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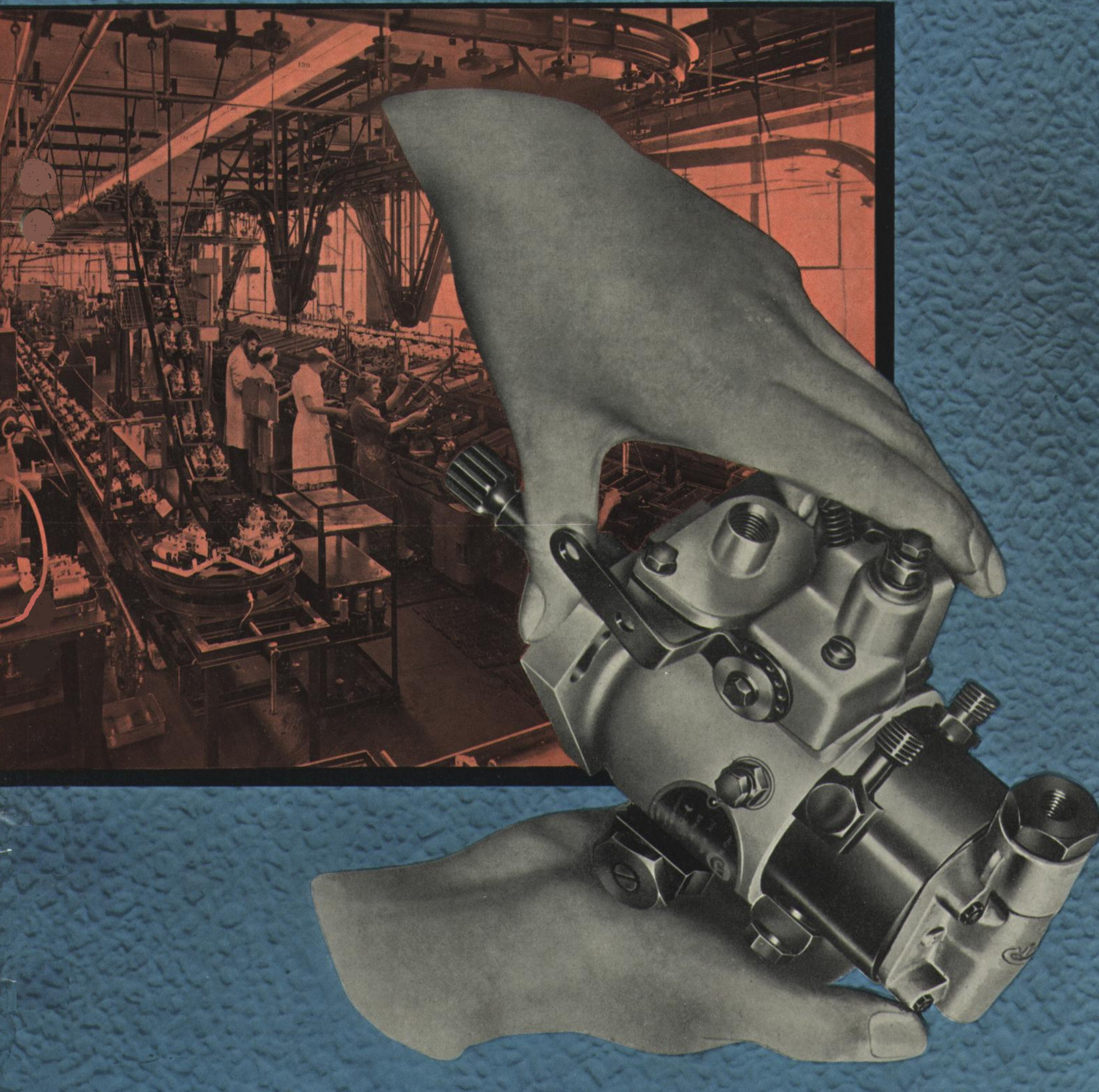
C.A.V.

FUEL INJECTION PUMP TYPE DPA



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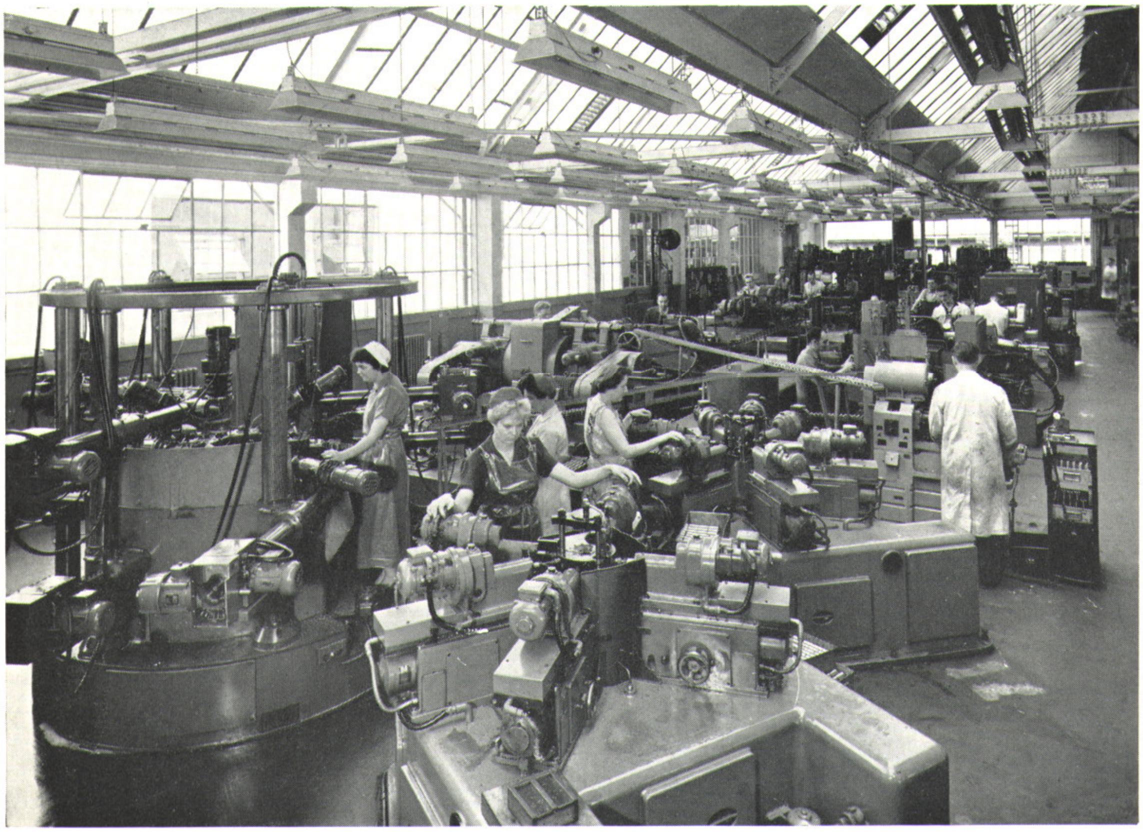
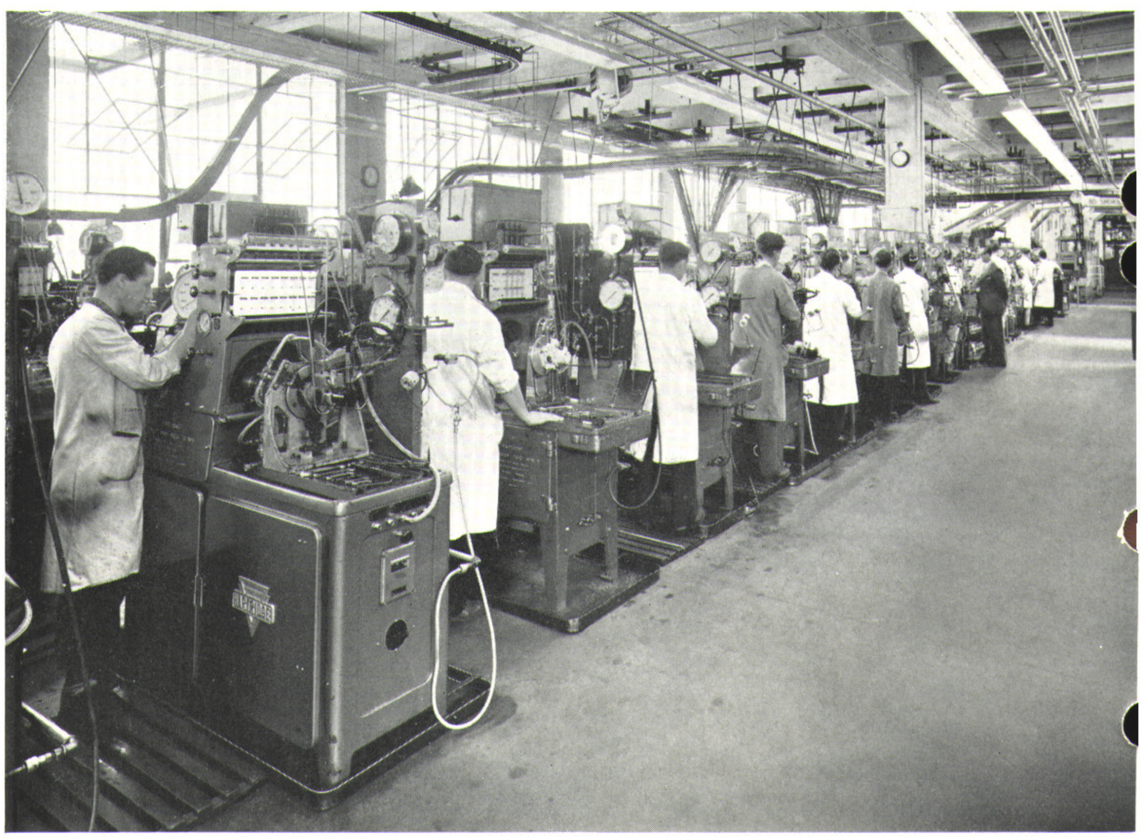


Special machine tools ensure geometrical accuracy in the drilling of ports in the 'DPA' pump hydraulic head and rotor.



Batteries of Hartridge test machines on which the 'DPA' distributor type pumps are thoroughly tested and set before despatch.

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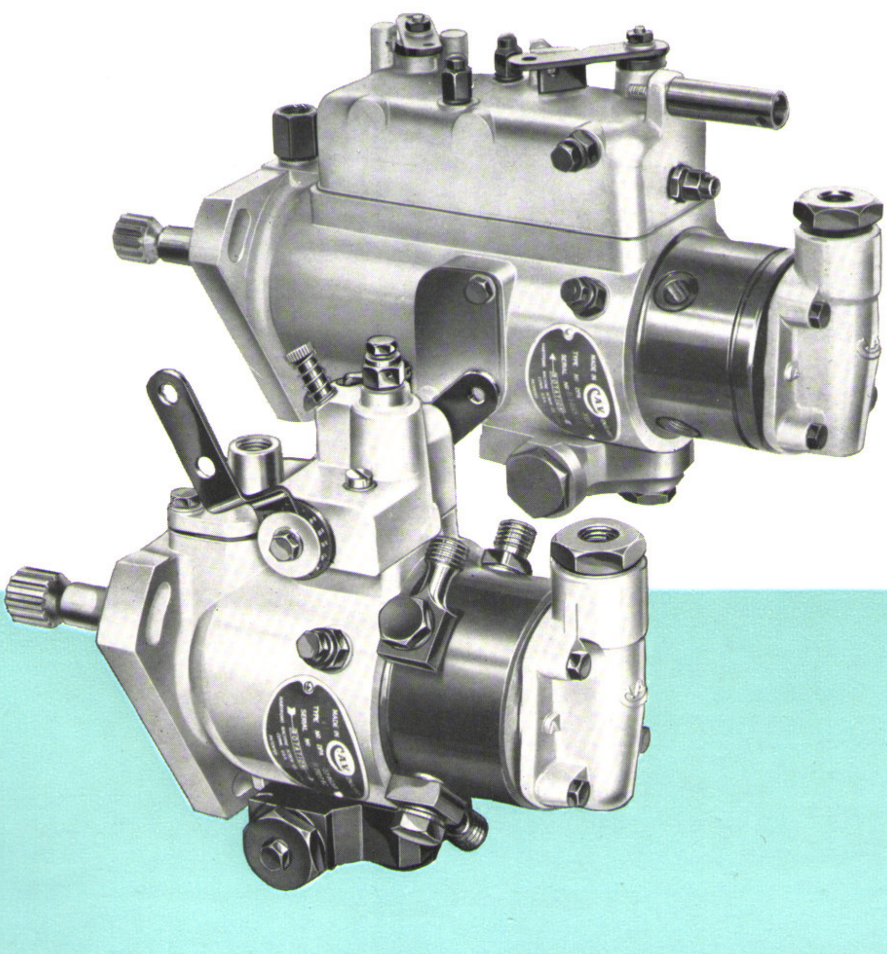
C.A.V.

'DPA' DISTRIBUTOR TYPE FUEL INJECTION PUMP



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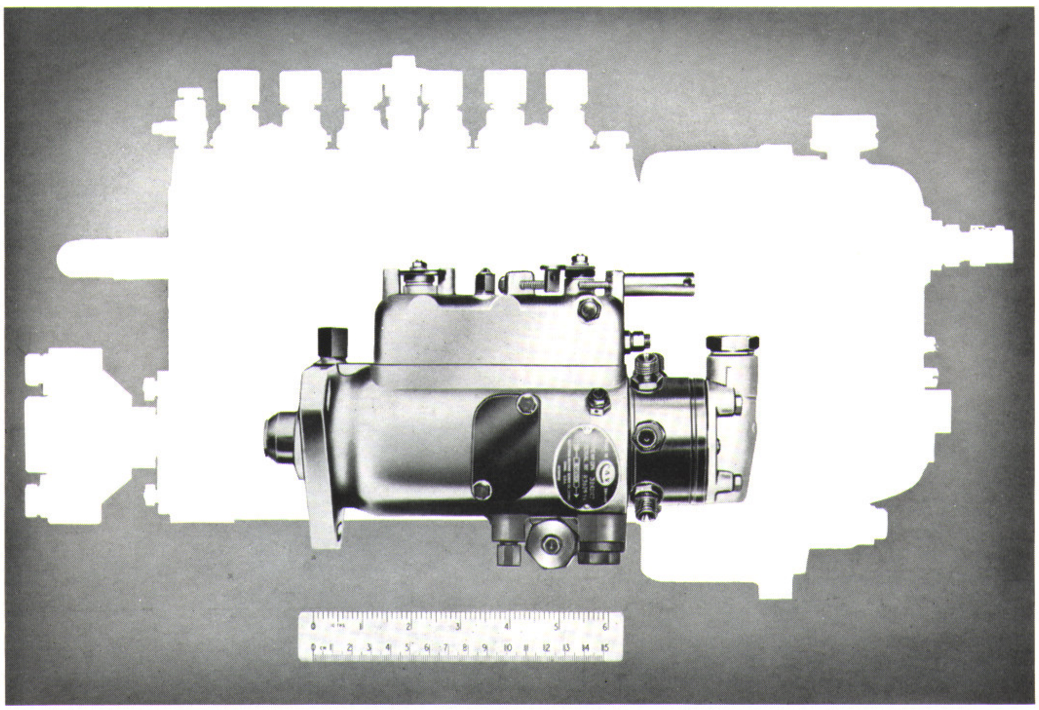
The C.A.V. DPA Distributor Type Fuel Injection Pump



Comparison of the size of the 6-cyl. 'DPA' Injection Pump with mechanical governor with that of an in-line pump for similar duty.

A brief description of its construction, its method of operation, and its advantages

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THE C.A.V. "DPA" DISTRIBUTOR TYPE FUEL INJECTION PUMP

THE chief attraction of a distributor type fuel injection pump is the opportunity offered by its simple design and construction, of reducing the cost of the equipment, and hence of the compression ignition or Diesel engine as a whole.

The desire for a substantial cost reduction is greatest and most urgent in the small engine field, where, because of various factors, the injection equipment tends to represent a higher percentage of the total engine cost than it does on larger engines.

In addition to the advantage the distributor type of pump has to offer in regard to reduced cost, however, it has also a number of technical points of advantage which considerably heighten its appeal. Apart from inherent simplicity, the design lends itself to compactness, ease of installation, convenient driving arrangements, and high speed operation.

The function of the fuel injection pump is to meter the fuel according to engine power requirements and to inject it at high pressure through nozzles into the combustion chambers of the engine at the correct timing intervals. The metering calls for careful design and construction, as it has to be carried out at a high speed and with great precision, in order to ensure even fuel distribution with smooth running, and sensitive response to power control. The timing of the injections must also be done with perfect precision, or high efficiency is impossible to achieve, and since the operating pressure may be as high as 6 000 lb/sq. in., the pump itself must be constructed with the utmost care, employing high-grade materials and the finest of working tolerances for the pump elements.

The employment of a separate pumping element for each engine cylinder, together with suitable means of output control, has been the general practice of fuel injection pump manufacture for some time. The idea of using one pump barrel and plunger to supply all cylinders in turn is a natural one, as it offers obvious savings; the pumping element operates more often (according to the number of cylinders), and is provided with a distributor or means of connecting the pump delivery to each of the injectors in turn.

The distributor type pump is thus an attractive proposition, since the number of pumping elements is reduced to one in all cases. Various designs have been produced and a great deal of development work carried out in the course of the evolution and perfection of the present model. The C.A.V. "DPA" type Distributor Pump is by far the simplest, and functionally the most satisfactory, yet devised. The design is based on one which has been used extensively in the U.S.A. on a wide variety of engines, with eminent success, and has now been developed by C.A.V. to meet the requirements of British and European engine designers. It is the result of a fresh approach to injection pump design, and incorporates many patented features. Its reliability and performance have been amply demonstrated under the most exhaustive tests.

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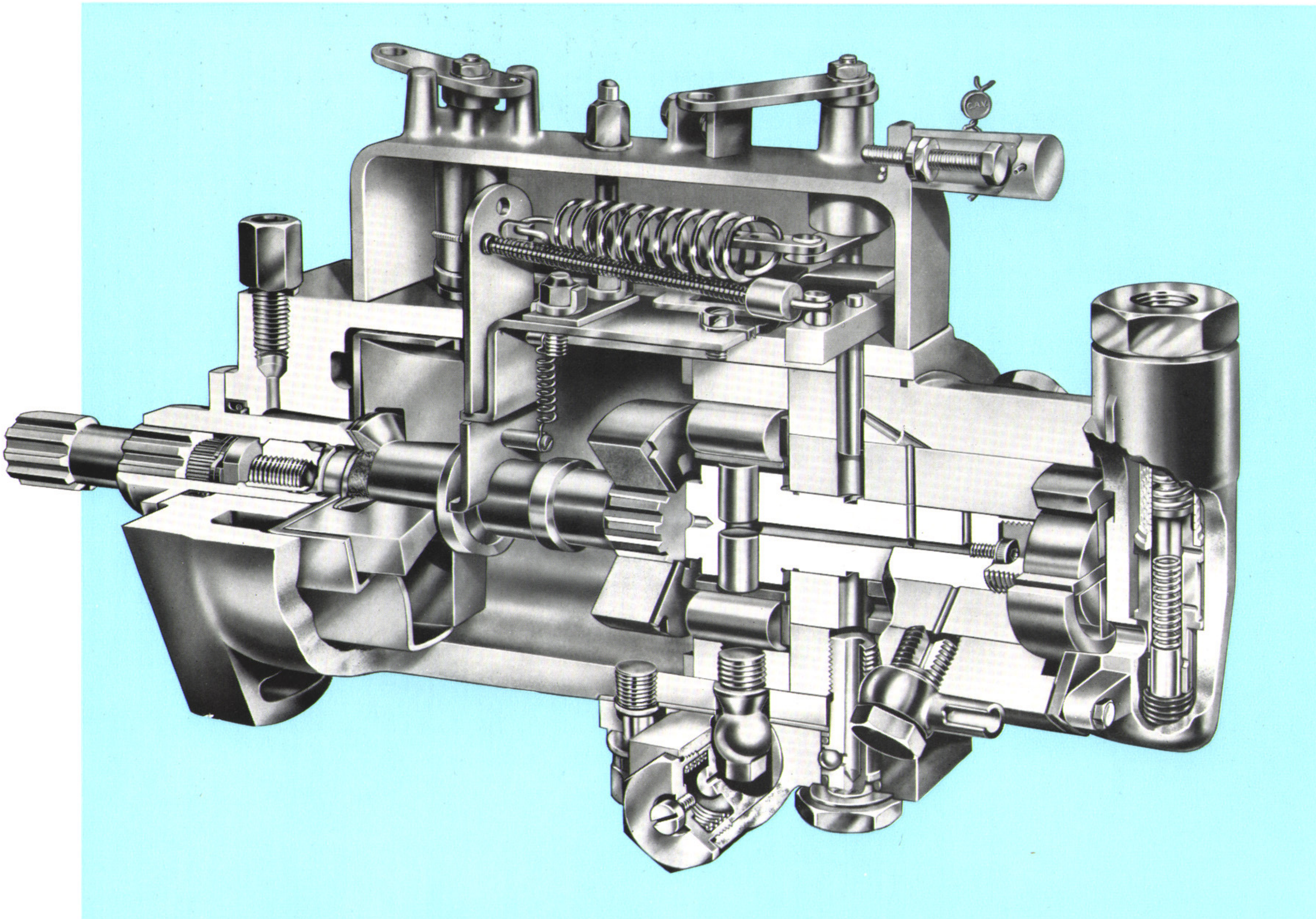

Fig. 1. *Pump with mechanical governor and automatic advance device.*

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Special Features

The C.A.V. "DPA" type Injection Pump may be briefly described as a single cylinder, opposed plunger, inlet metering, distributor type pump. It is relatively simple in design, contains no ball or roller bearings, no gears, no highly stressed springs. It contains the same number of working parts regardless of the number of the cylinders it serves.

A governor, incorporated in the pump, provides sensitive speed control at all engine speeds. This may be either the mechanical or the hydraulic type, the pumps being functionally identical in each case.

The pump, together with its governor, fuel transfer pump, pressure regulating piston, and automatic advance device (when fitted), forms an oil-tight unit in which pressure is maintained, thus preventing the ingress of dust, water or foreign matter during operation.

No special lubrication arrangements are necessary, since the pump lubricates itself with the filtered fuel oil it handles.

The problem of calibration—that is the balancing of the outputs of the several pumping lines one with another, necessary with all multi-element pumps—is non-existent with the "DPA" type pump, as equality of delivery to each injector is an inherent feature of the design.

Precise phasing, or the timing relationships between cylinders, and the functional sequences are also secured and are inherent, being determined by accurate manufacture, and cannot be maladjusted or altered during assembly of the pump. The design lends itself to the application of a simple automatic advance device, described in detail later.

The complete pump is a very compact unit, of convenient general shape for fitting to the average engine, and this, together with certain features of its driving torque characteristic, enables it to be mounted and driven on the engine in a manner that can result in notable simplification and reduction in engine cost, weight and bulk. The pump is suited for high operating speeds, such as are naturally to be expected in the lighter Diesel engines envisaged in the near future.

General Construction

The main features of the pump and the method of functioning are best seen in a part sectional illustration (Fig. 1). First, note that there are three main rotating units arranged on a common axis so that they rotate as one. These comprise a drive shaft, a pumping and distributing rotor, and a fuel transfer pump of the sliding vane type. The distributor rotor is driven by a drive shaft, splined at both ends, which couples the rotor to a drive hub located in the end of the pump housing. Master splines are provided on both ends of the drive shaft, the drive hub, and on the drive plate secured to the end of the rotor.

The rotor comprises two parts, a pumping section and a distributing section. The latter is a close rotating fit in a stationary steel cylindrical body called the hydraulic head.

The pumping section of the rotor is larger in diameter than the distributing section, and has a transverse bore or cylinder containing two opposed plain plungers; these are the pumping plungers, and their disposition in this way obviates any hydraulic side-thrust reaction on the rotor, which would be experienced with a single plunger.

The plungers are operated by means of a stationary cam ring, which is carried in the pump housing, through rollers and shoes which are carried in slots in the periphery of the rotor flange. Normally the cam ring has as many internal lobes as there are cylinders to be served, the lobes being equally spaced round the circumference.

The distributing section of the rotor contains a central axial passage which connects the pumping space between the plungers with ports which are drilled radially in the rotor and provide for fuel inlet and delivery. One radial hole in the rotor is the distributing port. As the rotor turns, the distributing port aligns successively with a number of outlet ports in the hydraulic head, from which the injectors are fed via external high pressure pipes. Only one outlet port is shown in the figure, but there are as many ports as there are engine cylinders to be served.

At an intermediate level, a number of radial holes are equally spaced round the rotor circumference; these are the inlet ports, and they are also equal in number to the number of engine cylinders. As the rotor turns, the inlet ports align successively with a single port in the hydraulic head. This is the fuel inlet or metering port, and is drilled obliquely.

The top end of the rotor carries the transfer pump, which is of the continuous discharge vane type. It is housed in the hydraulic head and is covered by an endplate which contains the inlet connection through which fuel enters the pump from

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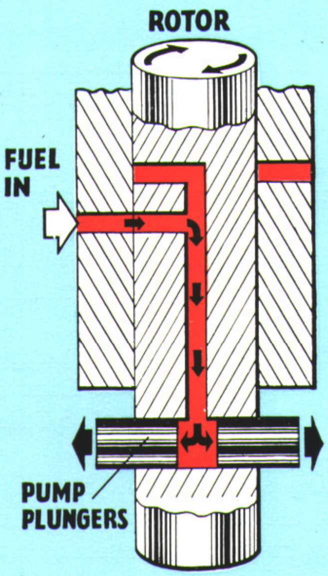

Fig. 2
Charging stroke.

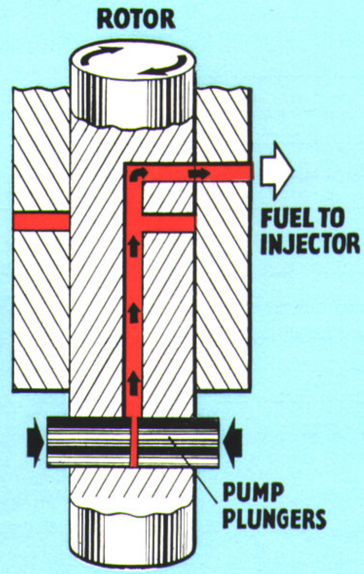

Fig. 3
Pumping stroke.

the main supply, and a by-pass valve which controls the transfer pump pressure.

Fuel entering the injection pump flows first to the transfer pump, which raises its pressure to an intermediate level, and then flows down a passage in the hydraulic head, round an annular groove in the rotor to a radial bore in the head which connects with the oblique metering port. The entrance to the metering port constitutes the metering orifice, and this is variable for control purposes by means of a metering valve which is a close sliding fit in the radial bore.

Principle of Operation

Before considering the operation of the pump in detail, the working of the main parts may be followed by reference to simple diagrams.

The action of the pump plungers is shown in Figs. 2 and 3. As the rotor turns, one of the inlet ports in the rotor opens to the metering port in the head, and fuel at metering pressure enters the rotor and separates the plungers (Fig. 2). (Note that there are no springs forcing the plungers out.) As the rotor continues to turn, the inlet port closes off again, and shortly afterwards the distributor port aligns with one of the fuel outlets. Both rollers then contact opposing cam lobes and the plungers are forced in towards each other (as in Fig. 3). This is the injection stroke, and the fuel trapped between the plungers is forced back through the rotor and out to the injector.

Fuel displacement ceases when the plungers reach the limit of inward travel imparted by the cam lobes, and shortly afterwards the distributor port closes, sealing off the fuel pipe to the injector.

As the rotation of the rotor continues, the cycle just described repeats itself, the pump discharging through each of the outlet connections in turn.

The functioning of the distributor rotor is also shown in the perspective diagrams, Figs. 4 and 5. In Fig. 4 the rotor is in the "charging" or inlet position. The rollers have reached their most outward position, being off the cam lobes, allowing the plungers to move as far apart as they can under the pressure of fuel from the transfer pump. The charging or metering port is shown in register with one of the inlet ports in the rotor, whilst the distributor port is well out of register with the discharge port in the head.

With further rotation of the rotor the relationships change, until the "discharge" or injection position is reached, as seen in Fig. 5. Here the rollers have reached the slopes of the opposite cam lobes, forcing the plungers together and discharging the fuel. Note that the distributor port in the rotor is in register with one of the fuel outlets, while the inlet ports in the rotor are out of register with the metering port.

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The Pressure Regulating Valve

A valve situated within the pump end plate performs two separate functions. Firstly, it regulates transfer pressure, maintaining the desired relationship between transfer pressure and the speed of pump rotation. Secondly, it provides means of by-passing the transfer pump, so that the fuel passages in the hydraulic head can be primed when the pump is stationary.

The end plate illustrated in Fig. 6 is cut away to show the working parts of the regulating valve, the nylon filter and the fuel inlet connection.

Fig. 7 shows the regulating valve in three different positions:—

(*a*) AT REST. The first diagram in Fig. 7 shows the valve in the static position. Since there is no fuel pressure within the end plate neither spring is compressed. The parts are annotated and can be identified by reference to the key.

(*b*) HAND PRIMING. For hand priming, the pressure caused by the hand primer across the transfer pump forces the piston valve down, compressing the spring and uncovering the priming port so that fuel by-passes the stationary transfer pump to fill the system via the passages in the hydraulic head.

(*c*) OPERATING. When the injection pump is in operation, the pressure of fuel from the transfer pump forces back the piston valve in the sleeve, uncovering the regulating port. The pressure on the piston is opposed by the regulating spring, and a position of balance is reached, the delivery pressure of the transfer pump being controlled by the rate of the spring used.

Transfer Pump

The transfer pump is of the positive type, with two vanes sliding inside an eccentric liner in the hydraulic head. The transfer pump rotor is carried in the end of the distributor rotor. The capacity of the transfer pump is considerably in excess of the requirements of the injection pump.

Metering Valve

A metering valve situated in the hydraulic head regulates the volume of fuel entering the rotor, under the control of the governor or of the hand throttle. The type of valve varies according to the type of governor fitted; with the hydraulic governor, a piston-like valve is used, and this is the type shown

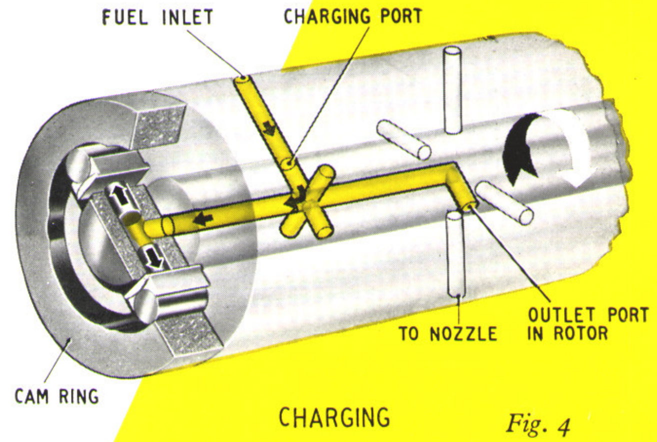

Fig. 4

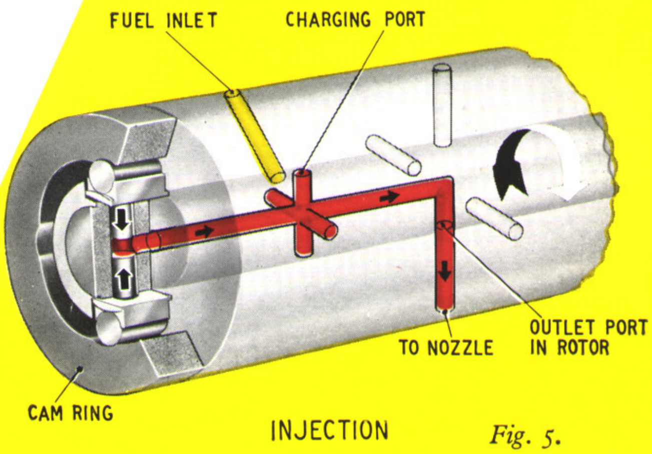

Fig. 5.

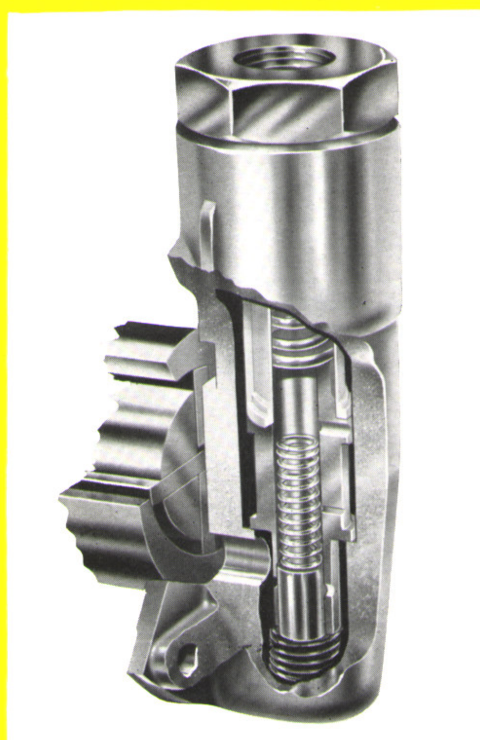

Fig. 6
Cut-away view of aluminium end plate showing regulating valve.

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in Fig. 11. The valve is spring loaded and controls the fuel according to its axial position. When a flyweight type governor is fitted, the valve is of a rotary type, with a slot cut in its periphery; the valve is rotated by the governor arm, to regulate the fuel entry.

Lubrication

Reference has already been made to the fact that the pump is self-lubricated. Oil reaches the annular groove of the rotor shank at the transfer pump pressure, and from there a slot allows oil to bleed to a part of the rotor of reduced diameter. From there, oil floods the cavity of the housing in which the cam rollers operate. An oil drain connection is provided in the pump housing to allow fuel to flow back to the main supply. This return line also serves to permit any air which may be in the fuel or originally contained in the pump to be carried out with the fuel.

Fuel Metering

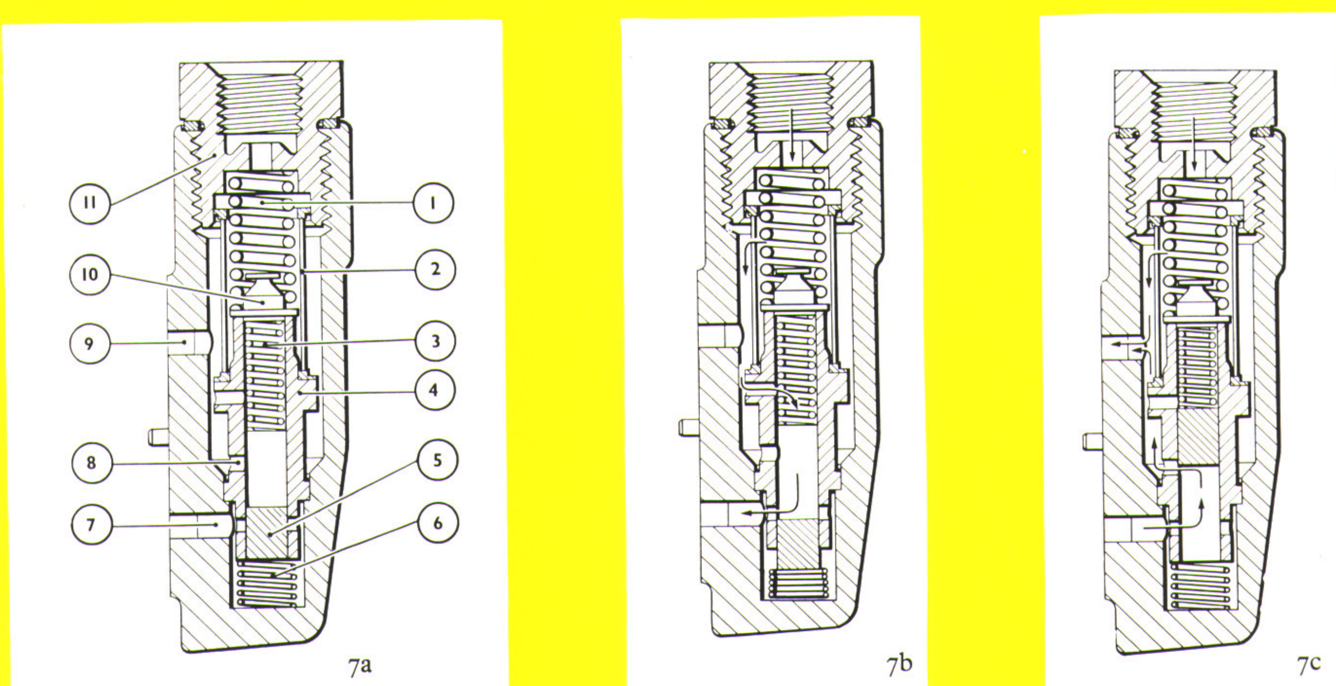
Apart from small losses which occur during the injection stroke, the total volume of fuel introduced into the element is passed to the injector. Metering is effected, therefore, by regulating the volume of fuel which enters the element at each charging stroke. The volume of the charge is governed by two principal factors: the fuel pressure at the inlet port, and the time available for fuel to flow into the element while the inlet ports in the rotor and the hydraulic head are in register. It is by controlling the pressure at the inlet port that accurate metering is achieved.

Fuel oil enters the fuel injection pump at feed pressure (green, Fig. 8) and passes into the transfer pump, which raises the pressure to a level known as transfer pressure (orange, Fig. 8).

Transfer pressure is related to engine speed, and rises as the speed of rotation is increased. A predetermined relationship between transfer pressure and the speed of rotation is maintained by a regulating valve situated in the end plate of the pump.

Fuel at transfer pressure passes through passages in the hydraulic head to the metering valve which controls the flow of fuel through the metering port. The effective area of the metering port is controlled by moving the metering valve, which is connected by suitable control linkage to the accelerator pedal and to the governor.

A pressure drop occurs as fuel passes through the metering orifice, reducing the fuel pressure to a level known as metering pressure (yellow, Fig. 8).



1. Retaining Spring. *4. Valve Sleeve.* *7. Fuel passage to Transfer Pump Outlet.* *10. Spring Guide.*
2. Nylon Filter. *5. Piston.* *8. Regulating Port.* *11. Fuel Inlet Connection.*
3. Regulating Spring. *6. Priming Spring.* *9. Fuel passage to Transfer Pump Inlet.*

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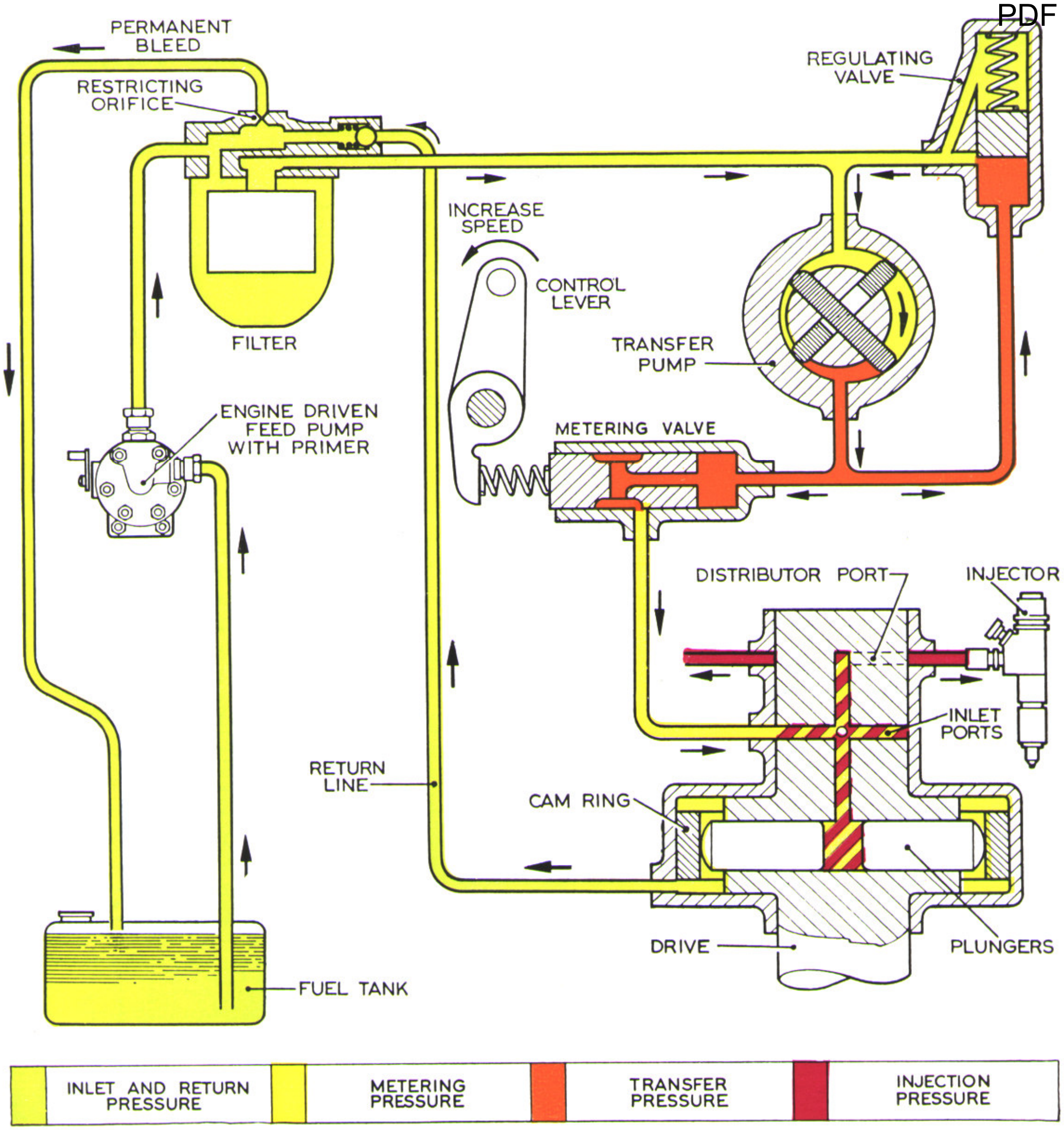

Fig. 8
Fuel Flow Diagram.

The smaller the metering orifice the greater will be the decrease in pressure and vice versa.

Fuel at metering pressure passes to the inlet port through an obliquely drilled passage in the hydraulic head.

At idling speeds both transfer pressure and metering pressure are at their minimum value. Depression of the accelerator moves the metering valve to a position where the effective area of the metering port is increased. This brings about an increase in metering pressure and consequent increase in the quantity of fuel entering the pumping element at each charging stroke. The engine will then accelerate in response to increased fuelling until a speed corresponding to the position of the accelerator pedal is attained.

If the pedal is then released, the effective area of the metering orifice is reduced, and engine speed will fall as the result of decreased fuelling.

When an engine is running at a fixed speed setting the governor controls the position of the metering valve, and maintains the selected speed within close limits by causing compensating changes of fuelling.

Maximum Fuel Adjustment

Maximum fuel settings are made by limiting the maximum outward travel of the pumping plungers at a point where the desired fuelling is obtained.

The cam rollers (A, Fig. 9) are carried in shoes (B) against which the pumping plungers (D) are pressed by the incoming fuel. The shoes slide in guides

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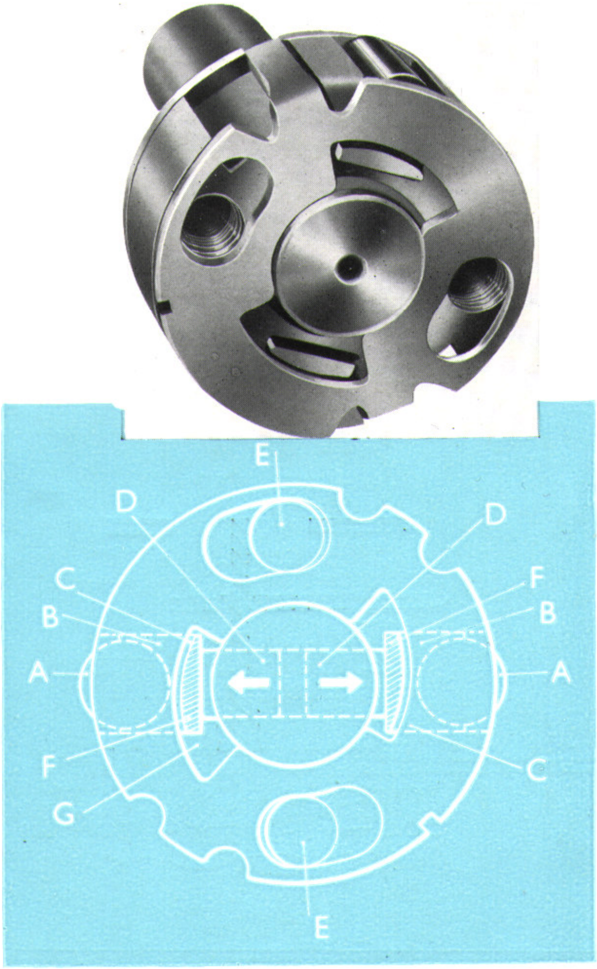

Fig. 9. Maximum Fuel Adjustment.

A. *Cam rollers*
B. *Roller shoes*
C. *Spaces between ears of roller shoes and adjusting plate slots*
D. *Pump plungers*
E. *Drive plate locking screws*
F. *Ears on roller shoes*
G. *Eccentric slots in adjusting plate*

machined in the rotor, and are retained in an endwise sense by top and bottom adjusting plates. "Ears" (F), integral with the roller shoes, project into eccentrically cut slots (G) in both adjusting plates. The "ears" are shaped to match the contour of the slots.

The bottom adjusting plate is rigidly clamped between the drive plate and the end of the rotor. The screws which secure the drive plates pass through elongated holes (E) in the adjusting plate so that the adjusting plates can be moved when the screws are slackened. Top and bottom plates are located one to the other by lugs integral with the top plate.

The plungers reach their outward limit of travel when the "ears" on the roller shoes contact the curved sides of the slots in the adjusting plates.

In Fig. 9 the adjusting plates are shown in the position which would provide the lowest possible maximum fuel setting. Since the slots in the adjusting plates are eccentric, maximum plunger travel will be increased if the adjusting plates are moved in a clockwise direction in relation to the rotor. The movement provided permits a wide range of maximum fuel settings.

Clearance "C" indicates the remaining travel to bring the plungers to the maximum fuel position.

Satisfactory adjustment of the maximum fuel setting can be made only while the pump is on the test bench, and should not be altered except on the recommendation of the engine manufacturer.

GOVERNING

Two basic models of the pump are manufactured, one incorporating a flyweight governor (Fig. 1), and the other, a simple hydraulic governor (Fig. 11).

Apart from the method of governing the two models do not differ in their principle of operation. The fundamental working parts—hydraulic head, rotor, drive plate, adjusting plates, end plate and regulating valve—are identical in both pumps.

The two pumps differ in appearance, the mechanically governed pump being somewhat longer, since the governor weight assembly is situated within the pump housing. The splined drive shaft, drive hub and quill shaft of the mechanically governed pump are replaced by a single splined drive shaft in the hydraulically governed pump. The metering valves differ as previously mentioned.

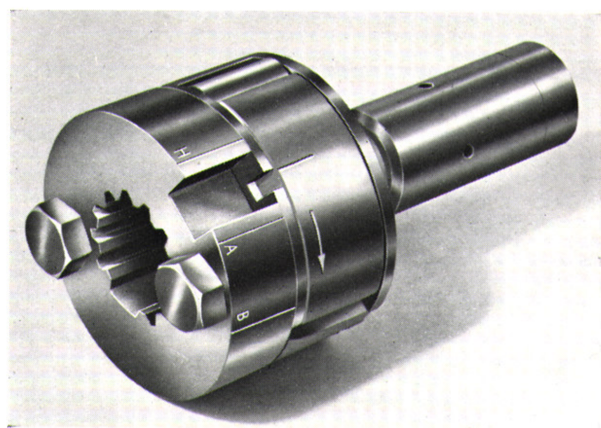

Fig. 10
Drive plate assembled to rotor, showing adjustment slots and pump timing marks.

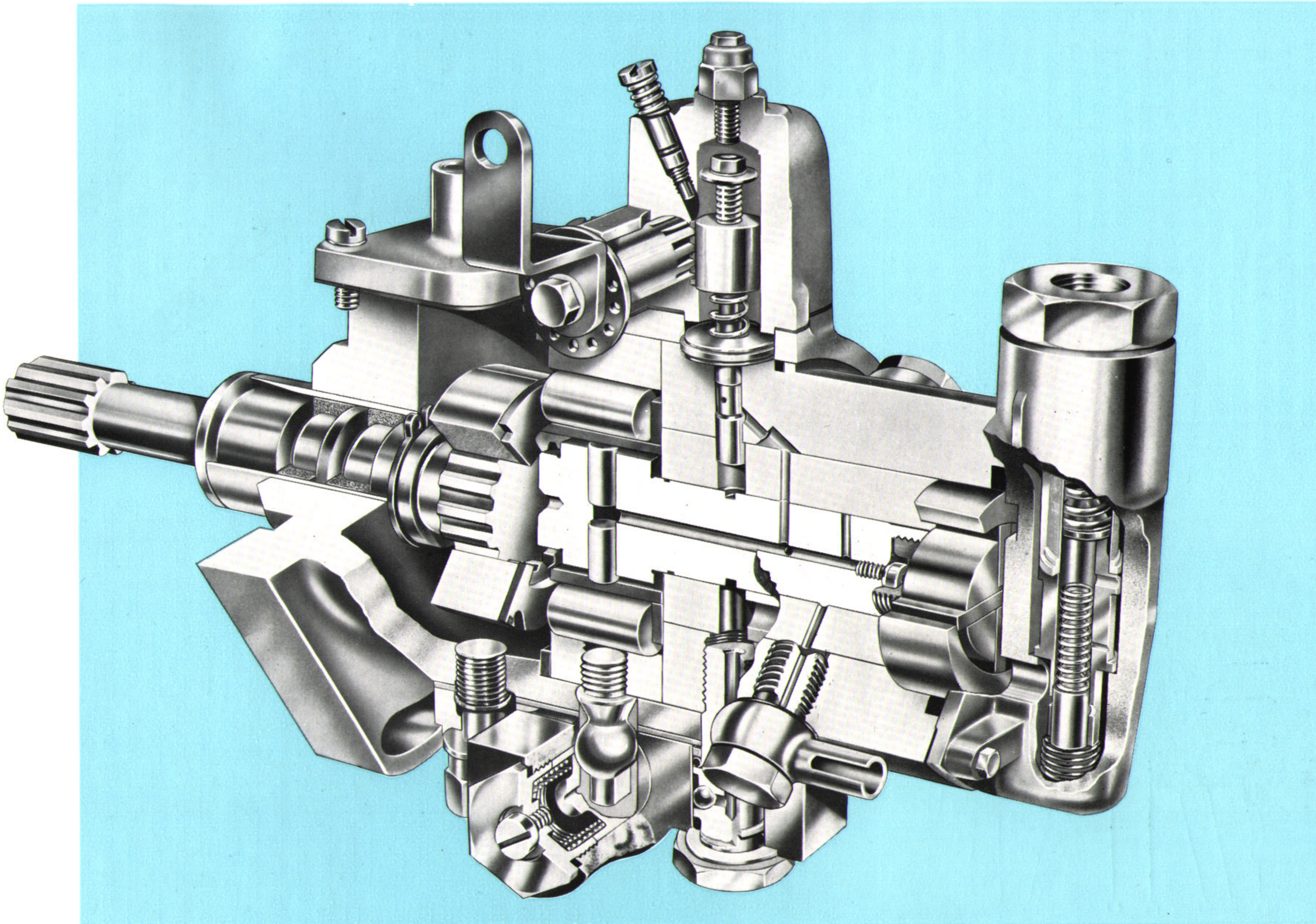

Fig. 11.
Pump with Hydraulic Governor and Automatic Advance Device.

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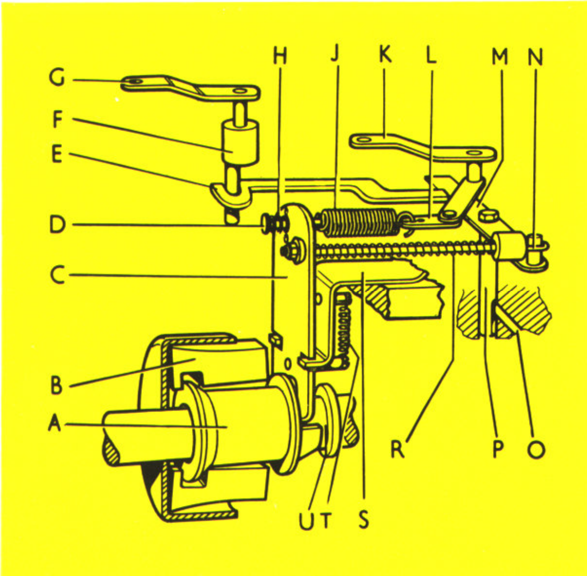

Fig. 12.
Mechanical Governor Linkage.

A. Thrust sleeve
B. Governor weights
C. Governor lever
D. Idling spring guide
E. Shut-off bar
F. Shut-off shaft
G. Shut-off lever
H. Idling spring
J. Governor spring
K. Control lever
L. Connecting link
M. Metering valve lever
N. Spring loaded hook lever
O. Metering port
P. Metering valve
R. Hook lever spring
S. Pivot plate
T. Retaining spring
U. Splined drive shaft

The Mechanical Governor

The DPA distributor type pump incorporating a mechanical governor is shown in Fig. 1. It will be noted that the flyweight assembly is carried on the splined drive shaft within the pump housing, and that the governor control linkage is enclosed by a cover fitted on the upper face of the pump housing.

Governor operation is best understood if reference is made to Fig. 12 which shows the governor and the control linkage diagrammatically.

The governor weights are housed in pockets in the weight carrier, which is rigidly clamped between the end of the drive hub and a step on the drive shaft. Drive hub, governor weight assembly and drive shaft thus rotate as a single unit when the pump is operating. The weights are so shaped that they pivot about one edge under influence of centrifugal force. Such movement causes a thrust sleeve, with which they are engaged, to slide along the drive shaft.

Movement of the thrust sleeve is transmitted to the metering valve by the pivoted governor arm and the spring-loaded hook lever.

Outward movement of the weights tends to close the metering orifice by rotating the metering valve, thus reducing the quantity of fuel reaching the engine cylinders at each injection.

The governor arm is spring-loaded by the governor spring; the tension of this spring acts in opposition to centrifugal force, tending to oppose outward movement of the weights and to hold the metering valve in the maximum fuel position. Spring tension can be varied by moving the control lever to which the spring is connected, tension being increased as the "throttle" control is moved towards the maximum speed setting.

When the control lever is moved to a position which calls for engine acceleration, increased spring tension is applied to the governor arm. This increased spring tension overcomes centrifugal force acting on the governor weights and the metering valve is rotated to the maximum fuel position. Engine speed builds up until centrifugal force acting on the weights is sufficient to overcome the increased spring tension. The weights then move outwards and reduce the fuelling by rotating the metering valve until the two opposed forces acting on the thrust sleeve are in equilibrium.

While running at a selected speed, spring tension opposing movement of the governor weights remains constant. When speed fluctuations occur the resulting change of centrifugal force causes movement of the weights and brings about a compensating change of fuelling—increased fuelling when engine speed falls below the selected speed, and decreased fuelling when it is exceeded.

It will be noted that the governor spring is not connected directly to the governor arm. It is coupled to the idling spring guide which passes through a hole in the governor arm and is spring-loaded by the light idling spring. At speeds outside of the idling range the tension applied to the governor spring is sufficient to compress the idling spring completely, thus rendering it ineffective. At idling speeds, when the tension of the governor spring is reduced to a minimum, the idling spring comes into action. Movement of the control arm by governor action is opposed by the light idling spring only, so that stable idling can be maintained when changes of centrifugal force are small.

Idling and maximum speed adjustment screws are provided on both hydraulically and mechanically governed pumps. Maximum speed and idling stops are adjusted while the pump is on the engine and should be set in accordance with the engine manufacturer's instructions.

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Fuel "shut off" is achieved by rotating the metering valve to a position where the metering orifice is completely closed. An eccentric on the "shut off" shaft engages a bar which is brought into contact with the arm on the metering valve when the control is operated. The hook lever connecting the metering valve to the governor arm is spring loaded so that the control can be operated at any engine speed without need for over-riding the governor.

The Hydraulic Governor

A DPA type pump incorporating a hydraulic governor is shown in Fig. 11.

The components of the hydraulic governor are contained in a small casting secured to the upper face of the pump housing, a speed control lever and a "shut off" lever being mounted externally.

Within the governor housing is a pinion which engages a rack carried on the stem of the metering valve. The rack is not secured to the valve stem but is located between two springs—the governor spring, and a light idling spring (see Fig. 13).

Movement of the piston type metering valve controls the effective area of the metering orifice, thus controlling the metering pressure at the inlet port in the rotor and regulating the quantity of fuel admitted to the pumping element at each charging stroke.

Two forces act upon the metering valve—spring pressure exerted by the governor spring and transfer pressure acting on the underside of the valve. The two forces are opposed, spring pressure tending to hold the valve in the maximum fuel position and transfer pressure tending to hold it in the idling position. During operation the valve assumes a position where the two forces acting upon it are in equilibrium.

When the control lever is moved towards the "full throttle" setting the metering valve moves to the maximum fuel position and the governor spring is compressed. Engine speed increases in response to increased fuelling, thus causing a rise in transfer pressure. When transfer pressure becomes sufficiently high to overcome spring pressure, the metering valve moves towards the idling position until the two forces are in equilibrium. The engine will then run at a speed corresponding to the setting of the speed control lever, and the selected speed will be maintained by governor action.

Any change in engine speed resulting from a change of loading is accompanied by a corresponding change of transfer pressure. This will cause movement of the metering valve and bring about a compensating change of fuelling; increased fuelling when engine speed falls below the selected speed and decreased fuelling when the selected speed is exceeded.

Within the idling range, the idling spring is compressed. The force exerted by it on the metering valve is in the same direction as the force exerted by transfer pressure. Thus, at idling speeds a reduced pressure exerted by the governor spring is opposed by transfer pressure plus the force exerted by the idling spring. This enables close governing to be maintained at low speeds where transfer pressure changes are relatively small.

The dished damper plate is immersed in fuel oil, and prevents violent metering valve movement by dashpot action.

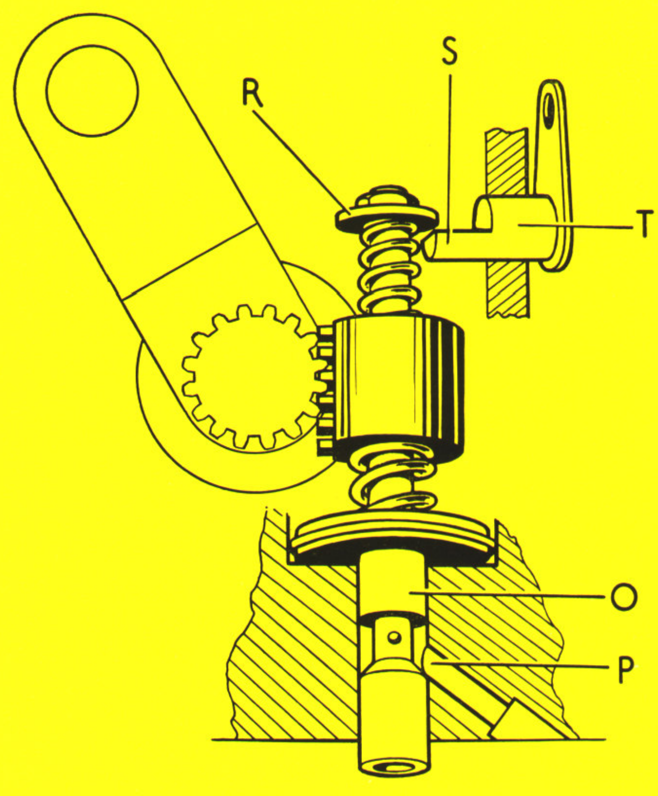

Fig. 13.
Hydraulic Governor Mechanism.

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A cam controlled by the "shut off" lever contacts the underside of a washer fitted at the upper end of the delivery valve stem. When the control is operated, the valve is lifted upwards and the metering orifice completely closed. The control can be operated at any engine speed thus enabling the operator to "shut off" the engine in an emergency.

AUTOMATIC CONTROL OF INJECTION TIMING

Inherent Timing Changes

Under full load conditions, the maximum amount of fuel is introduced into the element, and the plungers and actuating rollers are forced outwards to the limit of their travel. As the rotor turns, the rollers are brought into contact with cam lobes on the cam ring. The point of contact is near the base of the cams.

Under lightly loaded conditions, fuelling is decreased and the plunger travel is proportionally reduced. Contact between roller and cam is now made at a point near the cam peak.

It follows that injection timing becomes progressively retarded as fuelling is reduced, since contact between the actuating rollers and the cams occurs progressively later in the cycle.

Control of Injection Timing

The automatic speed advance mechanism illustrated in Fig. 14, is typical of several devices both manual and automatic, which are available to control the timing of the pump with respect to engine starting characteristics and loading requirements.

The piston (B) is free to slide in a cylinder machined in the body of the device (E). Movement of this piston is transmitted to the cam ring (C) by the ball ended cam screw (A), causing the cam ring to rotate within the pump housing (D).

Pressure exerted on the piston by the springs, tends to hold the piston and the cam ring in the fully retarded position.

Fuel oil at transfer pressure enters the device through a fuel passage in the screw which secures the device to the pump housing. Transfer pressure acts upon the piston and tends to move the cam ring against the spring pressure.

Transfer pressure increases progressively as the engine speed is raised, and the piston is moved along the cylinder to compress the springs and move the cam ring towards the fully advanced position. When engine speed is decreased, transfer pressure falls, and the piston and cam ring are moved towards the retarded position by spring pressure.

The impact of the actuating rollers on the cam lobes at commencement of injection tends to move the cam ring towards the retarded position. Such movement is prevented by a non-return valve situated in the fuel passage in the screw securing the device to the pump housing. Normal leakage between the piston and the cylinder permits the device to return to the retarded position when the engine speed falls.

Any desired timing advance up to a maximum of 12° (pump) is obtainable, and the engine speed at which this is attained can be varied by fitting stronger or weaker springs.

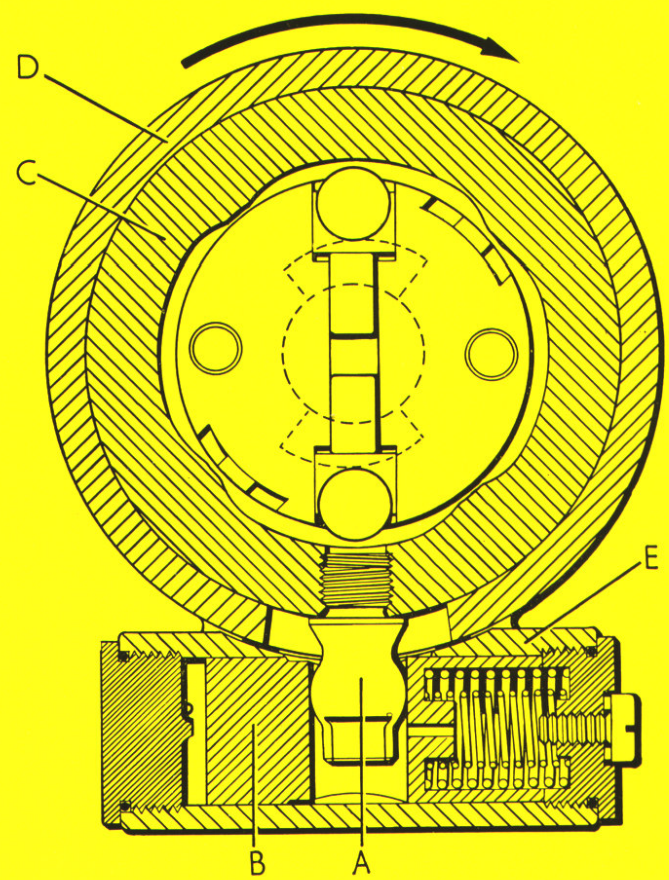

Fig. 14.
Automatic Speed Advance Device.

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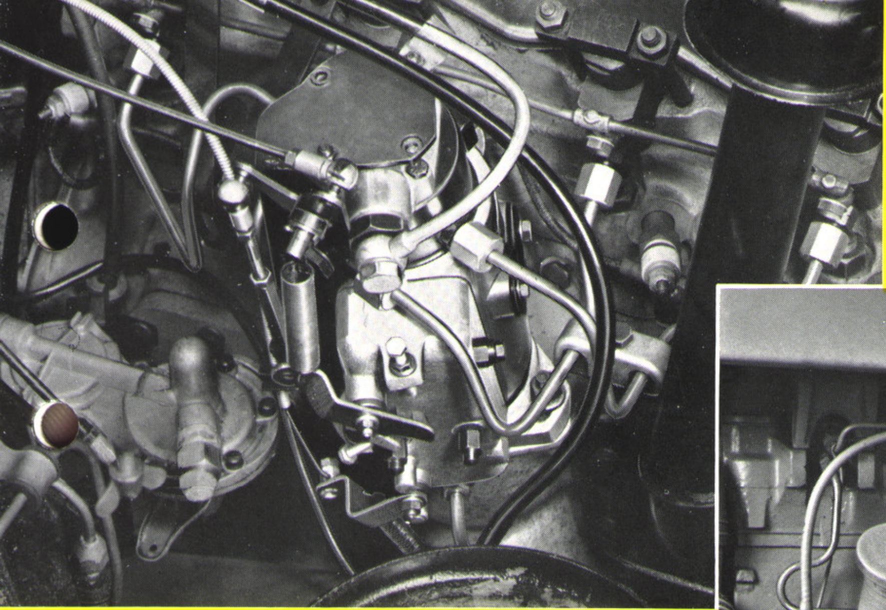

Fig. 15.
DPA Pump installed vertically.

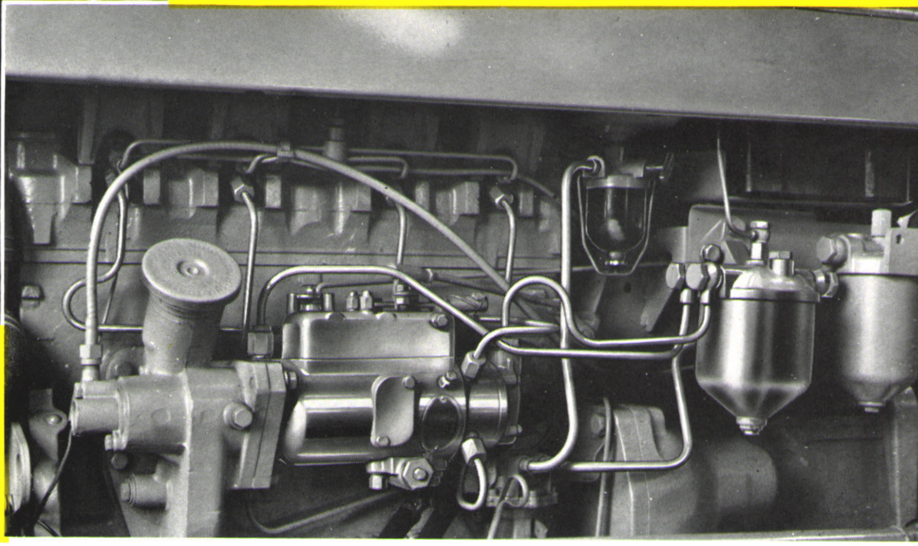

Fig. 16.
DPA Pump installed horizontally.

Some Attendant Advantages

The speed capability of the "DPA" pump is not limited by inertia and return spring loadings. The reciprocating masses are small, and hydraulic return of the plungers eliminates any spring problem. The pump has been tested up to 4 200 engine r.p.m. so far, but higher speeds are envisaged without any foreseeable difficulty.

The driving torque of the pump is always positive, owing to the absence of plunger springs and because of the continuous torque of the transfer pump. There is therefore a reduced tendency for torsional vibrations, and the pump can be driven, if required, direct from the engine camshaft. Such an installation is facilitated by the compactness and shape of the pump, and results in an important simplification of the engine and reduction of cost. Since the pump is completely filled with fuel oil during operation it can be mounted horizontally, vertically or in any intermediate attitude which might suit a particular engine installation. Pumps mounted vertically and horizontally are shown in Figs. 15 and 16 respectively.

The pump cannot inject if rotated backwards, since the transfer pressure immediately fails, charging becomes impossible and the port phasing becomes reversed relative to the cam. This feature obviates the need for any device to prevent the possibility of an engine being started running in reverse.

The pump operates under positive fuel pressure, a valuable feature in preventing cavitation in the pumping chamber during charging, contributing much towards obtaining regular injection and close calibration.

Unloading of the fuel lines is achieved by the provision of suitable retraction periods on the cam lobes, and no further provision is required for the prevention of "dribbling" of the nozzles.

The pump design allows a range of plunger sizes from 4·5 to 10 mm diameter in 0·5 mm steps; the maximum plunger stroke is approximately 2·2 mm. The maximum fuel output recommended is approximately 100 mm^3 per stroke for the present, though engines requiring considerably greater deliveries have been operating successfully.

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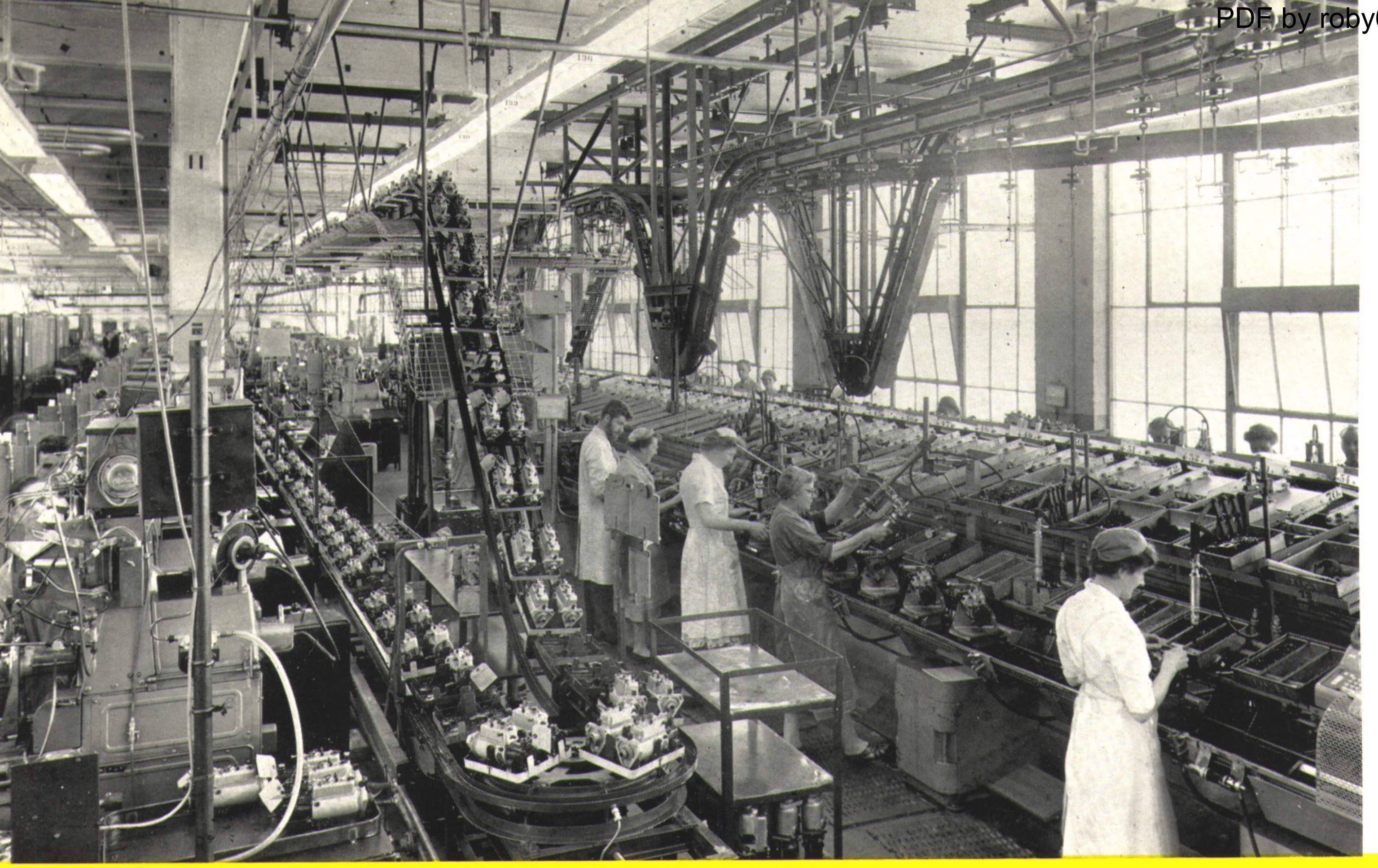
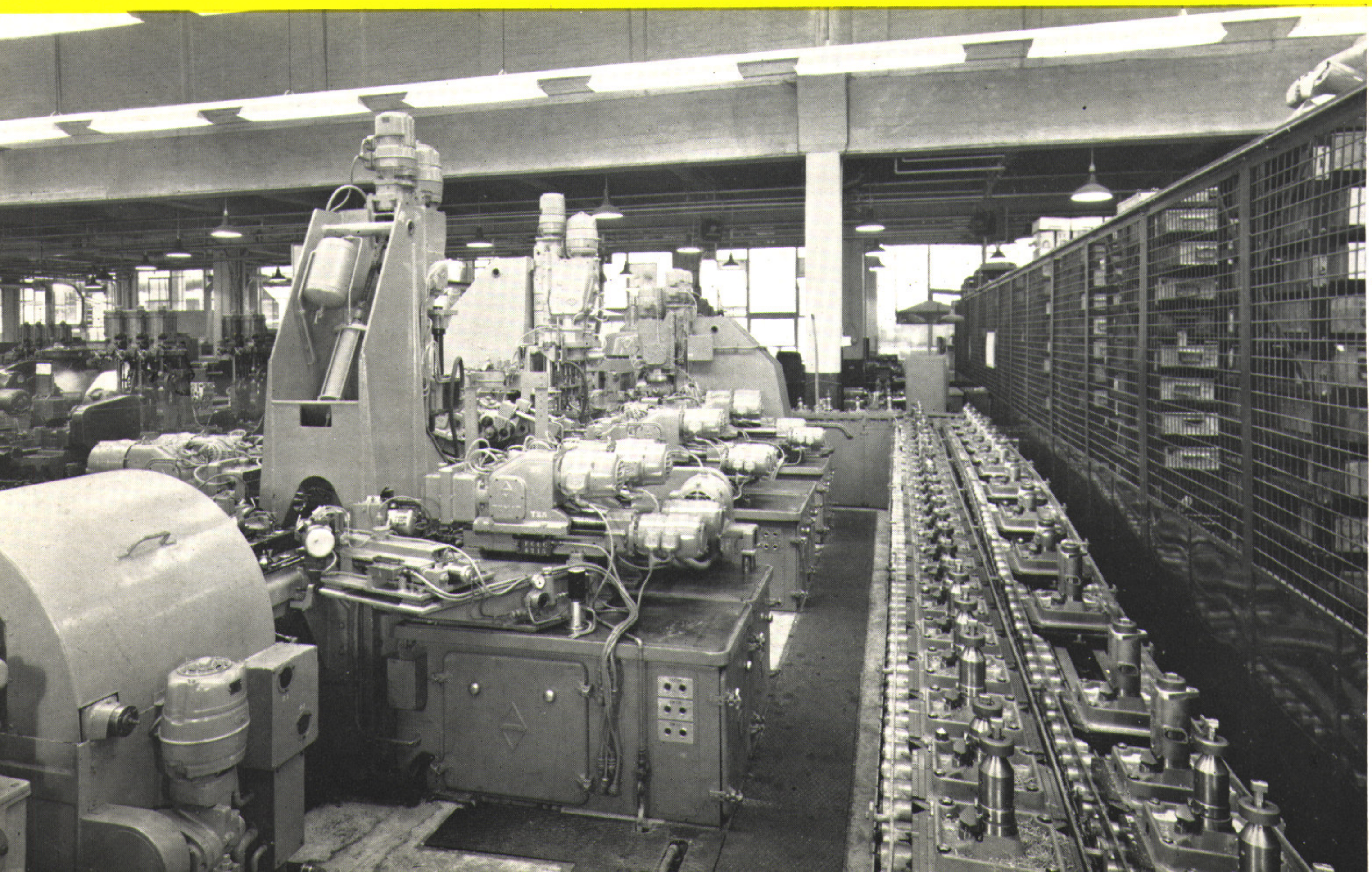


One of the assembly lines with conveyors, for DPA fuel injection pumps

Line of machines for complete machining of DPA injection pump housings



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