to the fuel inlet connection 166, through a tunnel in the barrel of the nozzle holder which terminates in an annular semi-circular groove at the ground face of the flange of the nozzle body 150*a*.

The nozzle valve 150*d* is kept on its seat by means of the valve spring 154 and the spindle 152. The pressure at which the nozzle valve will lift depends upon the amount of compression placed upon the spring 154, which is adjustable by means of the compression screw 158. A feeling pin 160 passes through the centre of the compression screw, which, as previously stated, enables the functioning of the nozzle valve to be felt with the finger while the engine is running.

Any slight leakage of fuel which may accumulate above the valve can be led away to a drain tank by means of a pipe connected to the leak-off nipple stud 162.

One important *practical point* in connection with the

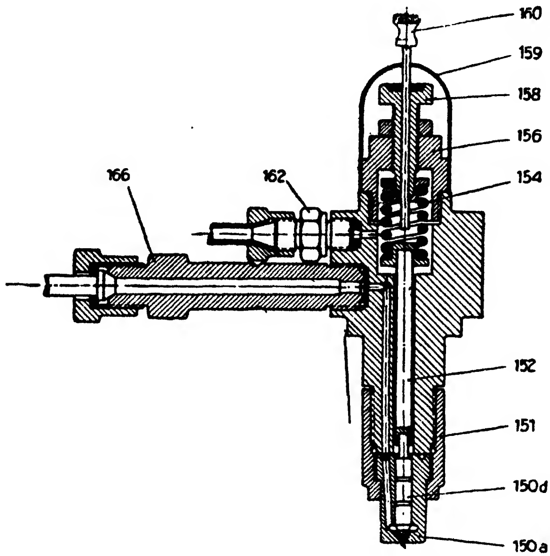


Fig. 57. The Complete Bosch Fuel Injector Unit

fitting of this nozzle holder into the cylinder head is to ensure that the lower end of its barrel is adequately cooled; a copper tube is often placed around the nozzle holder and the water in the cylinder head arranged to circulate around this tube.

CONTROLLING THE POWER OUTPUT

As has already been shown, the power output of a high-speed Diesel engine is controlled by varying the quantity of fuel injected; this is analogous to throttling the petrol-air supply of a petrol engine. All modern fuel pumps are provided with means for altering, between certain limits, the quantity of fuel pumped, a control rod or lever being provided for this purpose;

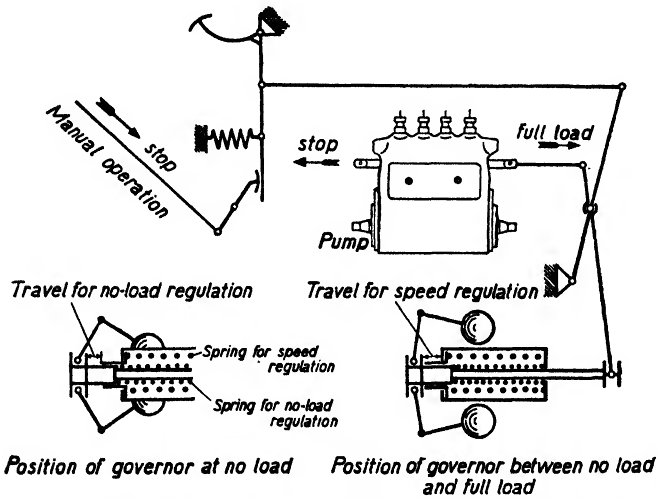


FIG. 58. SHOWING LAY-OUT OF FUEL GOVERNING SYSTEM TO CONTROL MAXIMUM AND "NO-LOAD" SPEEDS

examples of such controls have been mentioned in the descriptions of fuel pump given earlier.

The fuel pump control must also be provided with a means to shut off the fuel supply entirely, in order to stop the engine; this is the equivalent of the magneto switch in the case of the petrol engine. Many Diesel engines are fitted with centrifugal governors to limit the maximum speed, and in some cases to control the lowest idling speed, in order to prevent the engine from stopping.

The fuel delivery is sometimes controlled entirely by hand and foot "accelerator" pedal, in the case of certain motor vehicle engines. Fig. 58 shows one convenient method of governing a high-speed Diesel engine to provide regulation for both maximum speed and "no-load" conditions, whilst at the same time permit-

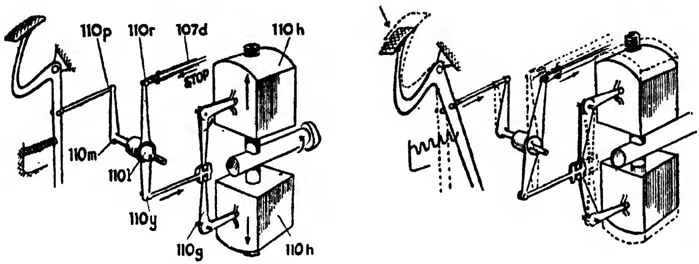


Fig. 59. Principle of Centrifugal Governor

ting the driver's manual control to operate between these two load limits; the arrangement shown is applicable to the Bosch fuel pump controls.

A Typical Centrifugal Governor. The principle of the C.A.V.-Bosch governing device, which is widely used on British engines, is given diagrammatically in Fig. 59. It will be seen that the two weights 110*h* are mounted on an extension of the fuel pump camshaft and that the bell-crank levers 110*g* are attached to these weights; these levers are in turn connected to the floating lever 110*r*. When the engine rotates slowly, the weights tend to fly outwards so as to move the control rod of the pump towards the "stop" position, thus slowing down the engine. These weights are retained by springs so that if the engine speed falls they move the weights inwards when the delivery of fuel is increased.

. The general lay-out of the complete fuel pump and governor is given in Fig. 60, a key to the various parts being given in the caption beneath; the parts marked in Fig. 60 are mostly the same as those given in Fig. 59.

With this type of governor the driver's accelerator pedal operates the fuel pump control rod independently

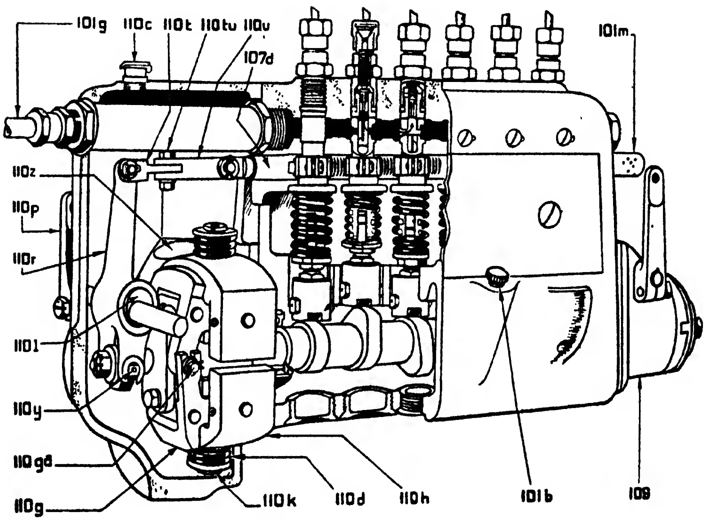


Fig. 60. The C.A.V.-Bosch Six-cylinder Fuel Pump with Governor

101*b* = Lubrication Dip Stick
101*g* = Fuel inlet connection
101*m* = Control rod stop
107*d* = Control rod
109 = Advance device
110*c* = Oil cup
110*l* = Eccentric
110*p* = Control lever
110*r* = Floating lever
110*t* = Outer link fork
110*d*, *e*, *f* = Governor springs
110*g* = Bell crank levers
110*ga* = Bell crank lever pin
110*h* = Fly weights
110*k* = Adjusting nut
110*tu* = Screw for link forks
110*u* = Inner link fork
110*y* = Coupling Crosshead pin
110*z* = Closing plug

of the governor. Thus when the pedal is depressed, the lever 110*p* (Fig. 59) and eccentric 110*l* are turned to the right; the floating lever 110*r* is thus moved by a corresponding amount and the control rod drawn from the "stop" position to accelerate the engine, irrespectively of the position of the governor weights.

The centrifugal governor in question employs no less than three concentric compression springs inside each governor weight in order to effect both the slow running and maximum speed controls.

Suitable adjustments in the form of screw control rod stops, and screw stops on the body of the governor housing itself, enable the maximum and minimum speeds to be controlled to the desired limits.

The Pneumatic Type Governor. The centrifugal governor, whilst satisfactory for the larger automobile Diesel engines is open to the objection of additional bulk and weight in the case of smaller engine models. In order to save both space and weight in such engines the pneumatic type was introduced; it subsequently proved to give a greater degree of flexibility, in controlling the very low and the maximum speeds, than the centrifugal pattern.

The principle of the C.A.V.-Bosch governing device which operates upon the pneumatic method is illustrated in Fig. 61. In this case speed control is effected by means of negative pressure variations in the throat of a venturi tube fitted into the main air intake to the engine as shown in the upper part of Fig. 61. It is claimed that this method does not involve any loss of volumetric efficiency.

The venturi unit in question consists of a flange for fitting it to the engine air suction pipe, the venturi proper and an air cleaner device (shown on the right).

At the smallest throat diameter of the venturi is fitted the butterfly valve *C* for controlling the air flow and actuated from the accelerator pedal through the lever *A*; the amount of movement for maximum and idling speeds being governed by two adjustable stops at *J* and *K*. On the same centre line as the butterfly valve is a screwed connection *D* through which the air is exhausted from the diaphragm unit actuating the pump control rod.

The diaphragm unit consists of a housing *G* mounted

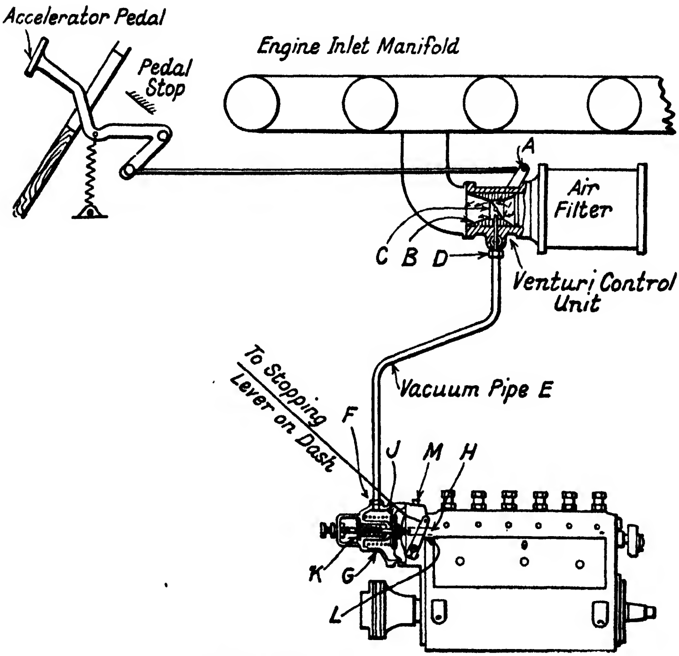


FIG. 61. ARRANGEMENT OF PNEUMATIC GOVERNOR

A = venturi valve control lever
B = venturi throat
C = venturi valve
D = vacuum pipe union
E = vacuum pipe
F = diaphragm housing union
G = diaphragm housing
H = main housing
J = diaphragm
K = main diaphragm spring
L = stop lever
M = oil cap

on a separate casting *H* with the special leather diaphragm clamped between the two, and providing an air-tight compartment within the housing *G*. A light spring *K* acting on the diaphragm is provided in order to damp out any undue oscillations in vacuum that may occur, and tends to keep the control rod in full open position, an additional lever *L* being provided for stopping the engine and operated from the dash-board by a steel wire or similar control.

The operation of the governor is as follows—

With the engine stationary and the lever *L* released, the spring *K* forces the control rod into the full-load position. Then by pressing the plunger on the excess-fuel device the control rod is allowed to open automatically to the extent of its travel and so provide an excess of fuel to be available for starting. When the engine has started it may be idled by releasing the accelerator pedal and thus closing partially the butterfly valve *C* when high vacuum is created in the connecting tube *E* and air-tight compartment *G*. As the air in the compartment *H* is at atmospheric pressure and therefore now in excess of that in compartment *G*, the diaphragm together with the control rod is moved towards the "Stop" position until the engine is running at the pre-determined idling speed required. The movement of the control rod towards the idling position releases the plunger of the excess fuel device which returns to its original position and forms a positive stop to the control rod, preventing it returning to the starting position.

To increase engine speed, the accelerator pedal is depressed, opening the valve *C* and so decreasing the air velocity past the mouth of the connecting tube. This results in an increased pressure in *G*, and the movement of the control rod towards the maximum-speed position. A maximum-speed stop is therefore provided in order to prevent the valve moving beyond this position, as further movement will tend partially to close the opening, thereby reducing rather than increasing the speed. It should be noted that the venturi in the valve unit amplifies the vacuum normally prevailing in the inlet manifold, but this is generally less than the vacuum in the air-inlet valves, so that no loss of efficiency results.

AUTOMOBILE ENGINE PRACTICE

The high-speed Diesel engine is now widely used on commercial and passenger vehicles, having to a large degree displaced the petrol engine on account of its lower fuel and maintenance costs.

Although for goods vehicles the petrol engine is still much used in the 1 to 3 ton class, yet in the 5 ton and over classes the Diesel engine is now almost exclusively used.

The most favoured types are the four- and six-cylinder ones rated at 25 to 45 h.p., but developing about three times the rated h.p.'s at their maximum outputs.

The ordinary maximum engine speeds vary from about 1,700 to 2,500 r.p.m., but certain examples of the smaller engines (15 to 25 h.p. rating) operate at speeds up to 3,500 r.p.m.; in one instance the maximum working speed slightly exceeds 4,000 r.p.m.

The B.M.E.P. values attained in the later engines range from about 90 to 120 lb. per sq. in. (maximum).

The fuel consumptions of these engines, under the most economical running conditions, vary from about 0·36 to 0.43 lb. per b.h.p. hour.

In regard to the minimum or idling speeds, these are rather higher than for the petrol engine, the actual values ranging from 300 to 400 r.p.m. as regulated by the governor idling control.

Regarding the weights of modern Diesel engines used on heavy vehicles these generally average about 12 to 16 lb. per b.h.p., but certain designs such as the A.E.C., which employs Elektron crankcases, weigh slightly less than 10 lb. per b.h.p.

Mention should also be made of the Perkins light six-cylinder Diesel engine, known as the "Panther," rated at 29·4 h.p., but developing over 100 b.h.p. which weighs only about 6 lb. per b.h.p. This engine has been fitted, experimentally, to motor cars with satisfactory results.

The smaller models of Diesel engines have also been fitted, commercially, to certain makes of motor-car, of which the Citroen and Hanomag cars are examples; each of these cars is fitted with a Diesel engine of about 1·6 litres capacity.

Actually, the Diesel engine does not appear likely to challenge the petrol engine either in the motor-car or light aircraft fields, owing to the high standard of development of the petrol engine, due primarily to the use of high octane value fuels which permit the use of higher compression ratios.

Another rival to the Diesel engine in these fields is the medium compression, fuel injection, spark ignition type, in which an intermediate value of the compression, between petrol and Diesel engine practice is used. The fuel is injected into the direct pattern combustion chamber, but owing to the compression heat being insufficient to ignite the fuel an electric spark is employed as in normal petrol engine practice for this purpose.

In regard to the performances of automobile type Diesel engines it may be mentioned that on 28th April, 1937, an adapted six-cylinder A.E.C. engine, of the type employed upon motor-buses, broke a world's speed record at 106·48 m.p.h. The engine used had a bore of 115 mm. and stroke of 146 mm. giving a capacity of 9,090 c.c.

Lower Fuel Consumption and Costs. We have already stressed the fact that the Diesel engine has a much lower fuel consumption per horse-power than the petrol engine. In this respect there is a saving in fuel of at least 25 per cent. In addition, owing to the present slightly lower cost of the fuel, there is a considerable saving in fuel costs.

Even with the Treasury tax of 8d. per gallon the

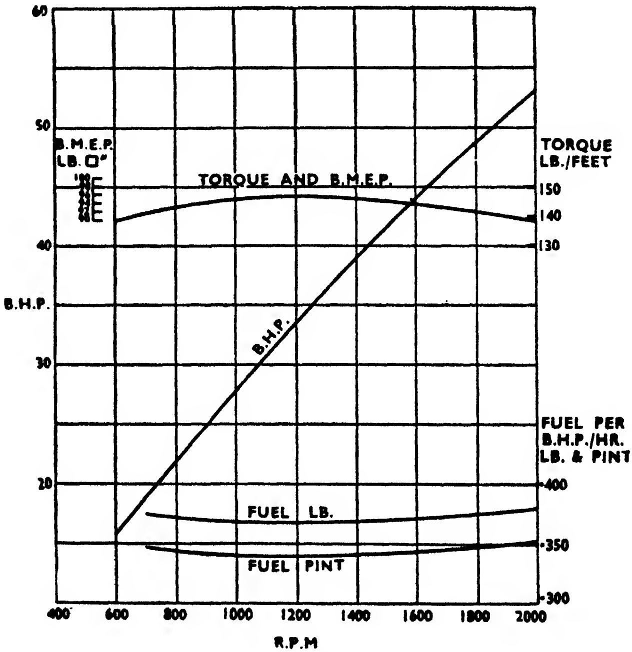


Fig. 62

Diesel engine still shows a 25 to 40 per cent advantage in reduced fuel costs per mile of road travel. There is another important fact, viz. that as a gallon of Diesel fuel weighs approximately 8·8 lb., as against about 7·3 lb. for petrol, and as these fuels are purchased by volume (i.e. by the gallon) there will be a greater number of B.T.U.'s in the former case, per gallon, and therefore a greater mileage per gallon.

Better Engine Torque. The engine torque curves for Diesel engines, when plotted on an engine r.p.m. base, are flatter and are more nearly constant over a given speed range than those of the petrol engine. Moreover, the Diesel engine has a higher torque at lower speeds so that the top gear speed range of Diesel-engined vehicles is appreciably better than for petrol-engined ones of similar maximum output.

The experience of drivers of Diesel-engine vehicles shows that these will pull better on top gear at low road speeds than petrol-engined vehicles; thus, it has been shown possible to turn Diesel buses around slowly, in top gear, on an ordinary road.

Starting Performance. Although some of the earlier Diesel vehicles were certainly more difficult to start from the cold, owing to the unsatisfactory design of the combustion chamber and fuel injection system, most modern engines are quite easy to start from the cold. In several cases decompression devices are fitted to enable the engine to be motored around electrically when starting. When the decompressor is cut out of action the engine at once commences to work. In many cases electric heating plugs (Fig. 63) are used in the combustion chamber for starting purposes; these are switched off when the engine starts.

These heater plugs consist of electric resistance elements which can be heated to redness by means of current from the low-tension battery of the vehicle. It requires about one-half to one minute's heating before an engine fitted with these plugs can be started from the cold; after starting the heaters are switched off.

The heater plugs are made in two patterns, namely, the *single* and *double pole* types (Fig. 63). The former, for a multiple cylinder engine are connected in parallel with a 2-volt battery. The latter are connected in series with a 12-volt battery. In most cases the heater plug circuit includes, besides the battery and switch, a resistance indicator having a light indicator to show

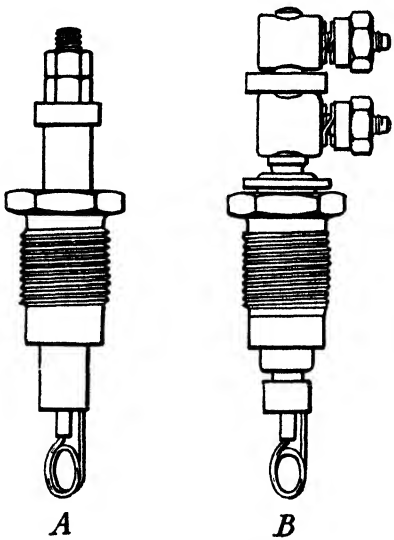


FIG. 63. TYPICAL ELECTRIC HEATING PLUGS USED FOR STARTING PURPOSES

when the heater plug switch is "on"; it is usually fitted to the instrument panel or dashboard.

There are certain British Diesel engines that can be started by hand in cold weather, without the use of heater plugs, more easily and in less time than a petrol engine of similar size. The principal advantage of the Diesel engine, however, is that it can be put under load almost immediately after it has been started, and it will pull satisfactorily whilst it is warming up. In the case of the petrol engine, however, it is necessary to allow several minutes for the engine to warm up sufficiently for it to pull satisfactorily. In very cold weather the petrol engine often "stalls" or misfires, due to the condensation of petrol in the inlet manifold; this trouble does not occur with Diesel engines.

Engine Temperatures. Owing to the better utilization of the fuel's heat energy, in the case of the high-speed Diesel engine, less heat is lost to the cooling water, and to the exhaust gases. The combustion chamber temperatures are lower and the exhaust valves are not subject to such high temperatures as in the case of petrol engine ones. There is also a reduced tendency for the pistons and cylinders to distort under the influence of the combustion heat.

In general, smaller radiators are needed for Diesel engines on account of the lower cooling water losses.

Maintenance. In the matter of running attention and maintenance, the Diesel engine is superior to the petrol engine in most respects, due to the absence of carburettor, ignition unit, and sparking plugs. It is true that in place of the ignition unit we have the fuel pump, with its accurately-made small working parts. but with modern methods of workmanship, and with present materials the fuel pump is as reliable as the magneto.

As an example of its reliability and exceedingly small rate of wear, one may mention the case of a Bosch fuel pump that was tested after 80,000 miles of running on a lorry engine. It was shown that after this period of running the delivery of the pump had diminished by only 3 per cent at normal loads, by 1·5 per cent for maximum loads, and 25 per cent at idling speeds.

The fuel injection nozzles do not require such frequent attention as the sparking plugs of a petrol engine for there are no electrodes to be burnt away. The only trouble that is likely to be experienced is that of a dirty seating, due to solid matter in the fuel; this is a most unlikely trouble, however, as there are two or more efficient fuel filters in the fuel feed system.

In the matter of cylinder and piston wear, it has been the experience of several Diesel engine firms that the cylinders require re-grinding after about 45,000–50,000 miles as against 60,000 miles in the case of the petrol engine. On the other hand, the valves last longer and require re-grinding less frequently.

Reduced Fire Risk. Owing to the fact that the Diesel engine uses a fuel of comparatively high flash point as compared with petrol, there is practically no risk of accidental fires due to the ignition of inflammable vapour. For this reason the usual official precautions regarding the handling and storage of Diesel oil become unnecessary.

SOME DISADVANTAGES

Having outlined the principal advantages of the high-speed Diesel engine, it is only fair to give equal prominence to its present disadvantages. Actually, however, these are not insuperable ones, and in view of the relatively short period during which this type of engine has been in use, it is probably only a matter of time before they are overcome. In any case, the advantages of the Diesel engine outweigh considerably its disadvantages.

Cost. Generally speaking, the Diesel type automobile engine costs appreciably more than the petrol engine, due at present to the smaller numbers manufactured and to the fact that the experimental work costs are spread over a smaller number of engines. With the increased production of these engines, however, the difference in first costs will gradually diminish and eventually disappear.

Weight. Many of the earlier motor vehicle Diesel engines were appreciably heavier and bulkier than the petrol engine of similar output. Although the petrol engine can always be made lighter than the Diesel engine, a good deal of progress has recently been made in the weight reduction of the latter type, so that some of the latest models are of practically the same weight as petrol engines of equal output.

Lubricating Oil Consumption. Many of the Diesel engines now in use show appreciably heavier oil consumptions than those of petrol engines, due probably to actual combustion of the oil on the one hand and to dilution of the oil by the fuel on the other. This heavier oil consumption is now the subject of investigation with the object of bringing it down to figures comparable with the petrol engine. With the use of two or even three oil control rings per piston it has since been found possible to keep the oil consumption down practically to that of corresponding petrol engines.

Greater Cylinder Wear. Experience has shown, as we have previously mentioned, that the cylinders of Diesel engines wear rather more quickly than those of petrol engines; the latter generally last about 20–25 per cent greater mileage before re-boring is required. With the use of centrifugally-cast alloy cast iron and nitrided steel liners, the actual period of running before re-boring becomes necessary has been extended considerably, so that this is not now a serious item.

The usual amount of wear for hardened cylinder liners is now about 20,000 miles per thousandth of an inch wear for many indirect injection engines, giving about 100,000 miles for only 5-thousandths inch, after which wear occurs more rapidly.

On direct injection engines the wear is generally found to be greater, but with hardened liners an average value of 14,000 miles per thousandth inch of wear is attained.

MISCELLANEOUS ITEMS

In some cases, where the injection timing has been over too long a period or has been too late in cut-off, smoky exhausts and pungent odours have resulted.

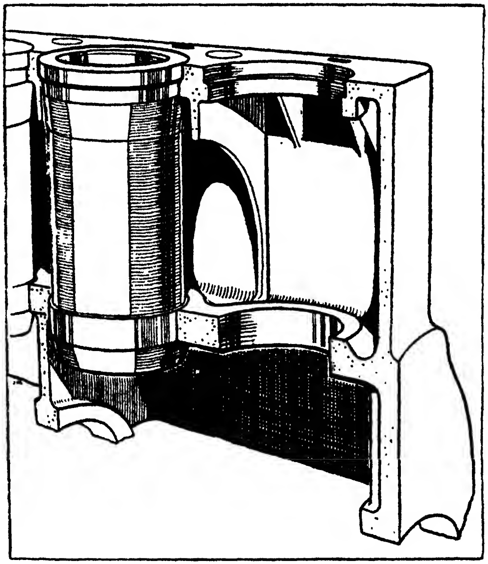


FIG. 64. A TYPICAL DIESEL CYLINDER LINER

This has been the cause of a certain amount of criticism on the part of the public, in the case of the earlier Diesel-engine omnibuses. With the later engines, however, it has been possible to obviate these difficulties by ensuring perfect combustion of the fuel; the Diesel engine exhaust is no longer open to this criticism.

In general, motor vehicles fitted with Diesel engines require rather larger diameter clutches to deal with the higher torques at the lower speeds. Similarly, in order to accommodate the power outputs at lower maximum engine speeds it has been necessary to increase the overall gear ratios, in order to maintain the same top speed performances as with petrol engine vehicles; the lowest gear ratio, however, is usually made the same as for the latter type vehicle.

When running at its lowest "idling" speed, the Diesel engine does not usually work so smoothly or so quietly as the petrol engine, and there is apt to be rather more vibration. In some of the earlier Diesels, the peculiar noise, known as "Diesel knock," occurred at idling speeds as well as at the maximum ones. This noise—comparable to detonation knocking—generally disappeared at intermediate speeds. It has now been overcome in the latest engines, as the combustion conditions causing this knock are now better understood.

CHARACTERISTIC FEATURES OF DIESEL ENGINES

Externally, the high-speed Diesel engine bears a fairly close resemblance to the petrol engine, but it is generally rather larger in size than a petrol engine of equal power.

In place of the sparking plugs and their high-tension cables, however, we have the fuel-injection nozzles and their small bore fuel feed pipes, whilst the fuel pump replaces the magneto or coil ignition unit.

Fig. 65 illustrates the principal external features of a typical high-speed motor vehicle Diesel engine of the six-cylinder type. The inlet system is more simple than that of the petrol engine, for in place of the complicated carburettor and inlet manifold unit we have direct openings to the air from the inlet valve ports; these openings are usually provided with air-cleaners, or filters.

Incidentally, rather higher volumetric efficiencies are

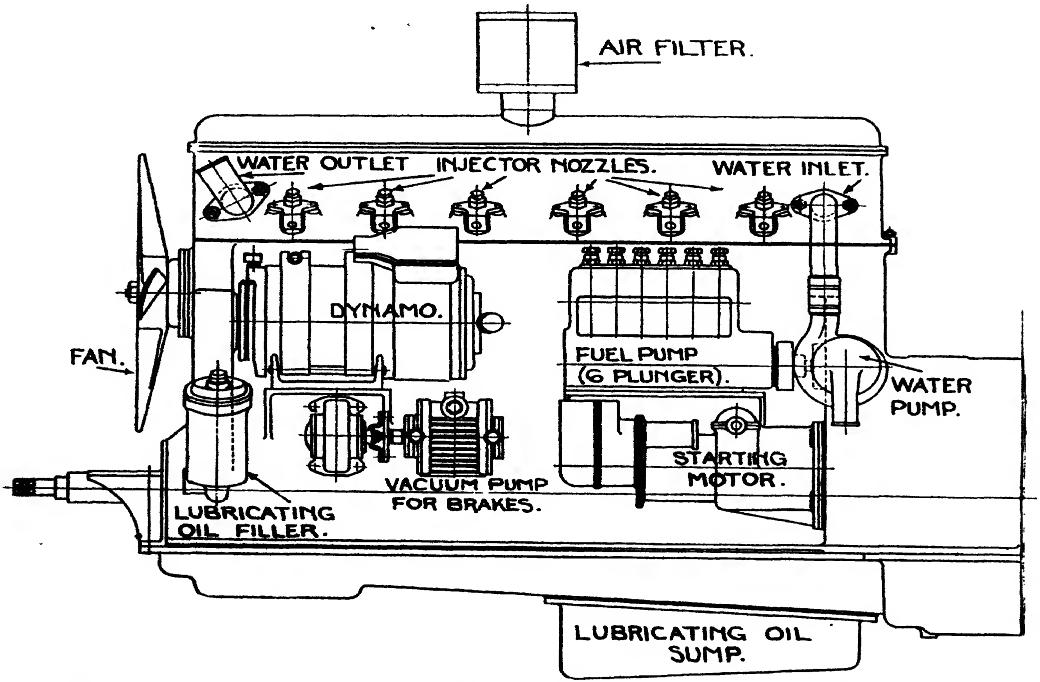


FIG. 65. SHOWING THE PRINCIPAL EXTERNAL FEATURES OF THE AUTOMOBILE DIESEL ENGINE

possible with the Diesel engine air inlet arrangement as there are fewer obstacles to the air flow. In some cases the inlet air is arranged to be heated by passing over the exhaust pipe, but means are usually provided to cut off the hot air supply when the engine has warmed up.

There is one important difference between petrol and Diesel engines, namely, *in the absence of any vacuum promoting device*, such as the inlet pipe of the petrol engine, it is not possible to work either the vacuum fuel feed or the vacuum brakes and windscreen wiper from the inlet side of the Diesel engine.

It is therefore necessary to fit an auxiliary suction type, or vacuum pump for these purposes. This pump is indicated in Fig. 65.

Internally, the Diesel engine, with the exception, perhaps, of the combustion chamber and piston design, follows petrol engine practice very closely. It employs the same types of cylinders, valves, valve-gear, connecting rods, and crankshafts; further, it uses the same materials. The lubrication and cylinder cooling systems are also practically identical in the two cases.

In view of the much higher compression pressures, and also of the higher working pressure in the cylinder, it is necessary to design the Diesel engine piston somewhat differently from the petrol engine type. In order to ensure freedom from gas leakage past the piston, it is now the practice to make the piston appreciably longer in respect to its diameter. Thus, most high-speed Diesel pistons have lengths of from 1·30 to 1·45 per cent of their diameters.

Incidentally, the stroke-bore ratios of modern Diesel engines range from 1·25 to 1·50, the average value being 1·38. The compression ratios vary from 13·0 to 16·0, but there appears to be a tendency to favour ratios of 13·5 to 15·0 in many British designs.

Reverting to the matter of the Diesel engine piston, it is usual to fit this with *five piston rings*, the common arrangement being four compression and one scraper ring; in one or two cases three compression and two scraper rings are used. It is necessary to prevent oil leakage past the piston so that the scraper ring is of greater importance in the Diesel engine, where higher differences of pressure between upper and lower sides of the piston exist.

We have already referred to the growing practice of fitting removable cylinder liners of special alloy cast iron, cast by the centrifugal process. This gives a much longer life to the cylinder liners and renders their renewal a comparatively simple matter; moreover, it is unnecessary to fit new pistons when cylinders are re-ground. It is now possible to employ valve seatings of the inserted type, using an alloy cast iron for this purpose. The valve seat inserts are so designed that they cannot become loose, but can be removed by means of a special screw-press tool.

In regard to the connecting rods and camshafts, these are made more rigid than in petrol engine practice in order to withstand the heavier loadings. The crank-shaft, in addition to being more massive, is so supported that there is one main bearing on either side of each crank pin. Thus, a four-cylinder type crankshaft will have five, and a six-cylinder type seven, main bearings.

Aluminium alloys are much used in modern high-speed Diesel engines, for parts such as pistons, crank-cases, and the castings of certain accessories. The well-known "Y"-alloy, R.R. (or Hiduminium) and certain aluminium-silicon alloys, are favoured by several engine builders for Diesel engine pistons.

As it is essential to reduce the weights per horse-power as much as possible in order to bring these figures into line with petrol engine practice, there is a growing tendency to use the light magnesium alloy, known as Elektron, for Diesel engine crankcases. This alloy has a density about two-thirds that of aluminium, whilst giving practically the same strength properties as those of the best aluminium alloys.

TYPICAL HIGH-SPEED DIESEL ENGINES

In concluding this section on high-speed Diesel engines for automobile purposes, it is proposed to describe a few typical engines that have been applied successfully to commercial vehicles. In view of the fact that there is now a relatively large number of such engines on the market, we have had to single out, for this purpose, those having certain outstanding features that are in actual use at the time of writing. New types are being developed, and at fairly regular intervals successful commercial types are superseded by new designs; this state of affairs is bound to exist during the development of what is a comparatively new innovation in automobile engineering.

The engines that were first used with success on certain motor buses and goods vehicles were mostly of the four-cylinder type, of 60 to 70 b.h.p. outputs. More recently the six-cylinder designs have been introduced; these are of greater outputs, viz. from 90 to 130 b.h.p. The engine speeds of the earlier motor vehicles seldom exceeded 1,500 r.p.m., whereas some of the latest types are capable of speeds up to 3,000 r.p.m.; maximum speeds of the majority of commercial engines, however, appear to be from 1,700 to 2,200 r.p.m.

In regard to engine weights, an analysis of the weights of twenty-five different commercial vehicle Diesel engines indicates that these range from 10 to 20 lb. per b.h.p., the average value being 15 lb. per b.h.p. In passing, it should be noted that the latest

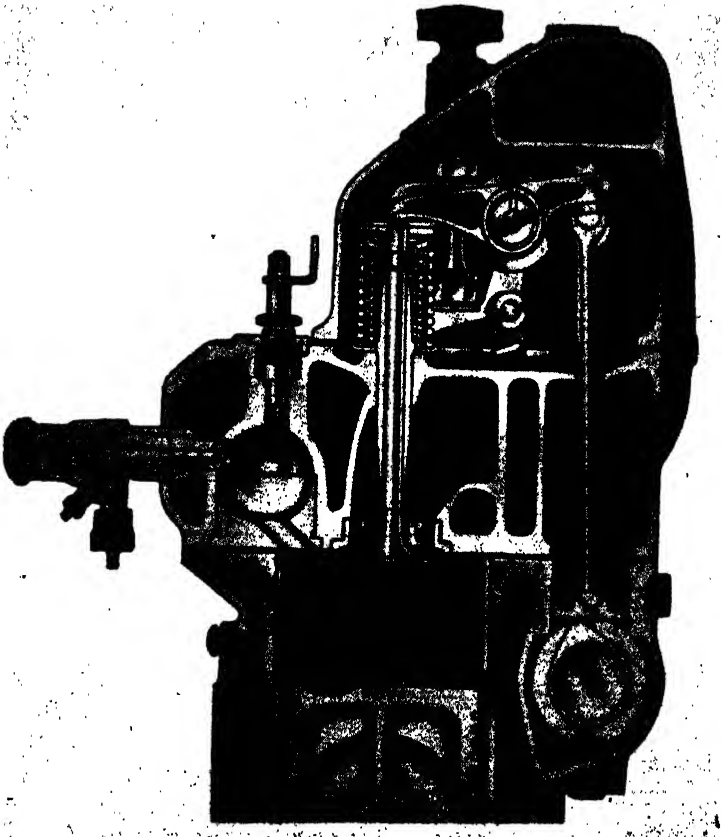


FIG. 66. THE A.E.C. CYLINDER HEAD WITH MODIFIED COMET COMBUSTION CHAMBER AND SLIGHTLY DISHED PISTON CROWN

type A.E.C. engine is no heavier than the lightest commercial petrol engines, for it weighs rather less than 10 lb. per b.h.p., with flywheel and accessories included.

The A.E.C. Engine. The earlier A.E.C. engine, introduced by The Associated Equipment Co., London, in 1929, was first used on an experimental lorry used for conveying employees. It was so promising in its performance that, subsequently, a number of these engines was installed in L.G.O.C. buses running on one of the regular London area services. in 1930. These engines employed the Acro type of combustion head, and had a maximum output of about 95 b.h.p.

Subsequently to 1932 all A.E.C. engines have been fitted with the "Comet" type of air turbulence cylinder head, with appreciable improvement in the power output and reduction in fuel consumption. The present A.E.C. engines for motor vehicles, e.g. passenger and goods ones, are made in the four-cylinder 6·6 litre, six-cylinder 7·7 litre and six-cylinder 8·85 litre sizes, developing maximum b.h.p.'s of 85 at 2,000 r.p.m., 125 at 2,400 r.p.m. (with just visible exhaust) and 135 at 2,200 r.p.m., respectively. In regard to the 7·7 litre model this has a bore and stroke of 105 mm. and 146 mm., respectively; it embodies certain improvements upon the other A.E.C. engine including the use of shallow cavity pistons giving improved combustion conditions (Fig. 66).

The engine, owing to various improvements in its design and materials, is both smaller and lighter than the original 8·8 litre engine for very nearly the same power output. The well-known Comet pattern cylinder head, with plain single hole pintle type nozzle, is employed.

The cylinder block is of cast iron mounted upon a cast iron crankcase and secured to the latter by means

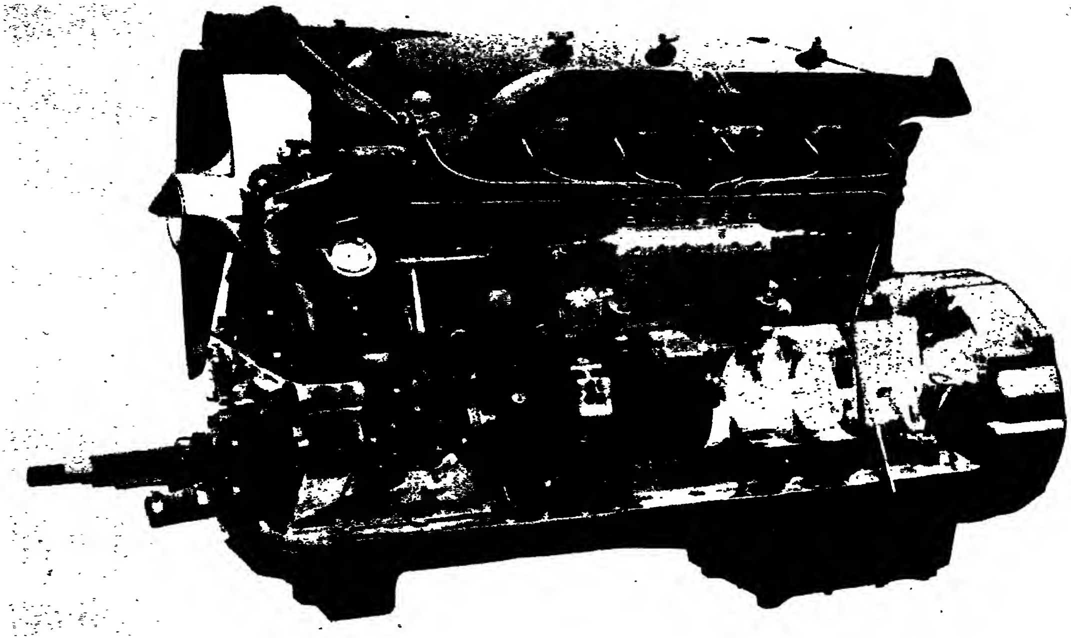


Fig. 67. The A.E.C. Diesel Engine
(A sectional view is shown in Fig. 66)

of the main bearing bolts which are carried upwards to the top of the crankcase for this purpose. Two cast-iron cylinder heads are fitted, each covering three cylinders; these heads have special jack-nuts to facilitate their removal.

The valves are operated by short push-rods and rocker arms, from the camshaft which is situated in a tunnel bored in the cylinder block. Flat-faced tappets of chilled cast iron are used. The position of the camshaft and auxiliaries makes possible a short neat timing chain lay-out, triple link roller chains with automatic spring tensioner being employed. The overhead valve gear is housed in a light easily removable cover.

The pistons are of aluminium alloy and employ four compression and three scraper rings. The gudgeon pins are fully floating in the piston bosses and small ends, circlips being used to locate them endwise.

The connecting rods are of H section alloy steel stampings. Lead bronze is used for lining the connecting rod big-end bearings, the main bearings being of whitemetal.

The alloy steel crankshaft is carried on seven main bearings.

In regard to the engine accessories a centrifugal water pump is used for circulating the cooling water; this is driven in tandem with the dynamo and is fitted with special graphite packing which eliminates the necessity of the usual water gland.

Lubrication is effected by means of a gear wheel pump provided with a device for supplying oil at low pressure to the valve gear.

A special vacuum pump, driven at one-half engine speed in tandem with the fuel injection pump, is employed for the vehicle brake operation. The fuel injection equipment is of C.A.V.-Bosch manufacture, the fuel pump being fitted with the governing device

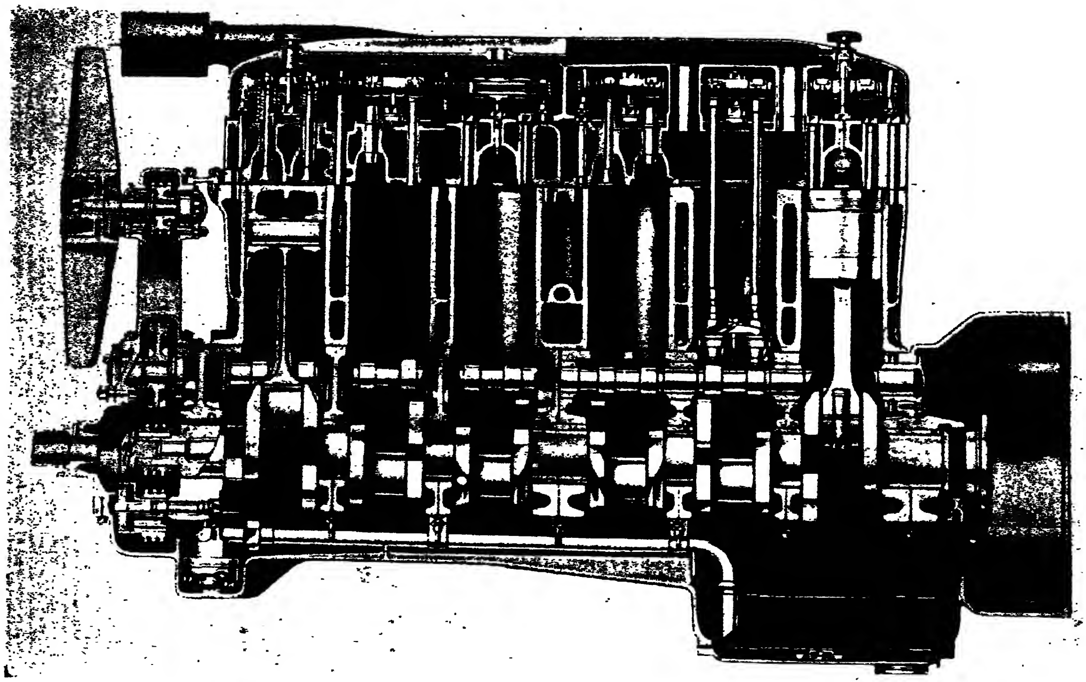


FIG. 68. SECTIONAL VIEW OF A.E.C. DIESEL ENGINE FITTED WITH COMET HEAD

described elsewhere in this section. It controls both the idling and maximum engine speeds; between these

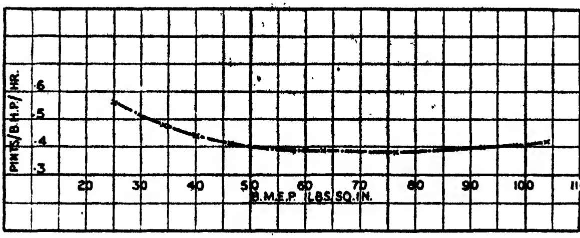


FIG. 69

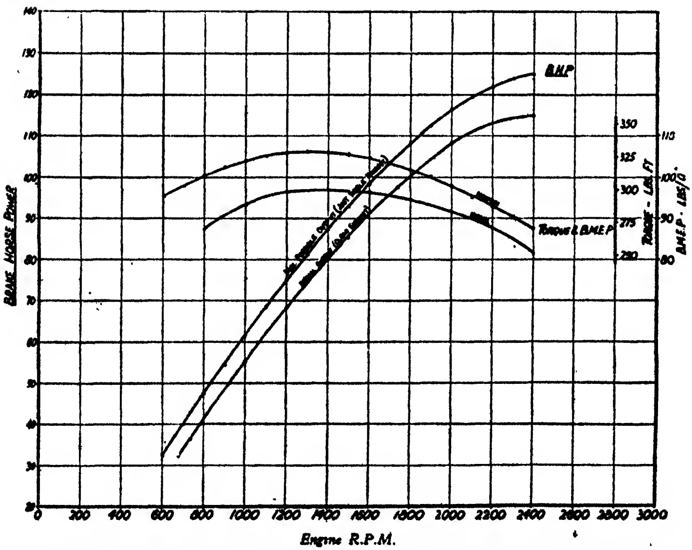


FIG. 70. PERFORMANCE CURVES OF A.E.C. 7·7 LITRE ENGINE

speeds the governor does not come into action, the engine being under the direct control of the accelerator pedal.

The fuel consumption and horse-power curves for this engine are reproduced in Figs. 69 and 70; from the former curve is will be observed that the minimum fuel consumption is 0·39 pints per b.h.p. hr., i.e. about 0·40 lb. per b.h.p. hr. with fuel of 0·815 density.

Another design of A.E.C. 7·7 litre engine now in use employs the direct injection system, using a double-lobe section cavity piston somewhat on the lines shown in Fig. 30. The injector is centrally located, but inclined at about 30° to the cylinder axis. This engine, whilst not giving so much power as the Comet type, has a 10 per cent lower fuel consumption.

The Dennis Diesel Engine. The engine illustrated in Fig. 71 is the later model Dennis Diesel type used in the Lancet chassis. It has four cylinders each of 117·47 mm. bore and 150 mm. stroke, giving a capacity of 6·5 litres. The engine is rated at 32·24 h.p. and develops a maximum b.h.p. of 82. The corresponding output at 1,000 r.p.m. is 50 b.h.p.

The direct injection system is employed with turbulence type cavity pistons in conjunction with duplicated inlet and exhaust valves in each cylinder on the lines of the "dual turbulence" method illustrated in Fig. 30. The fuel injection pressure is between 2,500 and 2,600 lb. per sq. in.

In regard to the constructional features of this engine, the cylinders and crankcase are formed as one unit, the cylinder heads being cast in pairs. The pistons are of aluminium alloy, with solid skirts, have four compression rings above the gudgeon pin and a single scraper ring at the bottom of the skirt.

The large diameter gudgeon pins are made a tight fit in the piston bosses, located by circlips.

The cylinders are fitted with wet-type liners, made a press fit in the block, leakage being prevented at the top by the fit in the block and at the bottom by means of a series of three grooves formed in the block; these are fitted with cork rings for the outer grooves, the centre one being open so that any water leaking past

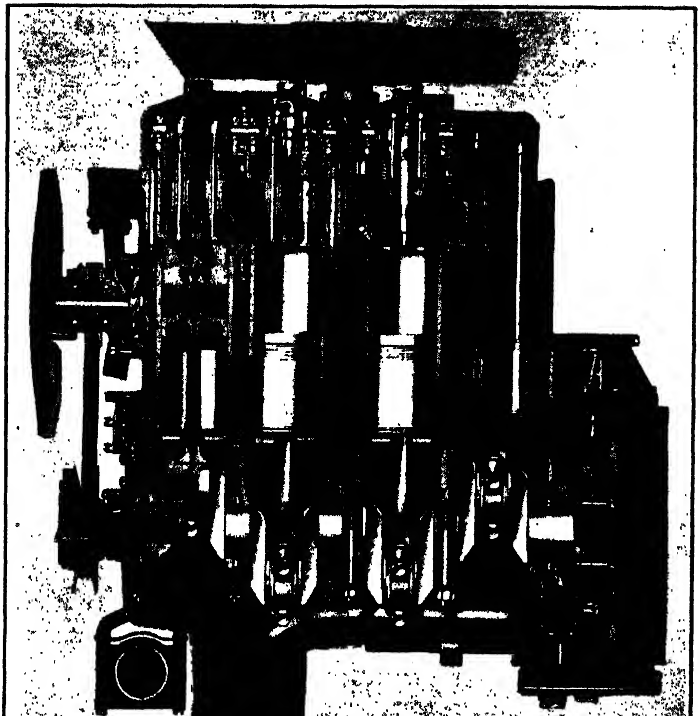


FIG. 71. THE DENNIS FOUR-CYLINDER ENGINE WITH CAVITY PISTON COMBUSTION CHAMBER

the top seal will drain out through two holes provided for the purpose.

The connecting rod follows normal design practice and has a bronze bush small-end bearing and lead bronze alloy big-end bearings. A feature of the big-end bearing is the provision of an oil-jet hole at 45° to the rod, such that when this hole registers, once every revolution, a jet of oil escapes from inside the drilled crank-pin on to the cylinder bores.

The main bearings, of which there are five, have steel shells lined with whitemetal.

The valves are operated by push rods and rocker arms from a relatively low camshaft; each pair of valves in a cylinder is actuated by a twin-armed rocker. The feet of the tappets are of mushroom shape, the tops being cupped to receive the spherical-shaped ends of the push rods. The top end of the latter are cupped to take the ball-headed peg of the valve rocker; a running clearance of 0·010 in. is allowed for each valve stem. Duplex valve springs are used. In regard to the valve timing of this engine the inlet valves open at 5° before T.D.C. and close at 47° after B.D.C. The exhaust valves open at 54° before B.D.C. and close at 19° past T.D.C.

The engine has a fixed injection timing of 25° before T.D.C. The compression ratio employed is 13·8 : 1.

Concerning the auxiliaries, these are all fitted on the nearside of the engine and include the vacuum pump and fuel injection pump driven in tandem from the rear or flywheel end of the engine. The dynamo (25 volt, 450 watt pattern) is also driven from the rear end of the engine.

The cooling system comprises a belt-driven multi-bladed aluminium alloy fan, the shaft of which also drives a water impeller in the front end of the cylinder block.

The Fowler Sanders Engine. The turbulent type Diesel engine having an auxiliary turbulence chamber connected to the cylinder head by means of a restricted passage, or throat, through which the heated compressed air is forced at a high velocity during the compression stroke of the piston is subject to the disadvantage of appreciable heat loss from the charge to the walls of the throat, during the compression and combustion processes. As a result the thermal efficiency is somewhat impaired, whilst starting from the cold is more difficult, necessitating the use of heater plugs.

These disadvantages are, however, overcome to a considerable extent in the instance of the Fowler Sanders

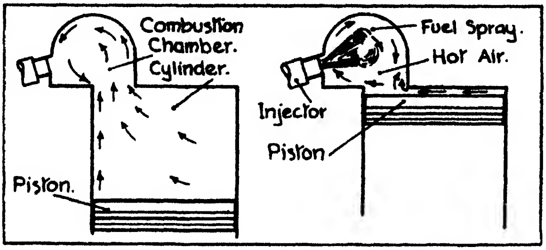


FIG. 72. PRINCIPLE OF FOWLER SANDERS ENGINE

engine, the principle of which is illustrated in Fig. 72. The engine in question works on the four-cycle principle and employs a high degree of turbulence so that, during the injection of the fuel, the latter is atomized and mixed with the air, thoroughly.

Referring to the left-hand diagram of Fig. 72, here the piston is shown moving upwards at the commencement of its compression stroke. As the air in the cylinder is compressed it flows upwards through the relatively large opening (as distinct from the usual narrow throat) at the side of the cylinder into the dome-shaped combustion chamber; the latter is designed so that the air is made to swirl round in an anti-clockwise direction.

The right-hand diagram (Fig. 72) shows the piston near the top of its compression stroke, when the compressed air is well heated and the injector has commenced to deliver its fine spray of fuel.

The direction of the air swirl is now clockwise, due to the fact that the piston when approaching the top of its stroke displaces the air in a sideways direction. This two-way swirl causes rapid and thorough mixing of the air and fuel, thus promoting very good combustion conditions over the working speed and load ranges.

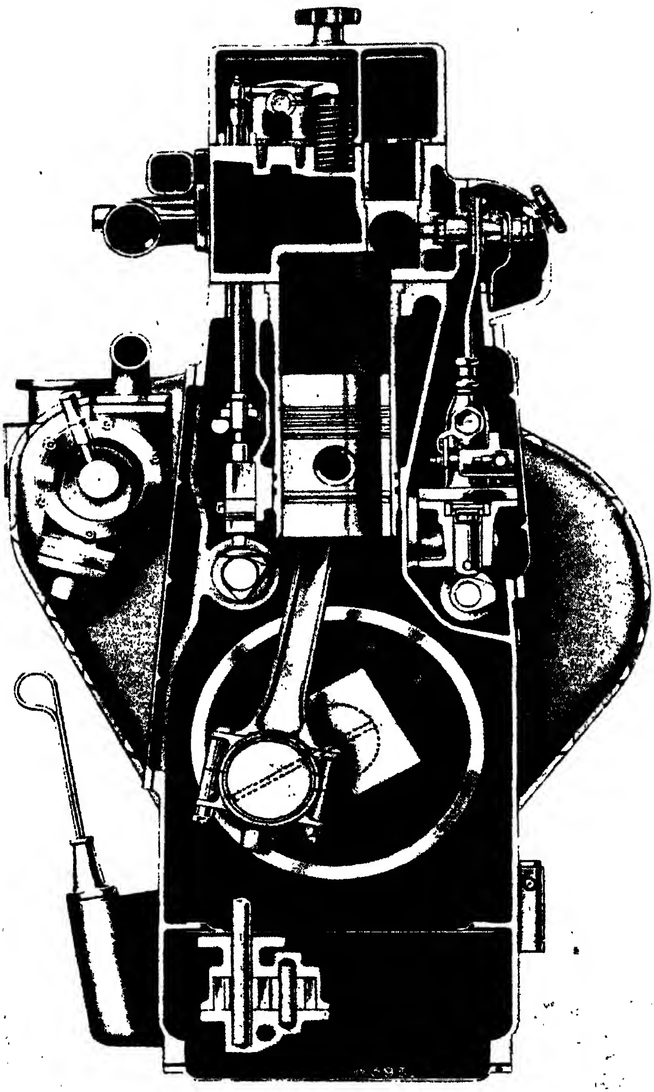
It ensures easy starting from the cold combined with smooth running and absence of "Diesel knock"; moreover, for the reasons previously stated the thermal efficiency is high and the fuel consumption correspondingly low. It should be mentioned that no heater plugs are required for cold starting purposes.

The engine is provided with a special design of decompressor gear which lifts one valve in each cylinder so that the engine can be cranked freely. A device is fitted for the purpose of returning the engine to full compression, automatically, after a few revolutions of the crankshaft. This device renders hand starting easier on the smaller sizes of engines, whilst on the larger electric motor started engines it ensures that no undue load is put on to the battery.

The engines in question show fuel consumptions of the order, 0·38 to 0·41 lb. per b.h.p. when operating at three-quarters to full-load; they employ C.A.V.-Bosch fuel injection equipment.

A typical four-cylinder engine of 4½ in. bore and 6½ in. stroke develops 57 b.h.p. at 1,100 r.p.m. and a brake m.e.p. of 115 lb. per sq. in. The compression pressure employed is 450 lb. per sq. in. and the maximum cylinder pressure developed is 780 lb. per sq. in.

The Gardner Engine. Although the larger LW type Gardner engines have been in use over an appreciable period, it is only more recently that the 4LK type has been put into production. This engine represents a smaller and much lighter version of the other type



and for this reason has not only been used in the smaller commercial vehicles but also in experimental motor-cars.

This engine (Fig. 74) employs the direct-injection, or open combustion chamber method with a plain

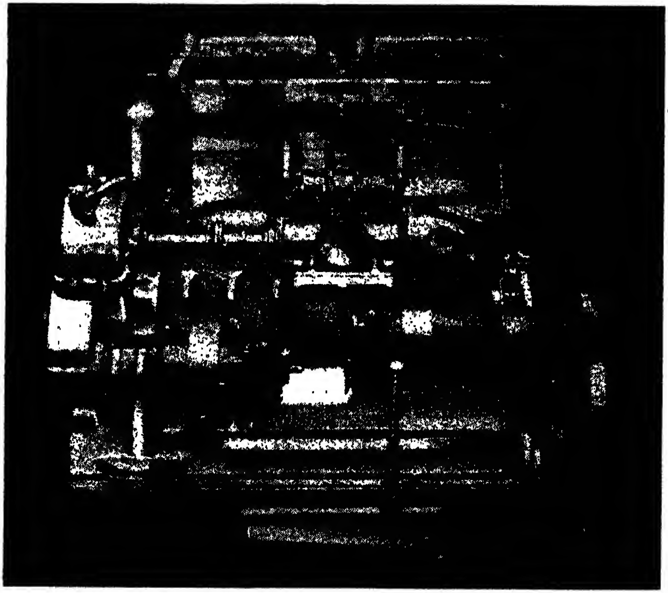


FIG. 74. THE GARDNER 4LK ENGINE

cylinder head, central fuel injector and a special shape of piston head, on the lines of that shown in Fig. 12.

The injection nozzle is of the multiple spray pattern with four small equally spaced holes spraying the fuel into the piston cavity. The compression ratio is 14 : 1. The engine is of the four-cylinder type, of 3¾ in. bore and 5¼ in. stroke giving a cylinder capacity of 3·80 litres. It is rated at 22·5 h.p. and gives 53 b.h.p. at

2,000 r.p.m.; the engine has a bare weight of 575 lb., which is equivalent to 10·8 lb. per h.p.

The fuel consumption is relatively very low, namely, 0·370 lb. per b.h.p. hr. at 1,000 r.p.m. and 0·380 lb. at 2,000 r.p.m. In this connection it is interesting to note that the manufacturers have found that the mileage per gallon of vehicles fitted with this engine is about 2·2 times that of the same vehicles with the same load but driven by petrol engines of similar outputs.

Special constructional features of this engine include the use of aluminium alloy cylinder heads in blocks of two, each carrying renewable valve seats and fuel injector seats. The overhead air inlet and exhaust valves are operated by push-rods and rocker arms. the whole of the valve gear is enclosed by neat aluminium covers which also form the air intake passage. The cylinders are cast in aluminium alloy in monobloc and are fitted with dry type cylinder liners of hardened metal. The crankcase is made of Elektron alloy throughout and carries the crankshaft in five whitemetal-lined bearings and the camshaft in six bearings. The rigid construction of this casting is supplemented by ten through bolts extending from above the crankshaft centre line up to the cylinder feet. A detachable Elektron oil sump is provided.

The aluminium alloy pistons, with their dished crowns, each carry five piston rings. H-sectioned connecting rods of forged alloy steel are used; these have whitemetal-lined big-ends and bronze bush small-end bearings. The large diameter gudgeon pins are arranged to float in the piston bosses and small-end bearings. The crankshaft is of nickel alloy steel machined all over from a forging.

The fuel injection pump and governor unit are mounted on the nearside of the engine. The drive is transmitted to the fuel pump camshaft by helical gears; the camshaft gear, sliding axially on its helically splined shaft, and coupled to the governor, provides automatic advance and retard of the injection timing. The fuel injection pump used is of the C.A.V.-Bosch design operated by Gardner cam gear and fitted with Duplex ram return springs. A simple centrifugal type of governor is used to control both the upper and lower speed limits.

Fuel is supplied to the fuel injection pump by means of an Amal diaphragm feed pump, an overflow return feed arrangement being fitted. Hand-operated priming levers are fitted to the fuel pump there being one lever per pump cylinder.

The General Motors Two-stroke Engine. This comparatively recent design of engine has been designed in a range of sizes from 85 h.p. up to 1,200 h.p. for various purposes. They are all built on the symmetrical construction system whereby with the aid of interchangeable components the engines can readily be converted from left- to right-hand rotation and the water and exhaust outlets arranged on either side of the engine.

The combustion method employed is the direct-injection one with a combined pump and injection valve in each cylinder head, operated by an overhead rocker arm gear (Fig. 75).

Each cylinder in the smaller units has two exhaust valves in the head, operated by rocker arms and push-rods from a high level camshaft.

An interesting feature of these engines is the fitting of a kind of dummy camshaft on the opposite side, but driven in the opposite direction so that the out-of-balance forces of the two shafts cancel each other. This arrangement also enables the camshaft positions to be interchanged.

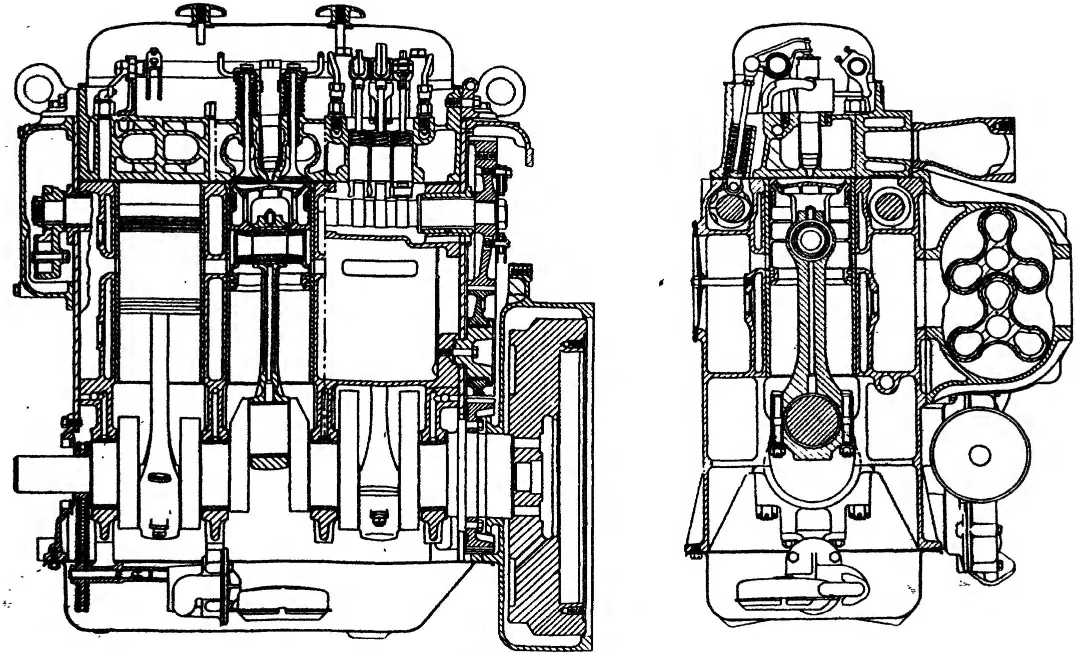


Fig. 75. The General Motors Two-cycle Engine

The fuel injection nozzle is situated centrally and sprays its fuel over a wide angle into the appropriately shaped cavity in the piston head. The piston is cooled by ribbing it inside and providing an oil jet from a nozzle fitted to the top of the small end of the connecting rod.

The two-stroke arrangement consists of piston-controlled ports in the lower part of the cylinder barrel and the exhaust valves, previously mentioned, in the cylinder heads. The fresh air charge—which also assists in scavenging the exhaust gases from the cylinders—is supplied by means of a Rootes blower having rotors of the three-lobe type. The blower is attached directly on to the cylinder block and delivers air under pressure through a large port into an air chamber or trunk surrounding the cylinders, thus providing an adequate storage of air for the cylinders; this air enters through the ports when the latter are uncovered by the pistons.

Interesting constructional features of these engines include the use of a rectangular cylinder block which incorporates also the crankcase; the cylinders are fitted with dry type liners. The cylinder heads are also made in block form and carry the exhaust valves and rocker gear, as well as the fuel injection units.

The pistons are made of special malleable iron, of thin section with the previously mentioned internal ribs. Four compression rings are fitted above the gudgeon pin and two oil-control rings below, namely, at the bottom of the skirt.

The auxiliaries, including the fuel transfer pump, governor and water pump are all mounted on the blower unit and driven from it.

The smallest engine made is the three-cylinder vertical type shown in Fig. 75. It has a bore of $4\frac{1}{4}$ in. and stroke of 5 in., developing 85 b.h.p. at 2,000 r.p.m. with a fuel consumption of 0·45 lb. per b.h.p. hr. The

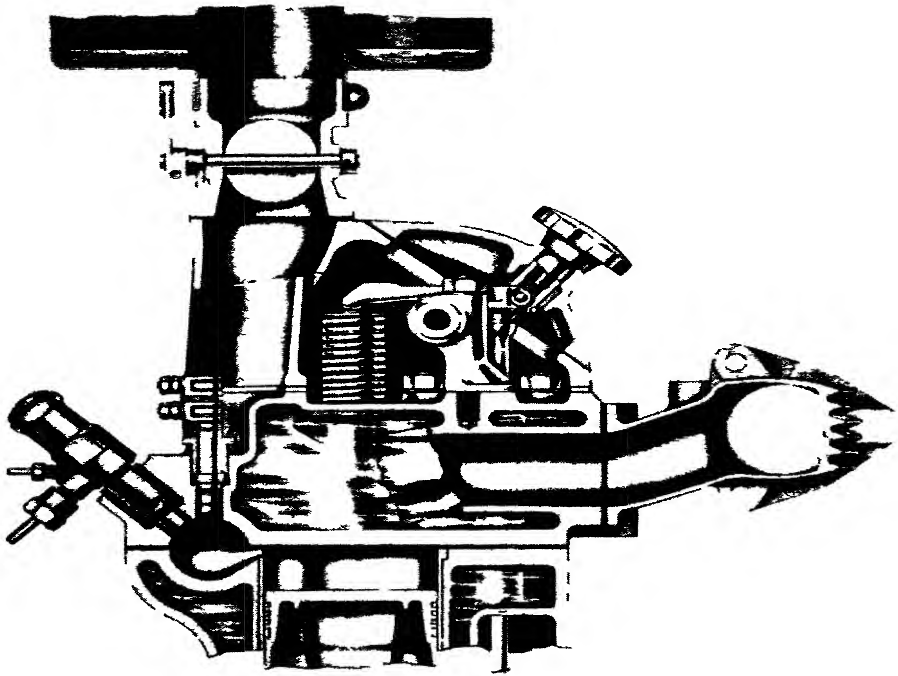


FIG. 76. SECTION THROUGH UPPER PORTION OF LEYLAND "LIGHT SIX" ENGINE

compression ratio employed is 16 : 1 and the maximum cylinder pressure is just under 1,000 lb per sq. in.

The Leyland Engines. The commercial vehicle petrol engines hitherto employed by the Leyland Motor Company have now been replaced, largely by Diesel ones for motor vehicles and rail-car power units.

Two different types of Diesel engine are made, namely, the direct injection and the turbulent ante-chamber ones.

The former type is made in both a four- and a six-cylinder model. The four-cylinder one has a bore of 114·3 mm. and stroke of 139·7 mm., giving a cylinder capacity of 5,731 c.c. It has an R.A.C. rating of 32·4 and develops 70 b.h.p. at 2,000 r.p.m.

The six-cylinder model has a bore of 114·3 mm. and stroke of 139·7 mm., giving a cylinder capacity of 8,603 c.c. It is rated at 48·6 h.p. and develops 95 b.h.p. at 2,000 r.p.m., which is the maximum governed engine speed.

The other type of oil engine referred to is known as the "Light Six" and has a bore of 88·9 mm. and stroke of 127 mm. giving a cylinder capacity of 4,731 c.c. It is rated at 29·4 h.p. and gives 90 b.h.p. at 2,500 r.p.m. This engine employs a spherical turbulence chamber, with a small throat leading to the main combustion space. One-half of the spherical chamber is formed in the top of the cylinder head and the other half in the cylinder block; this renders the chamber readily accessible for cleaning purposes. The chamber which is offset to one side of the cylinder head gives the usual turbulence to the air charge when the piston approaches its top dead centre.

The combustion chamber of the direct injection engine is shown in Fig. 77. It is formed as a simple offset open cavity in the top of the piston. No glow plugs are used for starting.

The fuel is sprayed towards the piston centre and is kept away from the cylinder walls where it may have a detrimental effect upon the cylinder lubrication. The

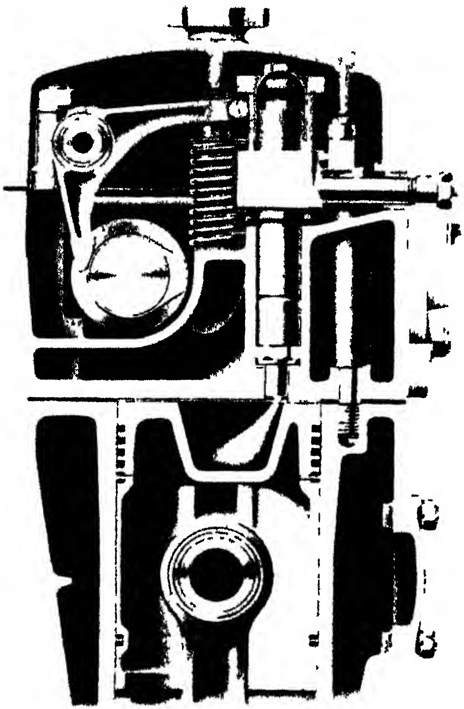


FIG. 77. SHOWING COMBUSTION CHAMBER OF LEYLAND DIESEL ENGINE

air flow in the cylinder head is non-turbulent and non-reversing, so that the combustion chamber is claimed to be free from the heat of the turbulent types.

Bosch fuel pump and injection nozzles are used with this engine. While the former is fitted with injection advance mechanism, the range of injection timing has been strictly limited to give practically fixed timing.

The engine in question follows petrol engine design in most particulars, but the working parts have been strengthened to take the heavier loads. Many parts, e.g. the flywheel, sump, and timing gears are common to both engine types of the Leyland firm; incidentally, the Diesel engine has been designed so as to be interchangeable with the standard petrol type used for commercial vehicles.

Special features of the Leyland Diesel engine include overhead exhaust and inlet valves (an air cleaner being fitted on the air intake), pistons of "Y"-alloy, a cast-iron cylinder head which carries all the valve gear, a seven-bearing crankshaft, and a deep, rigid crankcase of aluminium alloy.

The engine has a particularly clean exterior, all auxiliaries, such as the fuel pump unit, dynamo, and water pump being arranged on the near side for the convenience of access from the pavement of the road.

The Perkins Small Diesel Engines. The Perkins series of small engines range from the smallest four-cylinder "Wolf" model rated at 17·9 h.p. (developing 45 b.h.p. at 2,500 r.p.m.) to the largest, namely the six-cylinder "Panther" which is here described. These engines are of light construction and have been used for motor-cars, although more generally for light commercial vehicles. They incorporate the Perkins "Aeroflow" combustion system previously described in this section.

The Panther engine is a six-cylinder one of 88·9 mm. bore and 127 mm. stroke giving a cylinder capacity of 4.73 litres. It is rated at 29·4 h p., but develops 100 b.h.p. at about 1,500 r.p.m. and a corresponding torque of 193 lb. ft. at the same speed. The maximum engine speed of this model is 2,600 r.p.m. (governed)

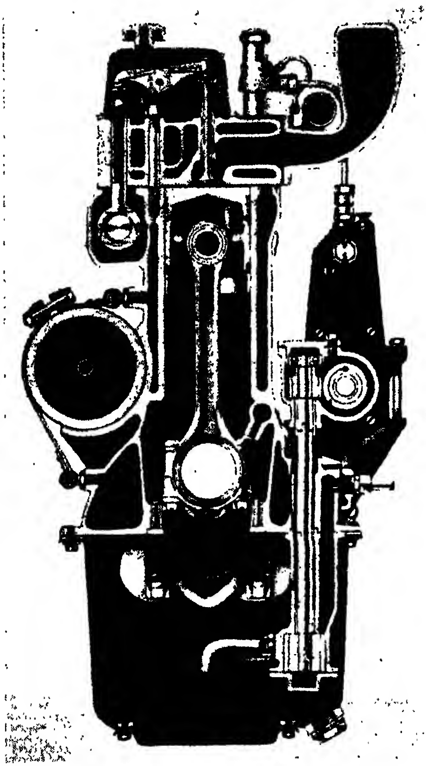


FIG. 78. THE PERKINS ENGINE

and the fuel consumption, 0·385 lb. per b.h.p. per hr. (minimum consumption).

The complete engine with fan, water pump, flywheel, electric starting motor, etc., weighs 712 lb. Without the flywheel and electric starting motor it weighs 580 lb.

The respective weights per b.h.p. are 7·12 and 5·80 lb. —figures which are comparable with those of the best automobile petrol engines.

It may be of interest to mention that one of these

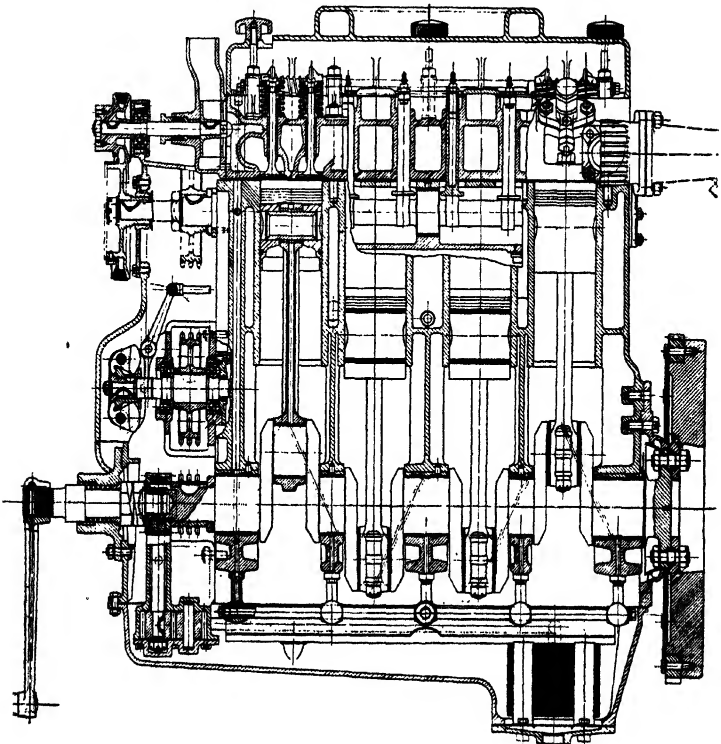


FIG. 79. SECTIONAL VIEW OF FOUR-CYLINDER PERKINS ENGINE

engines when fitted to a motor-car, weighing in all $37\frac{1}{2}$ cwt., gave a fuel consumption of 40 miles per gallon. The car was fitted with a top gear ratio of

$4\frac{1}{4}$: 1 and accelerated through the gears from rest to 50 m.p.h. in 14 secs.; from 10 to 60 m.p.h. the time was about 20 secs.

The engine can be started from the cold without the aid of heater plugs and it runs smoothly at all speeds.

Special features of the engine include the use of metal pressings for the crankcase, oil sump, valve gear cover and front timing case, and a crankcase ventilation system operating upon the continuous principle, not only when the vehicle is on the move but when it is stationary. Some of the air for combustion is taken through the large air cleaner above the engine and into the overhead valve casing, which is made airtight for this purpose, and thence it passes through a duct into the crankcase, whence it returns by another duct into the inlet manifold. The engine has a seven-bearing crankshaft and aluminium alloy pistons; the cylinder head which forms the inlet manifold is a one-piece chromium iron casting and is secured to the cylinder block by a large number of studs, the joint being made by a copper asbestos gasket. The valve gear is mounted in the head, the spherical combustion chamber forming half of the head and half is by means of a loose steel cap, which can be detached. An efficient cooling system is employed, consisting of a water pump delivering cold water from the bottom of the radiator to three ports in the cylinder head. This is directed on to the space around the six atomizers and the valve seats in order to keep these cool. Thermo-syphon cooling is used for the cylinder block, a fan being fitted behind the radiator. The engine employs the C.A.V.-Bosch fuel injection equipment and operates upon the pneumatic fuel control method, whereby the partial vacuum existing in the throttle in the inlet manifold is arranged to govern the fuel supplied to the engine in order to limit the maximum speed.

The Petter Harmonic Induction Engine. This type of two-cycle engine employs a novel method for inducing the fresh charge of air into the cylinder, in place of the usual crankcase compression, or a separate air compressor.

It is a well-known fact, readily demonstrable from

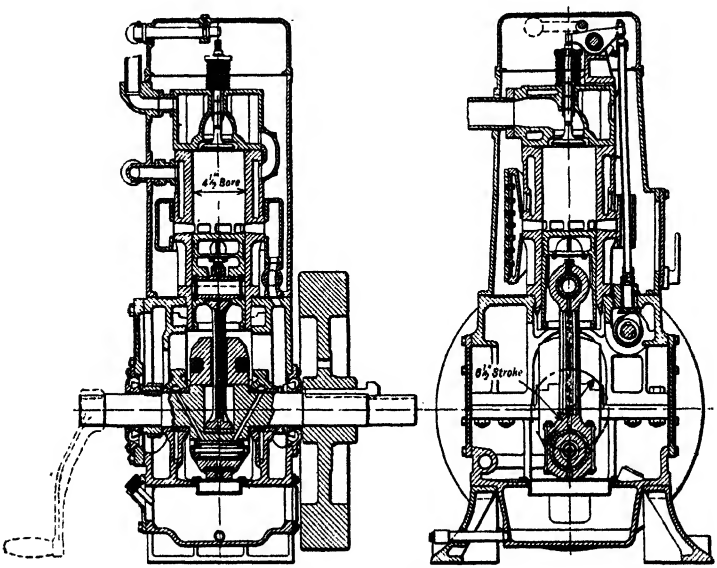


FIG. 80. THE PETTER HARMONIC INDUCTION ENGINE

a study of indicator diagrams taken from the exhaust manifold, that pressure waves occur in the exhaust system throughout the exhaust cycle; these consist of periodical compressions and rarefactions in the exhaust manifold, pipe and silencer.

The Petter engine takes advantage of this effect by arranging for the piston to open the air inlet ports near the bottom of the cylinder during the period that a rarefaction or partial vacuum occurs in the exhaust manifold when the overhead exhaust valve is open.

The sudden ejection of the exhaust gases when the exhaust valve opens is followed by the reduced pressure period and the coincident opening of the air ports by the piston towards the end of its descent.

Under the influence of the atmospheric pressure on the one side and the partial vacuum at the top of the cylinder fresh air is forced into the cylinder, tending not only to eject the remaining exhaust gases but also to fill the cylinder with fresh air at practically atmospheric pressure.

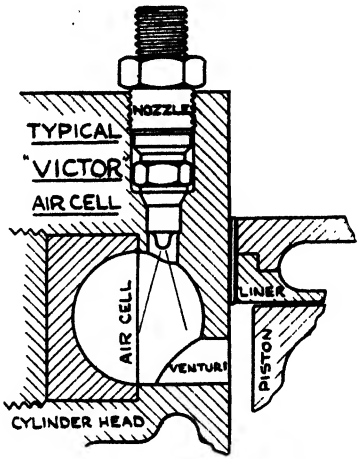


FIG. 81. VICTOR CYLINDER HEAD

This type of "resonance" charging is only effective, however, over a limited speed range, corresponding to the normal running speed, since it depends largely upon the dimensions of the exhaust pipe, the exhaust and inlet timing and the engine speed.

It is not satisfactory at low speeds, so that for starting and slow-running purposes crankcase compression is employed. In these circumstances an auxiliary throttle is used to place the crank chamber into direct communication with the air inlet port outer belt. The engine in question is made in both the single and multi-cylinder models. Fig. 80* illustrates, in front and side sectional views, an engine of $4\frac{1}{2}$ in. bore and

* By Courtesy of *The Engineer*.

6 in. stroke developing 16 b.h.p. at 1,000 r.p.m. It has been shown that the application of the inductive principle described results in a 50 per cent power increase over the crankcase-compression type.

The Victor Horizontally-opposed Engine. This small British engine has been used on light commercial

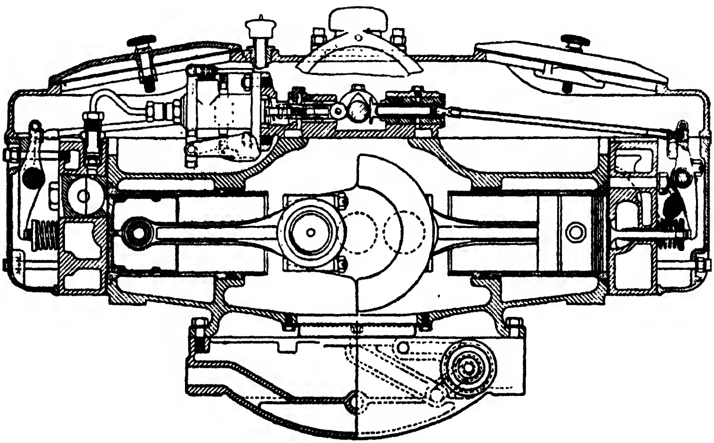


FIG. 82. THE VICTOR TWO-CYLINDER ENGINE

vehicles and for stationary purposes. It has a bore of 80 mm. and stroke of 100 mm., giving a total capacity of 1,000 c.c.

The two cylinders are arranged opposite to each other so as to obtain equal firing intervals. The engine develops 20 b.h.p. at 3,000 r.p.m. and weighs 280 lb.

The combustion system used consists of a spherical turbulent air chamber communicating with the cylinder by means of a venturi passage (Fig. 81). Fuel is injected vertically downwards towards the inner portion of the venturi.

Special features of the Victor engine (Fig. 82) are the two-throw built-up crankshaft; water-cooled cylinders fitted with detachable wet-type liners and detachable cylinder heads. The connecting rods are of

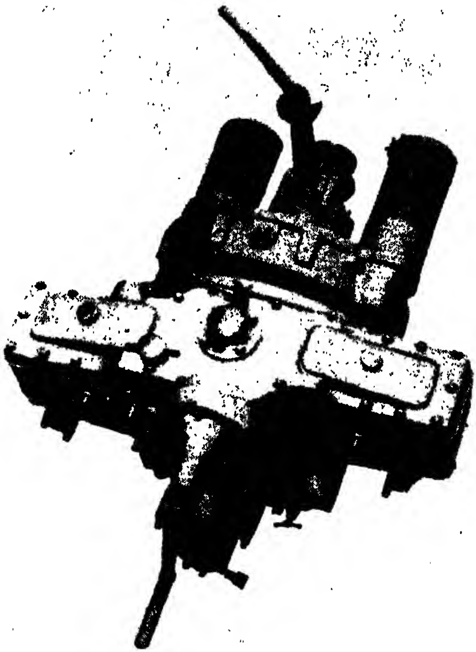


FIG. 83. THE VICTOR TWO-CYLINDER ENGINE

heat-treated high tensile steel machined all over. The big-end bearings embody rollers, there being 48 small diameter rollers in each big-end bearing. The main bearings also are of the roller type, whilst plain bronze bushes are employed for the small ends.

Cast iron pistons are used, each being fitted with three compression and one scraper ring. The crankcase, oil sump and cylinder head covers are of Alpax aluminium alloy.

The C.A.V.-Bosch fuel injection equipment is employed.

For starting purposes, from the cold, a spring-operated sliding type decompressor cam is used to

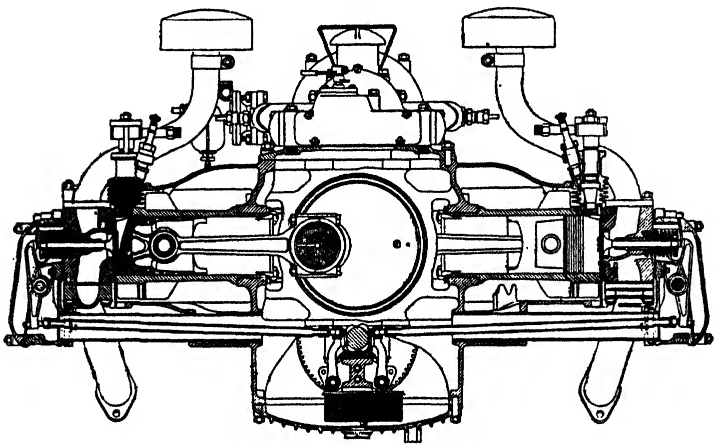


Fig. 84. The Krupp Four-cylinder Opposed Engine

hold the exhaust valves open until the flywheel has acquired sufficient momentum, when the cam is released and the engine rotates under full compression and with its fuel supply.

The Krupp Opposed Cylinder Engine. A four-cylinder air-cooled engine made by the German firm of Krupp is shown in Fig. 84. The cylinders are arranged as two opposing pairs with cranks at 180°.

The cylinder bore is 92 mm. and stroke, 130 mm., giving a capacity of 3,456 c.c.; it develops 14 b.h.p. at 2,300 r.p.m. The cylinders are air-cooled, the cooling arrangements including a centrifugal fan driven at engine speed, forcing air through suitable ducts along the finned and shrouded cylinder barrels.

The combustion method used is the direct-injection one, the piston crowns being cut away as shown in the left-hand sectional view. The semicircular opening of the piston cavity receives its fuel spray from an atomizer arranged in the pre-combustion chamber above.

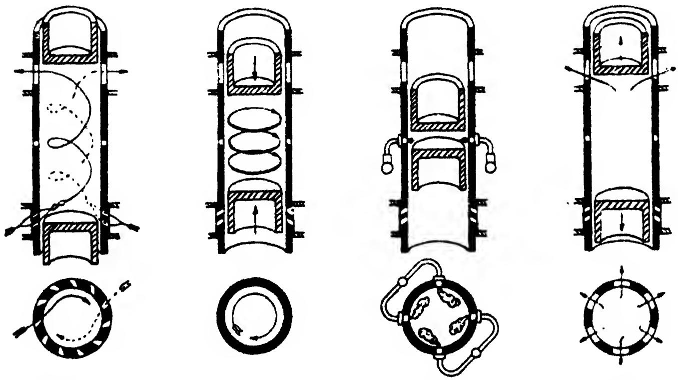
The engine has a compression ratio of 17 : 1. The compression and maximum working pressures are 570 and 780 lb. per sq. in., respectively. The fuel injection pressure used is 1,140 lb. per sq. in. The fuel consumption is 0·44 lb. per b.h.p. (minimum).

The Junkers Two-stroke Engine. This is probably one of the most original high-speed Diesel engines in present use. It employs a pair of opposed pistons in each cylinder, the pistons moving inwards and outwards symmetrically. The combustion chamber is formed in the cylindrical space left between the pistons when at their nearest approach to each other. It will be seen that with this arrangement it is necessary to have two crankshafts, one above and one below, as in the earlier Gobron Brille motor-car engine.

The upper crankshaft, in the original Junker engines, was coupled to the lower one (which transmitted the power to the clutch and gearbox) by means of long connecting rods. In the more recent aircraft engines these relatively heavy connecting rods are replaced with gearing, a train of gear wheels being used to connect the two crankshafts.

The air inlet and the exhaust ports are controlled by the pistons, the former being arranged at the lower, and the latter at the upper ends of the cylinders. In this way there is no opportunity for the exhaust gases to escape into the air inlet side. Moreover, the latest Junker engines employ air scavenging through the air inlet and out through the exhaust ports.

The four principal operations are shown diagrammatically in Fig. 85. Referring to Diagram *A*, the pistons are shown on their outer dead centres, the upper one having first uncovered the exhaust ports. The lower piston next uncovers its air inlet port to admit, through the tangentially-arranged ports (shown in



Scavenging. *A* — Compression. *B* — Injection. *C* — Exhaust. *D*

FIG. 85. SEQUENCE OF OPERATIONS IN JUNKER TWO-CYCLE ENGINE

lower diagram), the scavenging air; this effectively forces out the remnants of the exhaust gases. It will be observed that the air is in turbulent motion during the succeeding compression stroke (Diagram *B*) when both pistons are approaching after having closed the exhaust and inlet ports. This turbulent motion continues throughout the compression stroke until, when the pistons are practically on their inner dead centres, the fuel is injected through four equally spaced nozzles (Diagram *C*). This arrangement ensures efficient mixing of the fuel and air during the combustion process. Expansion then occurs and the pistons move outwards

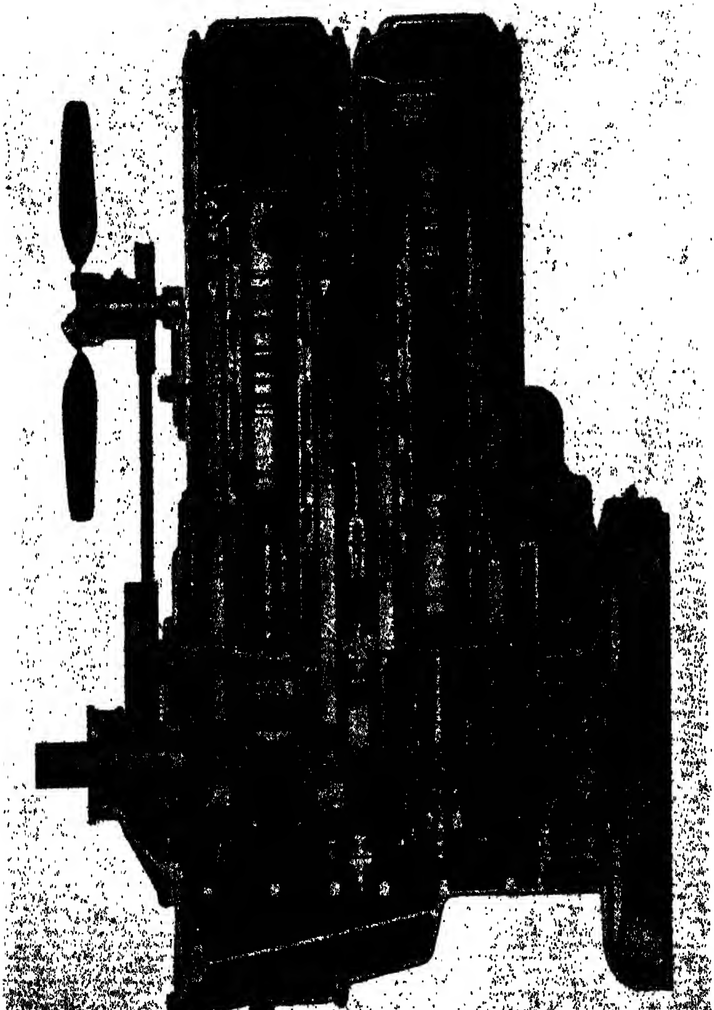


FIG. 86. DOUBLE-PISTON TYPE JUNKER ENGINE

until the upper one uncovers the exhaust ports, as shown in Diagram *D*, and the cycle of operations is complete.

A single-stage separate air compressor is used for the scavenging air.

The engine uses the special Junker's fuel pump and open-type injection nozzle. The model developed for motor vehicle purposes is a three-cylinder one, with three pairs of opposed pistons.

The Junkers engine employs roller bearings for the main bearings and needle bearings for the connecting-rod big ends. The gearing connecting the upper and lower crankshafts is mechanically lubricated and operates silently.

The engine, on account of the relatively small cooling area of its combustion chamber—and therefore of the smaller compression heat loss—starts readily from the cold.

A typical Junkers commercial vehicle engine of the 60 b.h.p. type is shown in Fig. 86. This engine has two double-piston cylinders, the upper pistons having crossheads and pairs of long connecting rods to transmit the power to the crankshaft below.

The Junkers aircraft engine has been designed especially for light weight per horse-power, liberal use of magnesium and aluminium alloys having been made. The heavier connecting rods, from the upper piston crossheads, as previously stated, have been replaced by gearing; in this case there is another crankshaft above, geared to the lower one and the propeller shaft drive is taken from the second gear of the five gear train, above.

The Jumo 204 engine is rated at 750 h.p. at 1,720 r.p.m. It has a bore of 120 mm. and stroke of 2×210 mm., giving a cylinder capacity of 28·60 litres. The compression ratio employed is 17 : 1 and the maximum value of the B.M.E.P. is 96 lb. per sq. in. The engine develops a maximum output of 800 b.h.p. at 1,850 r.p.m. for a total weight of 1,650 lb., which gives a weight of 2·06 lb. per h.p. The fuel consumption (minimum) is 0·34 lb. per b.h.p. hr.

The engine described is supercharged, centrifugal compressors being employed; these absorb a total of 80 h.p. at 1,200 r.p.m. The frontal area of the Jumo 204 engine is 8·5 sq. ft.

Flat Type Diesel Engines. A more recent innovation in commercial Diesel engines has been that of the multi-cylinder flat, or opposed arrangement of the cylinders. This may be regarded as a natural development of the two cylinder opposed types—similar to those shown in Figs. 82 and 84, which possess the important advantages of almost perfect engine balance and equal firing intervals.

Although the overall cylinder axial length of such engines is appreciably greater than for the usual vertical types, by arranging these engines with the cylinder axes transversely to the longitudinal axes of the vehicle and under the floor of the body or compartment, this no longer constitutes a drawback. Engines of 8 and 12 cylinders, giving outputs of 100 to 300 b.h.p., are now in use on the Continent, chiefly for high speed passenger vehicles; the same type is also being developed for a similar purpose in the U.S.A.

SECTION XIX

ENGINE LAY-OUT, SUSPENSION AND MOUNTING

BY

A. G. PENDRELL

SECTION XIX

ENGINE LAY-OUT, SUSPENSION AND MOUNTING

ENGINE LAY-OUT

ALTHOUGH only a perfunctory interest is normally shown in "what's under the bonnet," a very high performance standard is called for relative to portions of more popular interest. In the main, this call has been fully answered, and as a result there is a very high degree developed in the technique of engine building. It is now considered quite an ordinary procedure to prepare a lay-out, make the parts from detail drawings, and after assembly and running-in to have the engine firing almost at the first push of the starter button. Good draughtsmanship is essential in lay-out work, otherwise assembly may be impossible or the engine cannot be rotated without some portion fouling. Before commencing the lay-out there are certain preliminary matters to be gone into and decisions made, and it is here that skill and acumen applied to the various problems arising will be manifest in the product and help to secure success in operation and production. Preliminary questions may differ in value according to the case in hand, moreover they tend to be dependent on each other and are therefore, difficult to place in a definite order. Every decision will, of course, be governed by the fact that the engine must be in keeping or balance with the rest of the car. As an explanation, the 24 h.p. Ford and 25 h.p. Rolls are both excellent engines, and would no doubt perform each other's work with credit, yet no one with experience would recommend the change over. With an entirely new job it seems obvious that the first consideration is to have a clear idea as to the nature of the work the engine is to perform. Against this may be cited the fact that a medium sized car engine can be used in a truck, or that a baby car engine is used in a motor-cycle, but it will no doubt be found that if they had been prepared for their final jobs at the outset they would have performed more satisfactorily. In addition to the type, class, size, and weight of the vehicle, the work the engine has to do also involves the performance required, and the degree of assistance given by the driver chiefly by way of gear lever manipulation. A basis of comparison between cars and for estimating their performances is contained in what might be expressed as a figure of merit or constant K, which formulated is—

$$K = \frac{D^2 S a R}{TQ}$$

where D = diameter of cylinder in centimetres

S = Stroke of piston in centimetres

T = Driving wheel diameter in centimetres

a = Number of cylinders

Q = Weight in tons

$$R = \text{Gear ratio} = \frac{\text{Revs. of engine}}{\text{Revs. of road wheel}}$$

In other words, K is the power available (size of engine) multiplied by the time (gear ratio) and divided by the weight. When K is high the stress and trouble effect on an engine is low, providing the gear ratio has not been lowered to secure it, because the resulting increase in revolutions entirely offsets any gain, and in extreme cases may render the engine unsuitable for

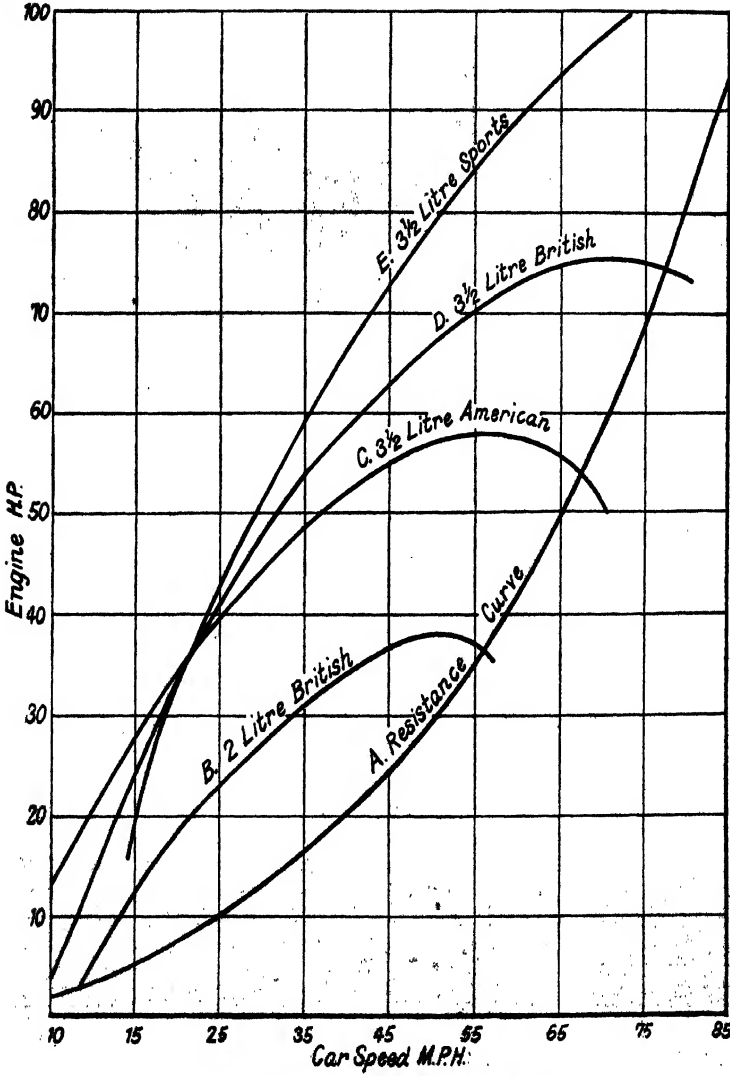


FIG. 1. POWER AND RESISTANCE CURVES

the job. In practice the value of K varies from 100 to 250, the lower value being general in British and the higher in American cars, due to the smaller engine in the heavier car, and *vice versa*. For an ordinary size vehicle we have as typical examples—

	Total Weight	Engine Capacity	Tax £	K value
	cwt.	c.c.s.		
British	$27\frac{1}{2}$	2,000	16	120
American	$24\frac{1}{2}$	3,500	22	200

Since the same performance is expected from both, it will be readily realized that the extra work required from the British engine necessitates different treatment in detail, design, and manufacture to give satisfactory service, than does the American. A high K value usually gives satisfactory maximum speed, but its chief advantage is good acceleration and hill climbing in top gear. The mental comfort given to the occupants by this feature accounts for its popularity, and, in addition, it provides a high average with a low safe maximum speed. Another method of dealing with the performance characteristic is to plot the resistance curve (power required at various speeds) and lay on it the power curve of the engine, and either or both may be actual or estimated. Such curves would be as shown in Fig. 1, where A is the resistance, B the power of a 2-litre British engine, C the power of a $3\frac{1}{2}$-litre American six and D and E $3\frac{1}{2}$ British sixes, ordinary and sports. Vertical distances between the power and resistance curves indicate the surplus available for hill climbing and acceleration, also the characteristic difference between the cars. The power curves also illustrate the fact of tilt about a common point and show that it is out of balance. This is unfortunate, since the amount that could be added at the low end of the 2-litre engine would give only a small advantage in performance to compensate for the large drop in power at the top, which would practically ruin the performance at that end. The student understanding this point will readily realize the importance of the fluid flywheel to the small British engine, since it takes care of the low speed power, and enables the fullest advantage to be taken of the high-speed output. The general methods of tilting the curve will be given later.

We now have to decide the number of cylinders with the bore and stroke necessary to give the required power, bearing in mind that it is affected by the type of combustion chamber, position of valves, and compression ratio. Since bore size forms the base of existing taxation, and quite naturally tax looms large in sales consideration, tax will be found to govern the size of the engine in the majority of cases.

The stroke in relation to the bore varies considerably, and there is no agreement on a definite ratio, while in practice it can successfully vary from 1 to 2. It is generally agreed that the lower value (short stroke) provides the cheaper engine, but a reflection that the tax is affected by the bore size will cause it to be high so to give greater capacity (piston swept volume). A convenient answer to all considerations may be found in making the stroke so that "cut outs" at the bottom of the bores to clear the connecting rods are just not required.

The number of cylinders in current practice are 2, 4, 6, 8, 12, and 16. The 2 is in opposed form, the 4, 6, and 8 in line, and the 12 and 16 in vee form. As a six can be made large enough to develop power sufficient for any ordinary vehicle, we may safely concern ourselves only with the 4 and 6. A larger number than 6 may be called for in racing, or to provide ultra smooth propulsion in luxury cars. These cases will be satisfied by a 12 in vee form, since the vee 8 does not appear to possess material advantages in smoothness, while the straight 8 needs considerable bonnet space, gives difficult carburation, and weighs and costs as much as the 12. The flat twin is simple, and provides more reasonable sized parts than does its competitive four, but any claim for reduction in costs needs to be backed up by careful estimate taken from completed designs. The change from 4 to 6 cylinders occurs in passenger work at about 12 h.p., more in response to the call of fashion than economy. For commercial work the 4 still holds place, but the 6 is now predominant for fast vans and heavier types of vehicles.

With the bore, stroke, and cylinders decided, the next questions are number of main bearings for the crank, length of connecting rod against stroke, height of piston above gudgeon pin, and position of the valves and gear when the overall dimensions are apparent; the lay-out can now be commenced. The number of bearings indicates the cost directly and indirectly by affecting the length and therefore weight of the crank and block; they also affect the smoothness of running, but not by any means alone. Four cylinders may have 2, 3 or 5 bearings, 3 being more general and 5 in larger sizes, with 2 rapidly losing popularity. With small dimensions and stamped webs there will be in service, with 2 bearing cranks, too large a percentage of breakages, and nearly always what is known as "two bearing thump." A generously proportioned crank machined all over, coupled with long rods and pistons, goes a long way towards securing the objective when the increased cost will remind us of the diction that a satisfactory 2 costs as much as a 3. Fortunately the owner of a small low priced car is not too exacting in the matter of thump, even if he is cognizant of it. One and a quarter litre engines should have 3 bearings, and one and a half litre must, is a safe and good generality, and here it is helpful to remember that the pin and journal diameter with thickness and width of webs for a 1 litre 2 bearing will be suitable for a one and a half litre 3 bearing. Five bearings are usually found in larger engines where 3 would lack durability or prove weak unless of uneconomical dimensions. In both the 3 and 5 there will be a temptation to make the centre one smaller than calculated size, but it should be resisted whatever the consequences.

Six-cylinder cranks may have 3, 4, or 7 bearings, and they are more difficult to design and cost much more than a four-cylinder crank of similar capacity, due in the main to their pins being in three planes. It will be found that the forging costs three to four times, and the machining twice, that of a four cylinder. Three bearings at one time held pride of place, and is still strongly supported in America to-day, its advantages being short length and low cost, but its disadvantage in a few words is that it has hardly enough strength to support itself. This causes roughness in operation and rapid wear of the centre bearing due to angular movement, causing rapid bell mouthing of the white metal on both sides. The troubles are minimized by making the webs stiff, reducing the stress by shortening the stroke, drilling the pins, counterweighting and making the centre bearing of large diameter and short length in the order of 4/1 ratio. Those who are inclined to favour 3 bearings will be helped by Taub in *S.A.E. Journal*, Sept., 1929, page 268, but those who have suffered under their use will read it more in sorrow than in anger. Three bearings can quite properly be ruled out and, it being difficult to detect the presence of a 4 or 7 bearing by difference in operation, it is necessary to consider any other points affecting the choice

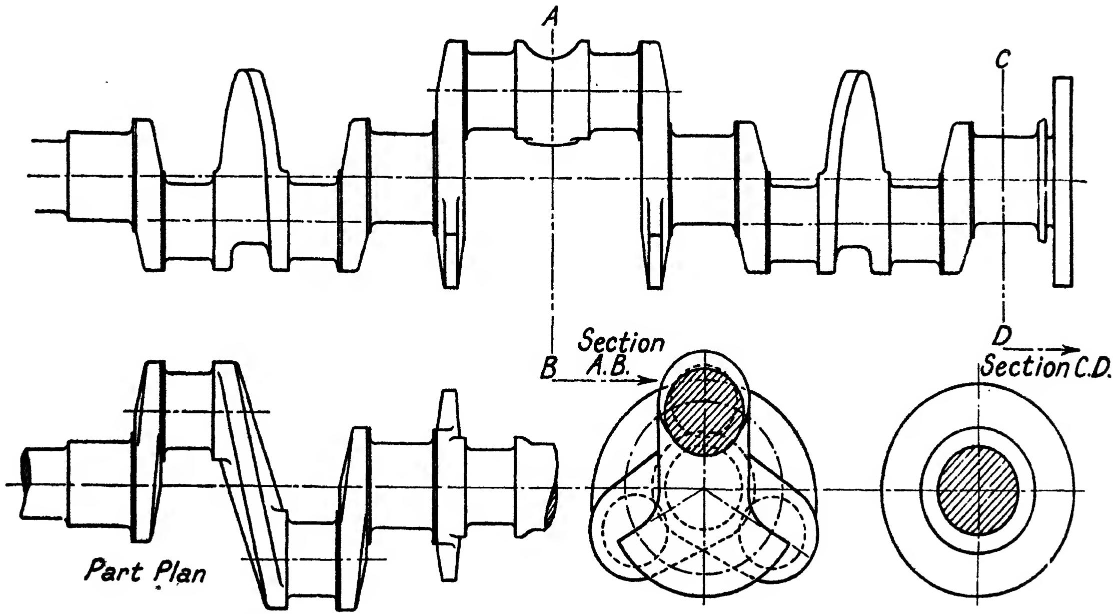


Fig. 2. Four-bearing Crankshaft with Internal Balance Weights

between them. Good proportion and economy indicate or almost demand 4 in a small engine, and 7 in a large one, while machining costs depend entirely on quantities which, if large, bring into use special machines for 7 bearings which cannot be readily applied to 4. Machining all over would appear necessary when full benefits from 7 bearings are desired, but it is not necessary with 4 where machining of the sides, on the short webs, covers most requirements or ordinary needs. Small sixes are nearly always subjected to hard driving, and since enlargement of bearings to cope with the resulting stress puts up costs, counterweighting to reduce loads is necessary and desirable. To keep the grain flow in the stamping the weights in a 7 will be better as separate pieces attached, but with a 4 they can be solid, providing a better job. Fig. 2 shows such a crank where 25 per cent relief of bearing load can be secured. Four bearings machined all over provides some difficulty in arranging, but that shown in Fig. 3 has proved highly successful under most exacting service, and is convenient to produce, being the product of a 60° set square. From the foregoing, there appears to be a balance in favour of the use of 4 bearings in a six-cylinder engine, but before we go on it must be remembered that owing to length and characteristics of any six-cylinder crank there is every chance of torsional vibration occurring, which is usually eliminated by a damper; this should always be placed on the lay-out even if its use is not required at the outset. Load and speed (P.V. factor) together with stiffness, both torsional and lateral, indicates the crank bearing dimensions. Stiffness has much to do with smoothness, and the lack of it may cause bearing failure due to spot loading, while the accepted P.V. factors are under suspicion with to-day's hard driving conditions, and here Heldt in *Automotive*

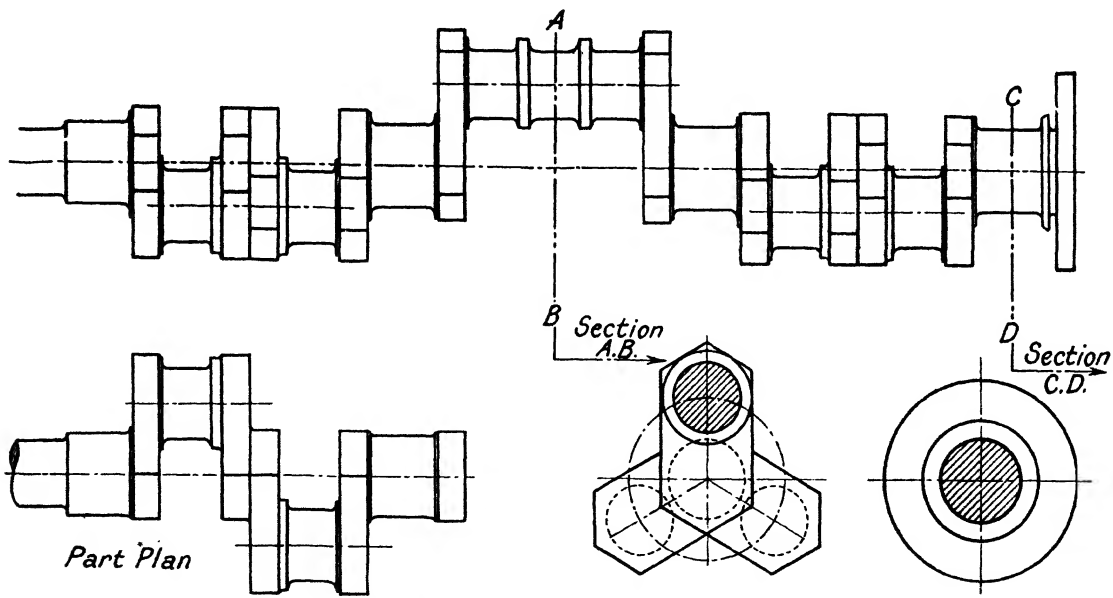


FIG. 3. A SUCCESSFUL DESIGN FOR A SIX-CYLINDER ENGINE

Industries, dated 11th June, 1932, page 842, may be read with advantage. Actual load factors remain unknown until full results are obtained from engine tests, but as a guide for lay-out purposes, which must be checked over on completion with the pressures obtained from a "made up" or theoretical indicator diagram, the big end bearing area should be one-third the area of the piston. This is for an ordinary high-duty engine having normal light alloy pistons, two cylinders between main bearings, and bore-stroke ratio of about 1·5. For a low-duty engine the area can be $\frac{1}{4}$, while a sports engine requires it to be increased to $\frac{1}{2}$ the piston area. The width or length of big end bearings varies from $\frac{1}{2}$ to 1 diameter; the former is suited to short high-duty engines having high oil pressure, and the latter to low-duty splash feed engines. Main bearings are usually greater than the pins in diameter, and 10 per cent will be found a satisfactory amount, while with regard to length the total usually exceeds that of the pins by 10 to 25 per cent. As already pointed out the centre bearings must be up to size and $1\frac{1}{2}$ to $1\frac{3}{4}$ times the pins should give satisfactory results. Flywheel weight is supposed to account for the extreme lengths of most rear bearings, but careful calculation will prove that the latter are in general too long, and that in most instances they need be no longer than the front, especially where this bearing has to take the pull of the chain or gear load driving the camshaft. Webs between mains and pins are most important, since they control stiffness in proportion to the square of their thickness which in turn contributes to the length, so that it is not surprising to find many engines being rough, due to thin webs. Where the use of thin webs is imperative the stroke should be reduced and the main and pin diameters increased so that their circles in end view "overlap." The "draft" necessary for stamping reduces the web thickness, especially if the flash has to be allowed for, but usual variations in this call for filing, chipping, or "running a cutter down," the cost almost paying for proper full machining on the sides which should be called for on all modern engines. Web thickness should be ·7 pin diameter for 2 bearing fours and 3 bearing sixes, ·5 for 3 bearing fours, ·4 for 4 bearing sixes, and ·35 for 7 bearings. Connecting rod length is a factor in the overall height of the engine, and, as may be expected, it is generally too short in present practice. The extreme limits are not worth while determining, and generally the length may be from 1·75 to 2·5 times the stroke, but it is advisable to keep it between 2 to 2·25, while in the low price class it could be reduced to 1·9. The high ratio of 2·5 will be found desirable for slow high-duty durable engines, but would be unsuitable for high speed on account of whip, which can only be prevented by increased section and, therefore, weight. A ratio of 2 is suggested as a reasonable practical economical figure, while 2·25 may be advisable in special cases, leaving racing out. With the length and big end dimensions fixed, it remains to decide on the gudgeon pin location, and if the white metal is to be direct or in a shell. The fixed pin is preferable for quantity work, since it is easier to secure and maintain the desired fit in the piston, whereas the opposite obtains in the floating pin whose advantage lies in reduced weight. The white metal direct in saves weight and cost, gives a better heat flow with a more durable bearing, but if failure occurs crankpin damage usually results, so that quality work demands shells when it ought not to.

However many problems there may be in piston design, the length above the gudgeon pin must be decided on to obtain the height of the block, while the length below completes the bore length. These dimensions vary in practice, and although the average total length may be said to be just practical, the length above the pin may be said to be expedient but inadequate, or in other words, the pin could be moved down with advantage. Modern trend indicates that the length or width of the top land is insufficient, and could be generally doubled, so that it is about $\frac{1}{10}$ of the bore diameter. If we assume that the piston skirt commences at the top of the third land, per contra bottom of the second ring, and is equal in length to the bore diameter, we will have with narrow type rings a total length of 1·2 diameters. With the pin in the centre of the skirt the distance to the top will be ·7, and to the bottom ·5 diameter. These proportions will add about $\frac{3}{8}$ in. to the height of the average 70 mm. engine, but the increased cost will be justified against future service demands, which may even call for their increase to 1·4, (·8 and ·6) against to-day's average of 1·1, (·55 and ·55).

Sufficient data are now available to lay down the cross-section as in Fig. 4, when the path of the connecting rod should be put in carefully and, if there are any tight clearance places, at a later stage they should be drawn out several times full size with the clearance determined if possible by calculation. This section should also include the piston and rod in its path of removal from the bore, which should be free from any interference which would damage the piston or make its removal awkward. Removal of the crank for this operation is very undesirable. This applies to the head even assuming that the big end has been made small enough to pass up the bore.

The student should make several lay-outs, varying the dimensions of the component parts within the limits specified, as by these means the effect of such variation is more readily realized and knowledge gained. The longitudinal section may now be commenced, and it is necessary to decide on the centre distance of cylinder pairs where they are between bearings, but with a full

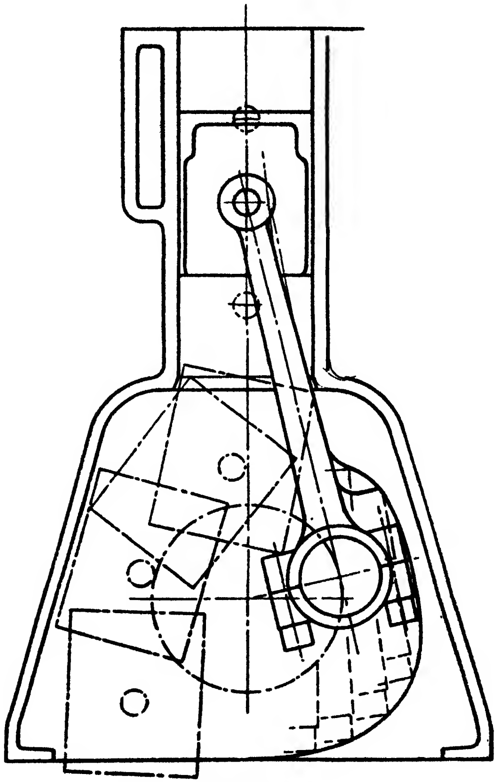


FIG. 4. ENGINE LAY-OUT

number of bearings they ordinarily fit in the crankpin bearing centres. Cylinder pair centres depend on the thickness of wall and space between; they may be 5 and 3 mm. in a small engine, and 6 and 5 in a large one. Increasing the space automatically increases the centres which weakens the block and crank, and must be avoided at all costs; this fact calls for special consideration before inserted liners are used. The smallest centre distance is that where the bores are siamesed, a procedure subject to much adverse criticism from all concerned, and to be avoided where possible. The resulting narrow spaces between bores and valve ports result in difficulties with the "gasket nip" being insufficient, causing blowing, which is becoming more prevalent with increase of compression ratios. Fig. 5 shows a longitudinal section with three different bore spacings and again the student should familiarize himself with the different bearing arrangements and proportions as in the cross section.

A most important decision has now to be made in the disposition and operation of the valves, which in turn controls the combustion chamber shape to a large extent, and divides engines into roughly two classes, namely, side and overhead valves. The latter is considered the high-class type and more of an engineer's job; it develops greater power, costs more, gives more economical service without attention which, when necessary, is easier to apply, but needs more care and skill than is usually available for service operations, and for this reason alone the side valve is often given preference. Until 1935 the overhead valve developed about 15 per cent more power than the side valve. Nowadays the gap has been closed considerably by the increasing outputs from the side type, consequent upon extensive development. With the call for still better performance, the use of overhead valves is increasing, especially in small cars. The inevitable development will no doubt again widen the gap in power difference. The costs difference for similar detail design should not be more than 10 per cent, but since

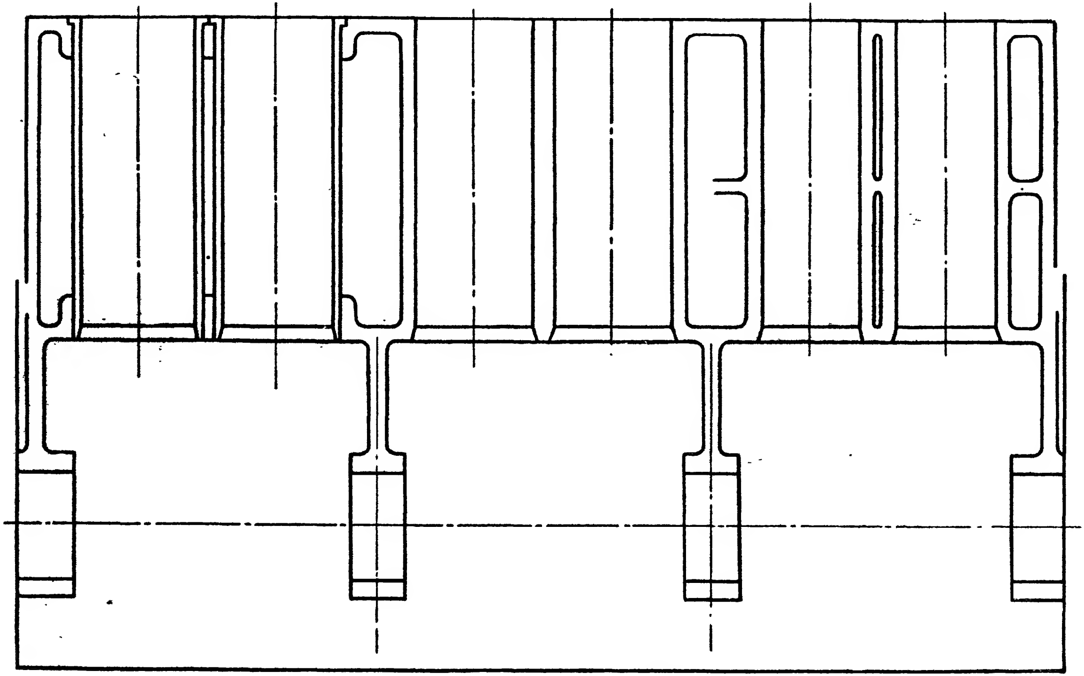


Fig. 5. Longitudinal Section

details are improved to keep up with the higher class it may be 20 to 25 per cent more. The usual valve arrangements are shown in Fig. 6: (*a*) hemispherical, full angle, central plug, (*b*) pent roof small angle, side plug, (*c*) in line central, side plug, (*d*) one over

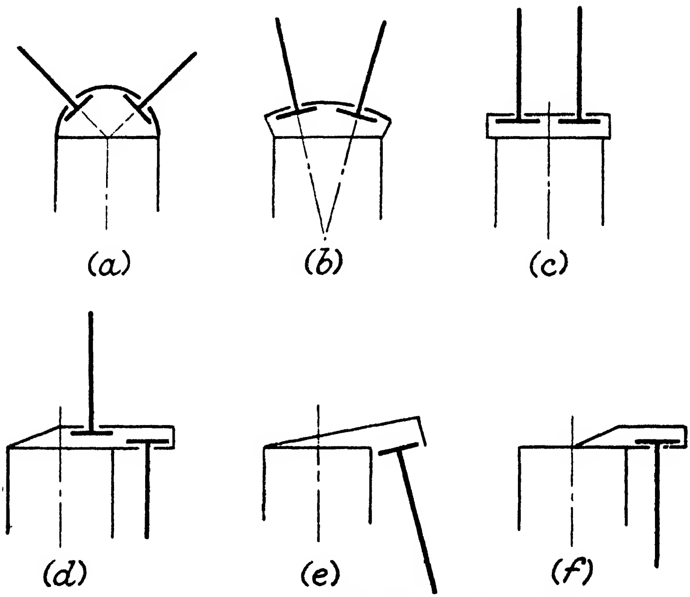


FIG. 6. ALTERNATIVE VALVE ARRANGEMENTS

the other, (*e*) side inclined, (*f*) side straight, and their usefulness in developing power may be taken as descending in that order. In (*a*) the valves are operated by direct cams, or push rods from shafts, (*b*) is usually from an overhead shaft between the valves, (*c*) by push rods and side shaft, (*e*) and (*f*) by side shaft and tappets, (*d*) a combination of (*c*) and (*e*). It is helpful to remember the sequence of valve arrangements by forming them into a mental vision in moving picture form, which should be completed by the full "L" and "Tee" form at the low end. Picturing the movement of valves in this order immediately indicates the stages in combustion chamber design, based on surface against volume, until the practical ideal of half sphere is reached. After due consideration, arrangements (*b*) and (*e*) will be found most suitable to represent the two main classes of engine. Before either can be placed on the lay-out the throat and lift must be decided, together with the type of tappet or rocker. It is here that the power characteristic or tilt previously referred to is controlled, being in the main dependent on the degree of filling (volumetric efficiency). It should be remembered that overhead valves fill better than side valves, increase of revs. causes a decrease in filling, and large valves cause difficulties in slow pulling. Valve diameters may be said to be decreasing on account of the smaller valve being naturally more durable; it stands up to modern treatment better in addition to answering the call for slow pulling without interfering with the top end, due to improved knowledge of port, passage, and chamber design. Valves differ in diameter, the inlet being the larger up to 15 per cent on area. Exhaust valves have been the subject of much study owing to the greater stress imposed by increased compression ratios. As a result it is wise to allow for inserted valve seats in case the means of cooling the ports prove inadequate. Maximum power is obtained when the engine speed corresponds to a gas speed of 190 ft. per sec., and if the ratio of valve to piston area is 1 to 5 the maximum torque will be delivered at about 1,800 r.p.m. Should it be desired to reduce maximum torque speed to 1,000 r.p.m. or lower, as in American practice, the valve area ratio will be 1 to 6, or lower. The question with rockers and tappets as far as lay-out is concerned is the type of follower, and since the mushroom pattern with cam gives the greatest area to the valve lift curve, and the lowest spring pressure and cost, strong reasons should be given if it is not adopted. One of these occurs if the valves have very close centres when the radius heel (solid roller) is the next best. When a rocker arm operates directly on the valve stem it must be positioned to avoid valve guide wear, and this important matter may be attended to after study of the article in *Automotive Industries*, dated 26th April, 1923, page 925. Mushroom tappets are generally offset say $\frac{1}{16}$ in. to ensure rotation by the cam action making best use of the large surface, but as this obviously cannot be done in the case of rockers the oil supply will need careful treatment. Tappet guide holes for side valves may again be formed integral with the cylinder block since cast iron camshafts with "chilled" noses are now available. These operate ordinary commercial case hardened tappets with freedom from the roughing up experienced with case hardened steel cams. This trouble caused engine dismantling, to avoid which the detachable tappet block was evolved, and this in turn should now be avoided since it weakens the cylinder block which needs strengthening to meet increased stresses due to output. Before discussing the integral tappet guide it will be advisable to lay-out both types and go through the pros and cons. It is not now considered necessary to form collars on the valve guides, and they may be plain tubes when the usual tap fit will be found sufficient to hold them in place with the spring location provided by a recess or a pressed steel collar which forms a heat break.

Valve opening and closing points depend on the performance required, and they also affect the power characteristic previously referred to. Ordinarily they may be, exhaust closes and inlet opens 5° past T.D.C., inlet closes 45° past B.D.C., and exhaust opens 45° before B.D.C. For fast work overlap is introduced at the 5° and the 45° are increased, while the opposite takes place for slow work. It should always be remembered that cams of the highest class (a product of most expensive machines used only on large quantities) are subject to a plus tolerance of 2° at each opening and closing point, so that 8° underlap will have to be provided to guarantee no overlap apart from variations due to clearance between tappet and valve. Therefore it is helpful to picture timing as the relation of piston position to the valve opening diagram which resolves in practice to valve fully lifted. It will be helpful to lay-out the valve, spring, and tappet on a separate piece of tracing paper (see Fig. 7), which can be offered up to the lay-out to determine the best camshaft position when the effect of the length of the parts on long and short stroke lay-outs will be interesting. The camshaft must, of course, clear the connecting rod boundary when the advantage of angular valves with certain proportions of

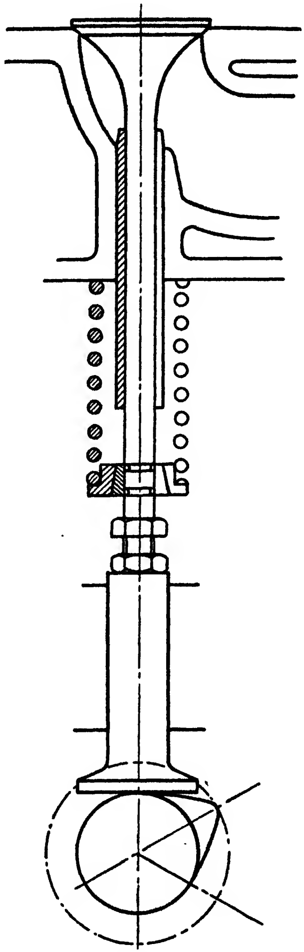


FIG. 7

bore, stroke, and rod ratios will be seen. The position of the cams along the shaft in relation to that of the valves, which in turn is thoroughly mixed up with the positions of bolts, ports, and waterways, is complicated by the necessity of avoiding "lumps" of metal and small cored spaces. It will generally be found that the valves cannot be spaced equally each side of the cylinder centre when the one farther away may be made the exhaust with advantage. Camshaft bearing position may coincide with that of the main bearings, since a supporting web already exists, but if auxiliary drives upset this later on, special supporting webs will have to be provided for the bearings, which, if integral with the block and bear directly on the shaft, never wear out.

With the overhead arrangement the main consideration is to set the valves in line across the head or stagger them, the former being preferable, but it makes a wider head than the latter. Staggered valves generally cause the combustion chambers to be siamesed, and form one lump right along the head; this also applies to vertical valves in line down the head centre. This feature together with attached parts, plug and bolt bosses, leaves very little vital surface for the cooling water to act on and is accentuated when push rods are used by the necesssary tunnels, and by fins due to the method forced on the coremaker in the design. In general it may be said that such heads prevent the full attainment of performance of which the type is capable. Valves in line across the head relieve these difficulties, and render sight holes at suitable places more useful, while underneath bolting (Fig. 8) may be advisable. In the interests of oil consumption it is inadvisable to place the cams in line with the valves in spite of the complication resulting from the necessary staggering of the rocker ends. With the usual arrangement of overhead camshaft bearings, difficulty is experienced in obtaining proper alignment which, moreover, will be interfered with at service or decoking, so that a complete bearing tunnel integral with the head casting, *à la* side valves, appears desirable, but resulting size and

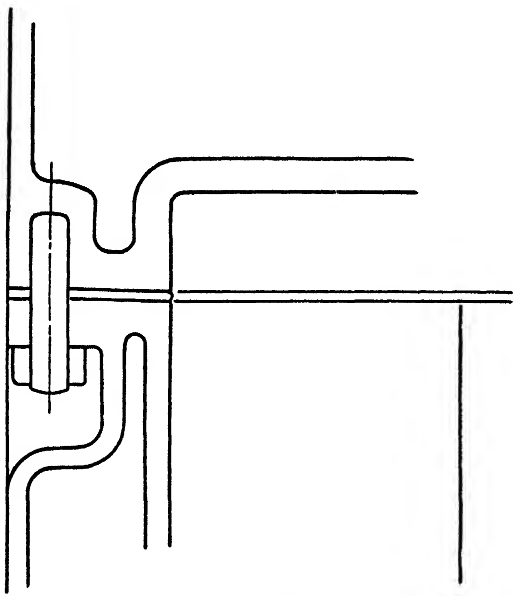


FIG. 8. UNDERNEATH BOLTING

weight makes the compromise of the lower half only integral, very acceptable.

There is now a distinct change in the work, because we have arrived at what may be termed the "accessory half" of the lay-out when it has to be decided what accessories shall be used, where they are to be placed, and how they are to be operated, and since the camshaft drive comes into the picture it will be considered with them. The lay-outs adopted are so varied that it is difficult to indicate a definite line on the matter, which is further complicated by being dependent on the size of the engine, as accessories do not usually scale down, i.e. petrol pumps are the same for all sizes of engine. A start may be made by listing the accessories required and cutting down the number of different positions by deciding if the camshaft drive is to be front or rear. Smoother running is claimed for rear drive due to it being close to the flywheel, but there is no serious proof in support. If there was, the demand for smoothness would surely sweep away the difficulties of accessibility, fouling the steering gear, and encroachment on body space. Front drive may be chosen with safety, since it follows general practice, and uses up the space above the track rod. Accessories include dynamotor or dynamo and starter, magneto or distributor, oil, water, and petrol pumps, fan, and revolution counter.

The dynamotor now practically defunct was a single unit, chain-driven from the crankshaft and functioned as a dynamo and starter.

The starter is placed adjacent to the flywheel, since teeth around the periphery form a convenient gear ratio with that of the pinion. It can lie alongside the cylinder block on either side of an overhead valve engine, but on the side opposite to the valves on a side valve engine. It is not always clear of the steering gear in these positions, and when this occurs, alongside the gearbox has to be resorted to. Since there is a danger of the starter pinion jamming, room must be allowed to reach the square spindle end provided for release. This trouble can be minimized by using a separate hardened steel gear ring on the flywheel. Room must also be allowed for the starter to be removed easily as a unit in case the trouble of "freewheeling" occurs. Instead of a spring controlled pinion the starter of large commercial engines has a longitudinally moving armature spindle to which the pinion is bolted direct.

This type seldom freewheels or sticks and the flywheel teeth need not be backed off.

The dynamo may be at the side of the block or on top of the head, but the latter position is seldom used as it may interfere with tightening down, thus causing the gaskets to blow. The side position is dependent on that applying to starters, and it is often convenient and economical to put it in the camshaft drive and have a chain around the triangle, swinging the dynamo out from one corner of a three bolt flange fixing to obtain adjustment, which is not often necessary. With this scheme it must be decided if the spindle is going to take the chain pull, when arrangements must be made to hold the pinion in approximate position when the dynamo is removed, otherwise the pinion has to be on a separate spindle running in an auxiliary bearing, connecting to the dynamo by a flexible coupling. This separate spindle lends itself to driving a vertical spindle by spiral gears when the distributor can be on top, and the water or oil pump at the bottom. In any case the spiral gears and bearings can be included in the engine lubrication system, which is a great advantage. The most popular method is to swing the dynamo in a cradle at the side to provide adjustment for the endless rubber driving belt which is triangulated round a combined water pump and fan unit to the driving pulley on the front end of the crankshaft. In this case the water pump body can be part of the head or block dependent on the desirability of pushing or pulling the water. If it is in the block a feed pipe along the side with holes at intervals can be integral or in a separate side hole cover.

The water pump may also be driven from the end of the dynamo spindle or from the front end of the camshaft.

The magneto needs to be in phase with the crank, and cross spiral gears on top of, or between, crank and cam gears are well tried methods. With the latter the shaft is at an angle, when the lower end of the spindle may be used to drive the oil pump, which would be higher than the position already indicated, but it avoids the use of outside oil pipes. The magneto may be driven by an auxiliary shaft in the chain line similar to the dynamo drive already given; in fact, it may be described as taking the dynamo's place in this instance. It must be remembered that the ordinary magneto should be horizontal and the ordinary distributor vertical or nearly so in both cases.

Two positions have already been given for the oil pump, but the ideal will be found in the centre of the mass of oil as low as possible, but avoiding slime on the sump bottom getting to the suction hole. A convenient drive is by angular shaft off spiral gears on a side camshaft when the top end may drive the distributor if the angle is not too great. This arrangement for overhead camshaft needs spiral gears from the crank, and although they are rather large the service will be found satisfactory. This shaft may be vertical instead of angular when the distributor may be in the cylinder head, often a very desirable position. If the spiral gears are on the outside a "bump" will have to be put in the block and sump, or the side face moved out to suit. The oil pump position is almost ideal when the gears are on the inside position, which is more difficult to arrange, as will be seen on looking up *Automotive Industries* dated 4th May, 1929, page 702. Other positions for the oil pump are direct on the side of the block, and a short shaft with spiral gears to camshaft, while for overhead shafts, in line with and driving directly from intermediate gears or sprockets, which are generally incorporated in the main drive. The former is preferable as the pump is lower, also outside oil pipes can be avoided. Petrol pumps are operated by mechanical, electrical, or hydraulic power, and all types should be allowed for, since they are gaining in popularity and finality has not been reached. The hydraulic pump needs a convenient place near an oil supply pipe and carburettor, but the mechanical is more difficult as it is operated by an eccentric which should be lubricated as part of the engine oil system. Existing patterns appear to be arranged for fitting to side camshaft engines when a machined face on the block, with eccentric on the cam, is all that is required, but for overhead shafts the placing is more difficult, but in both types the carburettor position may cause the petrol pipe to be festooned around the engine.

Overhead valves provide more room for accessories, and while the push rod type can be treated in the same manner as side valves, the overhead shaft needs different treatment, particularly as regards the main drive, since ready detachment of the head has to be allowed for. At one time a vertical shaft with spiral gears at the ends to the cam and crankshaft, with sets of spirals between to drive the accessories, was very popular, possibly due to the fact that it looked nice on the board; but difficulties in securing quiet operation modified the arrangement to use spiral bevel gears when the intervening shaft was formed into the dynamo, again giving a nice and workmanlike appearance. But, as with gears on side camshafts, variation in torque causes noise difficulties, especially with replacement parts, which is further complicated when the centres alter on crank main bearing adjustment.

Roller chain overcomes these camshaft drive difficulties, and are now in general use in single, double, or triple form according to the power transmitted and/or length. The drive may be in one stage shaft to shaft vertically, but since the camwheel would be large a 2/1 spur gear can be employed at the top, thereby providing a convenient break for head removal. In order to drive the accessories it will be convenient to triangulate the chain with the dynamo on the side of the block, and a vertical shaft from spiral gears off the auxiliary dynamo shaft, as in the manner for side camshafts already explained. To avoid the long length of chain between sprockets, which requires some form of automatic tensioner, the triangle may be made smaller by bringing the top sprocket down to the dynamo level, and replacing the top spur gear by a single step drive. The triangular step can be tensioned by dynamo movement, and the single top step by a tensioner or flat spring.

It will be realized by now that the accessory half of the lay-out is a somewhat stiff proposition, very complicated and confusing, making a final choice difficult, so that the student should study as many drawings and samples of the different arrangements as possible, or better still, make the drawings himself, putting in all the important details.

The enclosure for the front drive may be made integral with the crankcase, but when this is combined with the cylinders it results in an expensive and awkward casting. With combined block and case it is usual to make the enclosure separate and in two halves for a high-grade job, and a flat plate against the block with deep pressed cover for an inexpensive job.

The front housing forms part of the scheme for closing the ends and bottom when an entire absence of "T" joints and only plain flat faces should be called for. Fig. 9 shows a very bad example of "T" joints and curiously enough it is generally met with on expensive cars, and it will be apparent that if the case is of different material to the sump, as it often is, the expansion ratio will cause a great strain on the end covers. Where the front enclosures are in the form of castings there will be a "T" joint for the sump, but the usual cork gasket overcomes any inequality of the two surface levels. A method for closing the front end, which allows for both cast and pressed covers, is to fit a bar across the block as in Fig. 10, which also shows the oil thrower where the front end of the crank passes through.

Closing the rear end depends upon the gearbox

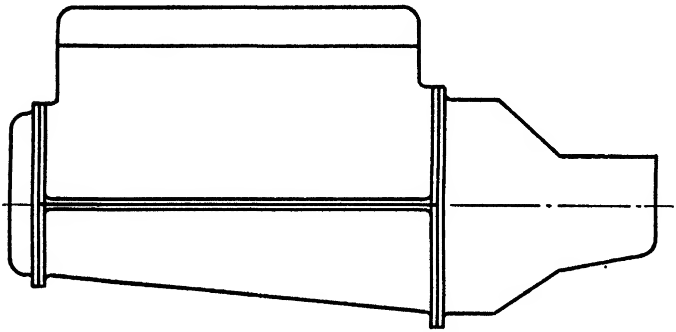


FIG. 9. BAD EXAMPLE OF "T" JOINTS

being either a unit with the engine or separate, and with the former upon the flywheel housing being integral with the block or separate. With engine and box as a unit a separate flywheel housing in cast form will be found most suitable when the machining of the rear face can be done at the same time as the front, and with no projecting arms, etc., the block is in rectangular form. Fig. 11 shows two forms of closure to suit the above arrangement and line up with Fig. 10. That at (*a*) is the cheaper, but (*b*) is preferable since it drains the oil back better with less chance of leakage. The small "T" joints in both cases cannot be avoided.

Both these schemes provide for a flat joint sump where depth depends upon the amount the block is

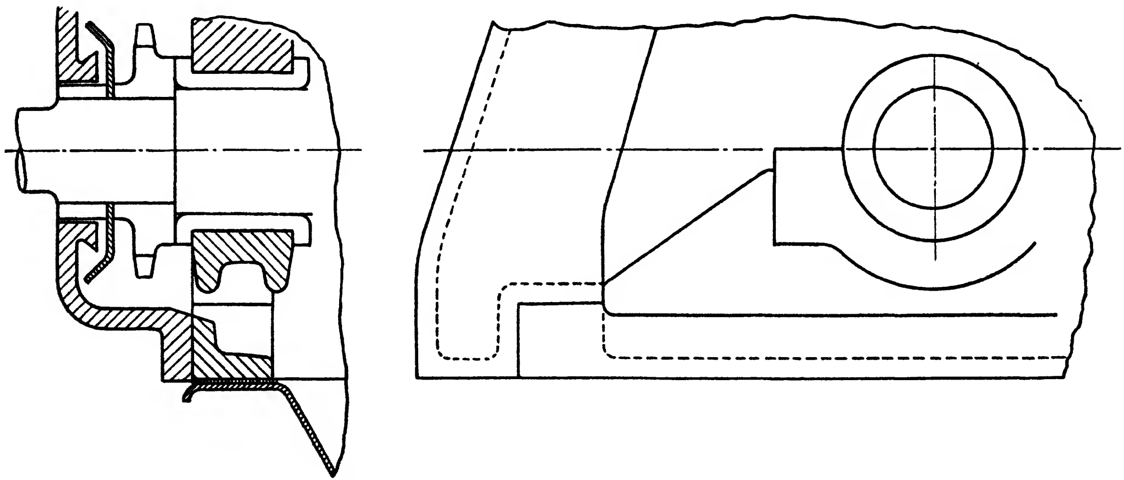


Fig. 10. Method of Front Enclosure

carried below the crank, and in extremes may become practically a flat plate, but it should not go below the flywheel housing level. When the block stops at the crank centre the flat sump joint is broken, "throats" have to be put in to fit round the bearing caps, and although this can be accomplished successfully when the sump is a casting, in pressed form it usually resolves

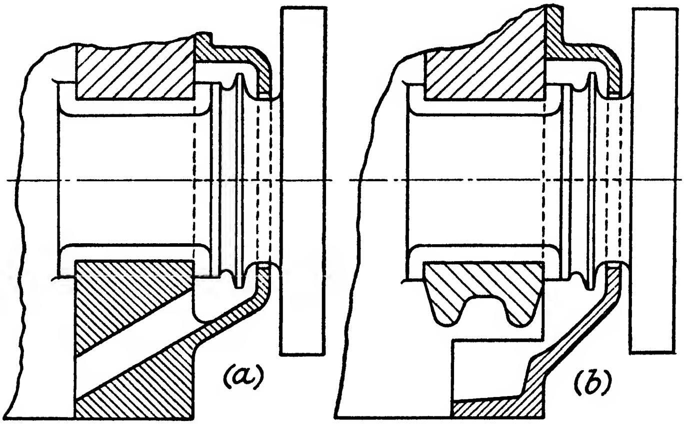


FIG. 11. TWO OTHER FORMS OF FRONT ENCLOSURE

itself into a conglomeration of pieces of steel and cork riveted together.

No difficulty will be experienced in placing the water inlet and outlet after the manner indicated in the paragraph dealing with "Cooling," and this applies to the manifolds, where long bolts, ordinary steel nuts, and two bolt outlet flanges should be avoided like the plague.

With the flywheel and its housing the lay-out is complete except for the bearers, which will now be dealt with in considering suspension, but a word may be added in regard to lubrication before we proceed further. The oil pressure should be kept low, 25 to 35 lb., since high pressure passes a quantity much out of proportion at high speed as will be seen from Fig. 12. In general,

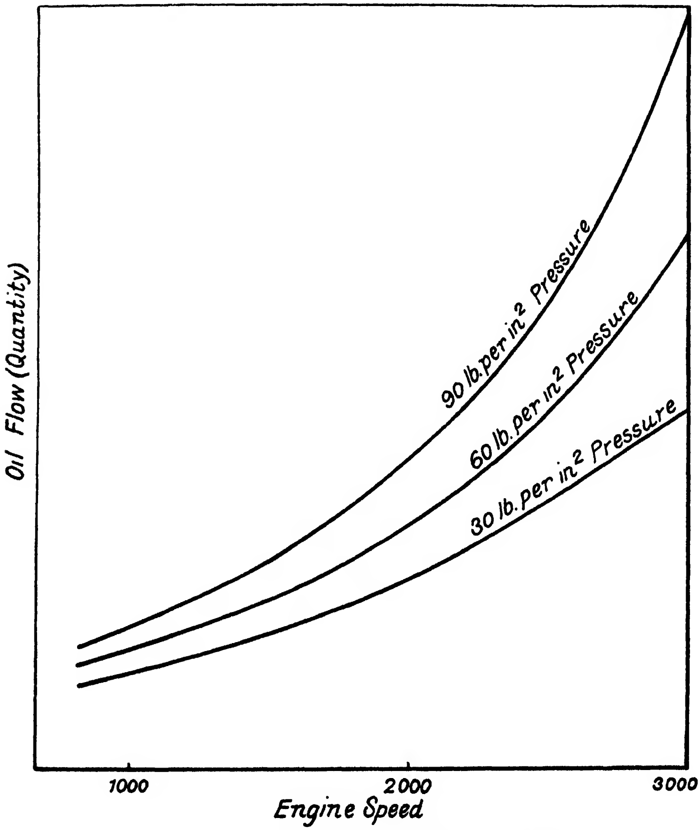
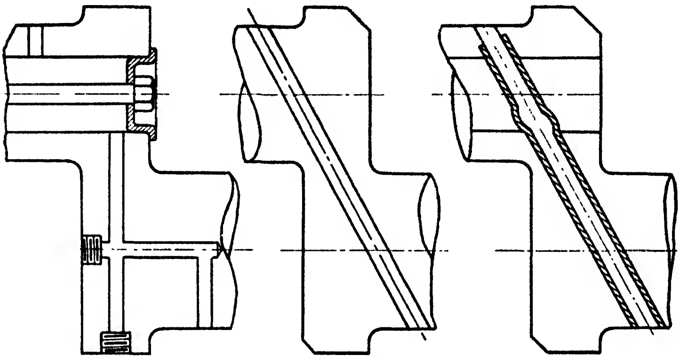


FIG. 12. RELATION BETWEEN OIL PRESSURE AND FLOW

oil pipes look as if they have been taken from a scrap heap, and only attain respectability by being polished to a high degree; moreover they probably contain a lot of solder. Therefore, the lubrication system should be arranged so that pipes are not required, and the oil distributed by drilled holes in the castings transferring from one part to another by plain face joints. These holes must be straight through, with no dead ends to detain dirt and swarf upon blowing out and



(a) (b) (c)

FIG. 13. CRANK DRILLINGS

cleaning. This applies specially to cranks, and Fig. 13 (*a*) shows a very bad but common example for cars in ordinary service, but (*b*) is much better and (*c*) provides for lightening holes in the pins. With the pump down in the oil the suction pipe is replaced by a plain hole drilled in the cover or body, the hole being provided with a wire guard to keep away any nuts and bolts, etc., that may be left in the sump. A gauze should not be used in this position, since it is far better to let fluff go into the bearings in "spread out" formation, where it will be destroyed harmlessly rather than "collect" against a gauze and form a pad to stop oil circulation. A gauze fitted over the whole sump area takes a long time to clog up, usually covering the period between rebores or overhauling, and since hot oil is dripping on it the mesh can be very fine indeed, almost like a petrol gauze, and therefore it effectively deals with any bits that may drop off or sweat out of the castings, etc., before it goes to the pump. The floating oil intake is being used in many designs and as the name implies it takes only the top oil which is likely to be free from dirt and water. The flexible joint is of course on the bottom of the sump. The outside oil filter on the "pressure" side of the pump may have come to stay, but unless the "element" is replaced or cleaned at regular intervals the large area gauze may be preferable. New engines are now prepared by coupling up the pump to an outside supply of "cleaned" oil, and the sump plug is left out or the sump removed so that the oil escapes back to the outside supply when the dirt, etc., removed from the engine in the process is abstracted in a machine. Thus the sump gauze can only deal with oil on service.

The lay-out can now be completed, and the following is a list of drawings nominally required: longitudinal and cross-sections, cross-section through front cover, front and rear elevations, side elevations and plan. Small important places such as section through dynamo pinion or oil pump, where they cannot be included in the section already mentioned, may be made up on a separate sheet or included in or around the other views.

From assumed or made up pressure and speed diagrams the various bearings and parts subject to load and stress are carefully checked over, also the drawings may with advantage be submitted to the producer for criticism.

The final procedure is to make small pieces of general lay-out showing any adopted modification and then prepare detail drawings, making up the final arrangements from these to provide a separate check.

ENGINE SUSPENSION

Examining past suspension arrangements, it is apparent that there has been an extraordinary lack of uniformity; indeed, with the engine and box separate it was usual to use different suspension systems for the two units on the same chassis. Two features emerge, however, in the general use of four arms from the corners of the rectangle, formed by the plan of most engines, and the alternating periods of rigid and flexible fastenings.

The history and facts of the problem will be better understood if we consider the structure to which the engine has to be fixed. In general, this consists of a portion of a rectangular framework whose length is about five times the width, the members being of small section compared with the overall dimensions. If this is laid on a flat plane, with weights at three corners, upon lifting the fourth, or free corner, the adjacent members will be bent and twisted, and if the lifting is continued the nearest weight will rise, and later the one on the opposite corner to it, when the remaining members will also be bent and twisted. In addition, the framework in a vehicle running on the road may also have its rectangle distorted, i.e. one side may be moved backward or forward in relation to the other. Thus we have a sort of moving bedplate to which the engine has to be attached.

Having in mind the usual fixing for a stationary engine, it is not surprising to find early suspension in the form of long webs from the case along the top of the frame, with fitting straps or packing to cover inequalities of the frame surface. It will be apparent that the system upset the normal distribution of stress and localized it at the rear end of the engine. The resulting fractures may have been the cause, but there was a general move to three and four point suspension with, in many cases, the use of a sub-frame which, as the name implies, is formed by two longitudinal members inside the main members fastened at one end to the front cross member and the other to a cross member that already existed or was specially provided. With a separate gearbox it was usual to extend the sub-frame to carry this also.

Then the tendency to separate the engine from frame movements entirely is revealed by the use of ball and/or shackle joints in conjunction with three and four point suspension, but due to vibration and rattle, or as with the sub-frame cost, we find a return to plain four arm fixing. The troubles due to the use of solid blocks may be avoided by replacing them with brackets, made from comparatively thin sheet metal, in a form to provide a certain amount of "give" or resiliency. This system is in use at the present time, and it is admitted that it holds the frame together at the point about the dashboard line. In some instances the front cross member is omitted when angular faced brackets riveted to the frame provide a bed for four angular pads on the engine, generally at the flywheel housing and front cover, held by the usual engine bolts. With the radiator located on arms projecting from the front cover it will be realized that this system provides for the mounting and removal of the complete unit with greatest ease. Breakage troubles were by now absent, but it is not surprising to find that vibration to the body, resulting from these systems, was beginning to be a sales deterrent, especially with the steering box bolted to the engine side. This was countered by the use of antivibrators to four-cylinder engines or the change over to six, together with a general refining in manufacturing processes which, while helpful, did not keep pace with the demand for smoothness; moreover these palliatives were all the time reaching a limit both commercially and technically. Then followed the fitting of leather between the arms and frame brackets, thereby avoiding metal to metal contact. The results being encouraging, leather was replaced by thick woven rubber-cotton fabric, and later by rubber pads. With the consequent release from restraint it was found that vibrations would couple up

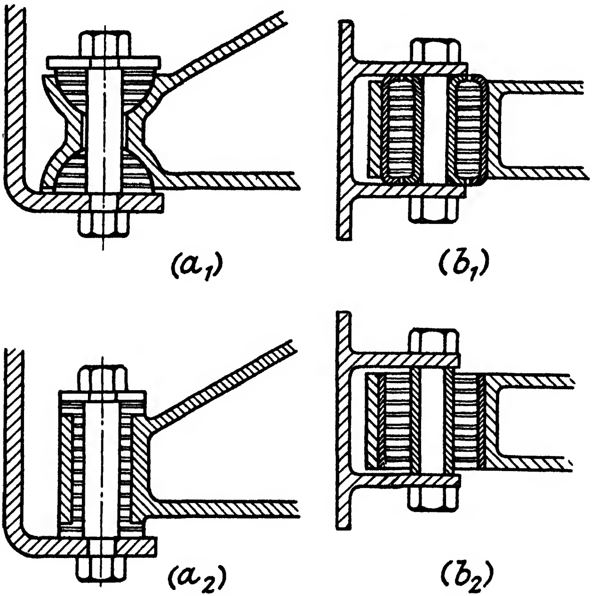


Fig. 14. Rubber Engine Supports

and result in very bad and often impossible vibration periods, which were only avoided by holding the frame sides with special cross and "X" members.

Rubber pads are now in almost universal use, and their design is based upon the fact that rubber while being deformable is for practical purposes incompressible, and that fixing bolts must not make contact with both the arm and the frame. Fig. 14 shows two styles of two different types of rubber mounting a_1 and a_2 used with four vertical bolts, and b_1 and b_2 for horizontal fixing.

It will be realized that the possible variations in placing the fixing points are great in number, and depend on several distinct features, chief of these involving

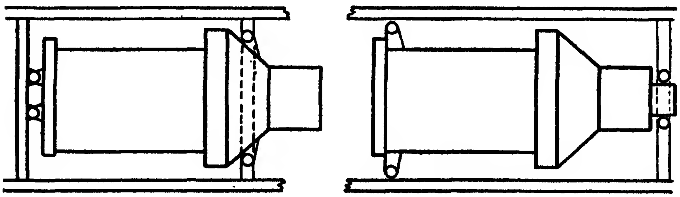


FIG. 15

the question of the rear fixing being on the gearbox or engine. With flexible mountings it is, of course, desirable to fasten the trays filling the gap between engine and frame to one part or the other, but never both, while due attention must be given to the pipes and control rods to allow for the usual movement.

Fig. 15 shows two schemes for the former and Fig. 16 two for the latter, both of which can use wide or narrow (three point) front. The type at (*a*) makes use of an "⌒" section cross member necessitating the engine being "dropped" for removal, while at (*b*) the engine should be lifted and perhaps the cross member detached in order to remove the sump.

The outstanding trouble with ordinary flexible mounting is a general vibration or roughness with a particular period at about 30 m.p.h. from four cylinders, and from six a period at about 16 m.p.h., spoiling the general smooth running obtained to-day. This is commonly assumed to be due to characteristic impulse feature with secondary out of balance in the former case, and torque reaction coupling with the chassis and body in the latter. The vibration can vary from bearable to unbearable by fitting different body styles on the same chassis, and by transferring one and the same engine from one chassis to another; meanwhile a similar variation in gearbox noise may be expected. Now this sort of thing cannot be tolerated in any organized industry

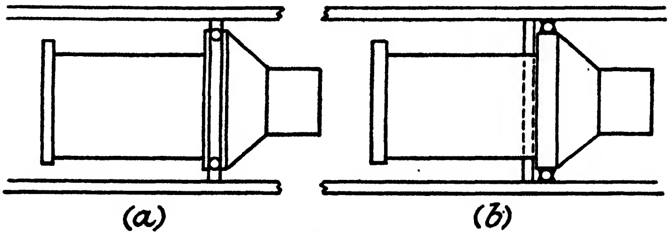


FIG. 16

for long, because the desired state is likely to be upset at any moment by any one of dozens of variations in all directions; therefore it is not surprising to find that a suspension system has been developed which upsets these vibration periods and prevents them coupling up or synchronizing. This system has been named floating power, and it allows the engine to rock to and fro around an axis on a vertical plane through the longitudinal centre of the crank, for several inches at cylinder head level, and is restrained by a leaf spring from engine to frame. This axis, it is claimed, does not coincide with the crank centre due to the power impulse moving along the block in the usual sequence of firing, acting on a body of irregular mass provided by the usual engine or unit. In the original form it is said that this line is from just below the gearbox at the rear end, to between fan and crank at the front, passing through the centre of gravity of the whole unit; this from casual knowledge would not appear unreasonable. If there is any difficulty in realizing this axis position it will be helpful to imagine the engine running at set variations in speed and power, while it is embedded in a large mass of transparent rubber, when stationary spots at the ends could be looked for and marked, since it will be realized that the engine will be rocking near these points and, therefore, the line or axis would vary. The mean providing the least movement or, may be, elbowing would be chosen.

In practice, the best position could be obtained by trials with suitable adjustable fixtures on the test bed and checked over by using the fixing as finally designed. Finding the line by calculation appears impossible. If it is found impossible or undesirable to use a single point at each end, a double point as close and near to the axis as possible could be used. This arrangement makes no design attempt at combating the out of balance secondary force, but it claims to deal with it, so that all forms of vibration are damped out in such a manner that they are undetected. The article by Helt, *Automotive Industries*, dated 6th Feb., 1932, page 184, gives a scheme for dealing with the unbalanced secondaries in a four, by arranging the engine to swing around a bar at right angles to the crank, and in such a position that the axis of the bar in relation to the vertical line between 2 and 3 cylinders corresponds to the centre of percussion. We are told that the centre of percussion corresponds to the position occupied by a door stop, which is so placed to prevent the hinges being stressed by the door crashing against it. This occurs when the stop is about two-thirds along the door from the hinge. Applying this, the axis in question would be across the frame in the region of the clutch fork shaft, but the best position could be obtained in the manner for finding torque axis. The rocking about the axis would be taken up at the front end by a spring or rubber pad, and it is claimed that this scheme prevents the secondary vibration being transmitted.

With such important and beneficial results in sight this latest phase of suspension will, as usual in such matters, develop very rapidly, and very soon definite and complete data on the subject will be available, so that past forms of suspension should only be adopted as a temporary measure.

ENGINE MOUNTING

In the past, most of the difficulties of mounting were caused by frame bolts not lining up with the engine. This was commonly supposed to be due to the frame maker using special rules marked only in $\frac{1}{4}$ in., but this is beyond doubt a gross libel on an excellent branch of the industry invented by a cunning draughtsman to cover up his dimensioning errors. There is no doubt, however, that many thousands of engine bolts have been fitted by means of springing the frame sides with a suitably formed and skilfully used crowbar. Mounting was further complicated by reason of it taking place at varying points in the assembly process.

Nowadays the actual mounting has been made one of the simplest processes, merely lifting the engine and dropping it into place. This ideal is secured by modern works lay-out and organization, which provides an engine at one particular spot at the correct time, with the necessary special quick operating tackle to do the lifting, and makes sure it will fit by drilling the holes in jigs and checking them with receiver gauges. It is customary to take the engines from the test bed, minus certain accessories, such as plugs and high tension cables, and to fit these (straight out of the suppliers boxes) at the point of assembly to chassis, when this is conveniently done on a raised track at right angles to the main track, so that the engine is swung out and dropped into the frame, only a very small distance.

COMMERCIAL ENGINES

For small vans it is general practice to use a passenger chassis with certain modifications, but the engine is generally the same. They perform their duties satisfactorily, except in hilly districts, when complaints of lack of power and boiling may be expected. The correct answer to these is to fit a larger engine with maximum torque at low speed, say, 800 to 1,000 r.p.m., but the usual answer is to lower the axle ratio, and where fan and pump already exist, to modify the radiator.

Fast vans of a similar size and type, and capacity of about 1½ tons, have become so popular that special attention should be given to their engines, which in general may be about 3½ litres capacity, developing about 55 h.p. maximum with high torque at low speed characteristic. The hard work the engines have to do, the manner in which they are operated, and the meagre attention they get is very similar to the ordinary car, so that very similar treatment during lay-out can be given. Two exceptions to this can be given, however, one that the engine being fairly large full advantage may be taken to place the accessories in the most accessible position, and the other that being operated by business people the engine is likely to be run to a standstill, and then taken to a service garage for treatment rather than continuous tinkering at home. Therefore, push rod operated overhead valves may be incorporated in these engines with advantage, since economy in fuel and upkeep, together with added liveliness associated with overhead valves will be very desirable. Improved acceleration, together with the effect of more even torque on the transmission, call for the use of six cylinders for this class of work, to which may be added the lower fatigue effect on the driver due to the reduction of vibration, although floating power suspension of four cylinders considerably reduces this advantage.

Lorry engines as a type may be said to be disappearing owing to economic reasons. In general, they had four cylinders with overhead valves (on account of accessibility), the bearing and oil pressures were low, the rods and pistons were long and heavy, and these, together with the natural limit in speed range, helped to make a very durable engine which, in fact, was the keynote of the design. Extra care was taken to provide for quick replacements, adjustments, and inspection of parts, as instanced in the doors provided for the easy removal of the piston and rod at the side of the crankcase, without disturbing the engine or sump in any way. It will be realized that they were very suitable for the small operator of the mechanic type and conditions of the past.

Coach engines may be said to represent the modern ideas for heavy commercial work. A necessary feature in the lay-out of these engines is to place all the accessories on one side so to allow the driver to be seated alongside the clear side, resulting in an increase of body space. In most instances they are operated by fleet owners who necessarily take long views of a highly competitive business so that first cost appears low down in the list of desirable features. Fuel being the largest portion of running costs make it very necessary to use only the most efficient type of engine, and since the maintenance staff are of a type that can look after almost any mechanical feature the design can be governed by fuel economy, and an overhauling period of, say, 40,000 miles or more. Fuel economy having almost reached a ceiling with the exception of compression ignition, development will be along the lines of an increase in mileage between overhauls and a reduction of their cost when necessary. Another development would be to provide some means to clear the dirty oil, say on the lines of the system used during manufacture, since it appears that fleet owners object to changing oil frequently even more so than car owners. Overhead valves are really necessary for these engines, while long connecting rods and counterweighted crankshafts, both machined all over, are very desirable features, together with a full number of main crank bearings owing to size.

Service difficulties and problems with regard to pistons and cylinders form a separate subject by themselves, but the lay-out should allow for the fitting of hardened wet liners for the cylinders and a very long or wide top land to the piston. The advantage of a circular combustion chamber with full angle valves will probably not be sufficient to necessitate its adoption, but as this arrangement usually calls for the use of two camshafts, an undesirable feature in many ways, this is rather fortunate. One camshaft can conveniently operate a line of valves which are parallel to one another, but the resulting combustion chamber will hardly be good enough, so that the remaining arrangement of valves at a small angle across the engine operated by rockers from a camshaft between is very welcome and efficient. This scheme will be found in general favour, and has already been recommended for passenger cars of the better class, and it only remains to add that the fitting of an additional sparking plug on the opposite side may be economical and at the same time relieve the pinking now usual when picking up speed. Six cylinders are generally employed especially in the large sizes, but as the public show quick and practical appreciation of comfortable running, additional cylinders may be considered. The resulting problems are not easy to solve, so that it would appear preferable to concentrate on the various flexible mounting schemes rather than increase the number of cylinders.

Coach engines (and their chassis) provide much more worth-while engineering than does the average car, and as there appears to be no slacking off in the general rate of improvement in standard the student will benefit by close attention to features in their lay-out.

SECTION XX

FRAMES, SPRINGS AND SUSPENSION

BY

J. BROWN, A.M.I.A.E.

SECTION XX

FRAMES, SPRINGS AND SUSPENSION

FRAMES

General. Large-scale production of popular motor-cars, together with rapid and comparatively recent advance in the technique of all-steel body construction, has inevitably led to the development of "chassisless" vehicles in the most important automobile producing countries. Whilst this section is not concerned with integral body and chassis or "chassisless" construction it is worth noting that the scope for reduced weight and economy of material is appreciably greater than in the case of separate chassis and body. But, even so, this manner of forming a unit is not without its disadvantages. It will be appreciated that a "chassisless" car for the competitive market is primarily a pressed steel product, therefore vital economic reasons (press tool costs, etc.) restrict the body types which can be applied to a given basic model. From such a point of view this disadvantage may be countered by providing also a separate chassis unit to accommodate several body types which would not lend themselves to "chassisless" construction. The range of body types that can readily be adapted to a normal frame constitutes a fundamental advantage, in fact, a separate frame is essential for coachbuilt and custom-built bodies.

Automobile design has reached a stage where the frame is exposed, wholly or in part, only on vehicles in the commercial class. It can be regarded, therefore, purely as an engineering structure without any thought for embellishment in the form of shapely curves on appropriate portions. Dimensional restriction, however, frequently complicates the general problem of design and its influence can be observed in many recent examples where expediency may also have made its contribution. The chief factors controlling the shape of a frame are engine position, method of car suspension, position of floorboards and seats, and position of final drive.

Frame Functions. A summary of the main functions is as follows—

1. To withstand normal dynamic and static loads without excessive deflection or stress in any direction.

2. To support the body, engine, and other units which may or may not be regarded as parts contributing toward structural rigidity of the complete vehicle.

3. To connect front and rear suspension systems.

Before it is possible to deal competently with design a consideration of the nature of applied loads is essential. These consist of—

(*a*) Heavy, suddenly applied loads of short duration; such as those occurring when a vehicle is traversing a road with a broken surface.

(*b*) Combined loads of momentary application at long intervals. For instance, the simultaneous application of brakes, negotiation of a curve, and striking a pot-hole.

(*c*) Vehicle overloading.

(*d*) Inertia loads of short duration; such as brake application.

(*e*) Impact loads externally applied; such as when a vehicle is involved in a minor collision.

Forces applied to the frame under (*a*) may be of a spasmodic or periodic nature and therefore require the fullest study in their effect. Loads under (*e*) need the least consideration, since they are mainly connected

TABLE I

SIDE-MEMBER STEEL SPECIFICATIONS (See page 238)

	Mild Steel Plate	Carbon Steel Plate	3 per cent Nickel Steel Plate
Carbon . .	0·15–0·20 per cent	0·22–0·27 per cent	0·25–0·35 per cent
Manganese .	0·40–0·60 per cent	0·50–0·70 per cent	0·35–0·75 per cent
Silicon . .	0·7 per cent max.	0·07 per cent max.	0·30 per cent max.
Phosphorus .	0·05 per cent max.	0·05 per cent max.	0·05 per cent max.
Sulphur .	0·05 per cent max.	0·05 per cent max.	0·05 per cent max.
Ultimate .	28–32 tons/sq. in.	32–36 tons/sq. in.	34–45 tons/sq. in.
Yield . .	14 tons/sq. in. min.	16 tons/sq. in. min.	20 tons/sq. in. min.
Elongation .	25 per cent min. on 4 in.	20 per cent min. on 4 in.	20 per cent min. on 4 in.
Tests . .	To be bent cold through 180 degrees on to its own thickness, the bend being made the worst way of the grain with specimen edges rounded. Material to withstand this treatment without cracking.	To be bent cold through 180 degrees on to its own thickness, the bend being made the worst way of the grain with specimen edges rounded. Material to withstand this treatment without cracking.	To be bent cold through 180 degrees on to 3 times its own thickness, the bend being made the worst way of the grain with specimen edges rounded. Material to withstand this treatment without cracking.

with the design of bumpers and their mountings, and even at that, is a point frequently disregarded—mere decoration being the reason for such fitments.

Magnitudes of road shock forces are beyond the range of practical calculation, but a general guide to their limits may be taken from the weight increment usually allowable before the frame strikes the axle buffers (see Table VI, page 264). Combined cornering and braking loads are possible and for a given vehicle depend for magnitude upon the friction available between tyre and road. Interlocking of tyre tread and road surface does in practice sometimes produce a coefficient in excess of unity.

Calculation of forces due to cornering can be made from an assumed limiting coefficient of friction, which, in order to maintain stability, is directly proportional to the vehicle's wheel track and inversely proportional to twice its centre of gravity height.

Expressed mathematically—

$CF = \frac{Wt}{2h}$ when vehicle is on point of overturning.

$\frac{Wt}{2h} = W\mu$ to keep vehicle in circular track.

$$\text{thus } \mu = \frac{t}{2h} \qquad (1)$$

where $CF =$ centrifugal force.

$W =$ weight on wheels.

$t =$ track.

$h =$ c.g. height.

With μ thus at its maximum and so limiting velocity at a given radius, the calculation of weight transfer from one side to the other during cornering can be followed by reference to Fig. 1 where the moment of the primary couple is

$$= \frac{WV^2h}{gr}, \text{ and moment of balancing couple}$$

$$= Ft$$

Hence $Ft = \dfrac{WV^2h}{gr}$

$$\text{and } F = \frac{WV^2h}{grt} \qquad (2)$$

Body weight side transfer

$$= \frac{W_bV^2(h_1 - h_2)}{grc.} \qquad (3)$$

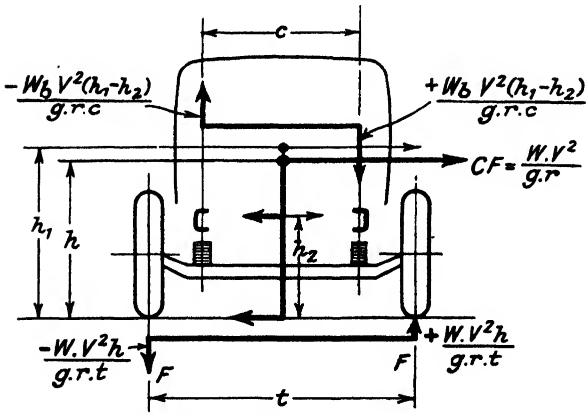


FIG. 1. FORCES ON FRAME DURING CORNERING

W = vehicle weight
W_b = body weight
V = velocity (linear)
t = track
h_2 = centre of gravity height of chassis
h_1 = centre of gravity height of body
h = centre of gravity height of vehicle
r = radius of turning circle (approx.)
c = transverse distance between body supports
F = weight transfer

The calculation of braking loads can be followed by reference to Fig. 2 which represents an ordinary touring car equipped with semi-elliptic springs and brakes on four wheels. Heavy lines denote primary couples and lighter lines secondary couples produced during braking. A retarding force $W\mu$ is equal and opposite to the

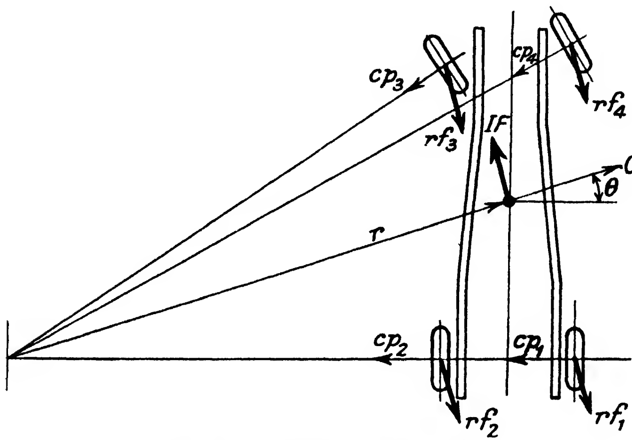


FIG. 2. COMBINED CORNERING AND BRAKING FORCES

cp = centripetal force IF = inertia force
rf = retarding force CF = centrifugal force

inertia force acting from the vehicle's centre of gravity and thus producing a couple of moment $W\mu h$.

This couple is balanced by an opposite couple of equal moment acting through front and rear wheels.

Weight transfer from rear to front

$$= \frac{W\mu h}{w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Secondary couples which are produced in this instance have the effect of transferring weight on the frame from rear to front. Assuming that the chassis centre of gravity lies on the neutral axis of the frame side-member, which is reasonably true for practical calculation, effective load transfer on the frame will be from the body, the inertia force of which is equal to the body retarding force. That is, $W_b\mu$. A couple is thus produced due to the body centre of gravity height being above that of the chassis, and is balanced by a

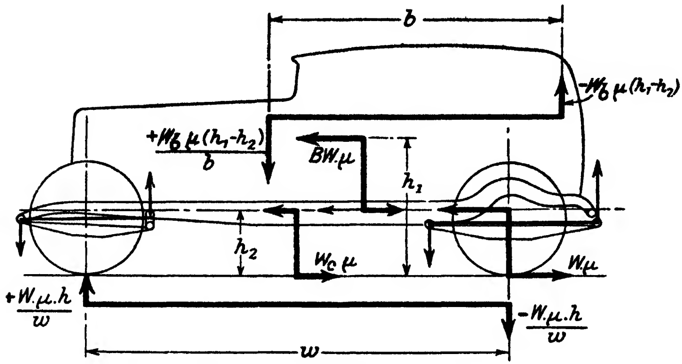


FIG. 3. FORCES ON FRAME DURING BRAKING

W_c = chassis weight
b = distance between body supports
w = wheelbase

couple of equal moment acting at front and rear body connections to the frame.

Since W_c and W_b together must equal W

$$W\mu h = W_b\mu h_1 + W_c\mu h_2$$

Moment of couple due to $h_1 - h_2$

$$= W_b\mu(h_1 - h_2)$$

$\therefore$ Body weight transfer

$$= \frac{W_b\mu\,(h_1 - h_2)}{b} \quad . \quad . \quad . \quad (5)$$

Although brake location is indicated by secondary couples in the particular case of Fig. 3, it should be clearly understood that weight transfer is absolutely independent of brake location, which means that for a given deceleration for a given vehicle, weight transfer from rear to front is constant, whether the brakes are applied through front, rear, or all wheels. If rear brakes only are used, the retarding force obviously will be directly proportional to the momentary rear axle weight and not the total weight. In this case weight transfer will be—

$$= \frac{W_r \mu h}{w + \mu h} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Another condition arising during braking is brake-drum torque reaction, which acts through the springs in such a manner that front eyes support less weight and rear eyes correspondingly more. Determination of reactions at the spring eyes is simply a matter of equating the moment of the couple ($W_a \mu r_e$) to that acting along the spring. (W_a = momentary axle reaction).

From weight transfer (4) momentary forces acting downwards at the wheels are—

$$\text{Each front } \frac{Fw}{2} + \frac{W\mu h}{2w} \quad . \quad . \quad . \quad . \quad (7)$$

$$\text{Each rear } \frac{Rw}{2} - \frac{W\mu h}{2w} \quad . \quad . \quad . \quad . \quad (8)$$

Couple produced by retarding force and tyre radius is of moment for front springs

$$= \left(\frac{F_w}{2} + \frac{W\mu h}{2w}\right) r_e \mu \quad . \quad . \quad . \quad (9)$$

Where r_e = effective tyre radius.

and for rear springs

$$= \left(\frac{R_w}{2} - \frac{W\mu h}{2w}\right) r_e \mu \quad . \quad . \quad . \quad . \quad (10)$$

∴ resultant downward load for each spring eye due to braking—

Front spring rear eye

$$= (7) + \frac{\left(\frac{Fw}{2} + \frac{W\mu h}{2w}\right)}{s_f} r_e\mu \quad . \quad . \quad (11)$$

Front spring front eye

$$= (7) - \frac{\left(\frac{Fw}{2} + \frac{W\mu h}{2w}\right)}{s_f} r_e\mu \, . \quad . \quad . \quad (11a)$$

Rear spring front eye

$$= (8) - \frac{\left(\frac{Rw}{2} - \frac{W\mu h}{2w}\right)}{s_r} r_e\mu \quad . \quad . \quad . \quad (11b)$$

Rear spring rear eye

$$= (8) + \frac{\left(\frac{Rw}{2} - \frac{W\mu h}{2w}\right)}{s_r} r_e\mu \, . \quad . \quad . \quad (11c)$$

Combined cornering and braking forces are also within the realm of practical calculation, but their determination is only necessary, as a rule, for detail stress analysis. Reference to Fig. 2 indicates a graphic vectorial method of determination of forces acting at any particular point. Centrifugal force acting from the general centre of gravity is balanced by centripetal forces at each road wheel, and the tangential inertia force also acting at the centre of gravity is balanced by parallel retarding forces at each road wheel. The slight diagonal transfer due to centrifugal force ($CF \sin \theta$) may be neglected for practical purposes. The resultant force of the resultants of cp, rf, c_1p_1 r_1f_1 etc., must equal the resultant of CF and IF, and act in a direction opposite.

Torque reaction upon the frame is produced by the engine and transmission gear. With an engine rotating clockwise viewed from the front, an upward reaction at the nearside mounting is equal in magnitude to a

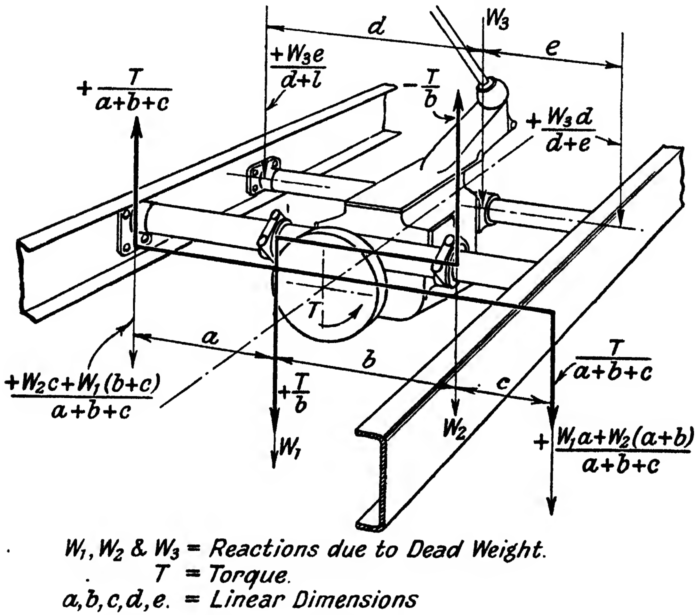


Fig. 4. Diagram showing Vertical Loads on Cross- and Side-Members due to Driving Torque Reaction and Static Weight of Gearbox

downward reaction on the offside mounting point. The gearbox, if independently mounted, will also produce reactions of a similar nature except that their magnitude is increased proportionally to the gear ratio concerned plus engine torque for reverse and minus engine torque for forward speeds. Fig. 4 diagrammatically shows reactions on side-members from gearbox torque.

Deflection. Until recent years the significance of general stiffness was not universally recognized. It is to be expected, therefore, that many variations in frame design are being produced, and no doubt will continue to be produced until designers, by considerations of weight reduction and economy, gravitate to less diverse forms of construction. In this country, progress in design to a large extent is restricted by less advanced production methods than those employed in certain other countries.

Fundamentally, the general problem is one of vibration and, in effect, it requires a consideration of disturbing forces set up by unbalanced forces whose periods must not be equal, or nearly so, to the free periods of various portions of the vehicle structure, of which the frame is a principal member. Unbalanced forces are produced by road-tyre reaction and are mainly the vertical loads acting upon the wheels whilst the vehicle is in motion. The designer, having no control over the magnitude and frequency of such dynamic forces, can therefore only guard against (*a*) a state of resonance and (*b*) disturbing forces of such magnitude that might otherwise incite forced vibrations of undesirable amplitude.

The point can be illustrated by reference to the all too familiar state of shimmy. The primary unbalanced force, which tilts the axes of the front wheels produces a disturbing force known as gyroscopic kick, which in itself is not periodic, its frequency being controlled by the road surface and the vehicle's speed. Gyroscopic kick may be accentuated by unsuitable geometry of steering connections, road wheel linkage and other causes not associated with the frame, which in general can only be regarded as a contributory and possible cause of shimmy. If the gyroscopic kick is not damped adequately and it has a period corresponding to, or approaching the free transverse or torsional periods of the front end, violent shimmy can be expected. The transverse period enters into the case owing to criss-cross component disturbing forces, and the torsional period owing to up and down components on either side. The frame designer's main contribution to avoid shimmy is a frame which is as rigid as practical both in torsion and two plane bending, in order to maintain free periods well above frequencies of the disturbing forces.

The body structure and its system of mounting have a profound bearing upon the degree of stiffness which is necessary in the frame. From this point of view a body may be an asset or a liability. An open top type of body is usually much less stiff than that of a closed body for the same weight and comparable size. A coachbuilt body also compares for stiffness very unfavourably with an all steel body of similar weight and size. For convenience various mounting systems can be roughly divided into three groups, namely—

(1) A closed body (private car) rigidly attached to the frame, where mutual stiffness is the object.

(2) A closed body flexibly mounted upon the frame.

(3) An open body, or one deliberately designed to be flexible, but rigidly mounted.

Group (1) typifies present-day practice and is no doubt the most satisfactory for private cars. Group (2) represents a practice fast declining in favour of the first system. Group (3) covers those vehicles which must depend entirely upon the frame for a sufficient degree of general stiffness. Commercial vehicles and private cars with open bodies, including drop head coupés are examples.

The term "rigidly mounted" is not strictly accurate as invariably some kind of insulating material is interposed between the body and frame. However, this is merely an anti-squeak device and the thickness of insulation is kept to an absolute minimum in order to avoid appreciable relative movement.

Lack of rigidity in the body, particularly its scuttle, or in the frame front end is usually manifest by a violent lateral vibration of the radiator relative to the scuttle under certain road and speed conditions. Even by tying scuttle and radiator by V stays and mounting the radiator flexibly upon the frame, it does not essentially follow that a satisfactory structure will thereby be accomplished. For instance, if the relative transverse movement be of such an order that the radiator has to be mounted upon one small central pad of rubber, to avoid wrenching the V stays off body or radiator, the resulting low transverse period of the front superstructure may well have a resonant effect and introduce shimmy, even though the frame itself is beyond suspicion at the forward end. Undue flexing of the rear and centre portions of the frame, in the event of the body not being stiff enough to resist, causes a lozenging effect upon the scuttle in relation to the radiator. It is, therefore, vitally important to consider fully general stiffness of (*a*) the frame, (*b*) the body, and (*c*) the scuttle structure, all in relation to one another.

A satisfactory degree of frame stiffness largely depends upon weight distribution and the relative positions of principal masses. For instance, the distance between the body and front axle is proportional to the amplitude and inversely proportional to the natural frequency squared of the front end system for a given frame section. Particularly if the body contributes towards mutual stiffness it is obviously advantageous to keep this distance to a minimum. For a private car the engine is invariably mounted flexibly, the natural frequency of such a system being designed to suit the engine's characteristics and although of a considerably lower order than that of the frame, possibility of resonance with road-wheel disturbing forces must not be overlooked. In order to reduce the amplitude of forced transverse vibrations in the frame from or to the engine unit, its centre of gravity should be as close as possible to the body dash.

A very approximate relationship between the torsional oscillation of the body-frame system and the torsional stiffness of the frame, is given by

$$\text{Time of oscillation} = 2\pi\sqrt{\frac{Wk^2}{T_c g}} \quad . \quad . \quad . \quad (12)$$

where W = Body and passenger weight.

k = Radius of gyration of the laden body about the frame polar axis.

and T_c = Stiffness in lb. ft. per degree of the frame.

The values of T_c may vary considerably in actual practice since it is here assumed that the body does not contribute toward general stiffness. A satisfactory order of frame torsional stiffness may be anything from 400 lb. ft. per degree to as much as 1,000 lb. ft. per degree, depending on factors previously mentioned.

To visualize the effect of torsion upon a frame structure it will be of assistance to imagine a frame with an infinite number of cross-members of constant depth; in other words a flat plate. If equal and opposite couples are applied at each end, deflection of a character shown in Fig. 5 (*A*) is produced. (A strip of celluloid sheet will serve as a demonstration of this deflection if twisted from the ends.) Proceeding a stage further by adding torsionally weak section side-members, it will be found that the angle of twist on the plate is reproduced on each of the added sides. It will also be seen that a comparatively small angle of twist produces an appreciable relative vertical displacement of the side members. If torsionally stiff sides were added, it is obvious that for a given torque,

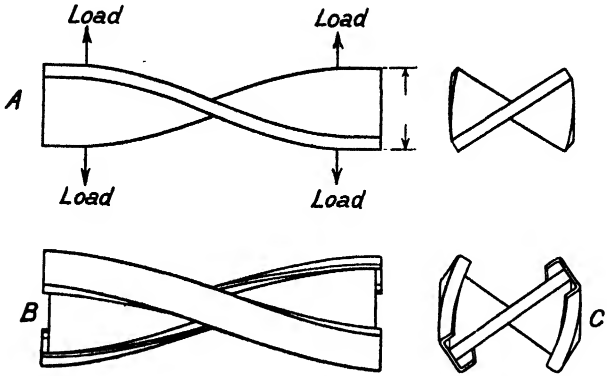


Fig. 5

deflection would be considerably reduced. But since bending (due to vertical forces other than end couples) is also present in a frame, efficiency is not necessarily served best by designing principal members to resist combined bending and torsion.

All end couples can be resisted by various members in combined torsion and bending, or bending only or torsion only. If the frame structure is to resist couples at either end by bending only, it is obvious that the structure should substantially consist of diagonal beams arranged as an X, where in effect one beam is loaded at its ends and supported in the middle by the lower beam which in turn is supported at both ends. Under these conditions both members are analogous to "freely supported beams" and should be proportioned to produce constant stress for efficient material distribution in obtaining maximum stiffness. Fig. 6 indicates diagrammatically the X frame and the effect of

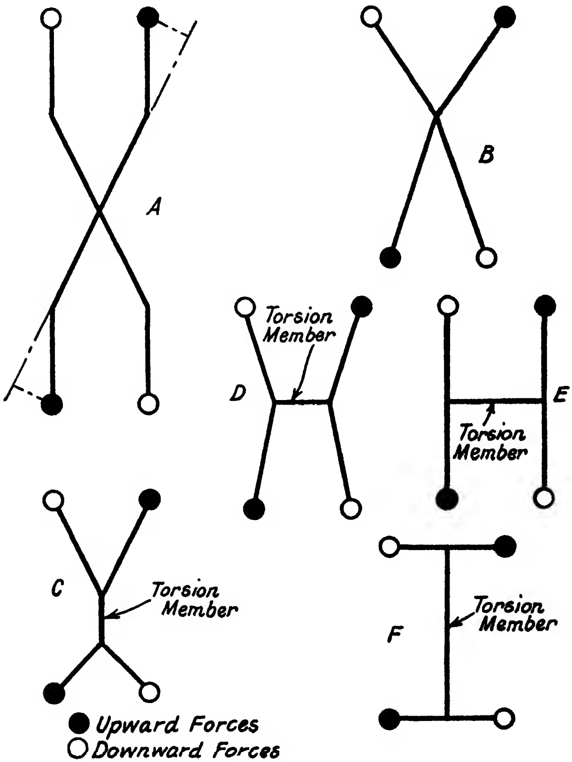


FIG. 6. THE X TYPE FRAME.

variations from the basic structure, which can be related to most present-day types. These variations are usually as follows—

(*a*) The addition of parallel extensions to each end of the X and the application of end couple forces to these extremities thereby introducing torsion into the X-members (Fig. 6 *A*).

(*b*) Mis-alignment of X-members with or without parallel extensions, as (*A*). This would appear to introduce torsion into the cross-members, but, since the junction of the two beams is in practice encastre rather than free, torsion does not occur if the centre tie-up is effective (Fig. 6 *B*).

(*c*) Mis-alignment of X-member in conjunction with a central torsion member having a longitudinal axis. In this case no torsion is applied to the X-member by reason of mis-alignment other than torsion in the central portion (Fig 6 *C*).

(*d*) Mis-alignment of X-member in conjunction with a transverse and central torsion member. In this instance again no torsion is applied to the X-member by reason of mis-alignment other than torsion in the central portion (Fig. 6 *D*).

(*e*) In effect (*D*) taken to an extremity where the central transverse torsion member represents a portion of the former conventional design of "ladder" frame (Fig. 6*E*).

(*f*) The so-called "backbone" type in which the centre torsion member (as *C*) is extended to one end and connected to a transverse member applying the primary torque (Fig. 6 *F*).

Another important structural type is the rectangular-plan box section frame where resistance to general twist mainly depends upon longitudinal torsion members and in a lesser degree upon their resistance to bending.

In order to carry the body, an X-member is usually supplemented by side-rails, i.e. bending members closing the side of the X. The effect of these additions is *incidentally* to increase resistance against general twist to a very inefficient and minor extent, since it will in this instance be a measure of the torsional resistance of the X beam sections. (Transverse members, for the same reason, will be of little value at the ends of the X.) When the side beams are extended, however, they are of major value in restricting torsion applied to the X legs arising from offset forces (Fig 6*A*), since this secondary twist is replaced by bending in the side beams (Fig. 7). It is important, therefore, that side-members should be increased in bending resistance, as such torsional moment in the X legs is increased in order to maintain a given degree of overall stiffness. Transverse X-members at the extremities of the parallel extensions to the side rails introduce torsion in the extensions and in the cross pieces themselves, but the *incidental* value of this depends purely upon the torsional stiffness of the sections concerned. Fig. 8 illustrates an early type of X frame.

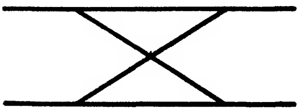


FIG. 7

Plano-vertical forces including those of a dynamic and static nature are usually supported by side-members either alone or in combination with body sills. Beyond torsional resistance offered by the X legs, the principal X-members resisting primary end couples do not add to stiffness in general vertical bending. By introducing a central transverse X-member connecting the side-members via the X tie-up, resistance to bending is materially increased owing to the distribution of bending moments over the X-member. In the case of "backbone" types the torsion member is also the principal bending member, sides being dispensed with completely.

Due consideration for transverse rigidity is essential, since it influences the choice of section for certain principal members. In this respect the body design and its frame connection are factors which have a bearing on the problem. At one extreme, where the body is comparatively rigid transversely and is mounted in such a manner that virtually no relative deflection can occur, the only frame portion which materially affects the issue is that which is not supported by the

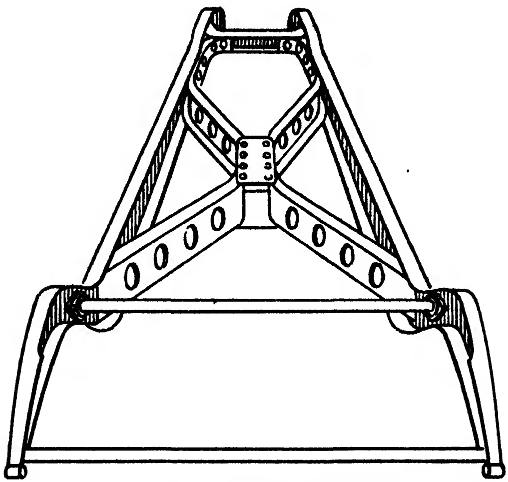


FIG. 8. EARLY X FRAME

body, usually the front end. At the other extreme, the system of mounting or maybe the body itself offers little assistance to frame transverse stiffness, in which case the complete frame must be considered from this angle. Where the front end only is to be dealt with, transverse stiffness depends upon the side-member section and its resistance to bending about the section's neutral axis, if the two sides are not connected by an effective web. A cross-member at the end does not constitute a web, and in this instance, each side front extension is analogous to two cantilevers with the point of contraflexure half-way between the first body connection and front cross-member. Therefore, maximum resistance is offered by a section in itself efficient for horizontal bending, and, as the sides are also to resist vertical forces, the best practical compromise is probably a box section narrow at the point of contraflexure and increasing in width rearwardly to the dash and forwardly to the front cross-member (Fig. 9).

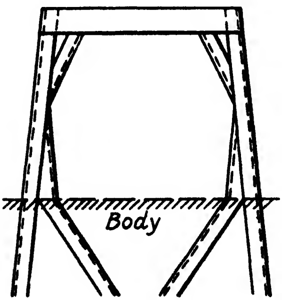


FIG. 9

If the whole length of frame is to be transversely stiff, the type of frame has an important bearing upon measures to be taken. An X-member type for example is inherently stiff in this respect, at least for that distance within the X, especially so if no mis-alignment occurs in the cross beams. One diagonal member becomes a strut and the other a tie, the order depending upon the direction of applied forces. "Ladder" and "Backbone" types represent a low degree of inherent transverse stiffness.

Local Deflection. In the foregoing paragraphs only an outline of deflection problems related to principal members has been stated. A study of frame details and methods of fabrication is of equal necessity, however, since the degree of overall stiffness depends also upon the effectiveness of main joints and the efficiency of detail attachments. The more common causes of local deflection in various members may be listed as follows—

1. Body mounting brackets overhung from side-members, thus introducing torsion if the sides are not adjacently braced by cross-member (Figs. 10 and 10A).

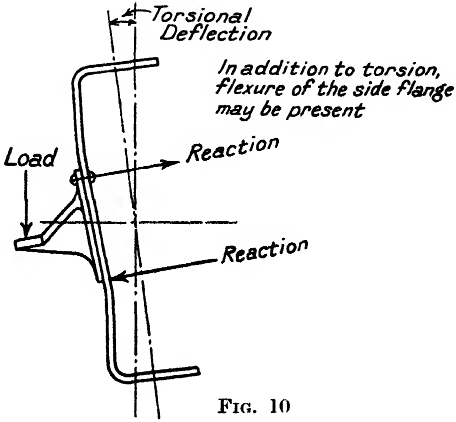


FIG. 10

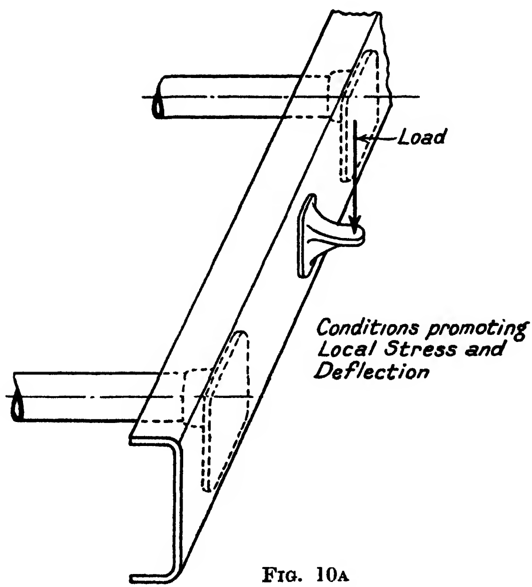


FIG. 10A

2. Body or engine mounting brackets, etc., with faulty attachment to the members concerned, whereby local flange distortion takes place in the members.

3. Overhung or offset spring attachments or wheel link anchorages with inadequate support to prevent local torsion or flange distortion..

4. Badly designed and overhung body mounting

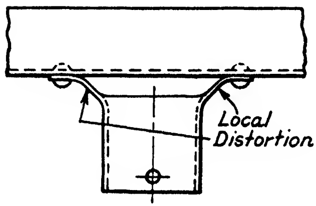


FIG. 11

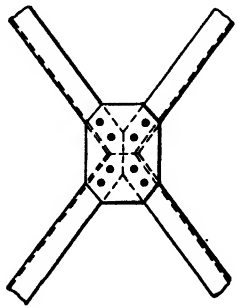


FIG. 12

brackets, which in themselves are subject to local distortion (Fig. 11).

5. Faulty connection of X legs in centre when frame is of X-member type.

It has been previously indicated that several variations of X-member design fundamentally affect centre tie-up requirements, and it follows that if torsion is avoided the conjunction can be of a very simple form highly resistant to bending (Fig. 12). If, however, the X design is such that great resistance to twist is essential in the tie-up, several methods of treatment are available, although each may be found more costly and of greater weight than in the case where torsion is not present. Fig. 13 illustrates one method of dealing efficiently with cross-member torsion, where the axis of twist is disposed longitudinally. Fig. 14

shows a typical X centre where the axis is transversly placed. Obvious disadvantages in the latter design lie in (*a*) form of section and (*b*) the provision of transmission clearance which further reduces torsional stiffness of the section.

Frame Sections and Fabrication Methods. Sections available for the main structural members are limited

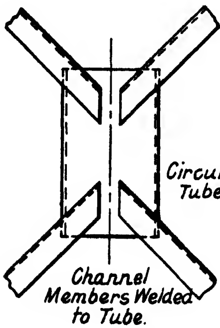


FIG. 13

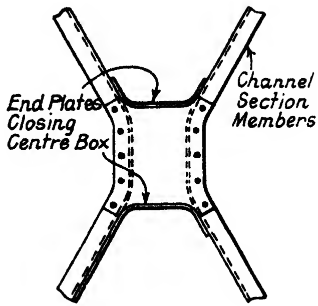


FIG. 14

to what is commercially and economically possible with pressed steel in combination with certain forms of welding. The open channel and built-up box sections are, for these reasons, the basic shapes at the engineer's disposal. A valuable addition is, of course, the welded or drawn tubular section. Summarizing the comparative merits of these sections—

1. *Channel.* As a consequence of its uniform thickness it is only moderately efficient in resisting vertical bending, and very poor in resisting transverse bending.

2. *Circular Tube.* Poor in bending but offers same resistance in all directions. Represents maximum efficiency in resisting torsion.

3. *Square and Rectangular Tube.* For combined bending and torsion this section is the best compromise

available. For a given wall thickness and cross-sectional area torsional efficiency is reduced when departing from square to rectangular shape, but increases in efficiency for vertical bending.

* Sherman has demonstrated the value of I beam sections in bending as compared with the orthodox channel and it may be taken that an I section represents maximum efficiency in vertical bending. The work of Sherman following that of other research workers emphasizes the need for caution in applying the beam theory to an automobile frame. Since stress is proportional to deflection, experimental data published by Bach in 1909 indicate that the channel is not an efficient bending member to the extent suggested by the beam theory. For a standard channel $4\frac{3}{4}$ in. deep he showed that the stress value obtained by measurements from his beam bending experiments was greater by 7 per cent than stress calculated by the orthodox method. The conditions of loading were in each case similar, namely, the load applied as in Fig. 15 and the beam freely supported at each end. The percentage differences for $8\frac{3}{4}$ in. and $11\frac{3}{4}$ in. channels, were 17 and 24 respectively, and by the sole modification of the loading point, that is, from the vertical neutral axis to the centre of the web the following differences were noted:—9 per cent, 24 per cent, and 31 per cent respectively.

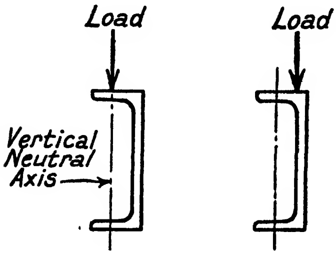


FIG. 15

Sections may be formed and members connected together by various means such as spot-welding,

* See *S.A.E. Journal*, November, 1936, p. 454.

continuous welding, riveting, and bolting. For sheet steel up to about 12 S.W.G., spot-welding is eminently practical and economical especially for building up box sections. It is less efficient than continuous welding which does provide section unity throughout the length of member. Welding in one form or another is frequently used in the assembly of the whole frame to the exclusion of bolts or rivets.

Commercial vehicle frames seldom include much welding since side-member material is usually nickel steel or medium carbon steel, neither of which is suitable for welding. Such frames are invariably bolted assemblies, the bolts fitting the holes in each mating part, thus providing positive location. In addition, adequate end pressure is also available to ensure a rigid joint.

Joints may be made by cold or hot riveting, and for speed of production and low cost, it is a method only surpassed by spot-welding. Hot riveting is superior to riveting cold because in the latter case, end pressure is not usually sufficient to prevent relative movement developing between the parts joined after comparatively short service. By contraction when cooling off, the hot rivet makes a firm joint and is therefore less affected by vibration.

Frame Materials. A steel for pressed frame construction must possess a fair degree of malleability and ductility in order that it will not be damaged during pressing, and to be economical in weight it must have a good yield point. Steels which more or less possess these necessary properties are mild sheet steel, carbon sheet steel, and sheet nickel alloy steel. Wrought steel is a material which is adversely affected by heat and mechanical treatment applied in frame manufacture, therefore it is usual to normalize the material in order to relieve any internal stresses developed in these processes. For ordinary private car frames where the material is comparatively thin, the frame is left "as worked." Mild and medium carbon steel side-members are invariably channelled cold for small and medium frame sides, but those for commercial vehicles and made from nickel steel are pressed hot for convenience of manufacture. The analysis and physical properties of typical frame steels are given on page 215. Members primarily designed for stiffness are usually made from mild steel, which is satisfactory for all normal private car frames where stresses are of a low order. Welded mild steel tube is commonly used for all types of frame where tubular cross members are incorporated.

Production and cost advantages of pressed steel mounting brackets, etc., are of such importance that stampings and castings are only used in special circumstances or when the design is of such a nature that it is beyond the scope of presswork. Malleable iron and mild steel castings, however, are frequently employed for the various minor details not strictly associated with the frame structure. From time to time cast frames have been introduced with the primary object of obtaining stiffness. One such example consists of several aluminium alloy castings including the dashboard all bolted together and so forming a single unit. When designing for a given degree of rigidity aluminium alloys do not provide the advantage of lower weight. Although aluminium is one-third the weight of steel bulk for bulk, it is practically cancelled because the modulus of elasticity for aluminium is also one-third of that for steel. The advantage of aluminium for a cast frame is a practical one from a foundry point of view because this metal can be readily die cast.

Small malleable castings have the merit of being easy to machine; they are castings of iron, which when first made are in the condition of cast iron, and

TABLE II

Materials Used for Frame Brackets

Material	Use	Physical Properties (Average)			
		Yield	Ultimate Tensile	Reduction of Area	Elongation
Mild Steel . .	Stampings	34,000 lb./sq.in.	50,000 lb. sq. in.	40%	20%
Med. Carbon Steel .	,,	56,000 ,,	90,000 ,,	55%	24%
1% Nickel Steel .	,,	68,000 ,,	94,000 ,,	50%	20%
Hard Steel . .	Castings	36,000 ,,	80,000 ,,	20%	15% on 2 in.
Soft Steel . .	,,	27,000 ,,	60,000 ,,	30%	22% on 2 in.
Iron (Black Heart) .	Malleable Castings	36,000 ,,	52,000 ,,	18·5%	13%

are made malleable by subsequent treatment without fusion, consisting of converting the combined carbon of white cast iron into an amorphous uncombined condition. Occasionally, annealed steel castings are employed for detail work, and can consist of bessemer, open hearth, crucible, or any other steel. The raw material ordinarily employed however, consists of steel scrap and pig iron, the bulk of the charge being scrap. A wide range of materials from which to choose is available for stampings and embraces mild, medium carbon and alloy steels, but practical limitations of forging impose considerable restriction upon design.

Stress. Although reservations must be made when applying the beam theory to a pressed steel structure such as a frame, it is the only theory which can be rationally applied when estimating stress and deflection in the principal members. The least that can be said for approximations so determined, is that comparative data are obtained and as such better than nothing. Stress usually reaches a very low maximum when a frame is designed for a high degree of rigidity, but in the case of commercial frame side-members they are often subjected to high stresses. Such stresses together with associated deflections are easily determined by orthodox graphical methods when applied separately to side-members and cross-members.

In the case of determination of side-member section, or conversely determination of stress for given sections, a recommended form of procedure consists of treating each side-member as a beam freely supported at four points. Reactions from the appropriate static loads are then calculated in the usual manner by equating moments about the axle centres and proportioning spring eye reactions inversely to their distances from the axle. Therefore, under static conditions now being considered, the spring eyes, if

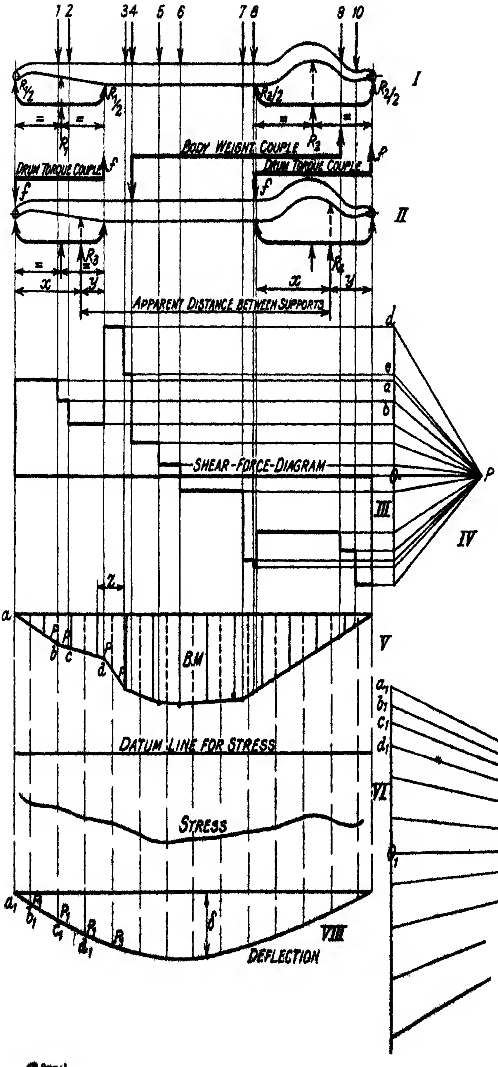


Fig 16 Static Load Diagram

Key	Static Weights
1	Radiator, etc., with water
2	Engine front support
3	Engine rear support
4	Dashboard body front
5	Gearbox front servo, etc
6	Gearbox rear body middle
7	Batteries, etc.
8	Body intermediate
9	Petrol tank with petrol
10	Body rear and luggage

symmetrical about the axle, will each support half the calculated reaction about the axle concerned.

For any loaded beam in equilibrium and freely supported the sum of the upward forces is equal to the sum of downward forces, and in the case of a side-member it is assumed that the loads are known and reactions are to be calculated.

Fig. 16 shows a static load diagram at I, and III represents a conventional shear force diagram obtained by plotting loads to a convenient vertical scale and their location to a horizontal scale. By means of a link and vector polygon shown at IV a pole OP is chosen and loads projected from III to a perpendicular from I. Lines aP, bP, etc., are produced parallel at ab, bc, etc., until a B.M. diagram is completed as at V. Braking loads can be dealt with in this manner: indicate modified body support loads on the load diagram (II) and, as the brake drum torque producing reaction f at spring eyes has the effect of moving the reactions R_3 R_4 rearwards, it is equivalent to having unsymmetrical springs. Under conditions where maximum braking on rear wheels only occurs, frame bending moments are clearly increased to an appreciable extent owing to apparent frame span modification due to effective rearward movement of rear supports. By additional front wheel braking, this effect is to some extent counterbalanced by likewise alteration of front supports. The approximate positions of R_3 and R_4 can be determined by the equation—

$$\frac{R}{2f} = \frac{x+y}{x-y} \left(\text{where } x \text{ and } y \text{ are distances of this ``reaction'' from each eye}\right). \quad (13)$$

Where $x + y =$ spring centres.

And $R =$ reaction on axle due to body weight transfer.

NOTE. Proceed by calculating body weight transfer and indicate in load diagram new frame loadings, and then find vertical forces at spring eyes due to drum torque. Calculate reactions on true axle centres due to new frame loadings, and for brake drum torque find by equation (13) apparent distance between supports, reactions for which must again be calculated because the apparent distance is not necessarily equal to the wheelbase.

It follows that by repeating the process of equating moments with new body reactions spring eye reactions are found, thus the braking shear force and bending moment diagrams can be drawn.

The BM vertical scale is —

BM at any section

$= m.n.d.\ OP$ where 1 in. on shear diagram $= m$ lb.

and 1 in. on horizontal scale $= n$ in.

d is depth of BM at that section.

OP is polar distance in inches.

Assuming that general stress in a side-member is reasonably constant in its whole length, the beam theory can be applied to derive the approximate general deflection at any point by simple graphical methods. Divide the B.M. diagram into strips of equal width (Fig. 16V) as indicated by heavy lines and then sub-divide these strips as shown by dotted lines, the lengths of which are pricked off to a reduced scale on the vertical line (Fig. 16 VII). The pole P, is chosen and both polygons can be completed by similar procedure as when constructing the B.M. diagram. In Fig. 16 datum lines are drawn horizontally, but these may be inclined depending upon the chosen position of P, or P.

The scale of a deflection diagram obtained by the foregoing method is as follows—

$$\text{deflection in} = \frac{3 \,.\, m \,.\, n \,.\, d\,, OP \,.\, O_1 \,.\, P_1}{E.I} \times S$$

where E = modulus of elasticity.

I = moment of inertia of section at S.

S = scale reduction (i.e. if a, b, $= \frac{1}{5}$ of a, on B.M. then $S = 5$).

d = depth of deflection diagram at S in inches.

Fig. 17 is a reproduction of an actual set of diagrams prepared for a frame subsequently produced for a 28 h.p. private car.

Moments of inertia and section moduli for sections covering normal pressed frame requirements are given in Table IV. Corner radii essential for working the

TABLE III

(See page 263)

Static Deflection	Periodicity
in.	
8	66
7·5	68
7	71
6·5	73
6	76
5·5	80
5	84
4·5	88
4	94
3·5	100
3	108
2·5	118
2	132
1·5	150

material without damage are neglected, as the differ ence involved is of no practical consequence.

Frame Fittings. The design of a frame must include provision for various units and their details of mounting. The radiator may be either solidly or flexibly mounted on a cross-member, one arrangement consisting of a single pad of rubber arranged centrally upon the cross-member, the idea being to avoid transverse radiator movement which might otherwise be transferred from the frame. In addition to this support the radiator is steadied fore and aft by V stays anchored at the body scuttle, the stays also preventing relative transverse movement between scuttle and radiator. It is occasionally the practice on private cars to mount front wings at their forward end upon stays attached to the radiator framing in order to reduce possibility of relative transverse movement between wings and radiator. Forwardly controlled heavy commercial vehicles are generally provided with radiators mounted solidly, since the driver's structure in combination with the radiator provides a superstructure of reasonable stiffness.

The subject of engine mounting has already been covered in Section XIX.

If the gearbox is separately mounted as in the case of many large private cars and commercial vehicles, it is often found advantageous to use rubber insulation in order to prevent vibration being transmitted to the body *via* the frame.

Where a draglink system of steering connections is to be provided for, the drop arm shaft is best pivoted either above or below the side member in such a manner that it is unnecessary to provide a large clearance hole in the side web of the section which would unduly weaken the frame. Fig. 18 shows a conventional type of steering box bracket, the flange of which is riveted to the frame channel, the latter being reinforced by an additional angle stiffening piece to prevent local deflection likely to be detrimental to good steering qualities.

Mechanical brake operating details as a rule, are mounted on the frame both for commercial vehicles and all types of private car, and as these fittings require rigidity so that lost brake pedal motion is not excessive, pedal pivot or servo motor overhang from the frame without bracing is better avoided. A combination bracket so designed to accommodate pedal pivot and servo motor reactions is worth consideration since by

these means no frame deflection can occur between the two units.

The spring to frame connections generally require some form of cross-member support, particularly if the spring base is wider than the frame. Overhang of the

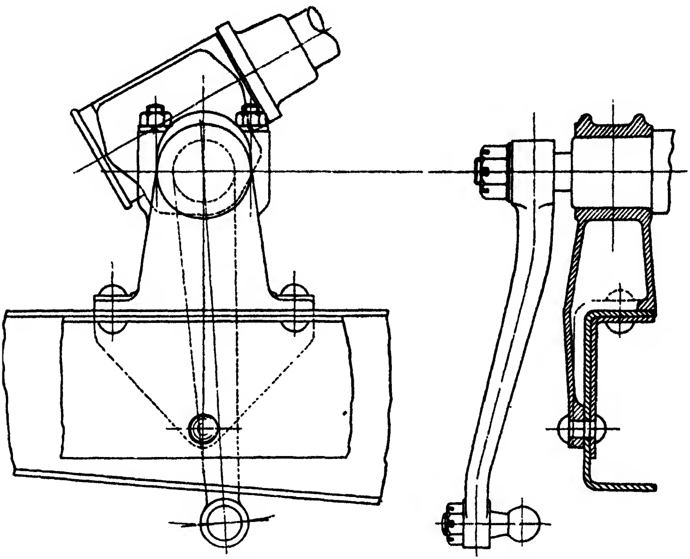


FIG. 18. STEERING-BOX BRACKET FIXING

frame supports can also be met but in a less efficient manner, by incorporating torsionally stiff side members.

SPRINGS

Road springs used for automobiles consist of one or a combination of three types differing fundamentally in the nature of stress they are each designed to sustain, namely—

1. Laminated springs—bending stress.

2. Helical springs—combined bending and torsion stress.

3. Torsion bars—torsion stress.

Laminated Springs. This form of spring is used in conjunction with both axle and independent sus-

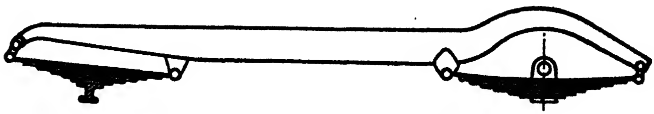


FIG. 19. SEMI-ELLIPTIC SYSTEM

pension. Its variety of application and design consists of the following when applied to axle suspension—

(*a*) Semi-elliptic, longitudinally disposed (Fig. 19).

(*b*) Semi-elliptic, transversely disposed.

(*c*) Quarter-elliptic, longitudinally disposed (Fig. 20).

(*d*) Cantilever semi-elliptic longitudinally disposed (Fig. 21).

With independent suspension the following types and arrangements are frequently used—

(*a*) Transverse semi-elliptic.

(*b*) Transverse quarter-elliptic.

(*c*) A combination of (*a*) and (*b*) for front suspension.

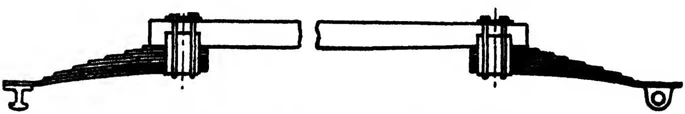


FIG. 20. QUARTER ELLIPTIC SYSTEM

(*d*) Longitudinal semi-elliptic springs (divided axle system).

Laminated springs are built up from a number of steel plates or leaves, the semi-elliptic consisting of what is variously termed an eye plate, back plate, or master leaf which is the longest plate, terminated by eyes or bosses for the purpose of anchoring the spring.

The eye plate is followed in descending order of length by intermediate plates, the shortest known as the short plate. In heavy duty springs one or two plates adjacent to the eye plate will often be extended around the eye in order to safeguard the eye plate against

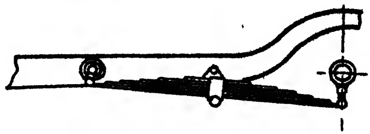


FIG. 21. CANTILEVER REAR SPRING

failure. Fig. 22 illustrates the various types of spring eye in common use.

For light and medium private car springs, leaves are held together by means of centre bolts.

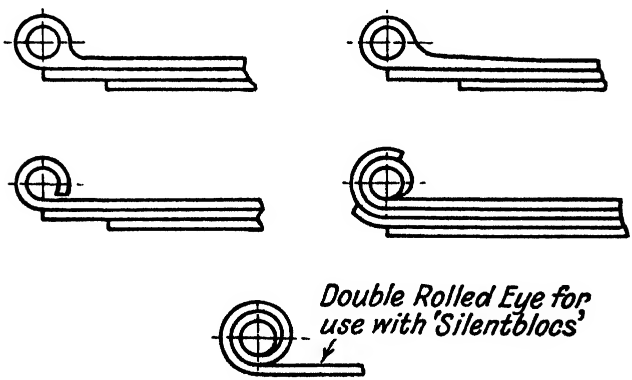


FIG. 22. TYPICAL SPRING EYES

Clamps, embracing the whole of the spring centre portion together with axle locating nibs, are usually in present day practice incorporated on heavier duty springs instead of centre bolts. In order to prevent separation of laminae during severe rebound, springs should be substantially clipped, particularly with respect to their eye plates. There are many designs of clip available; three of the more usual are shown in Fig. 23.

A form of clamping device which avoids a centre

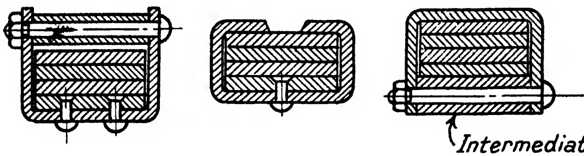


Fig. 23. Spring Clips

bolt by the use of nibs is shown in Fig. 24, where it will be noted that two set-screws are employed to hold the spring plates together, and that an integral dowel on the bottom clamp is provided to locate the spring on the axle pad.

Deflection. A semi-elliptic spring is clearly a beam which fulfils the conditions of "ends freely supported," Therefore, with certain reservations mentioned later, the beam theory can be applied in obtaining deflection and stress for a given loaded spring. If stress throughout the spring is constant, its deflected shape will be a curve of some true radius, and in order to obtain this condition the moment of resistance must be constantly proportional to the bending moment on any section normal to the direction of load. If we consider a hypothetical plate spring of constant thickness, loaded at the middle and supported at each end, it is obvious that the plate must be widest at the loading point to maintain a constant stress requirement. Actually the plan of such a spring takes the form of a rhombus and if this is cut into a number of narrow strips as indicated in Fig. 25 (B) and re-arranged as shown in Fig. 25 (*C*) and 25 (*D*) the nucleus of a perfect laminated spring is thus obtained.

In consideration of a spring's length in proportion to the magnitude of its deflection, the theoretical ideal does not hold by reason of the difference in rate of

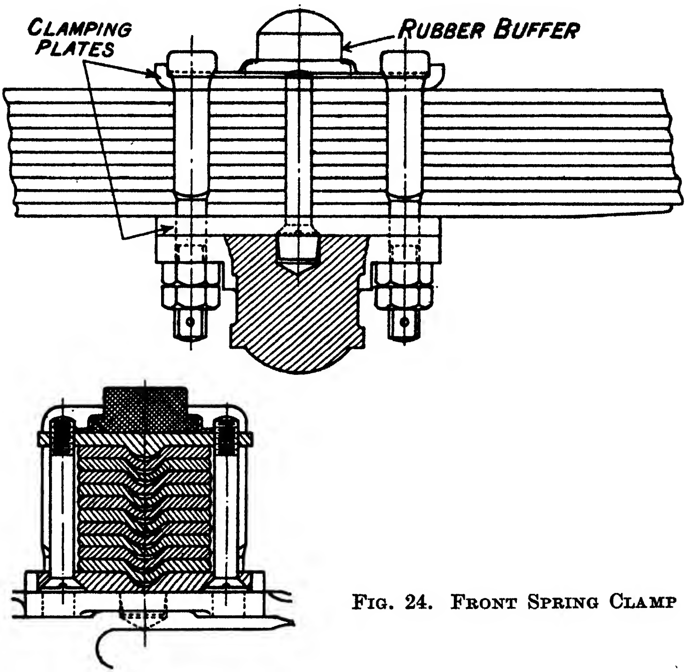


FIG. 24. FRONT SPRING CLAMP

increase of section modulii from loading point to supports. That is to say, as the spring is deflected, the section modulus at each end increases (if the spring is initially flat, Fig. 26), while at the middle it is constant. Further theoretical conditions of loading do not strictly apply since the load is not supported on a line but on an appreciable area, as represented by the axle pad upon which moreover it is firmly clamped. These are the essential departures from theoretical assumption.

Since for the same deflection of a spring of constant stress and a spring of constant section (Fig. 27) the latter is subjected to a much higher maximum stress,

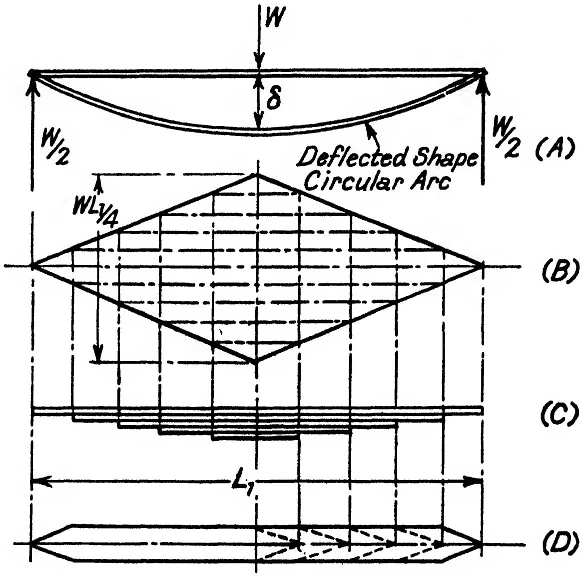


FIG. 25. IDEAL SPRING OF CONSTANT STRESS

it is obvious that economy of material is highest in the constant stress condition. This comparison is for two springs each having identical length, width, number and thickness of plates.

Although neglected, a discrepancy which should be understood is the considerable difference between actual deflection and ordinate deflection. Actual deflection derived from the following formula is the curvilinear path traced by the spring eyes, and ordinate deflection or camber as it is usually termed, is the

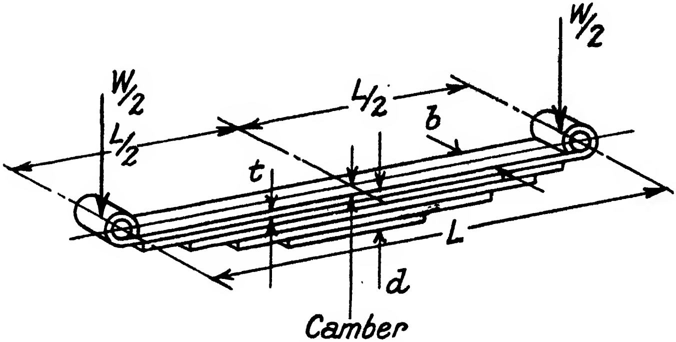


FIG. 26. SYMMETRICAL SPRING
NOMENCLATURE USED IN SPRING DESIGN

W -- static load
L = length between eyes (spring flat)
δ = deflection
t -- thickness of plates
d = total thickness of spring
b -- width of spring

perpendicular deflection measured from a centre line passing through both spring eyes, to the top surface of

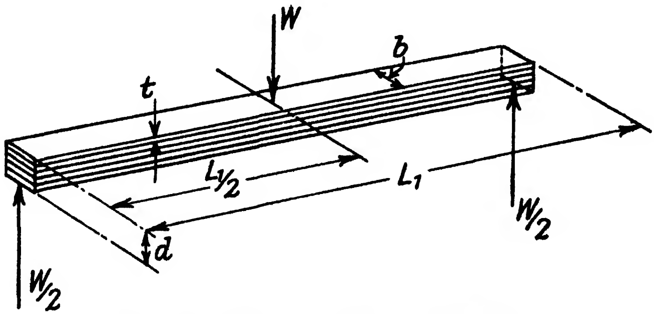


FIG. 27. SPRING OF CONSTANT SECTION

the eye plate. Another important point, more apparent but usually disregarded, is the difference in unladen spring span and laden spring span.

Seldom are springs so clamped to an axle that the centre portion is totally inert, therefore the length (L) includes the centre portion.

Deflection $d = WL^3/3Enbt^3$ . . . . (14)

where $E = 30 \times 10^6$

and $L =$ distance between eyes when spring is flat.

Stress. Several factors affect the stress sustained by a laminated spring apart from the question of load and variation from the rhomdoidal plan equivalent.

To assemble a spring it is easier and cheaper to abut

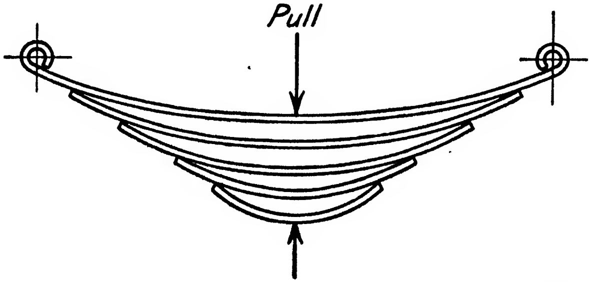


FIG. 28. NIPPED SPRING PLATES BEFORE BEING CLAMPED

ends only rather than attempt to fit the plates together for the whole of their lengths. Therefore nip is invariably introduced to obtain the former conditions. Machine fitted plates are of constant radius of initial curvature for a particular spring with the result that nip will be graduated from a maximum on the eye plate to a minimum on the short plate, this representing only slight pull, clearly not pre-loading the short plate to any appreciable extent. Fig. 28 shows, diagrammatically, nipped spring plates before being clamped, the amount of nip being purposely exaggerated in order that the effect of pull on plate stress may be more clearly indicated. Nip has no effect upon deflection.

The failure of an eye plate may have serious consequences, and in order to reduce bending stress the amount of nip is sometimes carefully worked out and

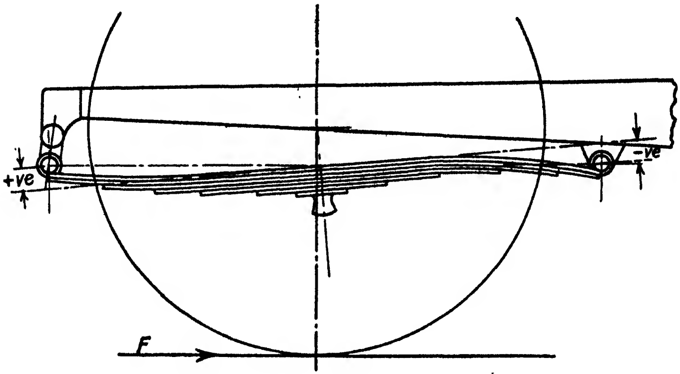


FIG. 29. DEFLECTION OF FRONT SPRING DURING BRAKING

arranged in such a manner that initial pull stress is negative in the eye plate and positive in the short plate, these stresses being algebraically added to the loaded mean stress. The otherwise resulting high stresses (due to static load) in the shorter plates are reduced by using thinner plates and more of them, but although plate thicknesses, lengths and nip can be so adjusted to give a uniform stress for a given load, plate stress for other loads will not be uniform throughout the spring.

Fig. 29 shows the effect of brake drum torque reaction on a front spring as applied to axle suspension, the deflection being such that there is negative camber on the rear half, and positive camber on the front half. In addition, the rear half of the spring is in compression due to the retarding force being transmitted through the springs.

$$\text{Stress} = 1{\cdot}5\ WL/nbt^2 \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where plates are of constant thickness.

$$\text{Stress} = 1{\cdot}5\ WL/b\Sigma t^2 \quad . \quad . \quad . \quad . \quad . \quad (16)$$

where plates are of different thickness.

Camber. The component force acting from the spring eyes in the direction of the plates when they are curved loads a spring without useful effect, therefore a spring can advantageously be so designed that it is flat under full static load, thus having slight positive camber when the vehicle is unladen and negative camber for shock loads. With increasing necessity for low loading, particularly for passenger commercial vehicles, the springs are commonly designed with a considerable negative loaded camber, providing the less obvious advantages of spring rate increase with a load increase due to shortening span, even more so than merely arranging the spring flat for static load. (Where any spring with rolled eyes is subjected to negative camber either for static or shock loads, adequate eye support is necessary because the component force acting parallel with the eye plate subjects the plate to tension, and so tends to force open the eyes.) Furthermore, ordinate and actual deflection are under these circumstances practically alike and so negligible error is involved when this is assumed in calculation. Free camber for a given rate is found by proportion, that is, free camber is equal to laden camber plus load/rate.

Plate Ends. It has been observed in considering the ideal spring of rhomboidal plan that the ends of each plate taper to a point from a full width at the adjacent plate point. Stress in the offset portion of each plate would also be constant if it was square cut, and its thickness between these limits varied in such a manner that the lower face presented a parabolic form. These two theoretical ideals are illustrated in Fig. 30 *a* and *b* respectively. In practice plates may be square cut with a drawn point, square speared with a bevelled point, or

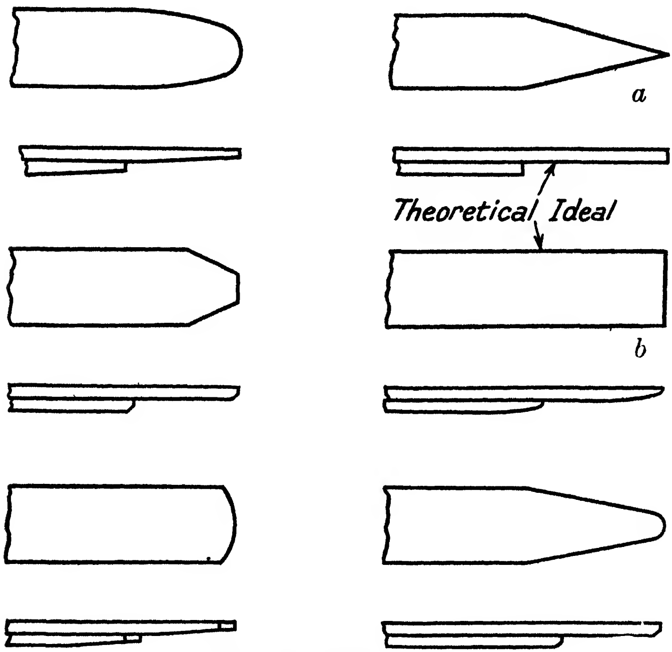


FIG. 30. PLATE ENDS

Gothic speared with a drawn point, or any combination. These do not represent all the shapes of plate ends in use, others being shown in Fig. 30. A speared end with no draw is standardized in America, an angle of 14° between the plate side edge and end being specified.

Spring Connections. Semi-elliptic springs are usually clamped to axles by means of U-bolts which are inverted for front springs. The distance between each is kept as small as possible in order to minimize restraint of the centre portion of the spring, and although provision is made for dowel registering of the

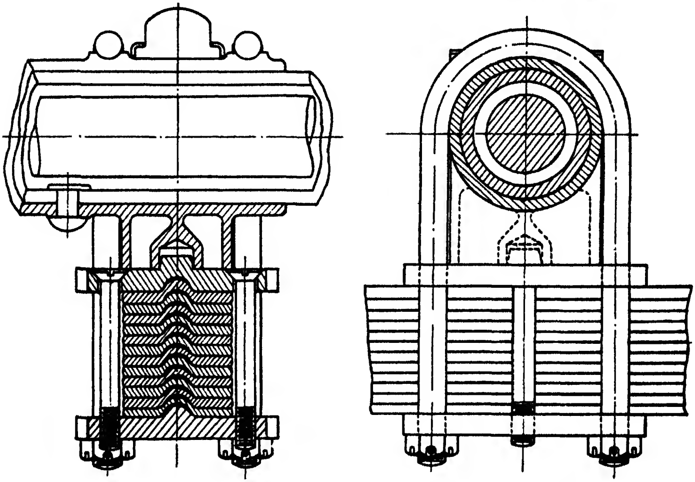


FIG. 31. REAR-AXLE TO SPRING FIXTURE

spring pad and spring, considerable duty is imposed on the U-bolts by spring deflection and brake drum torque reaction. These bolts are kept within reasonable proportions by the use of nickel chrome alloy steels having low carbon content in order to give a high degree of ductility. Figs. 24 and 31 respectively show representative types of front and rear spring axle fixings, the former including rectangular section bolts which are usually machined only on their shanks, to provide a fit in the axle pad holes. Rear axle spring bolts are more conveniently made from round bar and these are located in grooves on the axle brake bracket, their ends fitting the holes in both top and bottom spring clamp plates. Instead of the rear axle spring pad being rigid on the axle tube it can be made free to rock if used in connection with an enclosed propeller shaft drive, but when any torque reaction is transmitted to the springs such an arrangement is decidedly unsuitable.

Helical Springs in compression are mainly used in conjunction with independent suspension where each wheel is linked to the frame by swinging arms, the spring functioning solely as a resilient member, a condition seldom applying to a laminated spring. In most examples of independent suspension helical springs are subject to loads considerably in excess of downward hub loads, owing to leverage through wheel links. This is an important point in designing a spring for such an application, where, for a given load and deflection measured at the frame relative to the wheel, spring stiffness varies directly as the square of leverage.

The scope for design variation is limited to wire section shape and the possibilities of conical helix coiling in order to provide variable rate, but cylindrical helix coiling with round wire is sufficient to deal with ordinary requirements. Rubber seatings are often provided for the end abutments serving the purpose of insulation to prevent squeak.

Stress. In many instances of helical spring application axial displacement occurs during deflection, and precautions are usually taken to avoid such displacement when the spring is fully compressed. Attention to this detail is advisable if the stress otherwise reaches a very high maximum value for full compression along the normal coil axis, since further deflection perpendicular to the axis produces additional stress at a time when the spring is least able to withstand it.

Combined *bending and torsional* stress is given by the following formula—

$$\text{Stress} = \frac{16PR}{\pi d^3}\left(\frac{4x-1}{4x-4} + \frac{0{\cdot}615}{x}\right) \quad . \quad . \quad (17)$$

where R = mean radius of coil in inches.

P = load in lb.

d = wire diam. in inches.

$x = 2R/d$

For normal suspension spring proportions a value of 1·2 is fairly constant for the bracketed portion in the above equation, and the expression can therefore be reduced to—

$$\text{Stress} = \frac{6{\cdot}1PR}{d^3} \text{ lb./sq. in.} \quad . \quad . \quad . \quad . \quad (18)$$

(combined bending and torsion).

The maximum safe stress value depends upon material and surface finish. The bar from which springs are coiled is often ground in order to remove surface defects and to maintain a smaller tolerance on diameter, the object being to minimize the possibility of fatigue and to obtain uniformity of rate in a quantity of springs to one specification.

Deflection from the strain energy theory is given by

$$\delta = \frac{64PR^3N}{d^4 \cos\theta}\left(\frac{\cos^2\theta}{C} + \frac{2\sin^2\theta}{E}\right)$$

where N = number of active coils.

C = modulus of rigidity ($10^6 \times 12$).

E = Young's Modulus ($10^6 \times 30$).

θ = angle coil makes with planes perpendicular to axis.

$\text{Tan}\ \theta = p/2\pi R \qquad p$ = pitch of coils.

Pitch of the unloaded coils is usually of the order of πd, and since θ is generally small $\tan\theta$ is nearly equal to $\sin\theta$ and $\cos\theta$ nearly equal to unity, the unit deflection or rate for most practical purposes may be written—

$$\text{Rate} = \frac{187{,}500d^4}{R^3 N} \text{ lb./in. deflection} \qquad (19)$$

Torsion Bars. Anchorage methods of torsion bar springs are largely governed by the system of suspension, but in any case particular attention is paid to firmness of attachment to frame and avoidance of abrupt changes in section shape and size.

Torsion springs are invariably ground to fine limits and precautions are taken in most designs to ensure that no bending takes place.

Deflection is given by the formula

$$\alpha = \frac{583{\cdot}6\,T.L}{d^4\,C} \qquad (20)$$

where α = angle of twist in degrees.

T = applied torque = $A.P.$

L = length of bar.

d = diam. of bar.

C = modulus of rigidity.

A = length of arm.

P = load at end of arm.

$$\text{Stress} = \frac{5{\cdot}1\,T}{d^3} \qquad (21)$$

For large angles of twist—

$$\text{Torque } (T) = A\cos\alpha\,P$$

and ordinate deflection $= A\sin\alpha$ (at end of arm)

Spring Steels. There are two classes of steel which may be used in the manufacture of spring plates, carbon steel, and alloy spring steel, but the former is seldom used for heavy automobiles on account of its inferiority as compared with alloy steels. The alloy steels which are used for most springs made in this country are chrome-vanadium and silico-manganese steels, both giving greater uniformity in heat treatment and greater freedom from internal stresses in comparison with straight carbon steel of similar hardness. Particulars of the first two steels given in Table V are Dr. Hatfield's* recommendations for automobile spring steels extracted from his paper on "Steels for Automobiles and Aeroplanes." The analysis and physical properties of the carbon steel tabulated are included as a typical material applying to springs manufactured in this country. It would appear that chrome-vanadium is the best spring steel commercially available; its superiority over others is generally claimed with respect to its greater resistance to settling under repeated loads and its enhanced value in resisting fatigue and shock.

SUSPENSION

General. The suspension system of a vehicle may be defined as the combined relationship between elastic members and the various loads they support. Suspension is judged by the degree of comfort enjoyed by occupants when travelling over all conditions of road surface and during cornering.

The whole problem of suspension is far too complex to be dealt with briefly or simply because fundamentally we have to consider in one plane alone three interconnected dynamic systems involving altogether six

* Proc. I.A.E., vol. xxiv.

TABLE V

ANALYSIS AND PHYSICAL PROPERTIES OF TYPICAL SPRING STEELS

Steel	Physical Properties				Chemical Analyses						
	Yield	Ult. Tensile	Reduction of Area	Elongation	Brinell	C%	Mn%	Si%	Ni%	Cr%	Va%
	lb./sq. in.	lb./sq. in.									
Chrome-Vanadium	184,800	194,900	48%	16% in 2 in.		0·46	0·57	0·17	0·15	1·4	0·18
Silico-Manganese	172,200	197,600	36%	15% in 2 in.		0·52	1·05	1·95	—	0·05	—
* Chrome-Vanadium	196,000	216,100	38·6%	5·5% in 8 in.	456						
* Silico-Manganese	195,500	222,200	40%	6% in 8 in.	456						
Carbon (average)	145,600	123,200			3·5 dia.	0·55	0·6	0·2	—	—	—

* Steels manufactured by Samuel Fox & Co., Sheffield.

degrees of freedom at least. In practice there are so many variable factors, such as velocity, road surface, tyre pressure, damping and passenger weight to take into account that even if a complete mathematical treatment were possible, it would hold good only for the precise conditions assumed. The problem is further complicated by being closely linked with the dynamics of steering. However, knowledge of underlying principles does unquestionably direct experimental work and design along rational lines.

Of the resilent members (springs, tyres, and seats) road springs are the chief concern of the engineer and this section is therefore devoted mainly to suspension with respect to road springs.

Spring Flexibility. An obvious factor controlling riding comfort over uneven road surfaces is the flexibility of road springs. If the sprung mass is to simulate a "gliding" motion under such conditions energy imparted by vertical disturbing forces should be absorbed by the springs. Impulses arising from disturbing forces are opposed by inertia of the sprung mass so that if the time of duration is very small little or no vertical motion is transferred to the mass. Energy imparted is directly as the square of displacement and inversely as the time of application of force squared, factors governed entirely by vehicle speed and road surface. However, for a given set of such conditions it is clear that greater spring flexibility reduces the proportion of energy imparted to the sprung mass, and that conversely the absence of resilient members ensures that all energy is imparted to the body. Fig. 32 shows a weight supported by a spring and it is assumed that the weight is capable of vertical motion only, that is, the system possesses one degree of freedom. If deflection is proportional to load we have a case of simple harmonic motion.

The time of free oscillation is therefore

$$= 2\pi \sqrt{\frac{L}{gR}} \text{ where } L = \text{load}$$

and R = spring rate.

The value of L/R is equal to the static yield which is a measure of flexibility and entirely independent of bulk. By substitution of static deflection (d) the above equation may be written in its most useful form—

$$\text{Periodicity} = \sqrt{\frac{35{,}000}{d''}} \quad . \quad . \quad . \quad . \quad (22)$$

where d'' = static deflection in in.

Table III gives equivalent values of periodicity for static deflections in general use. For private cars such deflection frequently reaches 10 in. on rear springs and 7 in. on front springs, but since periodicity is inversely proportional to the square root of static yield small increments over 8 in. have little effect upon periodicity but are increasingly detrimental to constancy of tyre and road contact.

Inertia effects of a mass suspended as in Fig. 32 are readily demonstrated by moving a pendulum fulcrum to and fro. Above a certain frequency of fulcrum oscillation the bob remains in its original position with respect to the vertical zero line, and the arm oscillates in the sense that the pendulum is inverted. If the frequency of fulcrum disturbance is reduced to the natural frequency of the pendulum, the bob commences to swing with rapidly increasing amplitude. It is obvious that if the pendulum length is shortened, the disturbance frequency must be higher to maintain the inverted pendulum effect. Application of horizontal disturbing forces at the fulcrum is the dynamic equivalent of disturbances

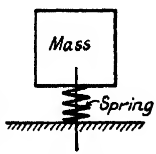


FIG. 32

TABLE VI (see page 216)

Type	Spring	Static Load	Static Deflection	Load Increment Full Bump	Ratio—Static Load / Load Increment
Car 10 h.p.	Front Rear	616 lb. 736 lb.	2·8 in. 7 in.	390 lb. 420 lb.	1·58 1·8
Car 18 h.p.	Front Rear	1,155 lb. 1,632 lb.	3·3 in. 8 in.	770 lb. 916 lb.	1·5 1·78
Coach 32 seater	Front Rear	2,800 lb. 4,900 lb.	2·875 in. 4 in.	2,450 lb.	2
Omnibus 52 seater	Front Rear	4,000 lb. 5,800 lb.	1·625 in. 3·8 in.	3,240 lb.	1·83

at the road wheel. Incidentally, the above analogy serves to show the "ironing out" value of high speed and also the advantage of low periodicity when a car is travelling over a "bumpy" road at low speeds.

Bounce and Pitch. In addition to actual periodicity of front and rear there are also two possibilities to take into account when an oscillating system, has two degrees of freedom (Fig. 33).

These factors are as follows—

(*a*) Magnitude of interference between one system of oscillation and the other.

(*b*) Resonant effect of periodic disturbing forces alternately applied at front and rear springs.

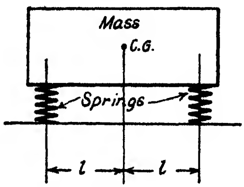


FIG. 33

For simplicity of dynamical similarity it is assumed that Fig. 33 represents a car in side elevation. If the centre of gravity is mid-way between the springs and they are of identical flexibility, similar disturbing forces, acting vertically and simultaneously upon each spring, produce a transitional vibration known as bounce. If the disturbances are a half-phase out or only one spring is disturbed, a rotary oscillation about the centre of gravity is produced, and is termed pitching. For any other phase relation a combination of pitching and bouncing is obtained. Interchange of energy between one system and the other occurs if the natural frequencies of the two systems are dissimilar. However, such interference does not take place when half the spring base is equal to the radius of gyration of the suspended mass (conditions of equal frequency).

* Expressed mathematically—

$$\text{Time of bounce (half swing)} = 2\pi\sqrt{\frac{m}{c}}$$

* See *Proc. I.A.E.*, Vol. XXX, p. 672 (F. W. Lanchester).

where m = mass suspended

and c = constant

Time of pitch (half swing) $= 2\pi\sqrt{\dfrac{mk^2}{cl^2}}$.

where k = radius of gyration.

and l = $\frac{1}{2}$ spring base.

By substitution when $l = k$.

Time of bounce = time of pitch.

When interference occurs between such two systems there are alternating intervals of pitch and bounce, the intervals undergoing cyclic changes. Interchange of energy can also occur even when $k = l$ if front and rear periods are unlike, since the axis of pitch does not pass through the centre of gravity. This condition introduces a secondary *periodic force* due to the oscillation of the mass (concentrated at the c.g.) about the pitching axis. Olley* has demonstrated that in practice the superimposing of the two oscillations produces a "whip lash" effect, presumably when the secondary periodic force is "heterodyning" with the pitch frequency. The greater the magnitude of the secondary periodic force the more pronounced the effect. (An increase of difference between front and rear periodicity will further remove the axis of pitch from the c.g. and therefore increase the magnitude of the secondary force.) This condition constitutes the reason for introducing what is colloquially referred to as a "flat ride," i.e. when front and rear periodicities are equal, but we have seen that this condition alone does not completely satisfy the above hypothesis.

Referring again to Fig. 33, if a disturbing force is

* See *Proc. I.A.E.*, Vol. XXX, p. 753.

applied at one spring to incite pitching and repeated at the other spring a half-phase out, a condition of resonance is produced. It is commonly met in practice when a car is travelling over an obstruction such as an abrupt change in road surface level. The front wheels first encounter the obstacle which forces the body upward at the front and compresses the rear springs. If the rear springs are thus compressed and just on the commencement of recoil when the rear wheels strike the same obstruction, impetus is added to the upward swing. (For obvious reasons the impulse is greater also, and the combined effect generally imparts a "surge" of energy to the springs beyond their capacity manifested by a "bump"). If the rear wheels are forced upward when the rear springs are just returning from maximum recoil the effect of impulse tends to cancel the first disturbance. If, in the former case, damping is insufficient or the magnitude of impulse abnormally large, reaction from the rear will cause nose dip if in phase with the downward swing of the front end.

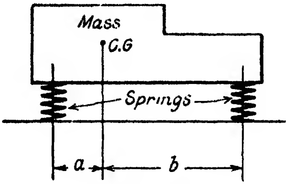


FIG. 34.

H. S. Rowell* has shown by analogy to the compound pendulum, that it is logical to bring a further general law to bear upon the problem, and one fairly simple to administer. If interaction is to be avoided, the radius of gyration (of the suspended mass) squared should be equal to the product of two horizontal distances from the centre of gravity to front and rear springs, i.e. $k^2 = a \cdot b$ (Fig. 34). Expressed differently the centre of one axle represents the axis of oscillation and the other axle the centre of percussion. Another

* See *Proc. I.A.E.*, Vol. 27, II, p. 480.

virtue in arranging for the centre of percussion to be located at one axle with respect to the axis at the other, is that passengers seated within the wheelbase cannot involuntarily leave their seats when the car is passing over a humped back bridge. This is due to a maximum possible acceleration equal to g at the centre of percussion and diminishing to zero at the swing axis. Beyond the centre of percussion acceleration can exceed g. The incidence of forward engines and radiators, spare wheels mounted well back, heavy bumpers at extremities, and so forth, together with a trend to shorten wheelbase all assists in improved suspension if an approach to the law $k^2 = a \,.\, b$ is any criterion, since with the average car of past years $k^2 < a \,.\, b$.

Rolling. Fig. 33 may also represent a simplified conception of the suspension system as viewed from the front of a car, except that the rolling axis is constrained usually well below the centre of gravity, which is therefore, subject to lateral displacement.

Excessive lateral C.G. movement caused by centrifugal force during cornering involves discomfort to passengers and excessive tyre wear. It is equally distressing for passengers to be tossed about sideways when the car is traversing rough roads.

With the usual laminated spring and axle suspension, rolling is resisted by the torsional stiffness of the springs in addition to their stiffness in vertical bending which is effective in resisting roll directly as the square of their distance apart. It is for the latter reason that spring track is made as wide as possible. With ordinary axle suspension the rolling axis is substantially parallel with the ground but with most forms of front independent suspension the rolling axis is inclined more or less from the ground at the front wheels to the height of springs at the rear wheels. The increased rolling moment is usually more than offset by the increased moment of resistance, since front spring track is equal to wheel track. With axle suspension steering lock limits spring track which is thereby of the order 50 per cent of the wheel track.

When very flexible springs are used side sway is reduced by introducing a resilient member which increases rolling resistance without affecting bouncing periods. A common device* of this nature consists of a torsion bar transversely mounted upon the frame and cranked at its extremities to which axle ends are linked. The angle of heel is proportionately reproduced in the bar as an angle of twist, energy thus stored is ultimately given out in righting the sprung mass.

Oscillation about a Vertical Axis. For various reasons there may be one degree of freedom with respect to the horizontal plane. If lateral location of one axle or the other is not rigid or nearly so, disturbing forces will cause an oscillation about a vertical axis removed from the centre of gravity. If the amplitude is appreciable and superimposing of rolling takes place, lateral motion would prove unpleasant when negotiating a rough road. Conditions associated with lateral flexibility are (1) the use of helical springs for rear axle suspension, and (2) long and narrow laminated rear springs with rubber bushes, inserts and mounting pads also in combination with axle suspension. In such cases, particularly the former, a transverse radius arm is attached at one end to the axle concerned and pivoted at its other end to the frame. This device thus eliminates any objectionable rotary oscillation by providing prescribed lateral location.

Damping. In the foregoing pendulum analogy and dynamic equivalents of motor-car suspension, we

* This device is variously termed, sway bar, anti-roll bar and roll stabilizer.

assumed that each system was free to oscillate at its own natural frequency after incitation and gradually die out until the initial energy imparted is dissipated via friction at the fulcrum and/or air friction acting upon the mass. Such a state would be intolerable in a motor-car having modern springing of high flexibility. Therefore, both forced and free vibrations must be properly damped, that is to say, the rate of energy dissipation must be as high as possible without detriment to the function of the resilient members, as it is important to avoid transferring energy to the suspended mass as far as possible. Perfect damping would only commence upon initial recoil. But vertical disturbing forces act upwards and downwards, therefore if damping is present in one direction only, disturbing forces in the opposite direction would transmit energy to the mass.

For this reason energy imparted by impulse must be dissipated in several damped swings; in effect a compromise between (*a*) complete dissipation on the first recoil, and (*b*) the maximum damping possible in both directions without substantially transferring energy to the mass.

The dissipation of energy by heat generation *via* friction is the principle underlying all types of spring damper. In the case of hydraulic dampers the laws of fluid friction apply, while friction dampers depend upon the laws of solid friction. The latter, in brief, are that friction is dependent upon pressure and is independent of velocity, and static friction is greater than sliding friction. Fluid friction is proportional to the square of velocity for moderate speeds but is independent of pressure. Viscosity also has some bearing upon fluid friction, but the desirable aim is maintaining a given viscosity over the working range of temperature.

Laminated springs possess damping properties by reason of interplate friction, but it is a variable quantity governed by surface condition and degree of lubrication.

Solid friction damping whether by interplate friction or external application is fundamentally at a disadvantage compared with hydraulic damping, owing to friction being highest while the system is at rest and therefore capable of transmitting some energy to the frame immediately a disturbing force is applied.

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