

SECTION 5, CHAPTER 6

COMPRESSION IGNITION ENGINES

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Introduction

1. Compression ignition engines are alternatively known as "diesel" and "heavy oil" engines, but "compression ignition" is now generally preferred, because the essential characteristic of these engines is clearly indicated in contrast to spark ignition engines. The description "diesel" came into use because Dr. Diesel, a German engineer, was responsible for much of the early in-

vention and development work on this type of engine; the nature of the fuel used gave rise to the use of the description of "heavy oil".

2. In many respects compression ignition engines are very similar to spark ignition engines, for the same basic components are used in the mechanical layout, e.g. cylinders, crankshaft, connecting rods, etc.,

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but some parts are made more robust, and therefore heavier, to withstand the greater stresses involved in compression ignition engines. Consequently, this chapter does not repeat the common information given in the preceding chapters, but is confined to the main differences in construction and operation of these two types of engines. The main difference between the two types is that the carburation and ignition equipment on spark ignition engines is replaced by the fuel injection pump and injectors on compression ignition engines.

Cycles of Operation

3. Four Stroke. The four strokes are, of course, completed in two revolutions of the crankshaft and, although very similar to the spark ignition engine there are important differences. On the induction stroke, air only flows into the cylinder, and it is then compressed on the compression stroke to a very high pressure. The compression ratio of a compression ignition engine is usually between 12 and 20 to 1, as compared with the 6 to 10 to 1 of the normal spark ignition engine. Thus, the high temperature to which the air is raised is sufficient, when injection commences just before top dead centre, to ignite the mixture of oil and air without the aid of an electric spark; the oil is sprayed into the combustion chamber through an injector. The burning of the gasoline/air mixture in a spark ignition engine is very quick indeed, but in a compression ignition engine the burning is spread over the period of injection of the oil. This prolonged burning imparts a more sustained pressure on the piston during the power stroke, as compared with the more rapid rise of pressure obtained in the spark ignition cylinder. The exhaust stroke is quite normal, the exhaust valve being opened some little time before bottom dead centre and closed just after top dead centre. The cycle of operations is illustrated at Fig. 1, the valve timing being that of a typical compression ignition engine.

4. Two Stroke. The cycle of operations for a two stroke compression ignition engine is illustrated at Fig. 2, and again, it will be noted, it is very similar to that of the spark ignition engine. On the exhaust stroke, the descending piston first uncovers the exhaust port and then the inlet port, the incoming charge being utilized to expel the

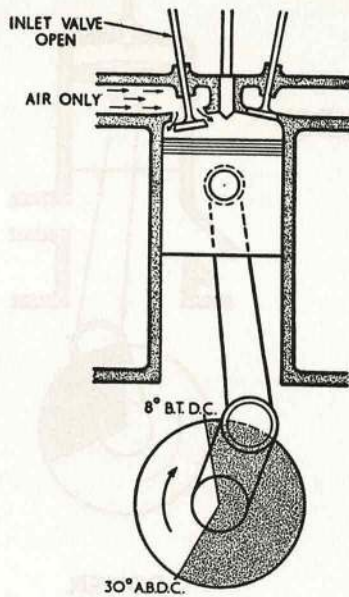
remainder of the exhaust gases. At this stage it will be realised that the compression ignition engine has a distinct advantage over an equivalent spark ignition engine, for whereas the incoming charge in the latter engine is composed of gasoline and air, and therefore some gasoline will be lost through the exhaust port, the induction charge in the compression ignition engine consists of air alone and thus it is of no consequence if some of the air is lost in the same way. Further, efficient scavenging of the exhaust gases is achieved on some engines by supplying the air under pressure from a blower or scavenging pump. On the compression stroke the rising piston closes the inlet and exhaust ports and compresses the air to a high degree, and just before top dead centre the injection of oil commences. Again, the temperature of the air is sufficient to ignite the mixture of oil and air, the burning being spread over the injection period. The piston is forced down on the power stroke until it uncovers the exhaust port, when the sequence commences all over again.

Comparison

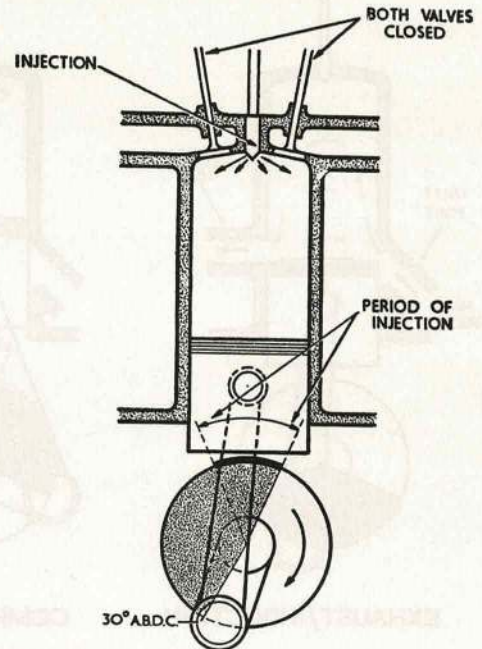
5. Although the mechanical layout of a compression ignition engine is very similar to that of a spark ignition engine, because of the very much higher cylinder pressures developed in the former, it has of necessity to be made more robust and consequently heavier. The outstanding difference between the two types is that in a compression ignition engine, air alone is compressed to a much higher pressure per square inch than is the gasoline/air mixture in a spark ignition engine. This compression causes the temperature to rise sufficiently high to ignite the injected oil, and thus an electric ignition system is not required. Further, a carburettor is also not required, for the correct amount of oil is forced through the injectors and into the cylinders by means of a fuel injection pump. This pump is the vital unit of the engine, for it not only performs the "timing" function of electrical ignition by varying the point at which injection commences, but it also controls the amount of oil sprayed at each injection.

6. There is also a noticeable difference in the running characteristics of the two types of engines. Because burning is spread over the period of injection of the oil in a

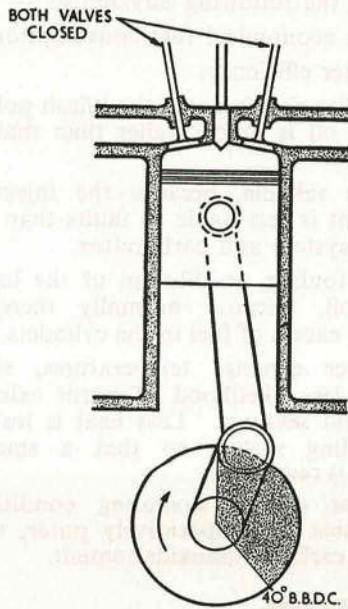
COMPRESSION IGNITION ENGINES



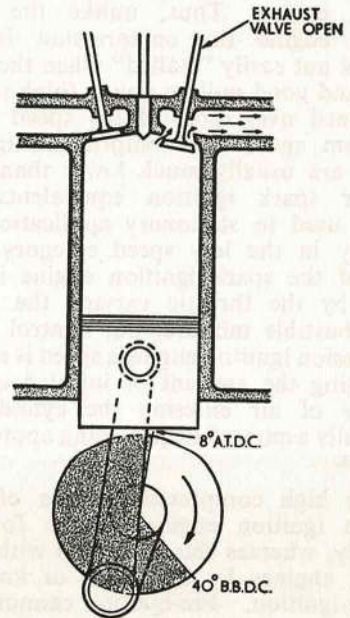
INDUCTION



COMPRESSION



POWER



EXHAUST

Fig. 1. FOUR STROKE CYCLE (C.I.)

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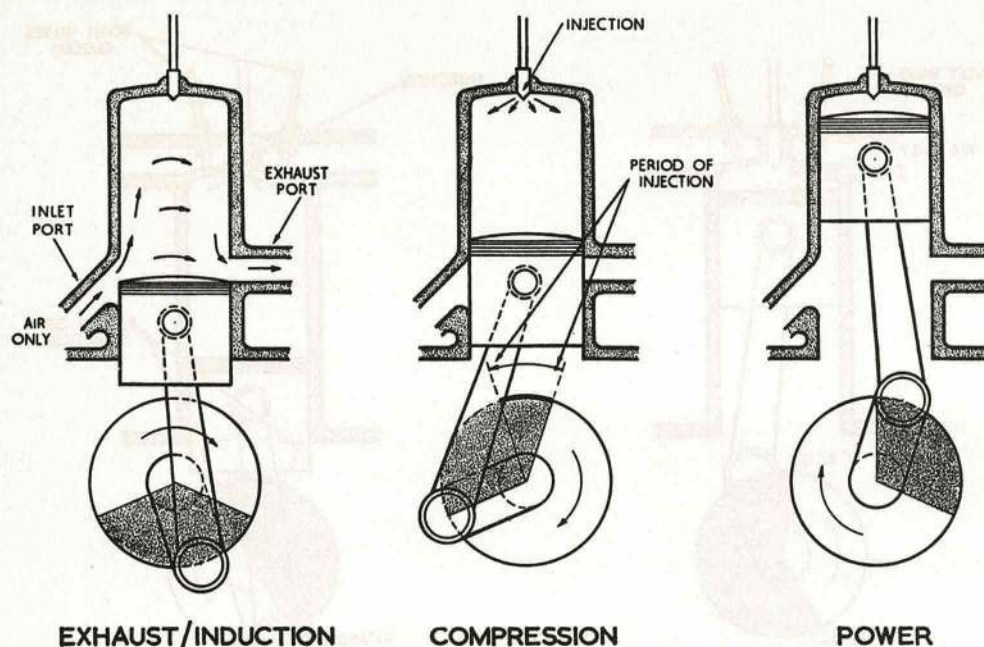


Fig. 2. TWO STROKE CYCLE (C.I.)

compression ignition engine, it is more sustained and enables power to be obtained at lower speeds than is possible with a spark ignition engine. Thus, unlike the spark ignition engine the compression ignition engine is not easily "stalled" when the speed drops, and good pulling power (high torque) is obtained over most of the speed range. Maximum speeds of compression ignition engines are usually much lower than those of their spark ignition equivalents, and engines used in stationary applications are normally in the low speed category. The speed of the spark ignition engine is controlled by the throttle varying the supply of combustible mixture, but control of the compression ignition engine's speed is effected by varying the amount of injected oil, the quantity of air entering the cylinders of a normally aspirated engine being appreciably constant.

7. The high compression ratios of compression ignition engines makes for high efficiency, whereas this is limited with spark ignition engines by the onset of knocking and pre-ignition. Pre-ignition cannot occur on the compression ignition engine because air alone is compressed, and, within limits, compression ratios are governed by the strength of the component parts.

8. **Advantages.** When compared with the spark ignition engine the compression ignition engine has the following advantages:—

- (a) More economical fuel consumption.
- (b) Greater efficiency.
- (c) Less fire risk, because the "flash point" of diesel oil is much higher than that of gasoline.
- (d) More reliable, because the injection equipment is less liable to faults than the ignition system and carburettor.
- (e) Less fouling or dilution of the lubricating oil, because normally there is never an excess of fuel in the cylinders.
- (f) Lower exhaust temperatures, thus, there is less likelihood of burnt exhaust valves and seatings. Less heat is lost to the cooling system so that a smaller radiator is required.
- (g) Under correct operating conditions the exhaust is comparatively purer, with very low carbon-monoxide content.

9. **Disadvantages.**

- (a) The initial cost of the engine is higher, largely due to the precision required in manufacture of the injection equipment.
- (b) The power/weight ratio is lower.

(c) More efficient filtering of fuel is required.

(d) Less power per unit of cylinder volume; this is because all available oxygen is not utilized during the short period of fuel injection.

(e) Requires much greater starting torque because of the high cylinder pressures; higher maximum cylinder pressures result in higher torque fluctuations.

(f) Creeping and saturation factors of the oil are injurious to materials such as are used in electrical installation.

Note.—Personnel should at all times take precautions to prevent diesel oil contacting their skin.

Injection and Combustion

10. The most noticeable characteristic of a compression ignition engine is the feature which is commonly called "diesel knock", which gives rough and noisy running. Much of the research carried out on this type of engine has been aimed at suppression of this feature, by finding the best methods of injection and the best shape of combustion chamber so as to attain smooth and complete combustion of the fuel.

11. The problem involved is that of injecting into the cylinder a very small volume of liquid fuel, say about as big as a pin head, in such a way that every tiny particle of oil immediately contacts its required amount of hot air, so that combustion will be as quick and complete as possible but without being so sudden as to cause rough running. Of the two general methods, either the velocity of the oil entering the combustion chamber is relied upon to get the oil to the air that it requires, or, the air is given swirl or turbulence so that it contacts the oil as it is injected. The former method is known as direct or open chamber injection, in which any movement of the air has not been specially designed except in certain applications where it may be caused by masking of the inlet valve. The second method is often referred to as indirect injection, and this covers many designs of swirl or pre-combustion chambers; more information on these two general types is given in later paragraphs.

Combustion Phases

12. Research has revealed that there are three phases in the combustion period,

although they may not, in general, be sharply distinguishable one from the other. The first phase starts at the commencement of injection and lasts for approximately 10° of crank angle; this is known as the "delay" period, during which burning begins but does not affect the normal rise in pressure caused by the rising piston. The second period is of slightly less duration than the first, and during this phase there is rapid burning and a quick rise in pressure; the sharpness of this pressure rise is a determining factor in causing rough running (diesel knock). The longer the delay period, the quicker will be the pressure rise during the second phase, and, in general, the shape of the combustion chamber has an important influence on this, although the nature of the fuel, the temperature and pressure of compression, and the initial rate of fuel injection are all factors in determining the length of the delay period.

13. Research has also shown that the delay period tends to be constant in time, irrespective of crankshaft speed, thus, the faster the speed, the greater is the crank angle over which the delay period extends, and thus there is an obvious requirement for injection advance with increased crankshaft speed. Roughest running is usually experienced on the majority of engines whilst slow running and on light loadings, and this is because of the lower compression temperature and smaller amounts of fuel injected under these conditions which increase the delay period. The second phase of rapid burning is followed by burning of the oil as it issues from the injector nozzles, and this is the only phase over which direct control can be exercised by controlling the rate of injection.

Combustion Chambers

14. There are various forms of combustion chamber used by the different engine manufacturers, but they can be classed into two main types.

15. **Pre-Combustion Type Chamber.** Applications using this type of combustion chamber are often referred to as swirl or ante-chamber types, or, as using indirect injection. The combustion chamber is formed in the cylinder head, and communicates with the cylinder through a narrow throat or passage as

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shown in Fig. 3, or through several holes. Only the minimum clearance is left between the cylinder head and piston at top dead centre; on some engines, each cylinder has a second combustion chamber which communicates with the first chamber (para. 17). The fuel is injected into the air in the pre-combustion chamber and partly burned, the spread of ignition being aided by the turbulence arising from the passage of the air through the communicating passage or holes during compression. As the piston starts to descend, the very high pressure in the chamber forces the products of the partial combustion and the remaining fuel through the communicating passage. The high turbulence thus created aids the final combustion in the cylinder, and the piston is protected from the high initial pressure arising from too rapid combustion of the whole of the fuel injected. With this method there are certain cooling losses which necessitates the use of heater plugs, or the use of a device to raise the compression pressure during starting to a higher degree than that required during normal running, if satisfactory starting is to be obtained. Engines using this type of combustion chamber are normally free from rough running and have a clean exhaust, but they

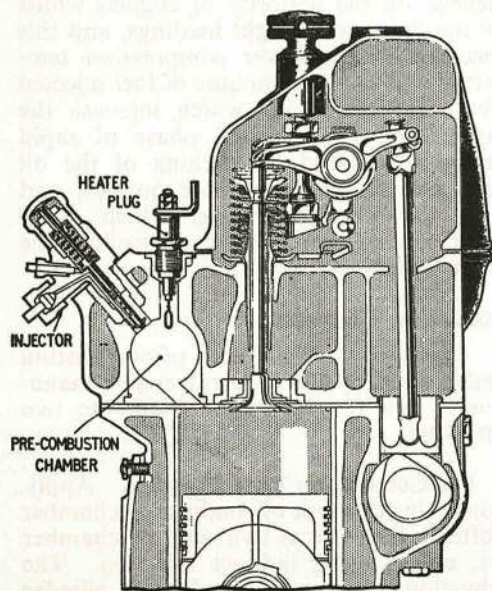


Fig. 3. PRE-COMBUSTION TYPE CHAMBER
Indirect Injection

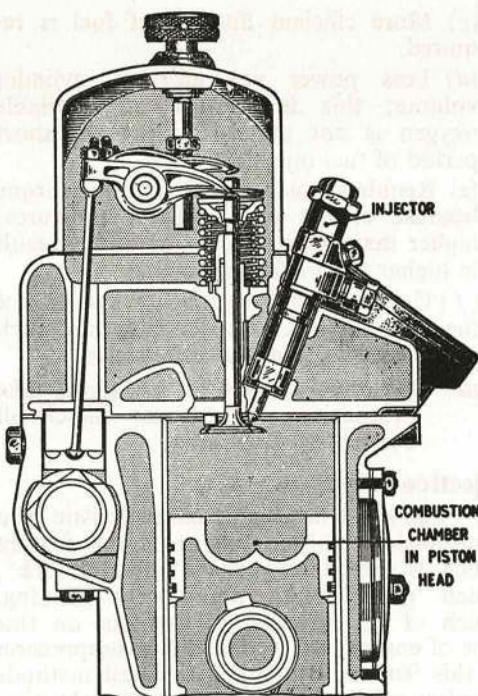


Fig. 4. OPEN TYPE COMBUSTION CHAMBER
Direct Injection

are not so economical nor do they give as much power as the direct injection types.

16. Open Type Combustion Chamber. Applications of this type of combustion chamber are formed largely as a hemispherical or toroidal recess in the crown of the piston, in a manner similar to that shown at Fig. 4. Thus, the charge of air is compressed into this recess, swirl or turbulence being aided by the shape of the chamber or, on some engines, by masking the inlet valve (Fig. 5). This type is also referred to as "direct injection", because the injector is fitted in the cylinder head directly above the combustion chamber, and the oil is sprayed at a high velocity, requiring a rather high injection pressure. Engines using this type of combustion chamber can have a moderate compression ratio, and no auxiliary starting devices such as heater plugs are required; they are also economical in operation, but have a somewhat greater tendency to rough running than the indirect injection type engines.

17. Compression Ratio Change-over Valve. It has been stated previously (para. 15)

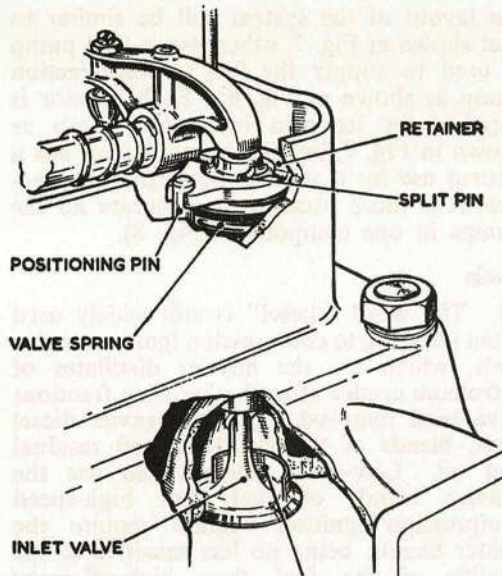


Fig. 5. MASKED INLET VALVE

that indirect injection type engines require some aid to starting; this is achieved on some engines by using a device which raises the compression ratio during starting to a higher degree than that used during normal running. This device is known as the compression ratio change-over valve, and a typical fitment is shown at Fig. 6. It will be seen that the cylinder head incorporates double combustion chambers for each cylinder, each secondary chamber being connected to its main chamber by a throat or passage, which can be opened or closed by operation of the hand operated change-over valve. Under ordinary running conditions the valve is held in the retracted position by a spring, so that the connecting throat between the two combustion chambers is open and both chambers are used. To provide the higher compression required when starting, the valve lever is lifted until the end of the plunger seals the passage between the two chambers, thus confining combustion to the main chamber.

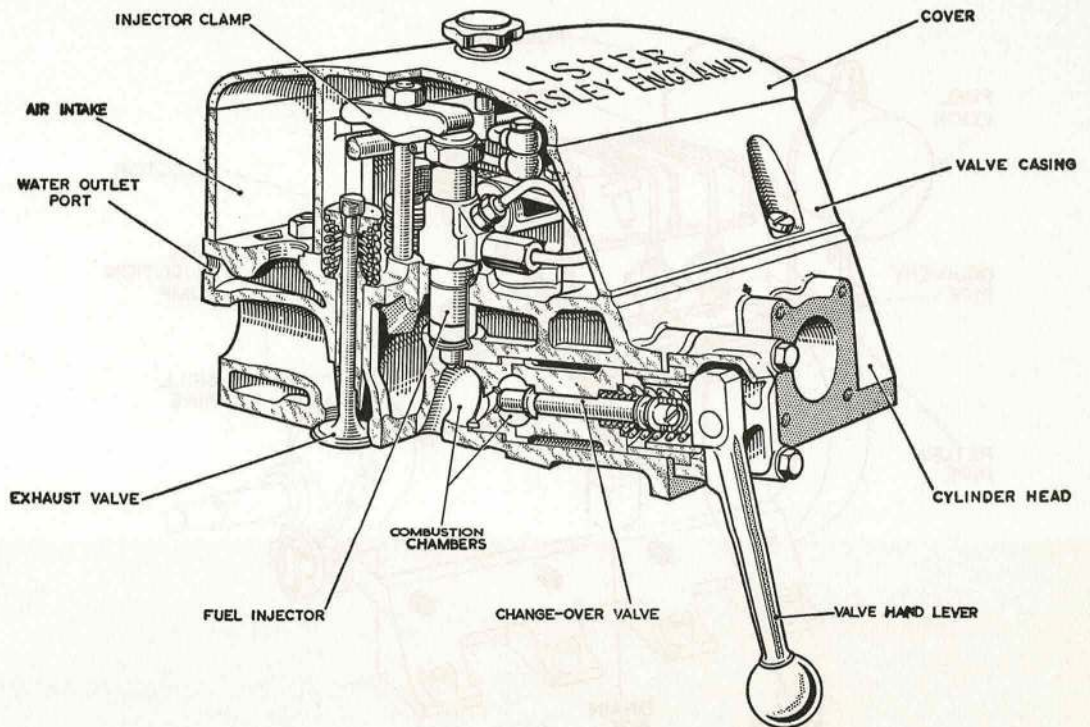


Fig. 6. CHANGE-OVER VALVE

FUEL SYSTEMS

Introduction

18. In general, the fuel systems for compression ignition engines are broadly similar to those used for spark ignition engines (Chap. 2), the requirement being to supply liquid fuel to the carburettor on the latter types and to the fuel injection pump on the former types. The differences between the two systems are:—

- (a) Different fuels are used.
 - (b) Whereas the carburettor supplies directly into the engine, the fuel injection pump supplies individual cylinders by means of pipes and injectors.
 - (c) A much finer degree of filtration of the fuel is required before it reaches the injection pump on compression ignition engines.
19. The system employed will depend mainly on the positioning of the fuel tank. When the tank is mounted high enough to be able to feed the injection pump by gravity,

the layout of the system will be similar to that shown at Fig. 7, otherwise, a feed pump is used to supply the fuel to the injection pump as shown at Fig. 8. Each injector is supplied by its own individual pump as shown in Fig. 7, but this actual layout has a limited use for it will be found that in practice, it is more usual to incorporate all the pumps in one component (Fig. 8).

Fuels

20. The word “diesel” is still widely used when referring to compression ignition engine fuels, which are the heavier distillates of petroleum crudes after the kerosine fractions have been removed, or, with heavier diesel fuels, blends of the distillates and residual fuel oil. Low-speed engines can use the heavier blends of fuels but high-speed compression ignition engines require the lighter blends, being no less sensitive to the quality of the fuel than high-efficiency gasoline engines. The fuel used in compression ignition engines is unlike gasoline in many ways:—

- (a) It does not vaporize at low temperatures.

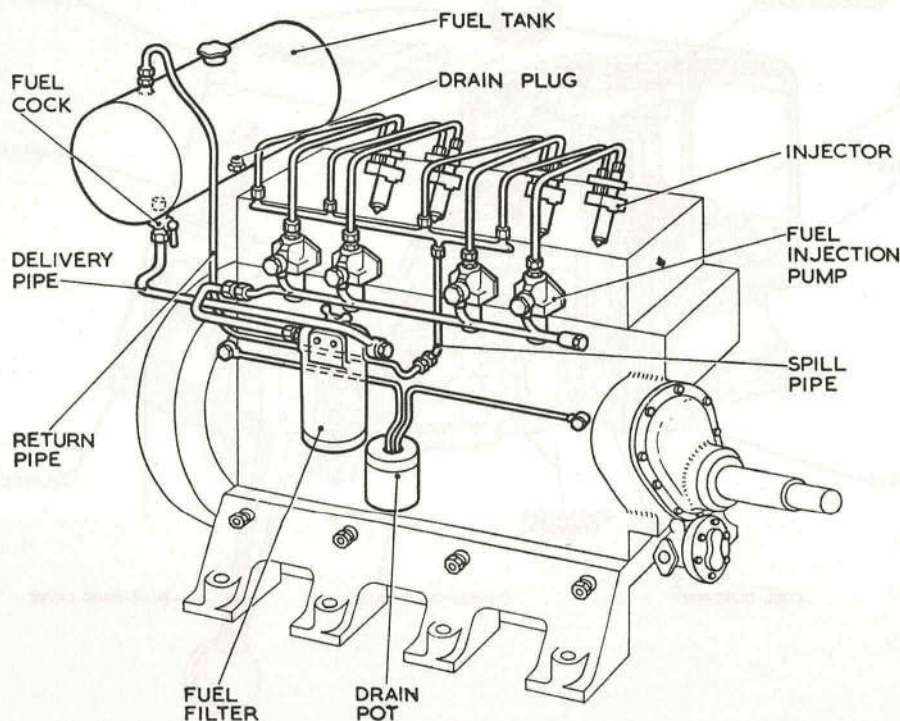


Fig. 7. GRAVITY FEED SYSTEM

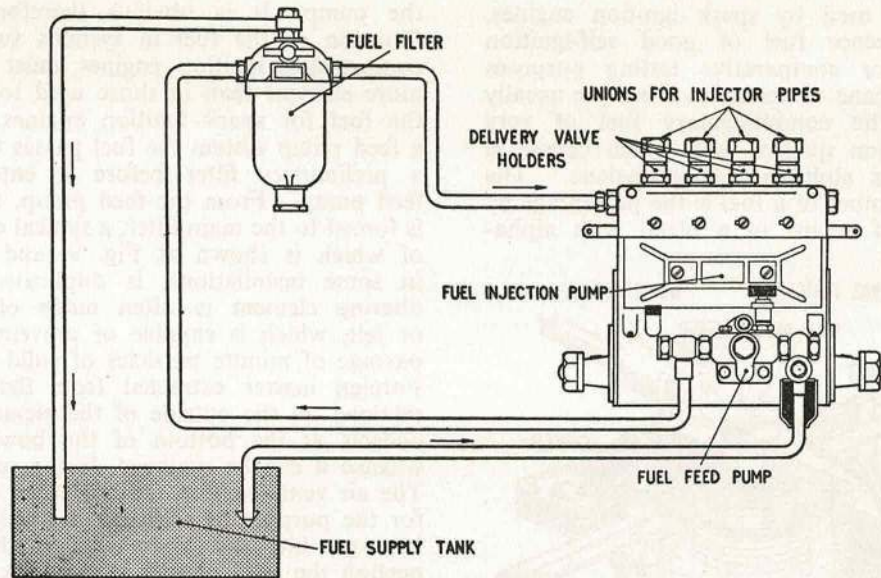


Fig. 8. PUMP FEED SYSTEM

(b) Its flash point is much higher than that of gasoline, thus reducing the fire risk.

(c) It has lubricating qualities which are utilized in the fuel pump mechanism.

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(d) Its specific gravity is ~~less~~ ^{higher} than that of gasoline, i.e. a greater weight of fuel for the same volume.

(e) It has greater penetrative power, so that joints, etc., must be a good oil-tight fit.

21. Ignition Quality. The most important quality of a fuel for a high speed compression ignition engine is its ignition quality, or its willingness for self ignition with the least delay after the beginning of injection. Almost all modern compression ignition engines are of the "solid injection" type, in which fuel only is sprayed into the combustion chamber, without the help of the compressed air which was used in many of the older engines. Injection commences before the end of the compression stroke, but the fuel does not ignite immediately it is injected; the reason for the delay or "ignition lag" is that although the air in the combustion chamber is hot, the fuel spray is relatively cold. Fuels giving short delay times are considered as having high

ignition quality, and fuel differences in this respect have considerable bearing on engine behaviour. Long "ignition lag" leads to combustion knock, because a comparatively large amount of fuel accumulates in the combustion chamber and, once ignition starts, all of this fuel burns very rapidly, causing a rapid rise of temperature and pressure which results in combustion knock. The delay time increases with reduction in temperature, therefore, if the temperature is very low, the delay time may be such that ignition cannot commence in the time available, thus, fuels with high ignition quality make starting easier. Engine operating conditions, such as compression ratio, degree of air turbulence and intake air temperature, coolant temperature, injection timing and engine speed, also have considerable bearing on the delay time, but fuel properties are equally important. The higher the speed of the engine, the higher is the ignition quality of the fuel required.

22. Cetane Number. The Service designation of a diesel fuel appears as a number and the words "Dieso", e.g. "47 DIESO", which indicates that the fuel has a cetane number of 47. The ignition quality of a fuel is measured in terms of a cetane number, which is similar to the octane number

for fuels used by spark ignition engines. The reference fuel of good self-ignition quality for comparative testing purposes is hexadecane or cetane as it is more usually called. The complimentary fuel of very low ignition quality with which cetane is blended is alpha-methylnaphthalene. The cetane number of a fuel is the percentage by volume of cetane in a blend with alpha-

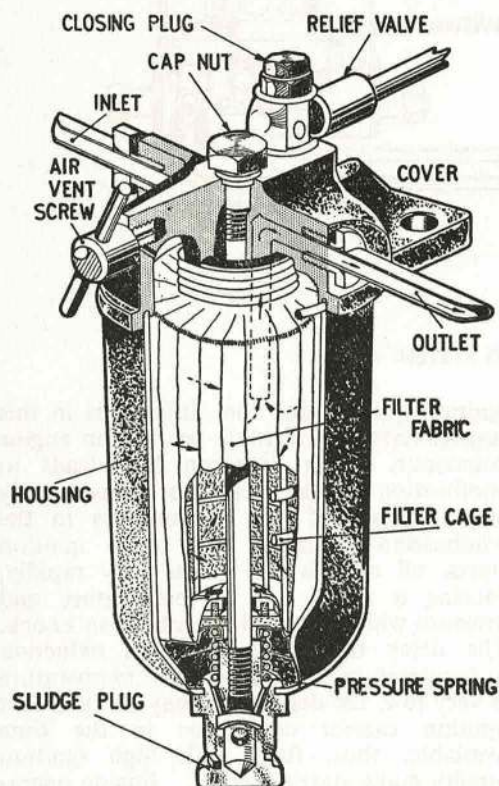


Fig. 9. FUEL FILTER

methylnaphthalene, which has the same ignition quality as the sample fuel when tested in the same engine under the same conditions; this number ranges from 30 to 70.

23. Filtration. Because the injection equipment of compression ignition engines is manufactured to very fine limits by precision methods, it is vitally important to keep out any foreign matter. A piece of grit, which, in a carburettor, might only block a jet, could, in a fuel injection pump, have abrasive effects that would seriously damage

the pump. It is obvious therefore, that filtration of the fuel in systems supplying compression ignition engines must be far more efficient than in those used to supply the fuel for spark ignition engines. With a feed pump system the fuel passes through a preliminary filter before it enters the feed pump. From the feed pump, the fuel is forced to the main filter, a typical example of which is shown at Fig. 9, and which, in some installations, is duplicated. The filtering element is often made of fabric or felt, which is capable of preventing the passage of minute particles of solid matter. Foreign matter extracted from the fuel is retained on the outside of the element and collects at the bottom of the bowl, from whence it can be removed during servicing. The air vent screw in the top cover is fitted for the purpose of releasing air which may have got into the system, and in this connection the level of fuel in the tank should never be allowed to get low enough to allow the ingress of air into the system. When fuel is pressure fed to the injection pump, the fabric type filter is fitted with a relief valve to prevent the build up of excessive pressure which would cause damage; the relief valve is connected back to the fuel tank or the fuel feed pump.

Fuel Feed Pumps

24. General. The fuel feed pump may be of the diaphragm type similar to that described in Chapter 2 of this Section, or a C.A.V. plunger-type pump as described in the following paragraphs. There are two plunger-type pumps which are basically identical, differing only in the type of inlet connection. Pumps with an identification ending in 'N62' have an inlet connection providing both a banjo and a nut and nipple connection, whilst those ending in 'N2' have a nut and nipple connection with a thimble-type filter. The additional connection on the N62 pump inlet provides for the draining back to the pump of excess fuel from the relief valve of the main fuel filter. The feed pump is flanged-fitted to a prepared face on the injection pump casing (Fig. 10). It is operated by a cam on the injection pump camshaft which imparts movement to the feed pump plunger, causing fuel to flow from the supply tank and to be delivered to the injection pump. A priming device is fitted to these pumps for use when it is required to prime the

pipng and expel any air from the system. This device is not a part of the feed pump, being supplied as a separate Stores item.

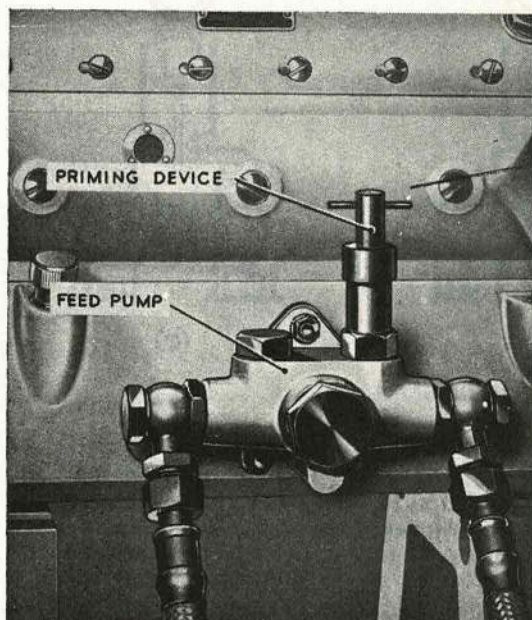


Fig. 10. C.A.V. FUEL FEED PUMP

25. Construction. Fig. 11 shows sectional views of the pump with the priming device fitted above the inlet valve. The pump body is a one-piece casting with machined chambers to accommodate the fuel inlet (11) and outlet (1) connections, inlet (9) and outlet (3) valves, and the pump plunger (15). Passages, interrupted by the spring-loaded inlet and outlet valves, connect the inlet and outlet chambers to the plunger chamber. Normally, the two valve chambers are sealed by plugs which also serve as guides for the valve stems, but, when fitted, the priming device replaces the plug over the inlet valve. The plunger chamber is sealed at one end by a plug (18) which also houses one end of the plunger spring (16), whilst the casing at the opposite end of the plunger chamber is drilled concentrically with the chamber to house a spindle (14), which transmits the movement of the cam roller (12) to the plunger. A circumferential groove in the spindle housing is connected by a passage to the inlet chamber; this ensures that the fuel which would other-

wise leak past the spindle into the injection pump casing, is drawn back into the feed pump on the inlet stroke.

26. Operation. As the cam moves from the position of maximum lift to that of minimum lift (Fig. 12A), the plunger spring, which was compressed on the previous

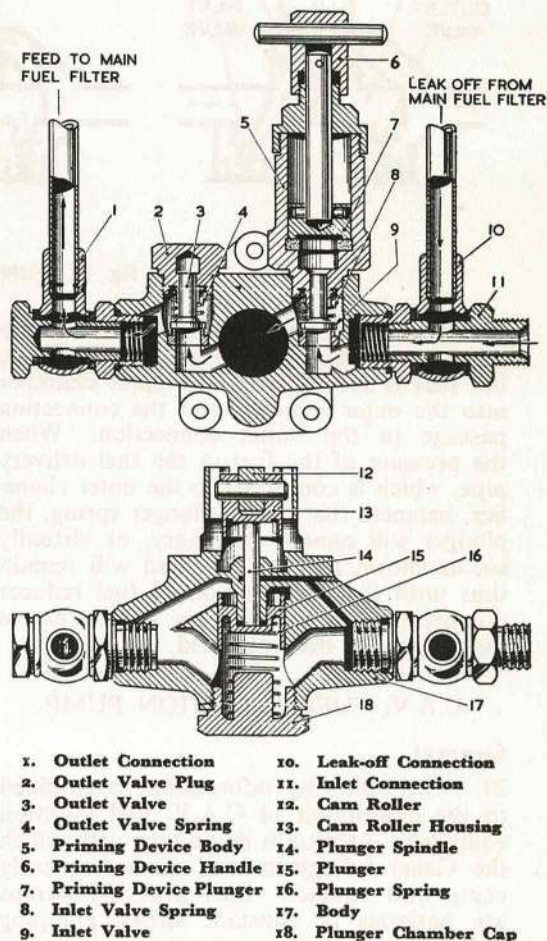


Fig. 11. FUEL FEED PUMP—SECTIONAL

stroke, reasserts itself and forces the plunger down the chamber. This movement forces fuel out of the outer chamber, and along the connecting passage to the fuel outlet; at the same time the inlet valve lifts off its seat, and fuel flows into the inner chamber. As the cam moves from minimum to maximum lift (Fig. 12B), the plunger movement causes the outlet valve to open by fluid

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MINIMUM LIFT POSITION

MAXIMUM LIFT POSITION

SUFFICIENT FUEL LIFTED

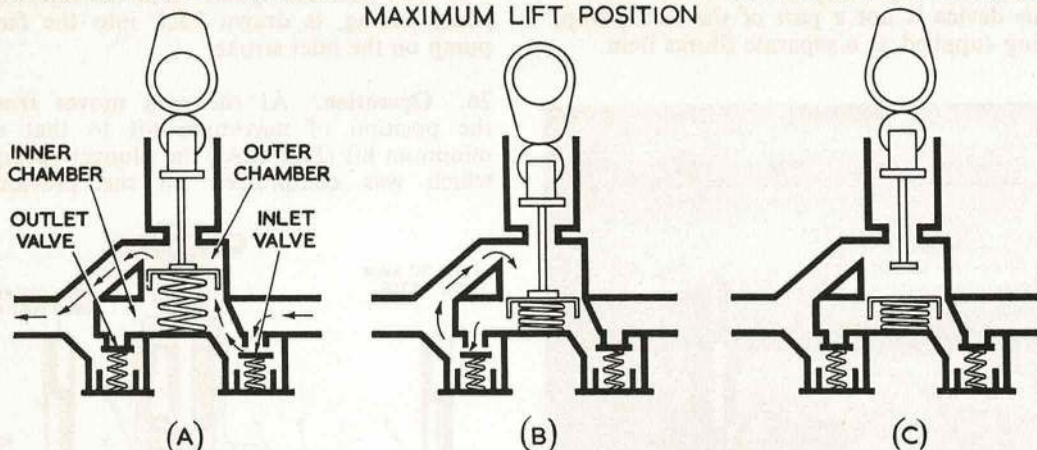


Fig. 12. FUNCTIONAL DIAGRAM

pressure, which also augments the spring-pressure to close the inlet valve. Thus, the fuel is forced from the inner chamber into the outer chamber and the connecting passage to the outlet connection. When the pressure of the fuel in the fuel delivery pipe, which is connected to the outer chamber, balances that of the plunger spring, the plunger will remain stationary, or virtually so, as shown in Fig. 12C, and will remain thus until the consumption of fuel reduces the pressure in the delivery outlet; excess fuel supply is thus obviated.

C.A.V. FUEL INJECTION PUMP

General

27. The following information is confined to the description of C.A.V. fuel injection equipment, because it is the type with which the General Engineer will most frequently come into contact. Fuel injection pumps are batteries of constant stroke pumping units, one unit for each engine cylinder, housed in one casting, although there are applications using a completely separate pump for each cylinder of an engine. Each pumping unit consists of a spring-loaded plunger working in the bore of a barrel and operated by a cam. The majority of fuel injection pumps have camshafts for operation of the plungers incorporated in the pump casing, but others which do not incorporate a camshaft are operated by the engine designer's own devices, e.g. by cams on the engine camshaft.

28. Each pumping element is calibrated to deliver the correct quantity of fuel, and is phased to ensure injection of the fuel into the cylinder at the correct time. The amount of fuel pumped by the plunger pumps is controlled by a rack or control rod. A governor may be fitted to the unit, and this may be of the centrifugal or vacuum type; it operates on the control rod, and may be fitted to either end of the casing. Fuel injection pumps are driven at half engine speed, and are timed relative to the engine crankshaft position in a similar way to magneto or coil ignition. Fig. 13 shows a fuel injection pump for an engine having six cylinders, and it is fitted with a centrifugal governor unit and a fuel feed pump; Fig. 14 illustrates a fuel injection pump for an engine having four cylinders, but without both a governor unit and a fuel feed pump.

Pump Casing

29. The pump casing (Fig. 15) is an aluminium alloy casting, formed with three longitudinal chambers. The base chamber houses the camshaft, which is supported in bearing housings secured to each end of the casing. Above this is a larger chamber which houses the control mechanism of the plunger pumps; it is fitted with an inspection plate over a full length opening on the front of the casing, thus providing access to the units for assembly and adjustment. A gallery which runs the full length of the upper part of the casing is the fuel suction chamber. This gallery terminates at each

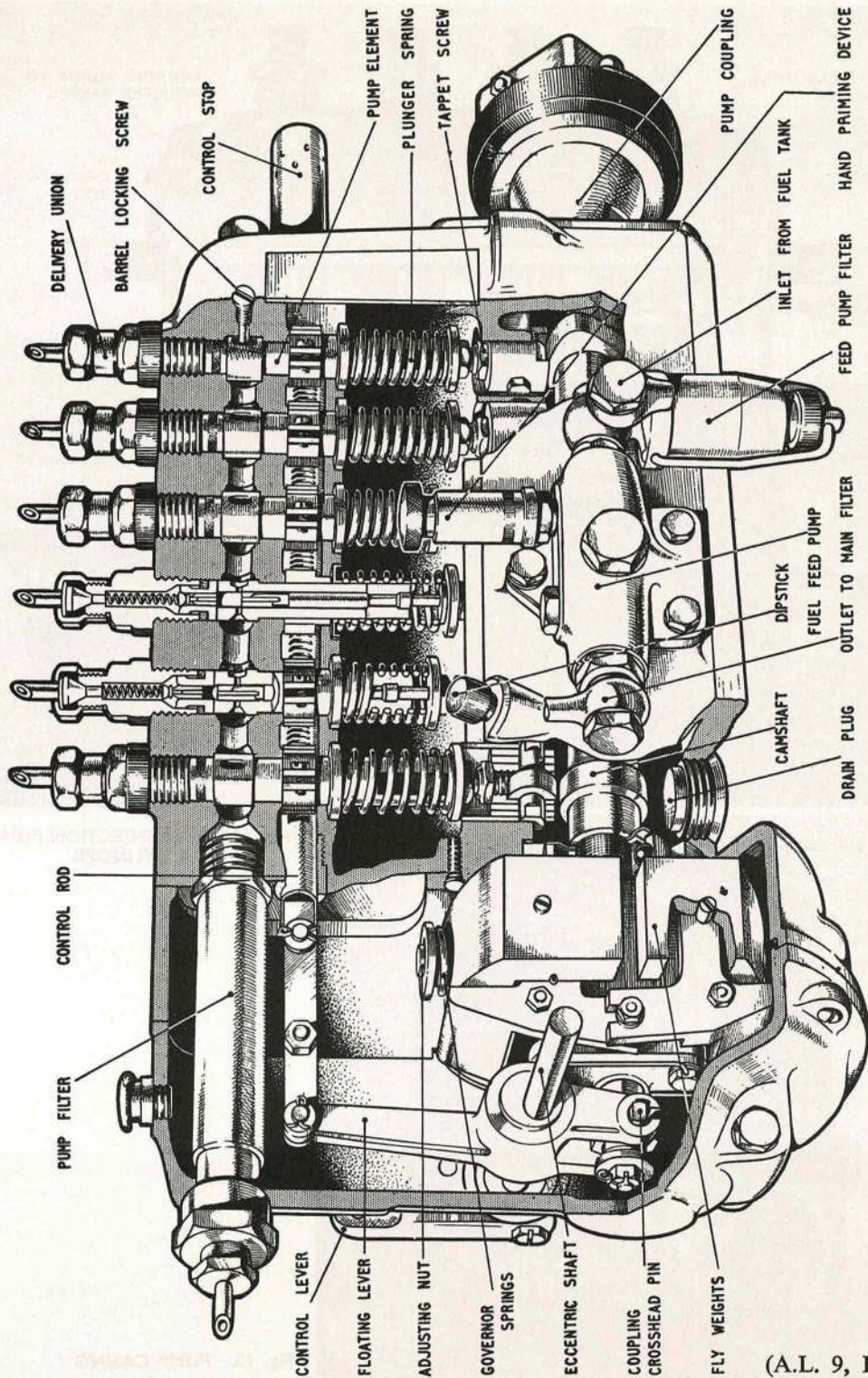


Fig. 13. FUEL INJECTION PUMP—6 CYLINDER

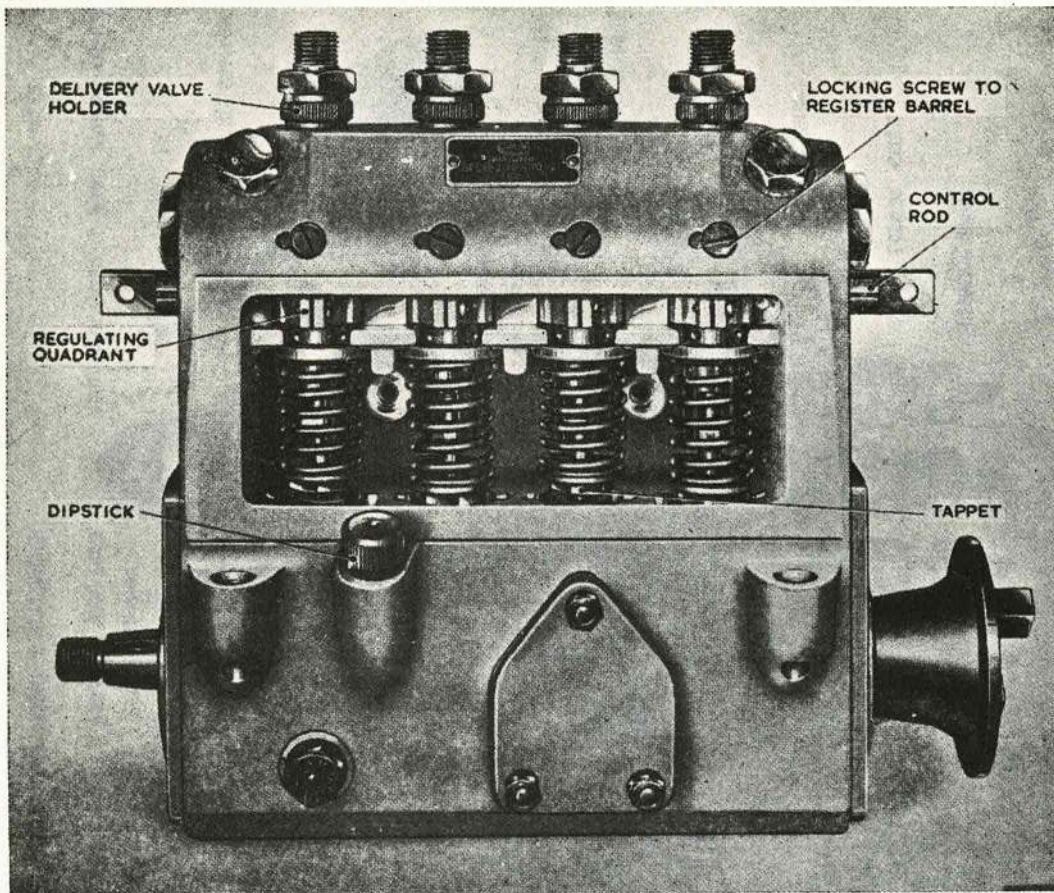


Fig. 14. FUEL INJECTION PUMP—4 CYLINDER

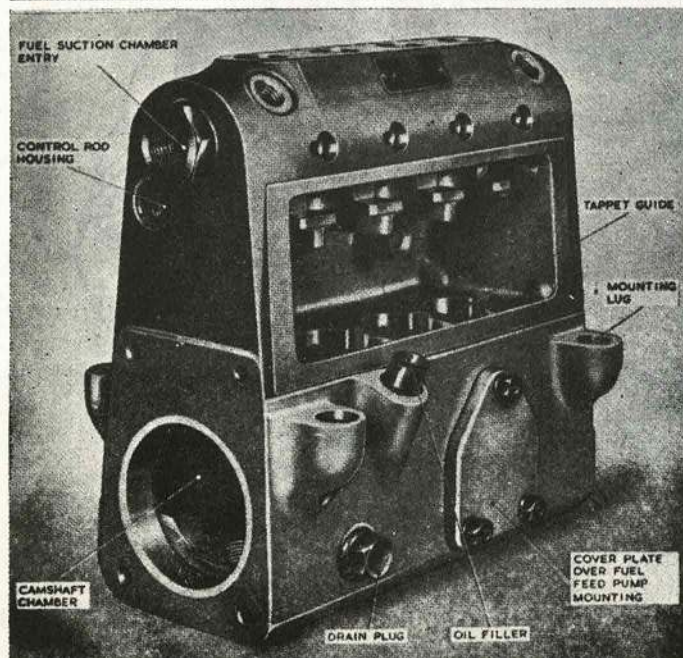


Fig. 15. PUMP CASING

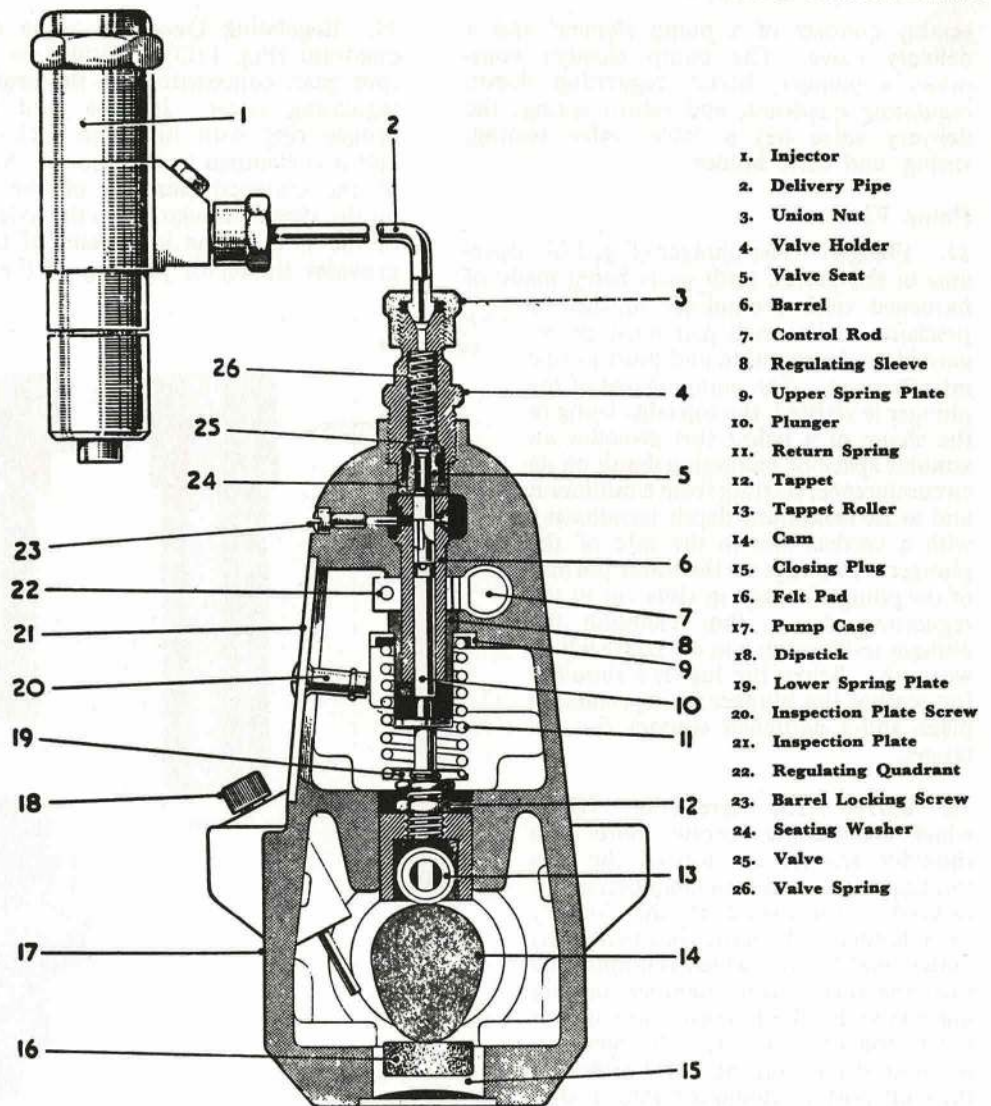


Fig. 16. SECTIONED PUMP

end of the casing with a screwed plug hole, to one of which the delivery pipe is fitted; either end may be used for fuel delivery, the other end being plugged. The pump casing is suitably bored for the fitting of plunger pump assemblies, camshaft and control rod. A heavy section web above the camshaft chamber incorporates the tappet guides, these being bored and slotted in the web. Communicating with the sump of the camshaft chamber, the casing is fitted with a dipstick and oil filler, as well as a

drain plug. Mounting for a fuel feed pump is provided near the base at the front of the casing and, if a pump is not fitted, the opening is closed by a cover plate. Four lugs, one near each corner of the casing, are for mounting the injection pump on the engine.

Plunger Pump Assemblies

30. Fig. 16 shows a cross section of an injection pump through one of the plunger pump assemblies. Each plunger pump as-

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sembly consists of a pump element and a delivery valve. The pump element comprises a plunger, barrel, regulating sleeve, regulating quadrant, and return spring; the delivery valve has a valve, valve seating, spring, and valve holder.

Pump Element

31. Plunger. The plunger (Fig. 17A) operates in the barrel, both parts being made of hardened steel, ground and finished to precision limits; each pair must be regarded as inseparable and must not be interchanged. The pumping end of the plunger is waisted, the top edge being in the shape of a helix; this provides an annular space of increasing depth on its circumference, starting from a minimum and at its maximum depth terminating with a vertical slot in the side of the plunger. Two lugs on the lower portion of the plunger engage in slots cut in the regulating sleeve, thus enabling the plunger to be rotated in the sleeve whilst working. Below the lugs is a shoulder for seating the plunger spring retaining plate, and a hardened contact for the tappet.

32. Barrel. The barrel (Fig. 17B) in which the plunger operates, butts on a shoulder against the top of the web forming the fuel suction chamber, and is secured in the casing by the delivery valve holder. The barrel has two ports, "inlet" and "spill", which communicate with the fuel suction chamber and are uncovered by the plunger when it is at the bottom of its stroke. The inlet port is countersunk, but the outer orifice of the spill port is elongated into a slot, which, together with a locking screw, forms the means of axial location for the barrel.

33. Regulating Sleeve. The regulating sleeve (Fig. 17C) fits over the lower end of the barrel about which it is free to oscillate. Two slots at the lower end of the sleeve engage the plunger lugs to impart axial movement to the plunger, such movement being applied by the regulating quadrant which is clamped to the top end of the sleeve. On assembly, the marked lug of the plunger must be fitted into the slot in the sleeve which has a short line inscribed just above it (Fig. 18).

34. Regulating Quadrant. The regulating quadrant (Fig. 17D) is formed as part of a spur gear, concentric with the centre of the regulating sleeve. It is a split phosphor bronze ring with lugs and locking screw, and it is clamped to the sleeve. Adjustment of the clamped position of the quadrant on the sleeve, in relation to the axial position of the slots in the lower end of the sleeve, provides timing or phasing of the injection

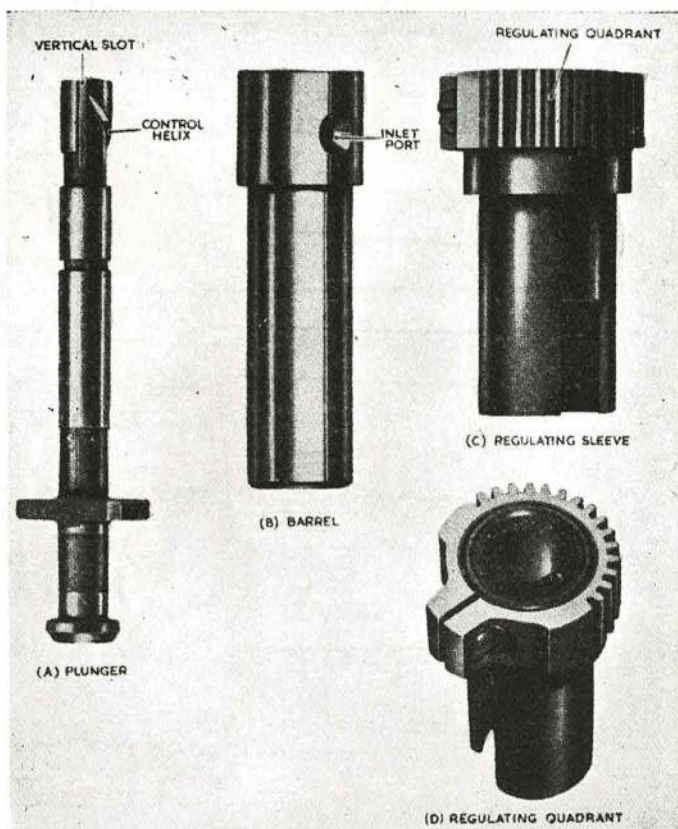


Fig. 17. PUMP ELEMENT COMPONENTS

cut-off; a timing mark on the quadrant must coincide with one on the sleeve (Fig. 18). Teeth of the quadrant mesh into teeth of the control rod or rack which is common to all pump units.

35. Return Spring. A spiral spring, fitted between the upper spring plate which butts on a shoulder on the upper end of the regulating sleeve, and the lower spring plate which butts on a shoulder at the base of the pump plunger, provides impetus for the return or intake stroke of the plunger (Fig. 16).

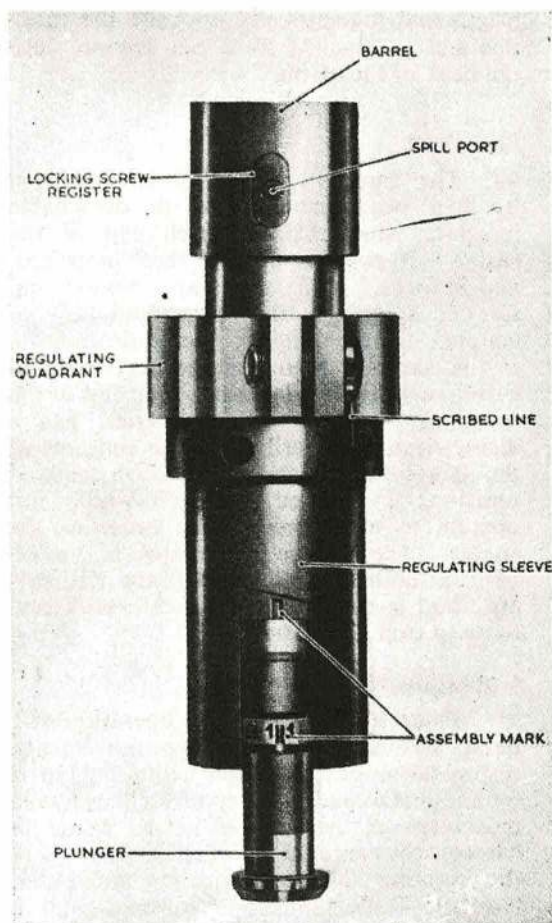


Fig. 18. ASSEMBLY OF PUMP ELEMENT

Delivery Valve

36. Valve. The valve (Fig. 19) is mitre faced with a cruciform guide, and is topped by a spigot for the valve return spring. Separated from the valve face by an annular groove, the top of the valve guide is formed as a shallow piston which fits accurately in the bore of the valve seating. This device provides a sharp cut-off of fuel injection at the termination of the pump delivery and is termed the anti-dribble device. The light coil spring fits above the valve, and helps to return and keep the valve on its seating during the non-delivery period.

37. Valve Seat and Valve Holder. The valve seating (Fig. 19) has its lower surface

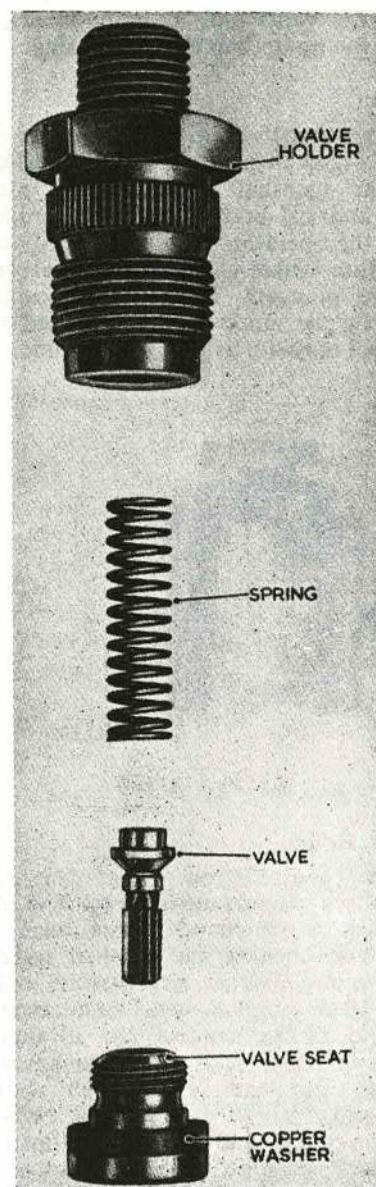


Fig. 19. DELIVERY VALVE COMPONENTS

ground to form a pressure face joint with the top of the pump barrel; its top is externally threaded for extraction purposes, and it is internally bored to house the delivery valve guide. It is clamped on to the top of the pump barrel by the delivery valve holder (Fig. 19), which screws into the top of the casing. Thus, the valve holder

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houses the valve and its components, and secures the pump barrel in the casing.

Tappet

38. The tappet (Fig. 20) operates in a guide formed in a heavy web in the casing over the camshaft chamber. It consists of a cylindrical body fitted with a roller to engage the operating cam, and an adjustable tappet face which engages with the base of the pump plunger. The tappet is positioned axially in the guide by an extended roller pin which operates in slots in the guide.

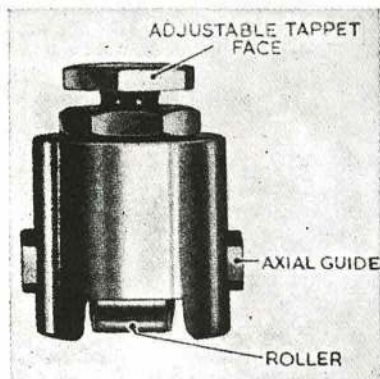


Fig. 20. TAPPET

Control Rod

39. The control rod (Fig. 21) passes through and extends outside each end of the casing. It is machined in the form of a rack which engages the toothed quadrant of each pump element, and permits uniform control of all pump elements simultaneously. Each end of the control rod is similarly formed for fork and pin attachment of controls. One end is inscribed with an arrow and "STOP", and both ends are marked with a centre-dot to indicate the centralized position of the control rod travel. The rack is located radially, on assembly, by a slot in the middle of its

length and diametrically opposite the rack; this slot is engaged by a pin screwed into the back of the casing.

Camshaft

40. The camshaft (Fig. 22) is supported by ball bearings mounted in detachable housings which fit into each end of the casing. It extends outside these housings, and is formed with keyed taper shanks and screwed spigots so that either end may be coupled to the engine; the end opposite to that coupled to the engine may be fitted with a centrifugal governor. One end of the camshaft, varying with the series, has a diametrical mark scribed on the end face of the spigot to indicate its correct assembled position; a reversed camshaft would not operate in the correct firing order of the engine. The camshaft operates in an oil bath in the base of the casing, and felt pads are fitted in closing plugs beneath each cam to assist cam lubrication.

Operation of Pump Unit

41. **Pump Movements.** The operation of a pump element comprises two movements, reciprocation of the plunger within the barrel for the intake and delivery of fuel, and axial movement of the plunger in the barrel to control the amount of fuel delivered to the injector. The reciprocating movement (stroke) of the plunger is constant, and is operated by the camshaft and return spring. Commencement of injection is also constant, but the amount of fuel injected depends upon the particular positioning of the plunger helix, i.e. whether pressure relief of fuel occurs earlier or later in the plunger stroke, thus affecting the time of "cut-off" of delivery of the fuel. Axial movement of the plunger is obtained by the regulating quadrant being operated by the rack on the control rod, and, because all pump elements in an injection pump, irrespective of number,

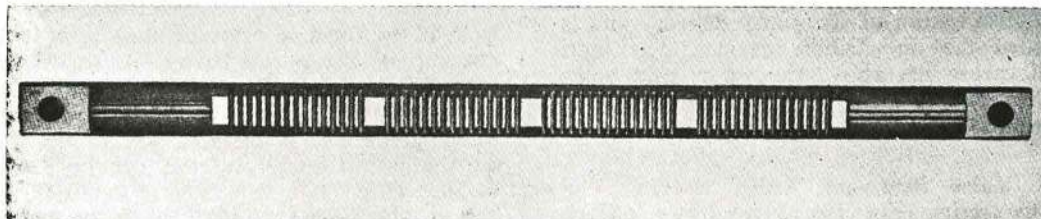


Fig. 21. CONTROL ROD

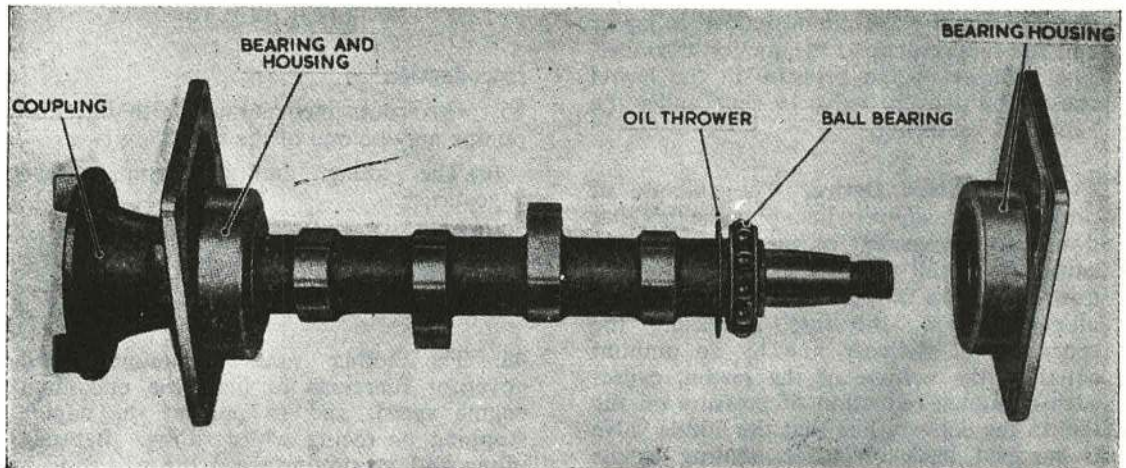


Fig. 22. CAMSHAFT

are operated by the same rack, it follows that uniform control is obtained.

42. Fuel Delivery. When the plunger is at the bottom of its stroke (Fig. 23A), the top of the plunger uncovers both the inlet and the spill ports, so that fuel flows into the barrel above the plunger and into the cavities formed by the vertical and helical grooves in the plunger. On the upward stroke some fuel is forced back through the ports until the plunger closes them; further upward movement of the plunger exerts pres-

sure on the fuel trapped in the barrel and causes it to lift the delivery valve, whence the fuel passes into the pipe connecting the pump to the injector (Fig. 23B). As the system is already full of fuel, this extra fuel causes the pressure to rise and lift the injector valve (nozzle needle), and fuel is sprayed into the combustion chamber. At this moment, fuel is being pumped into the delivery pipe at the pump end, and an equal quantity is being sprayed out at the nozzle end. Delivery continues until the edge of the helix uncovers the spill port (Fig. 23C), when the

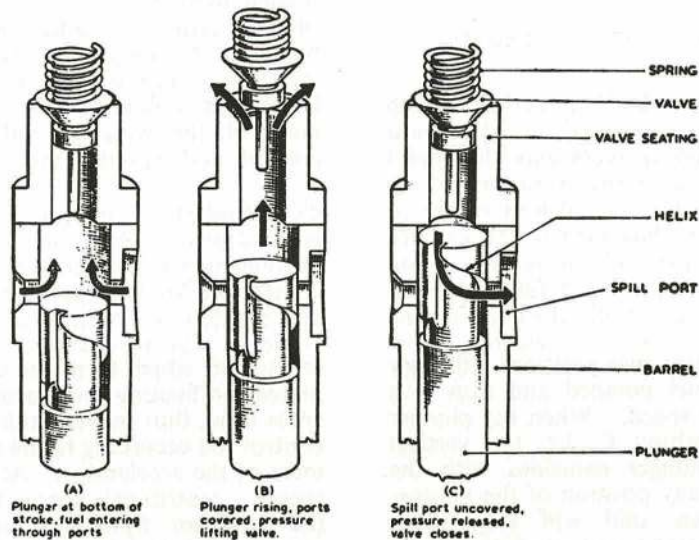


Fig. 23. OPERATION OF PLUNGER PUMP

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pressure on the fuel immediately collapses and the fuel returns to the suction chamber. The collapse of the pressure in the barrel causes the delivery valve to shut under the action of its spring.

43. Anti-Dribble Device. The collapse of pressure in the delivery line, plus the fact that downward movement of the delivery valve causes the small piston (para. 36) to sweep down the bore of the valve seating with a plunger action, therefore increasing the space in the delivery line by an amount equal to the volume of the piston, causes such a sudden reduction of pressure on the fuel in the delivery line that the nozzle valve is snapped back on to its seating by its return spring, thus terminating the fuel spray into the cylinder without dribble.

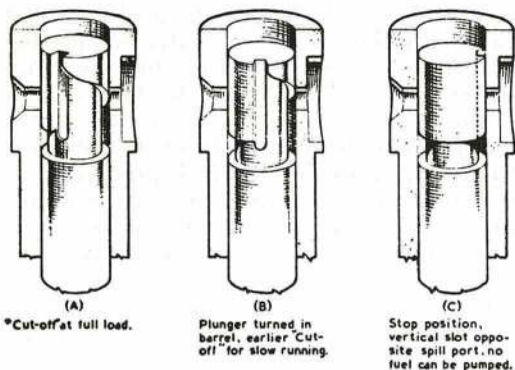


Fig. 24. VARIATION OF FUEL DELIVERY

44. Delivery Control. The period during which the spill port is closed on the upward stroke of the plunger represents the useful pumping stroke, therefore, rotation of the plunger will uncover the spill port earlier or later in the stroke, thus varying the amount of fuel pumped. Fig. 24A shows the position of the plunger for cut-off at full load, and B shows an earlier cut-off when the engine is slow running. Axial movement of the plunger within these two positions will vary the amount of fuel pumped and thus give control of engine speed. When the plunger is rotated to position C, i.e. the vertical groove in the plunger coincides with the spill port during any position of the plunger stroke, continuous spill will take place on the upward stroke and no fuel will be pumped; this is the stop position for the control.

GOVERNORS

Introduction

45. Governors incorporated with injection pumps may be one of the following types:—

- (a) The idling and maximum speed governor.
- (b) The variable range speed governor.
- (c) The pneumatic governor.
- (d) Combinations of (a), (b) and (c).

46. The idling and maximum speed governor functions to limit the maximum engine speed, and to prevent the engine stopping or racing whilst idling. Between idling and maximum speed the governor is ineffective, control of engine speed between these limits being the responsibility of the operator. The variable range speed governor and the pneumatic governor are intended to maintain a pre-set engine speed under varying load, such as is required of an engine coupled to a generator or other stationary equipment; these types of governor permit any engine speed between idling and maximum to be selected and maintained.

Idling and Maximum Speed Governor

47. This type of governor (Fig. 25) is a centrifugal unit bolted directly on to the pump housing, and having two spring-loaded flyweights mounted on an extension of the injection pump camshaft. Movement of the weights is transmitted to the injection pump control rod by bell-crank levers and a floating lever, outward movement of the weights tending to pull the control rod toward the "stop" position.

48. Principle. The principle of operation is illustrated at Fig. 26. Two independent movements which operate the injection pump control rod are incorporated in the governor, the accelerator control and the governor control. The accelerator moves the control cross-shaft which turns the eccentric and displaces the floating lever which pivots on the cross head, thus moving the injection pump control rod according to the amount of movement of the accelerator. At excessive engine speeds, centrifugal force tends to cause the governor flyweights to fly outwards, causing the bell-crank levers to pull the link pin and crosshead, thereby pivoting the floating lever on the eccentric and moving

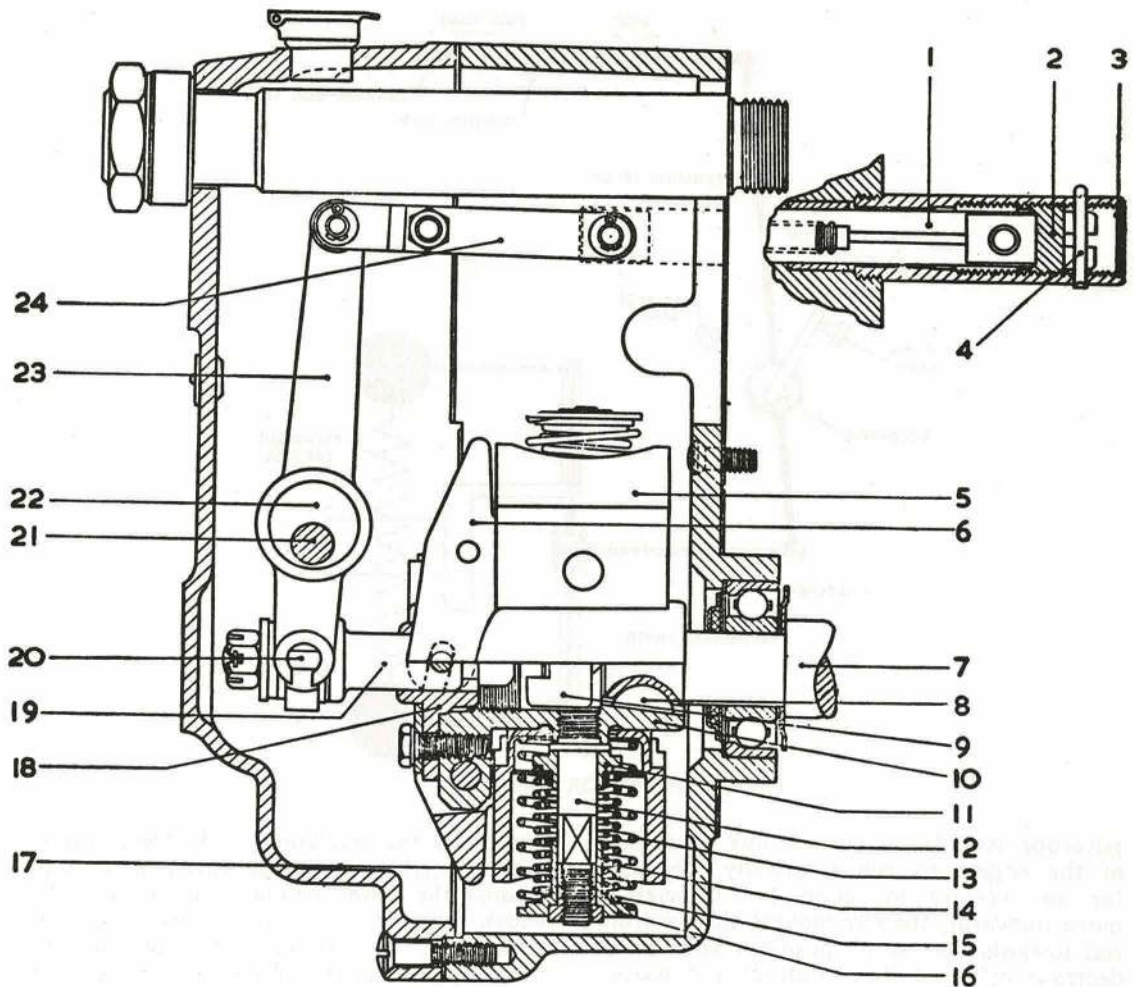


Fig. 25. IDLING AND MAXIMUM SPEED GOVERNOR

- | | | | |
|---------------------|-------------------------|------------------------|----------------------|
| 1. Control Rod | 7. Camshaft | 13. Spring | 19. Link Pin |
| 2. Control-Rod Stop | 8. Key | 14. Spring | 20. Crosshead |
| 3. Dust-cap | 9. Nut | 15. Spring | 21. Cross-shaft |
| 4. Split Pin | 10. Fixing Sleeve | 16. Outer Spring Plate | 22. Eccentric |
| 5. Flyweight | 11. Bottom Spring Plate | 17. Flyweight | 23. Floating Lever |
| 6. Bell-Crank Lever | 12. Spring Pillar | 18. Guide Bush | 24. Control-Rod Link |

the control rod towards the stop position. The weights are, however, retained by springs which, as the engine speed falls, move the weights inwards to maintain a steady running position. Thus, the fulcrum for the floating lever during accelerator movement is the crosshead, and for governor controlled movement the eccentric is the fulcrum. These movements can operate separately or together, hence, any move-

ment of the control rod imparted by the accelerator to produce excessive engine speed can be corrected by the governor.

49. Idling Control. Minimum speed control is effected by the flyweight outer springs (Fig. 27A and B), for at low speed the flyweights float at a position determined by centrifugal force and the pressure of the outer springs. Thus, at low speeds the

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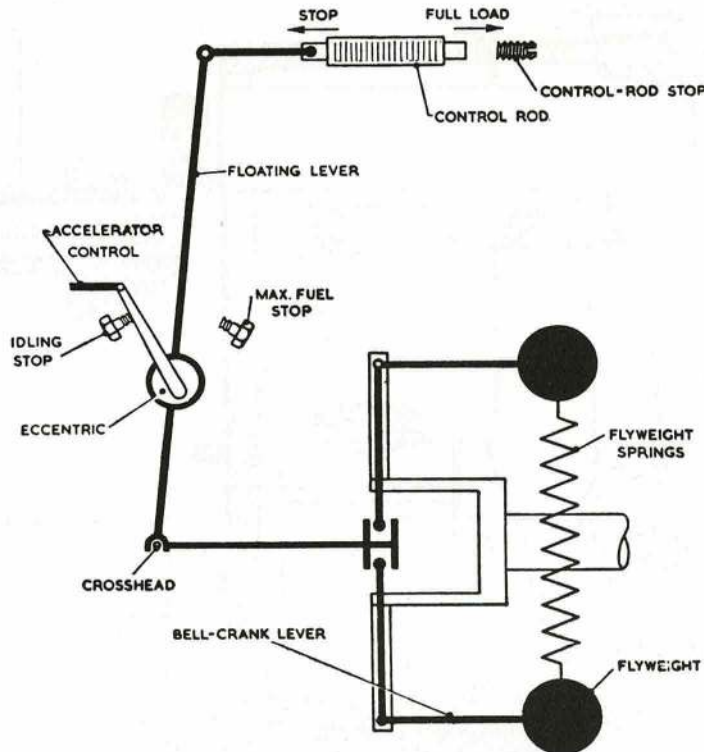


Fig. 26. GOVERNOR PRINCIPLE

governor is able to correct any tendency of the engine to run erratically, because for an increase in speed the flyweights move outwards, thereby moving the control rod towards the "stop" position, and for a decrease of speed they contract and move the control rod in the reverse direction to inject more fuel.

50. Maximum Speed Control. Engine speeds between idling and maximum are con-

trolled by the accelerator. At these speeds the flyweights move outwards and bear against the lower spring plates (Fig. 27B) which support the inner springs. They remain in this position throughout the normal speed range of the engine, because the inner springs are strong enough to resist compression at these speeds, thus, the governor is inoperative. If the accelerator is moved so far that the injection pump delivers more fuel than the engine needs for

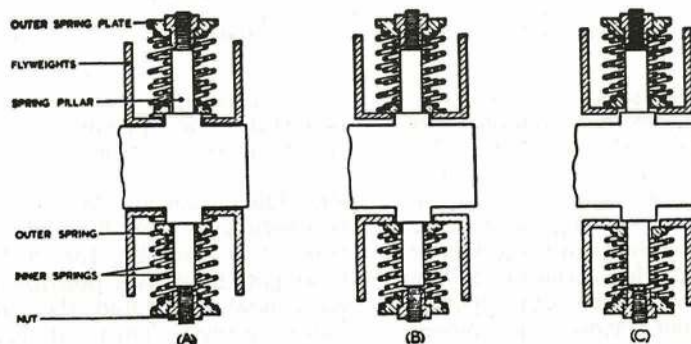


Fig. 27. OPERATION OF FLYWEIGHTS

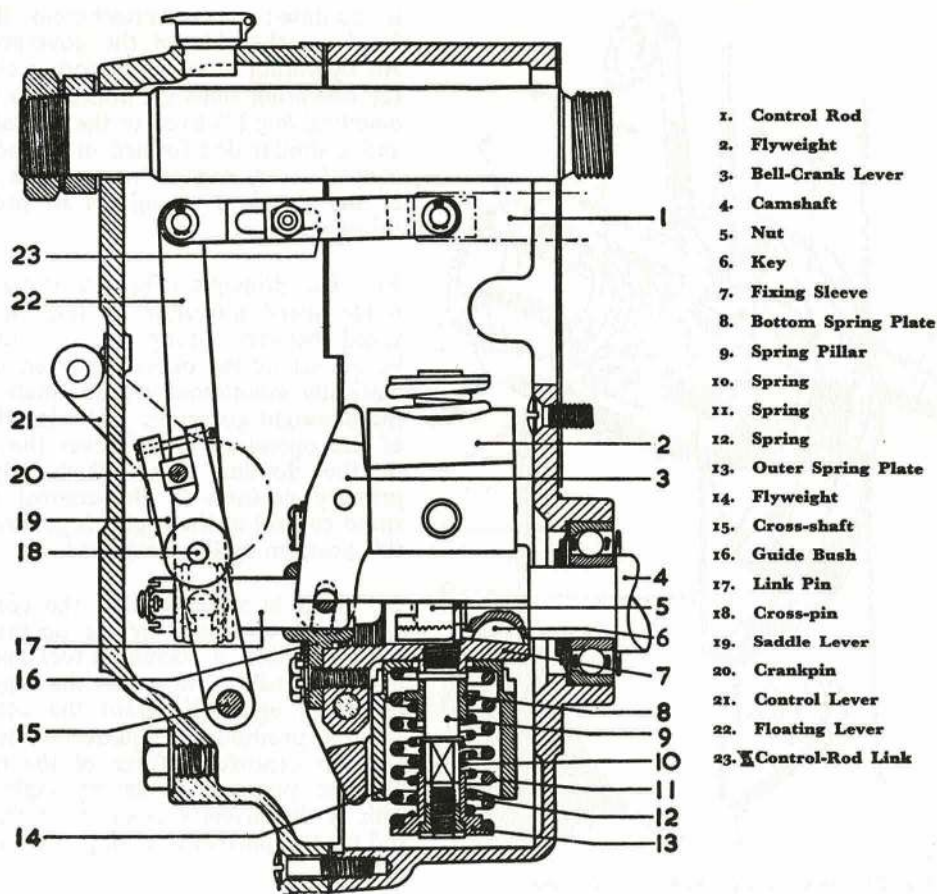


Fig. 28. VARIABLE RANGE SPEED GOVERNOR

the load, the speed will exceed the predetermined maximum and centrifugal force of the flyweights will overcome the pressure of the inner springs (Fig. 27C). This outward movement of the flyweights will move the injection pump control rod towards the "stop" position to hold the engine speed at its predetermined maximum, and any further movement of the accelerator will be similarly countered by the governor.

51. Control Rod Stop. To limit the control rod travel when starting the engine, the control rod is fitted at its free end with an adjustable stop. This is needed because when the engine has stopped, the flyweights, under pressure of the outer springs, will have moved inwards to rest on the extension shaft, thereby pushing the link pin outwards

to pivot the floating lever on the eccentric, and push the control rod to a position beyond the normal maximum fuel delivery. This is due to the high leverage ratio between the crosshead to eccentric and eccentric to control rod. To obviate this excessive movement of the control rod, the control rod stop is adjusted to give the best amount of excess fuel for easy starting; as soon as the engine starts, the flyweights close the control rod to the idling position.

Variable Range Speed Governor (Figs. 28 and 29)

52. The construction of the variable range speed governor differs from the idling and maximum speed governor only in the linkage (19 and 22) between the crosshead and the injection pump control rod, and also that

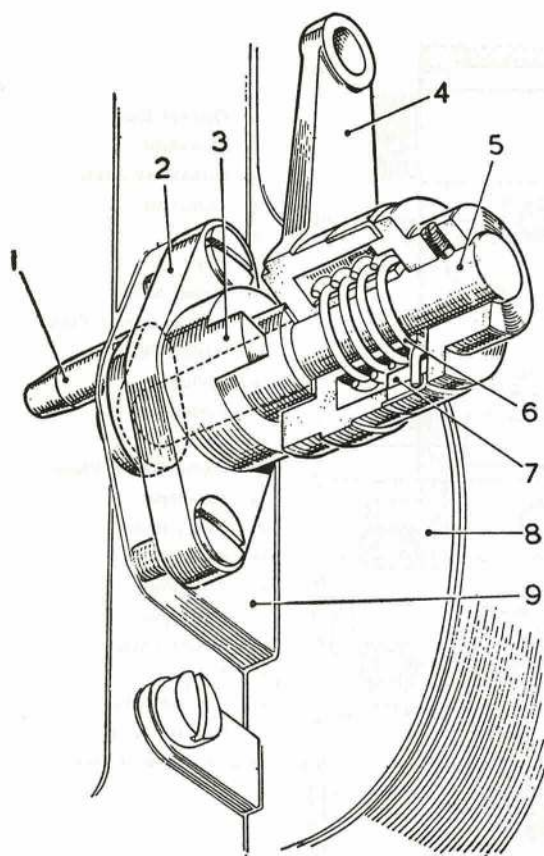


Fig. 29. SPRING COUPLED OPERATING LEVER

- | | |
|----------------------|--------------------|
| 1. Crankpin | 6. Coupling Spring |
| 2. Attachment Flange | 7. Coupling Dog |
| 3. Stop | 8. Pump Casing |
| 4. Control Lever | 9. Governor Casing |
| 5. Operating Shaft | |

each of the flyweights are fitted with three springs (10, 11 and 12) which come into operation consecutively as the engine speed is increased.

53. This linkage comprises a floating lever (22) having a forked end which engages with the crosshead; its other end connects with the control rod, and it pivots just above the crosshead on a cross-pin (18) between two arms of a saddle lever (19), the axis of which is a cross-shaft (15) secured in the base of the governor casing. One arm of the saddle lever is slotted, and this slot is engaged by the pin of a crank (20) on an operating shaft (5 in Fig. 29) which

is mounted in an attachment flange (2) fitted on the side of the governor casing. An operating lever (4) is spring coupled to the operating shaft (5) outside the casing; a coupling dog (7) fixed to the operating shaft and a similar dog formed in the boss of the control lever, engage between the two ends of the coupling spring (6) to provide the coupling.

54. The principle (Fig. 30) of the variable range speed governor is, that any engine speed between idling and maximum can be pre-set by the operating lever, and automatically maintained within small limits by the flyweight governor. Altering the setting of the operating lever moves the cross-pin of the floating lever which adjusts the primary position of the control rod, and speed control of the engine is governed about this position by the flyweights.

55. If it is assumed that the control rod is initially advanced by the operating lever to the position of maximum fuel delivery, the flyweights will move out as the engine speed increases, and will retard the control rod until a position of equilibrium is reached between centrifugal force of the flyweights and the pressure of the flyweight springs. This is the primary position of the control rod for this particular setting of the operating lever.

56. When this position has been reached, however, the control rod has been partly retarded from the initial setting, and the engine speed could be again increased by a further advance of the operating lever. A fresh state of equilibrium for this new speed will be assumed by the flyweights, and the control rod will have another primary position according to the new position of the operating lever.

57. Thus it will be seen, that most pre-set positions of the control lever will initially advance the control rod to its maximum stop, hence the necessity for a spring coupling on the operating lever, in order that the lever can be advanced to the maximum and the coupling stressed to counter the retarding effect of the flyweights until the selected speed has been reached. As this speed is approached and the control rod tends to be retarded by the governor, the stress of the spring is progressively released,

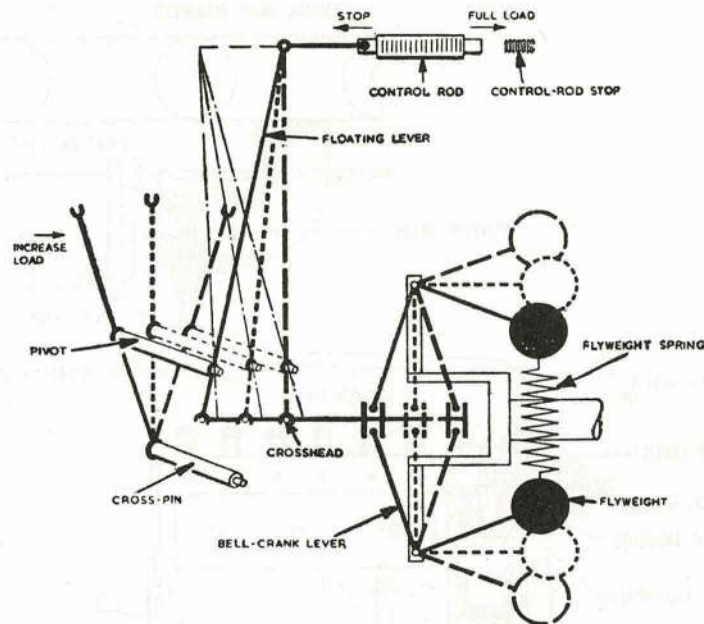


Fig. 30. FUNCTIONING OF GOVERNOR

adjusting the position of the cross-pin until, when the primary position of the control rod is reached, there is no stress on the spring coupling, and a condition of equilibrium is attained by the governor components.

58. When the primary position of the control rod is reached for any particular setting of the operating lever, the flyweights control the engine speed within a limited range according to the variation of the engine load.

Pneumatic Speed Governor

59. This type of governor controls the fuel injection pump output, and therefore the engine speed, at any given speed between the idling and maximum limits; maximum and idling speeds are predetermined by stop screw adjustment. The pneumatic governor (Fig. 31) comprises a diaphragm unit which is mounted on the injection pump and connected to the control rod, and a venturi control unit mounted on the engine air intake. The venturi unit is connected to the diaphragm unit by means of a flexible pipe.

60. **Diaphragm Unit.** The diaphragm unit (Fig. 32) housing is made in two parts (2).

and a leather diaphragm (1) clamped between them forms an airtight chamber on one side, to which the pipe from the venturi unit is connected; the other side of the diaphragm is open to atmosphere. The diaphragm is connected to the control rod (12), and a light spring (7) acting on the diaphragm tends to keep the control rod in the full open position; the spring also serves to damp out undue oscillations of the diaphragm when the engine is running. For slow running another spring (8) adjusted by the screw (9) gives additional loading to the diaphragm through the plunger (10). An externally fitted stop lever (11) operates a short lever inside the unit which connects with the control rod, and this is used for operating the control rod in the stop position.

61. **Venturi Control Unit.** This unit (Fig. 32) comprises a body formed with a venturi (3) to suit the particular type of engine. At the smallest throat diameter of the venturi, a butterfly valve (5) is fitted to control the air flow. This valve is fitted with an operating lever (4) which is connected to the engine control lever, the movement of the lever being limited by two adjustable stops which control the maximum and idling

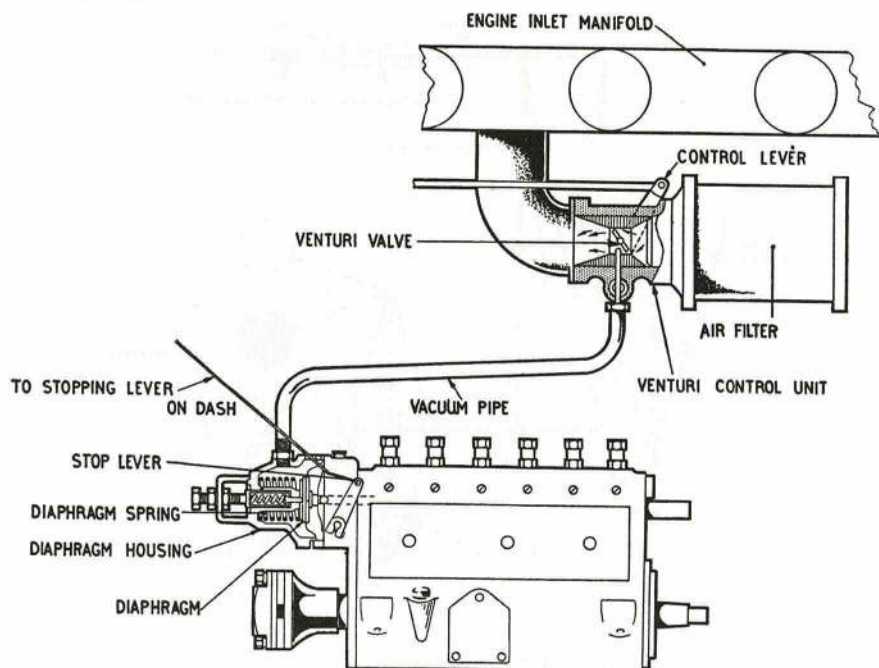


Fig. 31. PNEUMATIC GOVERNOR LAYOUT

1. Diaphragm
2. Housing
3. Venturi
4. Operating Lever
5. Butterfly Valve
6. Connecting Tube
7. Diaphragm Spring
8. Slow-Running Spring
9. Slow-Running Adjustment Screw
10. Plunger
11. Stop Lever
12. Control Rod

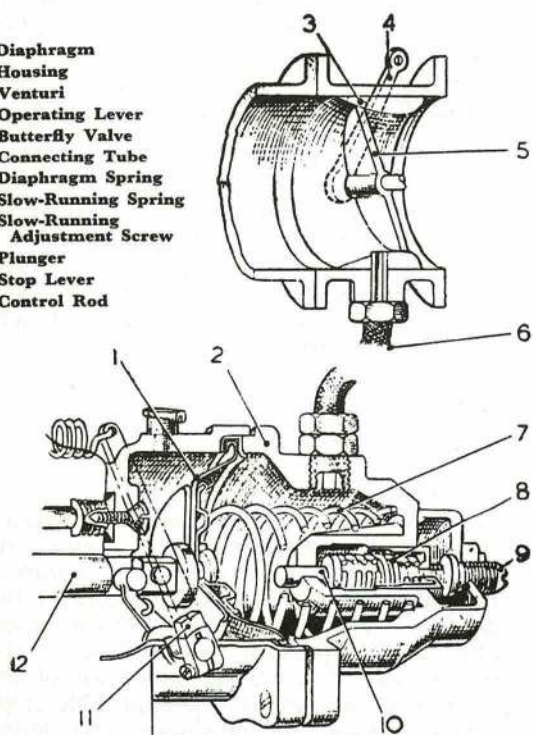


Fig. 32. PNEUMATIC TYPE GOVERNOR

speeds. On the engine side of the butterfly valve and communicating with the depression in the venturi, there is a small bore pipe that passes out to a pipe connection on the body, to which is connected the pipe to the diaphragm unit (6).

62. Operation. When the engine is not running, and the stopping control is released, the diaphragm spring forces the control rod to the full extent of its travel, i.e. the starting position. Once the engine is started and the butterfly valve is back to the idling stop, the existing depression is communicated to the air-tight compartment of the diaphragm unit. This causes the diaphragm and control rod to move towards the stop position, until the engine is running at the speed determined by the adjustment of the idling stop. Increase of engine speed is obtained by opening the butterfly valve, and so decreasing the velocity of the air flow past the mouth of the connecting tube, thus decreasing the depression and allowing the diaphragm and control rod to move towards the maximum speed position. Hence, engine speeds between idling and maximum are determined by the opening of the butterfly valve causing a corresponding

movement of the diaphragm and control rod on the fuel injection pump.

C.A.V. FUEL INJECTORS

Introduction

63. There are three essentials regarding the delivery of fuel which are necessary to ensure satisfactory running of a compression ignition engine. The fuel must be delivered to the cylinders at the right time, in the right quantity, and in the right condition for complete combustion. The first two requirements are met by the correct adjustment of the fuel injection pump, and the third by a device known as an injector.

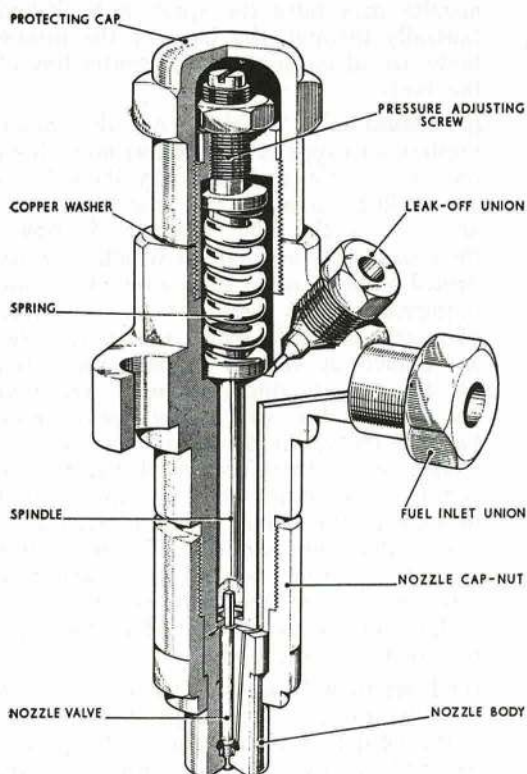


Fig. 33. C.A.V. FUEL INJECTOR

64. Each cylinder is fitted with an injector (Fig. 33), which is connected to its relevant delivery valve of the pump by a pipe, and it comprises two assemblies, viz., a nozzle and a nozzle holder. The nozzle atomizes the fuel for combustion, and the nozzle

holder provides the means for securing the nozzle to the engine cylinder and connecting the fuel supply pipe from the pump; additionally, the holder houses a spring which is required for the loading of the nozzle valve.

Nozzles

65. **General.** The nozzles described in the following paragraphs are all "closed" types, so called because after each injection of fuel, the delivery aperture in the nozzle is closed by a valve. They are functionally the same, but differ in design details to suit the requirements of different types of engines.

66. **Construction.** A nozzle consists of two parts, the nozzle valve and the nozzle body; these two parts are mated in manufacture and must be kept as such throughout their service life. The valve is of plunger form (Fig. 34), the main portion being lapped to very fine limits into a central boring in the nozzle body. Each end of the valve is reduced in diameter, one end, known as the stem, being machined to a valve face, and the other, the stalk, being a plain portion on which a spring-loaded spindle is located. The nozzle body houses the valve in the central boring, and incorporates the valve seating on which the valve seats to seal off an injection aperture in the lower part of the nozzle body. Additionally, the body has a series of small-diameter passages, which lead from a concentric groove in its top face to a gallery or reservoir immediately above the valve seat. The top face of the body is ground to a mirror-like finish, to mate with a similar face on the lower part of the nozzle holder.

67. **Types of Nozzles.** To ensure the efficiency of engines which, because of their design require a different kind of spray, both the nozzle valve and nozzle body vary for different types of engines. For this reason, it is imperative that unserviceable nozzles are replaced by identical ones. There are five classes or types of nozzle and each of these may also have detail differences. Typical examples are shown in Fig. 35 and are as follows:—

(a) **Pintle Nozzles.** This type of nozzle is used on engines which incorporate a pre-combustion chamber or "air-cell" in

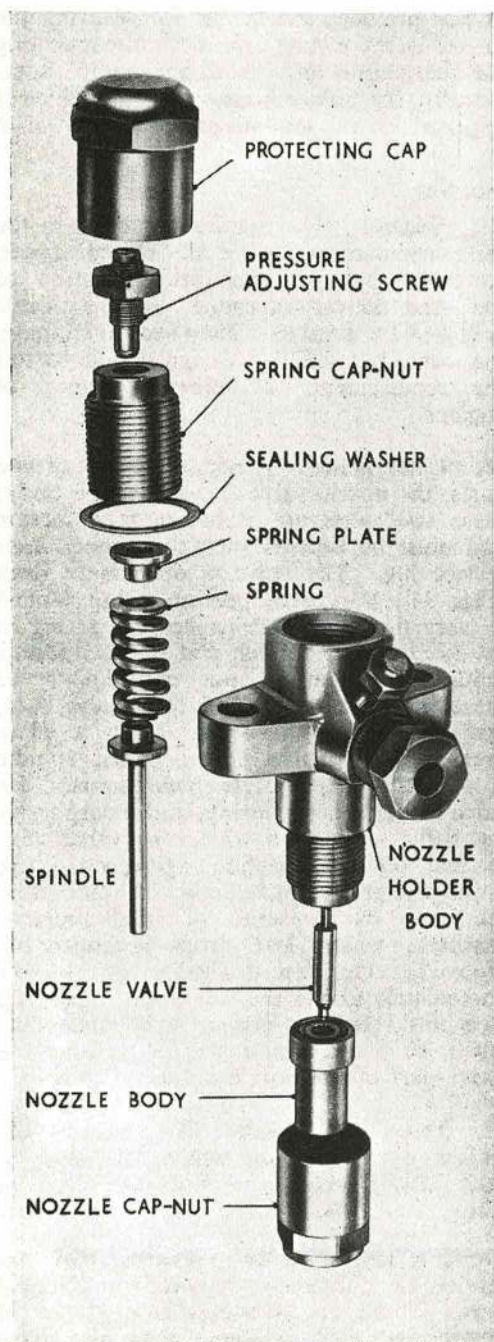


Fig. 34. FUEL INJECTOR DISMANTLED

each cylinder. The stem of the valve is extended to form a pin or pintle which protrudes through an injection orifice in the base of the nozzle body. By varying

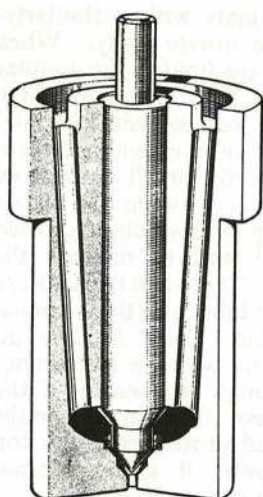
the size and shape of the pintle, variation in the spray is obtained; the form of the spray may vary from one of pencil form, to one of cone form with an included angle of 60 degrees or more.

(b) **Delay Nozzles.** Delay nozzles are, in general, a variation of the pintle type; they improve the idling of some engines, usually those of the pre-combustion chamber type. With this type of nozzle the pintle is designed to increase the rate of injection towards the end of each delivery, and this has the effect of lengthening the period of injection at idling speeds without interfering with combustion at higher speeds.

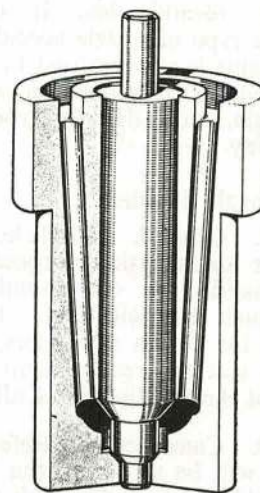
(c) **Single-hole Nozzles.** These types of nozzles may have the spray hole drilled centrally through the base of the nozzle body, or, at an angle to the centre line of the body.

(d) **Multi-hole Nozzles.** As the name implies, this type of nozzle has more than one spray hole, and generally these holes are drilled radially from the centre on an even pitch. The number of holes, their size, and the angle at which they are drilled, depend upon the particular engine requirements. A variation of this type of nozzle is one in which the spray holes are drilled at various angles, and often on an uneven pitch. This is required on engines for which the spray must be in a particular direction; this type of nozzle must, therefore, be fitted to the engine at all times with the spray holes located in the same position relative to the combustion chamber. To ensure this correct location, two short dowels are fitted in the joint face of the nozzle holder, and mating holes are drilled in the joint face of the nozzle body.

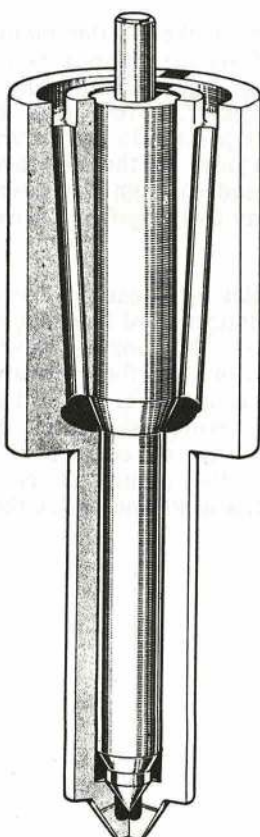
(e) **Long-stem Nozzles.** One of the main considerations in the design of all nozzles is the need to maintain the working parts as cool as possible. For direct-injection engines where, because of the limited space between the valves in the cylinder head, adequate cooling is not possible for a normal nozzle, a longer nozzle is used. The extra length allows the critical portion of the nozzle to be more remote from the source of heat, and, by a reduction in diameter of the extended portion, a more efficient dissipation of heat is obtained.



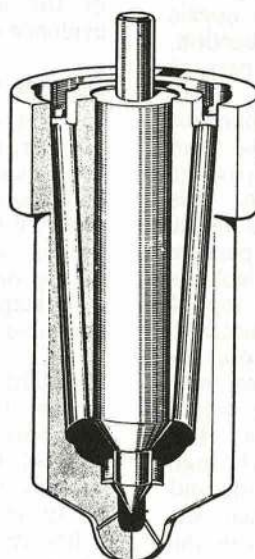
SINGLE HOLE



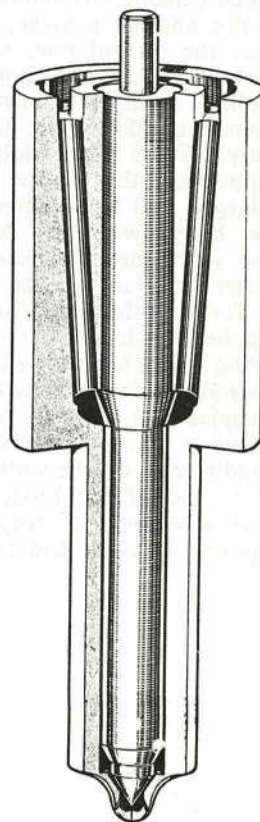
PINTLE



LONG STEM



MULTI-HOLE



LONG STEM

Fig. 35. TYPES OF NOZZLES

(A.L. 9, Feb. 61)

68. **Identification.** It is imperative that the type of nozzle specified for a particular engine is always fitted to that engine, and to facilitate identification every nozzle has a code, indicating its type, stamped on the body.

Nozzle Holders

69. **General.** Nozzle holders are also coded for identification purposes, being made and classified to correspond with the nozzles which they will hold. There are variations in the design of holders, in all size groups, to suit the requirements of engine designs, but the functioning of all is the same.

70. **Construction.** Referring to Fig. 33, it will be seen that the body of the nozzle holder is provided with an inlet connection, a leak-off connection, and a central passage which extends throughout its length. There is also another passage, smaller in diameter than the central one, which extends from the bottom face of the body to a short passage leading from the inlet connection; this passage is offset from the centre line of the body. There is an additional short passage connecting the centre drilling, which is enlarged and screw-threaded at the top of the body, with the leak-off connection. The screw-threaded recess at the top, together with a cap-nut which screws into it, forms a housing for the spring which supplies the load to the nozzle valve. The spring load is transmitted to the valve through a spindle which extends from the spring and occupies the central passage. When the nozzle is fitted to its holder, the end of the spindle seats on the stalk of the valve. The lower end of the holder is screw-threaded externally for the reception of the nozzle cap-nut, and the end face is ground to a

mirror-like finish to mate with a similarly-prepared face on the nozzle body. When the nozzle is fitted to the holder, the annular groove in the nozzle face aligns with the passage in the holder leading from the inlet connection. The leak-off connection is provided as an outlet for the small amount of fuel which finds its way between the nozzle valve and nozzle body, and which, if allowed to accumulate, would tend to increase the pressure on the valve. This small fuel leakage is beneficial, because it lubricates the frictional surfaces of the nozzle. Some holders incorporate a feeling pin, which is a clearance fit in a drilling through the centre of the pressure adjusting screw. The pin rests on the spring lower plate and protrudes at the top of the adjusting screw; it gives external evidence of the nozzle valve pulsations.

Injector Operation

71. At each upward stroke of the pump plunger, fuel is forced via the pump delivery valve and connecting pipe, to the inlet connection of the injector. From here it passes down the offset passage in the nozzle holder, around the groove in the top face of the nozzle body, and through the series of passages in the body to the gallery above the valve seating.

72. The pressure builds up at each delivery stroke of the pump plunger until it balances the spring-loading on the nozzle valve. At each subsequent stroke the fuel pressure exceeds that of the spring, lifts the valve off its seat and ejects, through the nozzle in a fine spray, an amount of fuel equal to the volume displaced by the pump plunger; the spring then reasserts itself and closes the valve.

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