3

Mechanical Injection

In developing the diesel engine for higher speed and lighter weight, it became necessary to discard air injection with its bulky and power consuming compressor, and to replace it with mechanical injection. In this latter system the fuel is forced through a spray nozzle and into the combustion chamber by hydraulic pressures of 2000 psi or more.

Diesel Injection Systems

Three general systems of mechanical fuel injection have been developed: the constant-pressure or common rail, the spring pressure or accumulator type, and the jerk pump. The latter type is the most popular.

COMMON RAIL

In this system the fuel is maintained at constant pressure in a manifold connected to cam actuated nozzles, or with a timing and distributor valve and pressure operated nozzles. Substantially constant injection pressure of 4000 to 8000psi are obtained by: (1) making the fuel manifold large and utilizing the compressibility of the fuel oil , (2) using a pump of excess capacity and delivering fuel between each injection, and (3) by-passing the excess fuel from the accumulator through a manually or governor controlled pressure regulating valve. The fuel quantity discharged per injection depends upon the injection pressure, total nozzle orifice area, and time that the nozzle valve is lifted.

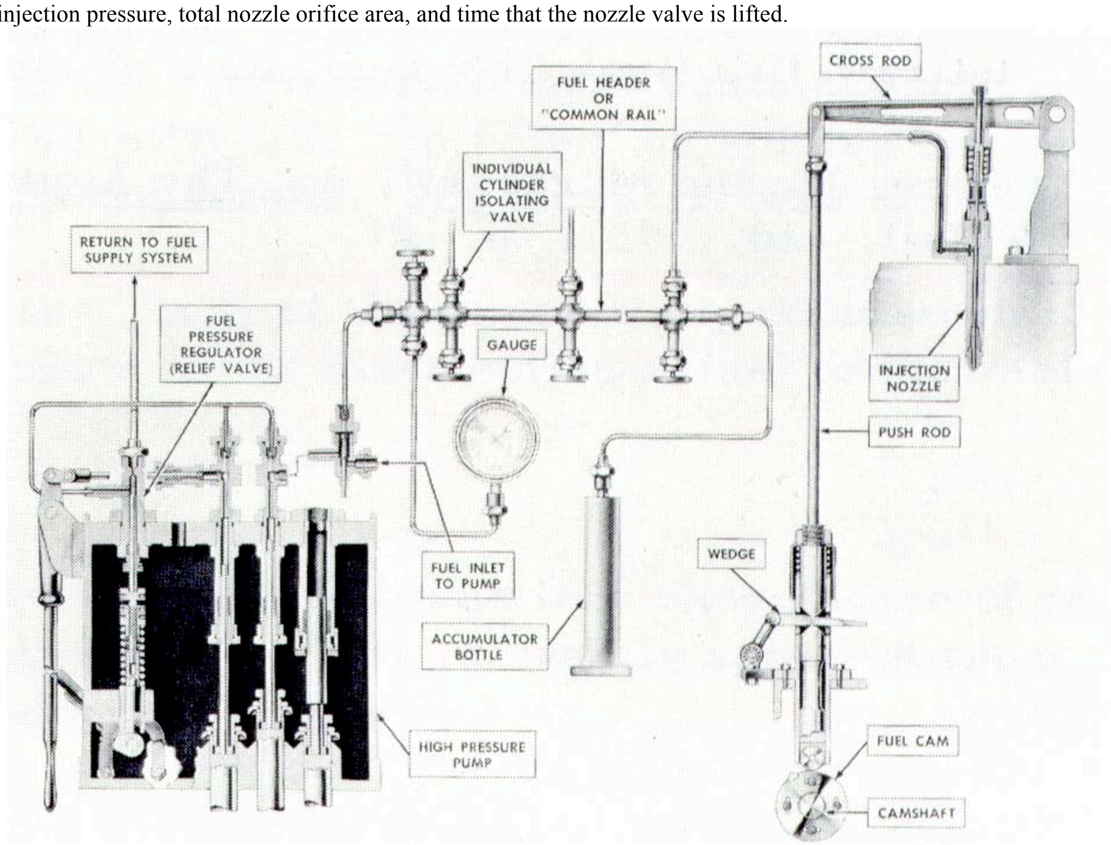


Fig. 22. Common rail system (Atlas-Imperial)

Cam Actuated Nozzles[1]

The conventional common rail system, as shown in Fig. 22, comprises an untimed, multiple plunger, high pressure pump which delivers fuel to a header and accumulator, a spring loaded relief type pressure regulator, and mechanically operated nozzles connected by branch tubings from the header. The spring loaded nozzle valves are lifted mechanically by push rods and levers actuated by timed cams. Short injection durations are obtained by small triangular projections on the cam lobes, and further control of durations at part loads is effected by governor positioned wedges varying the clearances between the cam followers and push rods. Thus, the beginning and ending of injection varies with the spray duration or load. At low loads and idling the injection pressure is generally reduced to prevent the duration from becoming unduly short.

For equal fuel delivery to all engine cylinders there should be no flow restriction past the valve seat, even at minimum lift, and the orifice areas of each nozzle should be equal. It is essential that the valves are tight when seated, as otherwise fuel will leak into the engine cylinders out of time and detonation and smoky exhaust results.

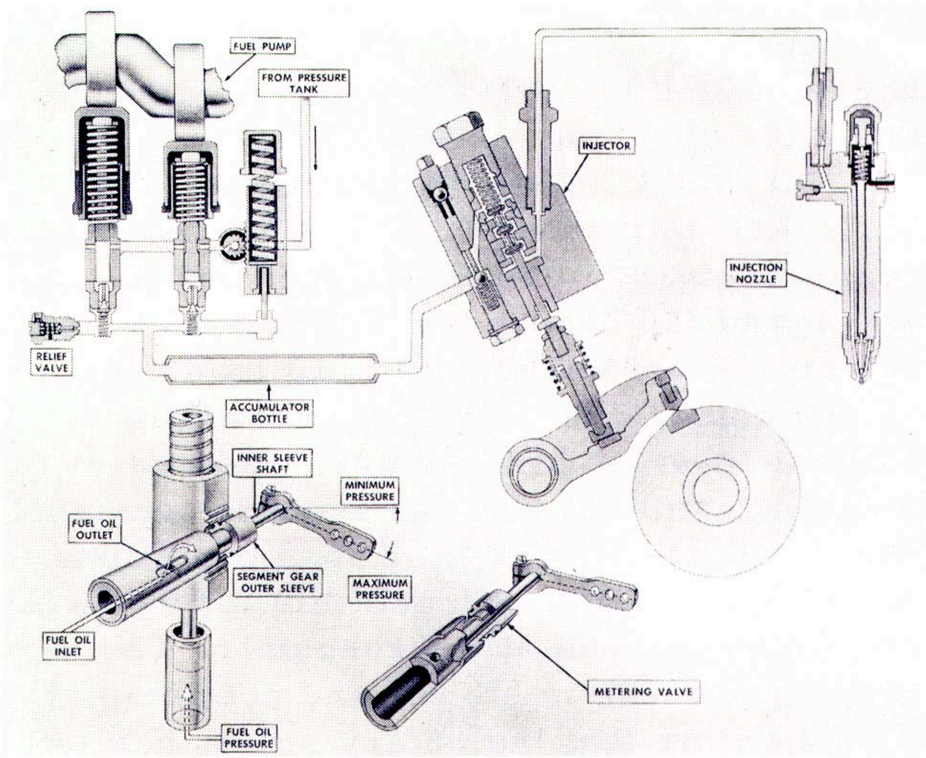


Fig. 23. Controlled pressure distributor system (Cooper-Bessemer)

Distributor[1]

Cooper-Bessemer modified the common rail system by introducing a distributor to time and meter the injected fuel and by replacing the mechanically operated nozzles with conventional pressure operated, differential-valve nozzles. As shown in Fig. 23 the distributor element for each cylinder consists of three disc valves actuated by a plunger from a timed can, lever, and lifter. High pressure fuel is supplied above the top valve, and all three valves must be lifted by the plunger before fuel flows to the nozzle. The injection duration is determined by the length of time the valves are held open. This is governor controlled by the eccentric shaft which raises or lowers the cam lever to vary the clearance between the valve lifter and cam lever. Atmospheric relief of the injection line from the distributor to the nozzle to prevent dribbling is effected at the end of each injection by the residual pressure lifting the lower valve off the plunger to expose an axial vent hole. A variable capacity pump is used, instead of by-passing surplus oil, with the inlet fuel throttled by a rotary sleeve valve controlled by pressure and speed.

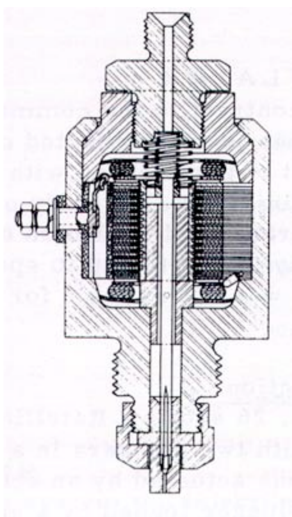


Fig. 24. Magnetically actuated nozzle (Atlas-Imperial)

Electrically-Operated Nozzles

A further development by Atlas-Imperial was a common rail system with electro-magnetically lifted injection valves to time and meter the fuel from a constant pressure accumulator. The nozzle shown in Fig. 24 consists of a soft steel body encasing the solenoid structure, valve assembly, and spray tip. The stator is composed of alternate laminations of iron and brass riveted together, and it has a control bore is which the similarly laminated plunger operates. The magnetizing coils surround the stator, and when energized they induce opposite poles in the plunger laminations. When the valve is seated the plunger laminations are displaced toward the tip relative to the stator, and when the coil is energized the resultant strong magnetic flux pulls the laminations into register. The plunger contacts the valve collar after .005 inch travel to lift rapidly the valve off its seat, and the spring reseats the valve when current and magnetic flux drop off. Both plunger and valve are light in weight, the valve is loosely guided in the plunger, and only the valve seat is lapped. The coil is impregnated so that it is not affected by fuel oil.

Control of the fuel quantity by the time that the valve is open is accomplished by the simplified electrical circuit shown in Fig. 25. The rotary switch alternately connects the condenser across the battery for charging and then across the nozzle coil for discharging and opening the valve. Between these periods the condenser is grounded to discharge it completely.

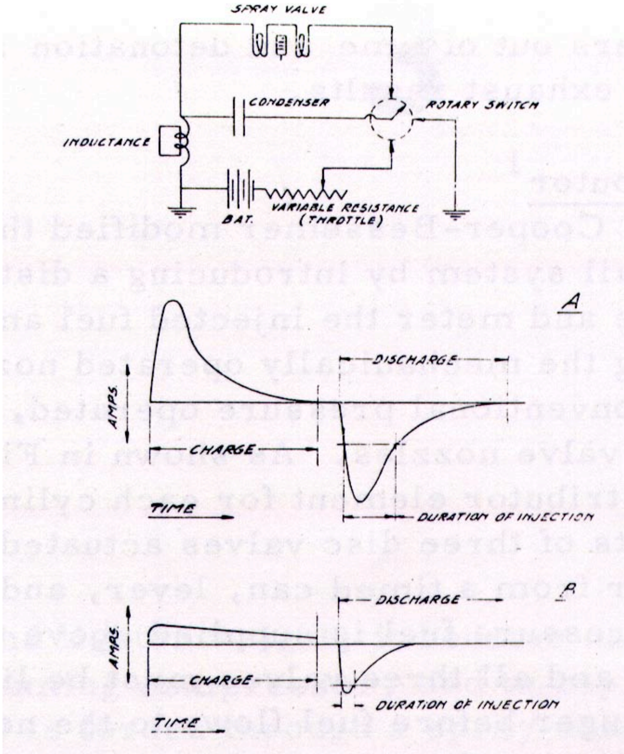


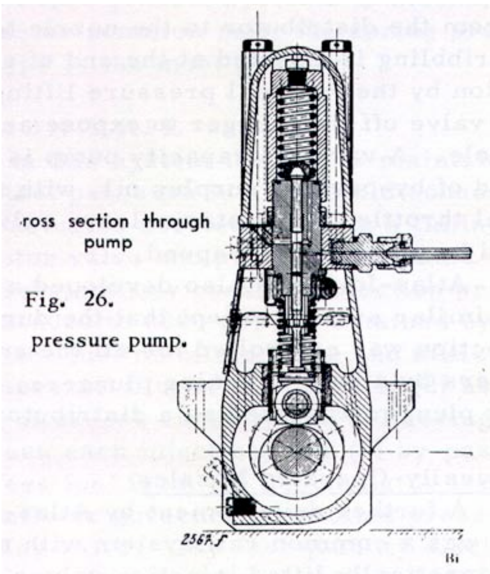
Fig. 25. Simplified electrical circuit (Atlas-Imperial)

Duration of valve opening depends only on the condenser charge, which is controlled by a small rheostat in the charging circuit.

Typical curves of charge and discharge are also shown in Fig. 25 Diagram A represents complete charging and discharging of the condenser with low throttle resistance corresponding to full load. At low resistance charging and discharging is complete even at high speeds, and consequently the time of valve opening is constant. Diagram B illustrates part load conditions where the charging current is limited by the increased resistance of the rheostat. The charging process is slower and not completed by the time that the rotary switch has left the "battery" segment, so that the total charge and quantity of fuel injected are reduced. The operating characteristics can be varied over a wide range by changes in the constants of the discharge circuit, and because of the low mechanical and electrical inertia of the nozzles very short durations of injection are possible.

ACCUMULATOR

In contrast to the common rail system, the fuel quantity injected can be made independent of pump speed with spring or accumulator injection. In early pumps of this type, the crank angle duration of injection was directly proportioned to speed so that the system was not suitable for a wide speed range.



Spring Injection

Fig.26 shows a Ratellier pump of this type with two plungers in a common bore, the lower one actuated by an eccentric and the upper plunger loaded by a spring. During the upward stroke of the lower plunger the fuel trapped between the two plungers increases in pressure, depending upon the characteristic of the upper plunger spring, until the delivery groove in the lower plunger indexes with the outlet passage. Injection then continues as the energy of the spring forces the upper plunger downward.

In the Ratellier pump, made at one time by Scintilla of Switzerland, the injection pressure and rate of injection at high speeds is increased by enclosing the upper spring in a fuel filled chamber vented by a small orifice. The fuel fed to this chamber during the suction stroke is sealed off during the initial lift of the lower plunger, and thereafter it is compressed by the motion of the upper plunger. The fuel quantity is varied by rotation of the lower plunger, which has a helical upper edge.

Hydraulic

In this system fuel discharge occurs during the expansion of fuel from an accumulator volume, usually located in the nozzle holder as shown in Fig. 27. Metered fuel from an eccentric cam driven pump is delivered through the check valve into the accumulator volume as well as through the spill duct into the nozzle spring chamber. No delivery valve is used in the pump so that when the plunger starts to by-pass the check valve closes, fuel in the spring chamber is vented through the spill duct back to the pump, and fuel in the accumulator passes through the discharge duct to the nozzle. Since the accumulator pressure is higher than the nozzle opening pressure, the nozzle valve lifts and injection continues until the accumulator pressure drops to the nozzle closing pressure. The maximum injection pressure, which is the accumulator pressure at the

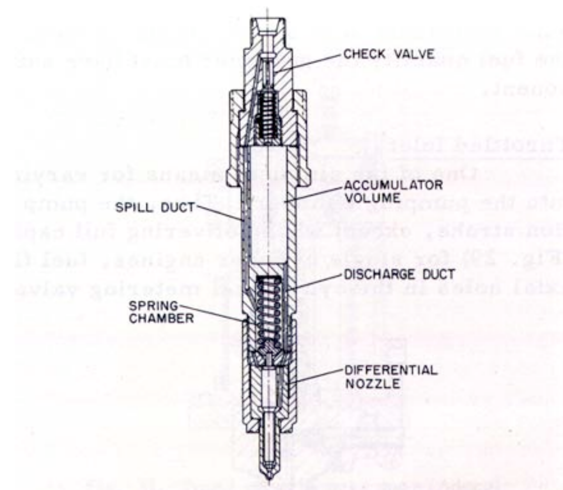


Fig. 27. Accumulator nozzle holders (American Bosch)

starts of injection, depends upon the accumulator volume and the quantity of fuel metered to it by the pump. It is, therefore, independent of the pump speed and nozzle orifices. Since the fuel delivered to the spring chamber is spilled back to the pump, the volume of this chamber should be as small as possible. The accumulator volume is a compromise to avoid excessive pressures at full load and inability to deliver idling fuel quantities. A simple equation for relationship of the variables in an accumulator system is:

$$q = \frac{V}{K} (P_1 - P_2)$$

Where:

- **q** = discharge quantity, cu. mm.
- **V** = volume of accumulator, cu. mm.
- K = bulk modulus of fuel, 280,000 psi.
- P_1 = peak accumulator pressure, psi.
- P_2 = nozzle closing pressure, psi.

JERK PUMP

In this system the injection pump times, meters, and forces the fuel at high pressures through the spray nozzle. Plunger pumps are used exclusively, and the plunger is actuated by a cam whose contour exerts considerable control of the injection characteristics. The spray duration in crank degrees increases with speed and fuel quantity, but not to the extent of the common rail system, so that the jerk pump system has been widely adopted for high speed engines as well as for those of low and medium speeds. Numerous methods have been developed for controlling the fuel quantity of these pumps.

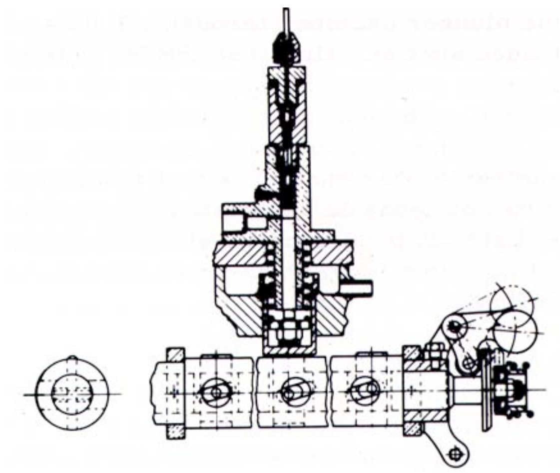


Fig. 28. Variable stroke jerk pump (Sheppard)

Variable stroke

Fig. 28 shows a simple pump of this type used on the Sheppard precombustion chamber diesel engine. The plunger stroke is varied to change the fuel quantity metered by sliding the contoured end cam plate in or out of its slot in the hollow camshaft. The governor shaft inside of the hollow camshaft carries a pin which engages the angular slot in the cam plate, and axial movement of this shaft produces radial displacement of the cam plate. For regulating the fuel quantity the governor must have sufficient power to overcome the driving torque component.

Throttled Inlet

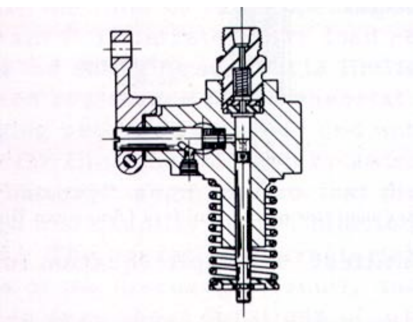


Fig. 29. Throtted inlet pump (Demco)

One of the simplest means for varying the fuel discharge is to throttle the flow of fuel into the pumping cylinder. Thus, the pump does not receive a full charge of fuel on its suction stroke, except when delivering full capacity. In the Demco IPFN throttled inlet pump (Fig. 29) for single cylinder engines, fuel flows into the plunger bore through transverse and axial holes in the cylindrical metering valve. By rotation of the metering valve the port opening to the plunger bore can be varied. This pump is actuated by a separate cam and tappet mechanism in the engine. Fuel delivery commences when the plunger covers the inlet port on the upward stroke of the plunger, and it terminates when the spill groove in the plunger uncovers the inlet port.

Advantages of the throttled inlet control are its simplicity, very low control forces, and declining fuel delivery vs. speed characteristic which facilities governing. It is not suitable for multi-plunger pumps because of the difficulty of uniformly controlling the throttling of several valve over the entire range of fuel deliveries. It has been successfully applied to the Roosa Master distributor pump.

Throttled Bypass

In this method fuel in the plunger chamber is simultaneously discharged through the nozzle and by-passed through a throttle valve back to the inlet. In the pump shown in Fig. 30 a full charge of fuel enters the plunger chamber through a light spring loaded suction valve, and the by-pass port opening is varied by a governor operated needle valve to control the quantity of fuel spilled back into the fuel inlet chamber. One advantage of this control is that the duration of injection tends to be constant regardless of fuel quantity so that quieter combustion is obtained at light loads. A serious shortcoming is the sharply increasing delivery vs. speed characteristic, which is an obstacle to good governing. Calibration of a multi-plunger pump is also diffcult.

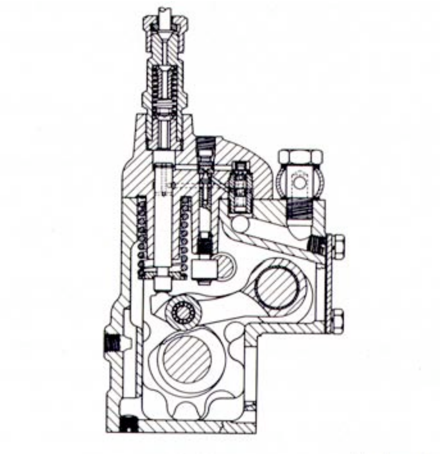


Fig. 30. Throttled by-pass pump (Adeco)

Timed Valve Bypass

The fuel quantity in this method is controlled by spilling the excess through a mechanically operated by-pass valve. Fig.31 shows a pump design with this type of fuel control which has been used successfully on some medium and large size diesel engines. Fuel enters the plunger chamber through the spring loaded suction valve, and delivery begins on the upstroke of the plunger. The fuel delivery ceases when the pressure- balanced by-pass valve is lifted by contact with the rocking lever, which is actuated by plunger follower. The quantity of fuel discharged is varied by rotating the eccentric shaft on which the rocking lever pivots. With this type of control the plunger and by-pass valve have long lapped lengths to minimize leakage and the use of a suction valve gives a slightly declining delivery with increase in speed. Disadvantages of this design are the large dead volume of fuel under compression and starting fuel delivery at the beginning of plunger lift when the plunger velocity is very low.

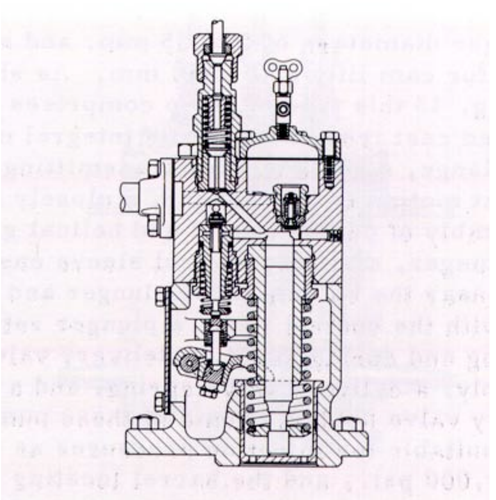


Fig. 31. Timed valve by-pass pump (Adeco)

Port Control

In this method of fuel metering, a portion of the reciprocating plunger serves as a valve in covering and uncovering ports in the plunger barrel during filling, discharge, and by-passing of the fuel. By providing a helical groove or land on the plunger and arranging to rotate it, the effective plunger stroke can be varied to control the quantity of fuel delivered per stroke. Plungers are made with metering lands having lower helices, upper helices, or both to give constant port closing with variable ending, variable port closing with constant end, or both variable beginning and ending, respectively. Four steps in the pumping sequence of a port type pump are illustrated in Fig 32.

Beside their simplicity, ported pumps have minimum fuel volume under compression so that good control of injection characteristic is possible. A disadvantage of conventional port control pumps is the rising delivery characteristic with increasing speed. This is the result of fuel throttling through the ports so that less fuel is by-passed before port closing and after port opening as the pump speed increases.

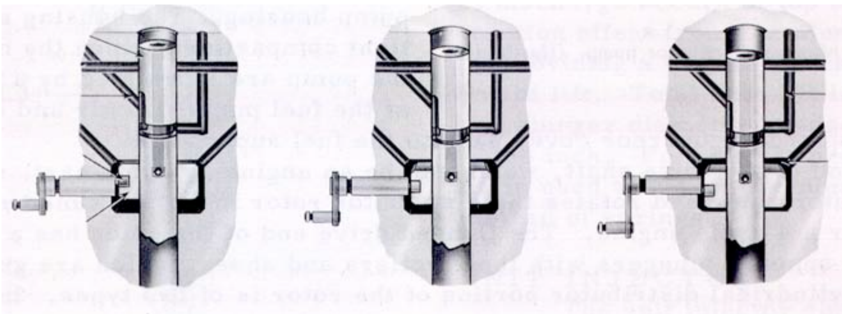


Fig. 32. Pumping sequence with port control (American Bosch)

MODERN INJECTION PUMPS

In the preceding section various methods of mechanical injection and metering control have been described. Some of these have been discarded, others have been improved by the injection equipment manufacturers, and there have been important developments in distributor pumps and unit injectors for small, high speed engines. This section will, therefore, describe some of the pumps and injectors being manufactured and used in the United States. Similar pumps are also being produced in Europe.

SINGLE PLUNGER UNITS

These pumps of the single plunger type, one for each cylinder of the engine, are flange mounted directly over the engine camshaft which has a fuel cam for each pump. The tappet assemblies are also part of the engine. These pumps are popular for small single cylinder engines and for medium and large size engines, since each pump can be located adjacent to each engine cylinder. This has the advantage of permitting short discharge tubings. Practically all of these pumps are now of the port control type

American Bosch

This manufactures produces several sizes of their single plunger APF pumps with plunger diameters of 5 to 35 mm and suitable for cam lifts of 7 to 40 mm. As shown in Fig. 33 this type of pump comprises a rugged cast iron housing with integral mounting flange, a guide cup for transmitting the tappet motion to the plunger, a closely fitted assembly of ported barrel and helical grooved plunger, a slotted control sleeve engaging lugs near the bottom of the plunger and meshing with the control rack, a plunger return spring and spring seats, a delivery valve assembly, a delivery valve spring, and a delivery valve holder. Some of these pumps are suitable for injection pressures as high as 15,000 psi, and the barrel locating screws serve the additional function of absorbing the impact of the spilled high pressure fuel.

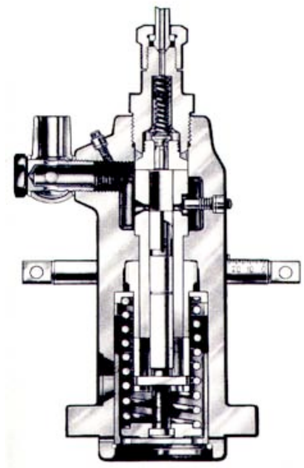


Fig. 33. Single plunger unit pump (American Bosch)

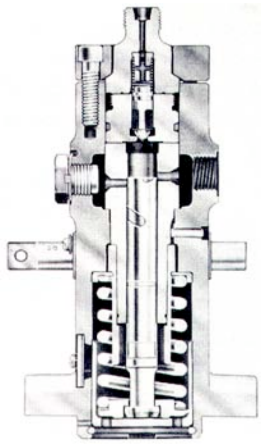


Fig. 33. Single plunger unit pump (Bendix)

Bendix

Similar pumps are also produced by this manufacturer. The significant differences of the Bendix pump (Fig. 34), as compared with others, are that its plunger barrel has axially spaced inlet and by-pass ports and the plunger has axial and radial holes communicating with the helical grooves instead of vertical slots. This construction reduces the unsupported area near the top of the plunger, thus improving its manufacturing and operating conditions. During operation port closing occurs when the top of the plunger closes the inlet port and the end of delivery when the helical groove starts to uncover the by-pass port.

MULTIPLE PLUNGER PUMPS

These pumps, which have as many pumping elements as there are cylinders, have been popular for small, high speed engines. They are generally of en bloc construction with self-contained camshaft and tappet mechanism. This simplifies the installation of injection equipment on the engine, and it enables the pump to be calibrated and serviced as a unit.

American Bosch

This company makes their series of APE pumps in one to eight cylinders and in three basic sizes to satisfy the requirements for all sizes of high speed diesel engines. All are of the ported type with helix metering plungers availed in various helix configurations and

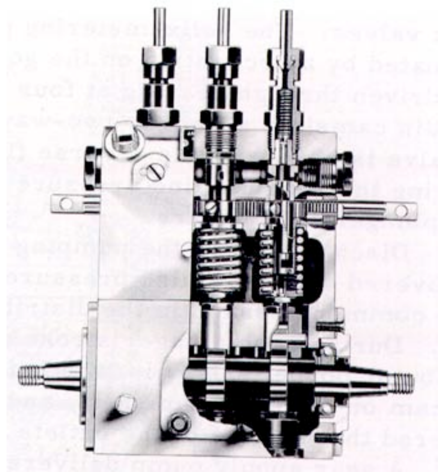


Fig. 33. Multiple plunger pump (American Bosch)

diameters from 5 to 13 mm. Camshafts are also available with several different cam shapes. The pump (Fig. 35) comprises an aluminum housing dividing into the camshaft compartment, plunger and spring compartment, and header section. The lower compartment contains the camshaft and serves as a reservoir for lubricating oil. End plates containing ball bearings and oil seals support the camshaft.

The partition between the lower compartments contains the bores for tappet assemblies. Timing adjustment for each plunger is provided by tappet screws. In addition to containing the plunger-barrel assemblies and return springs, the control sleeves with adjustable gear segments meshing with the control rack are also located in the middle section. The upper part of the housing contains the fuel supply sump, retracting delivery valves and springs, and discharge outlets.

These pumps are available for base mounting or flange mounting in accordance with S.A.E. standards. They are usually equipped with fuel supply pumps and governors. They may be driven at half engine speed or engine speed for four-cycle and two-cycle engines, respectively.

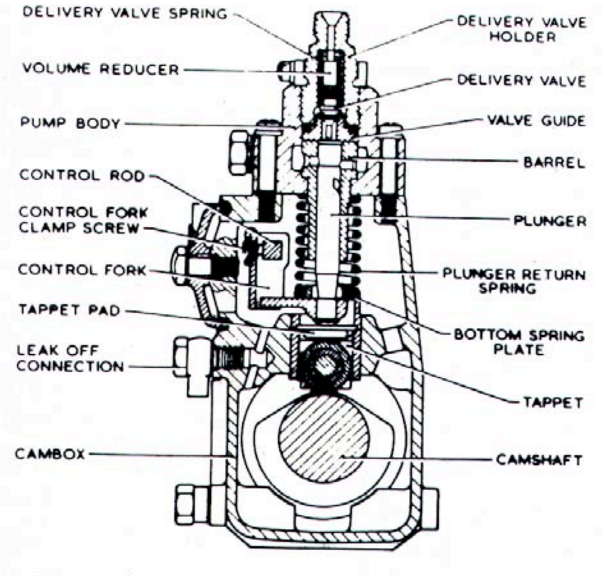


Fig. 36. Multiple plunger pump (Simms)

Simms

The newly designed SPGE..M series (Minimec) pumps of this English manufacturer are suitable for engines up to 90 cu. in, per cylinder. For ease of servicing the camshaft, bearings, tappets, control rod, and governor are contained in the aluminum alloy housing; and the individual pumping elements with plungers of 6 to 9 mm. diameter are assembled to a steel body which contains the fuel passage and is bolted to the housing (fig. 36). Compactness is achieved by using serrated delivery valve holders to permit closer spacing of the cylinders, and by using control arms attached to the lower ends of the plungers of engaging adjustable forks fastened to the control rod. Tappet height is reduced by using shims instead of adjustment screws. The variable speed governor incorporated at the drive end of the pump features roller weights, a torsion speed control spring, and an excess fuel device which only operates on starting.

Robert Bosch

In addition to its extensive line of single and multiple plunger pumps, this German manufacturer has introduced a reduced size model M pump (Fig. 37) for 4 and 6 cylinder engines up to 55 cu. in. per cylinder. For compactness, the plungers are rotated by links connected to the control rod, and the tappet adjustment screws are eliminated by selection of proper tappet roller diameters. Plungers have axial holes and helical slots instead of milled helices to reduce the unsupported area for longer plunger life. The governor housing is integral with the die cast pump housing , and it is deigned to accommodate either a pneumatic or mechanical governor.

DISTRIBUTOR

Lower cost by reduction of parts and inherent calibration are features of distributor pumps for small, high speed engines. One or, at the most, two pumping elements are used in combination with a suitable distributing means to meter and deliver the fuel to all the engine cylinders.

International Harvester

This company has developed and manufactures a distributor pump for some of its four and six cylinder engines. As shown in Fig. 38 the single pumping element serves four outlets through a poppet valve type distributor. In the six cylinder engine two pumping elements are used to deliver fuel to the six outlets through a distributor with six poppet valves.

The helix-metering plunger is actuated by an eccentric on the governor shaft driven through gearing at four times the main camshaft speed. A two-way delivery valve is used with the reverse flow valve relieving the residual line pressure to 600 psi when plunger spill occurs.

Discharge from the pumping element is delivered

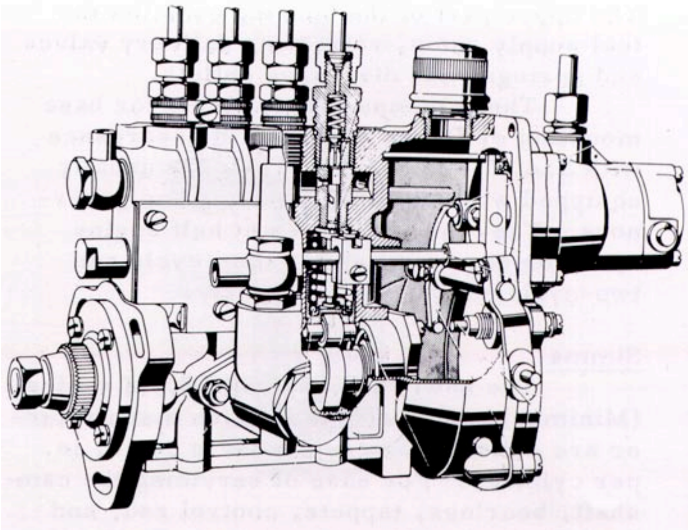


Fig. 37. Small multiple plunger pump (Robert Bosch)

through a high pressure tubing to the common passage in the distributor block. During each plunger stroke one of the distributor poppet valves is lifted off its seat by a cam on the main camshaft, and fuel is delivered through one of the outlets.

A gear supply pump deliverers fuel to the pumping element. There are two by-pass valves, one at the supply pump for maintaining the supply pressure constant, and the other for venting the pumping element. Vented fuel is returned to the supply tank, and leakage past the distributor valves is returned to the supply pump inlet. The centrifugal governor actions is transmitted to the control rack through levers whose motion is resisted by tension springs. A torque control feature is provided by the action of leaf springs in the governor linkage.

American Bosch

In their PSB type pump, the single plunger has the dual functions of reciprocation for pumping and rotation for distribution,. It is built with either four or six outlets and for either crankshaft or camshaft speeds, although the former is most popular because it simplifies the engines drive and flange mounting. As shown in Fig. 39 the complete pump consists of a housing and drive mechanism, a hydraulic head, and a governor.

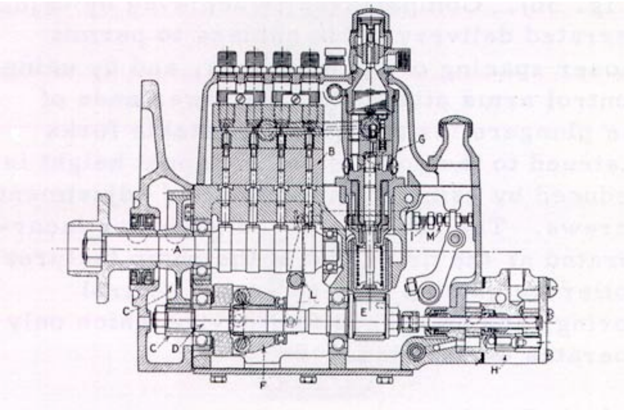


Fig. 38. Single plunger distributor pump (International Harvester Corp.)

The pump housing is an aluminum die casting with an integral flange for mounting to the engine. The camshaft is supported in a ball bearing at the front and a sleeve bearing at the rear, and its cam has two lobes for four outlets or three lobes for six outlets for 4-stroke cycle engine application. A spiral gear on the camshaft meshes with a vertical shaft for rotating the plunger through a pair of spur gears. The gear type supply pump is also driven by the spiral gear. A roller tappet is interposed between the cam and plunger. Pressure lubrication is directed to the tappet, bearings, lower portion of plunger, and drainage is provided through the flange.

The hydraulic head contains a plunger of 7 to 10 mm diameter with its lapped control sleeve, a plunger return spring, a plunger drive gear, and a delivery valve assembly. The discharge fittings communicate with the plunger bore through equally spaced ducts, and other drillings connect the broached sump with inlet and outlet of the fuel supply.

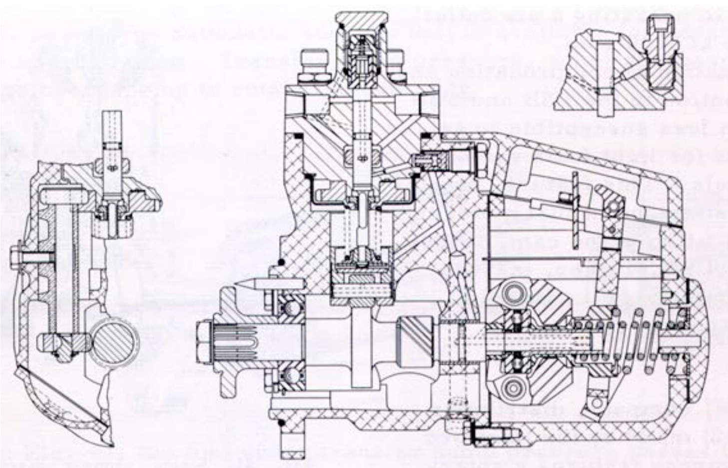


Fig. 39. Single plunger distributor pump (American Bosch)

In operation, fuel enters the space above the plunger when the two inlet ports are uncovered on the downward stroke of the plunger. When the ports are closed on the up stroke fuel is delivered through the delivery valve and passage to the upper plunger annulus, which has a connecting vertical groove that communicates on successive discharge strokes with each of the outlet ducts. Fuel metering is varied by axial positioning of the plunger sleeve which is connected by a control lever linked to the governor fulcrum lever (Fig. 40). The axial position of the sleeve determines the lift of the plunger at which spill occurs through the axial and radial spill holes in the plunger.

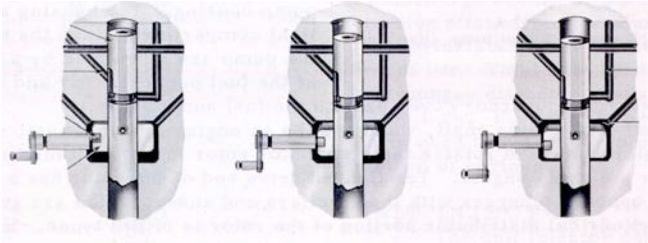


Fig. 40. Fuel control sleeve positions.

Thus, maximum delivery occurs when the sleeve is in its uppermost position, and no delivery takes place when the sleeve is at its lowest position. Two types of mechanical governors are available, one of simple extension spring type and the other having internal compression springs and torque control.

PSB pumps of V-type construction with two hydraulic heads have been built for 8 and 12 cylinder engines, and a larger PSB-B T pump having two hydraulic heads in tandem is also available. Maximum capacity of the standard PSB pump is 150 cu. mm. per stroke, and about double that for the PSB-B T.

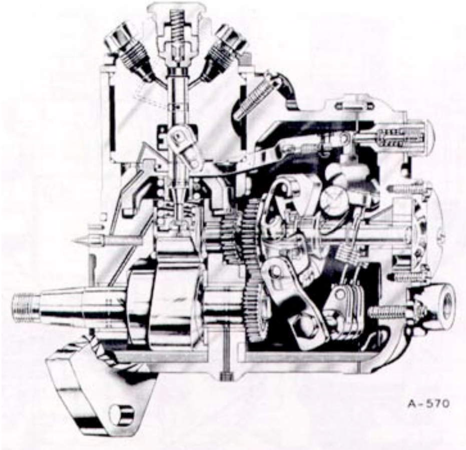


Fig. 41. Single plunger distributor pump (American Bosch)

A more compact edition is the PSH pump (Fig. 41) with reduced cam lift, simplified tappet, flexible vane fuel supply pump, and smaller governor driven through gearing. The parasitic high pressure fuel volume of the hydraulic head is reduced to a minimum for optimum injection characteristics in small open chamber engines.
This pump is obtainable in either engine or half engine speed applications by selection of suitable gearing and camshaft, and it is available in both flange and shank mountings. V-type and tandem head PSH pumps have also been built, as well as single head 8 outlet executions. Maximum capacity of the present PSH pump is 100 cu. mm. per injection at full load.

For a single plunger distributor pump the maximum operating speed depends upon the cam lift, inertia of reciprocating parts, spring force, fuel inlet time, and fuel supply pres- sure. Present ceiling for the PSB and PSH pumps is set at 10,000 injections per minute.
This is equivalent to operating a six outlet engine speed pump at 3200 rpm. The combination of reciprocating and rotating plunger motion in the PSB and PSH pumps makes them less susceptible to seizure with close fits for light fuels such as gasoline and jet fuels. This feature, together with the positive method of fuel metering and pressure lubrication of the cam, tappet, and lower portion of the plunger, makes these pumps particularly well adapted to multi-fuel engine applications.

Roosa-Master[5]

This simple, compact, distributor type pump (Fig. 42) made by the Hartford Machine Screw Company features a rotary distributor with an integral, single cylinder, opposed plunger, inlet metering pumping element actuated by an internal cam ring. The standard model DB pump has a die cast aluminum housing with 2 or 3-hole flange to permit mounting in vertical or horizontal position. It supports the drive shaft in a sleeve bearing and contains the governor springs and levers, the internal ring cam, and the distributor assembly including the fuel transfer pump and governor fly weights. Accessories such as electrical solenoid shut-offs are also contained within the pump housing; and where an automatic timing advance unit is used, it can be incorporated at the underside of the pump housing. The housing serves as an oil tight compartment since the moving parts of the pump are lubricated by a continuous flow of the fuel pumped. Air and excess fuel are vented from the top of the governor cover back to the fuel supply tank. The tang end of the drive shaft, which can be an engine or pump part, engages with a slot in the distributor rotor and rotates the distributor rotor in proper timed relationship at camshaft speed for a 4 cycle engine.

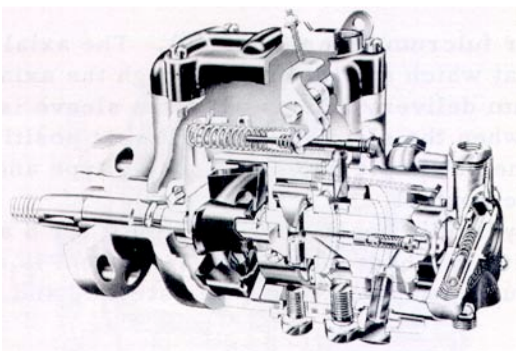


Fig. 42. Opposed plunger distributor pump (Hartford Machine Screw Co.)

The flanged drive end of the rotor has a diametric bore containing the two opposed plungers with their rollers and shoes, which are guided in slots in the flange. The cylindrical distributor portion of the rotor is of two types. In one a single angled passage functions as a common passage for both inlet and discharge. The second type has an angled passage for the fuel inlet and an axial bore with a retraction type delivery valve for the discharge. The end of the rotor has cross slots for the two blades of the vane type transfer pump.

The hydraulic head of the distributor contains the axial bore in which the rotor revolves, the charging ports and discharge outlets, the metering valve bore, and the transfer pump liner. The cylindrical metering valve has a triangular milled slot by which the inlet passage to the pumping chamber can be varied by rotation of the valve. The end plate bolted to the hydraulic head contains the fuel inlet connection, fuel strainer, and transfer pump regulating valve.

The centrifugal governor mounted on the drive end of the rotor consists of a stamped retainer housing the knife edge pivoted flyweights, which actuate the thrust sleeve in opposition to the control spring to rotate the metering valve. Speed regulation is adjustable by varying the effective length of the control spring. Torque control is adjustable to some extent by means of a torque screw limiting the wide open position of the metering valve. A hydraulic, servo type automatic advance unit is available for advancing the cam as a function of either speed or load. Transfer pump pressure, which increases with speed, actuates a piston against a spring to rotate the cam ring.

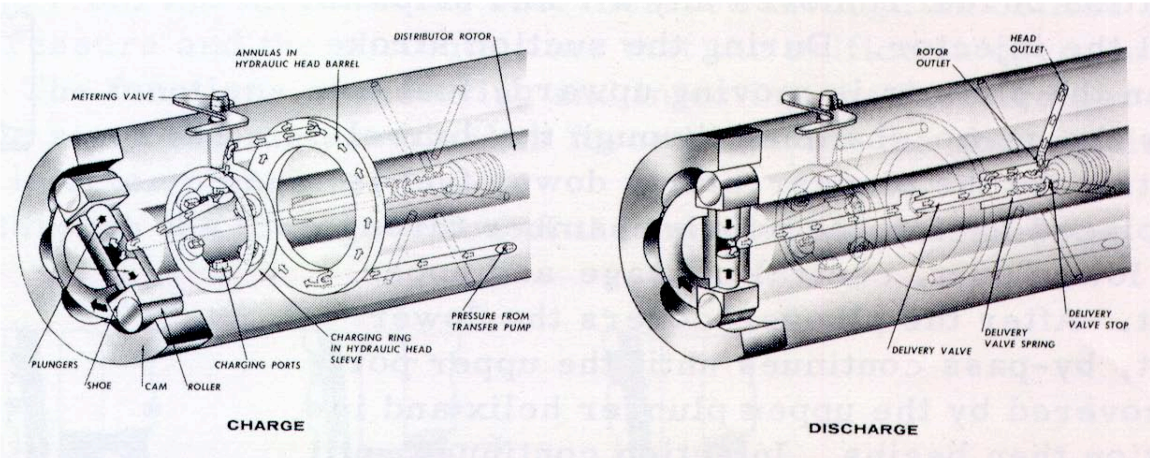


Fig. 43 Fuel distribution principle (Hartford Machine Screw Co.)

As shown in Fig. 43, the fuel under transfer pump pressure passes through the drilled passage in the head to the large annulus. From there it flows through the metering valve to the charging ring and charging ports. As the charging port of the rotor registers with a charging port in the head, fuel flows into the pumping cylinder and forces the plungers apart in pro- portion to the quantity of fuel metered. As the rotor continues to revolve, the charging port passes out of registry, and the rollers contact the cam rise to begin injection. The plungers are forced toward each other, and the fuel trapped between them is forced through the delivery valve and out one of the outlet ports to a discharge tubing and nozzle. As the rollers leave the cam rise, the rotor outlet port passes out of registry, injection is terminated, and the charging and discharging cycles are repeated for successive engine cylinders. Maximum fuel delivery is adjusted by limiting the outward travel of the cam roller shoes by means of a leaf spring.

Since inlet throttling is used for fuel control, the fuel delivery decreases with in- creasing speed to help governing. End of fu- el discharge occurs at top of stroke, and re- traction effect from the plungers is obtained by providing a step in the cam just after the end of lift. Total cam lift is only .072 inch and plunger diameters range from 0.250 to 0.390 inch. Fuel supply pressure and inertia is used to return the plungers and rollers instead of springs.

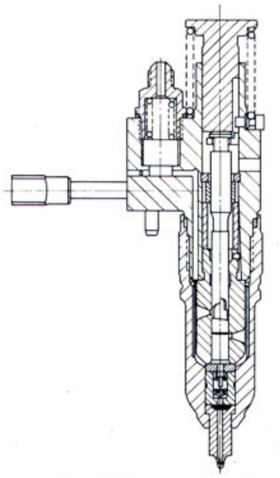


Fig. 44. Unit Injector (General Motors)

UNIT INJECTOR

The unit injector combines the pump and spray nozzle in a single unit, which is T mounted on the cylinder head. This design eliminates the problems of pressure waves and fuel compressibility in long discharge tubinge, but it requires that means be provided on the engine for actuating the injectors.

General Motors[6]

The injector for their Series 71 engines, shown in Fig. 44, consists of a forged steel body and long retaining nut in which are housed the inlet filter, follower, helix type plunger with ported barrel, rack and gear, check valves, and spray tip. The injector is seated in the cylinder head on the bottom taper of the long retaining nut and is clamped down by a crab and bolt. The pump plunger follower is depressed by a roller at the end of a cam actuated rocker and returned by the follower spring.

Fuel oil at 20 psi enters the injector through the inlet filter and flows through the annular chamber around the barrel before passing out the outlet. This continuous circulation of fuel removes any air and helps to cool the injector. During the suction stroke when the plunger is moving upward, fuel enters the pump chamber through the barrel ports. As the plunger starts down, fuel is displaced back to the supply chamber through the lower port, central passage and upper port. After the plunger covers the lower port, by-pass continues until the upper port is covered by the upper plunger helix and injection then begins. Injection continues until the lower helix of the plunger uncovers the lower port. This terminates the injection, and by-pass continues through the central passage of the plunger and lower port for the remainder of the downward stroke. To prevent erosion the spilled fuel impinges against the hardened spill deflector sleeve, which is loosely fitted in the annular chamber. Quantity control is effected by rotating the plunger by the control rack and gear. This rotation varies the injection quantity as well as the injection timing, depending upon the type of plunger helices. The metered fuel passes through the check valve, opens the spray tip valve, and is discharged through the orifices in the spray tip. The check valve prevents combustion gases from entering the injector if the spray tip valve becomes temporarily in active. Three types of spring loaded spray tip valves are used: (1) a spherical check valve opening at 500 to 2000 psi, (2) a crown check valve opening at 350 to 850 psi, and (3) a differential valve opening at 3000 psi. The number and size of the spray holes depend on the engine model requirements and vary from 6 to 12 holes of .006 to .017 inch diameter. The peak injection pressure at full load varies from 15,000 to 40,000 psi.

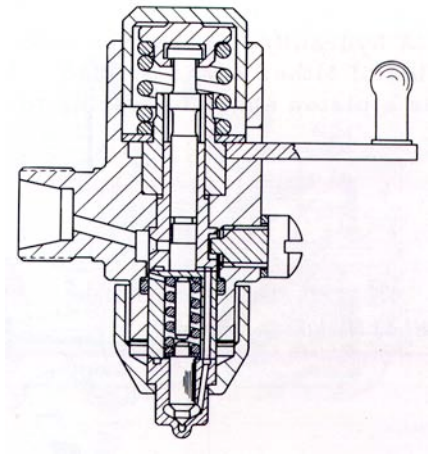


Fig. 45 Unit Injector (American Bosch)

American Bosch

These unit injectors, as shown in Fig. 45, use many of the pump and nozzle components of the conventional jerk pump system. In some cases a delivery valve between the plunger and nozzle has been found beneficial in getting sharper end of injection, but it adds to the over-all length of the unit. Differential valve type nozzles are used for best results.

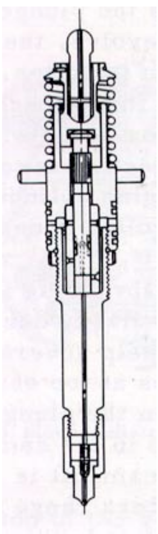


Fig. 46 Unit Injector (Murphy Diesel Co.)

Murphy

Fig. 46 shows the simple construction of the latest unit injectors made by Murphy Diesel Company for their own engines. In operation fuel enters the injector at 20 psi, flows into the annular space around the barrel, and enters the plunger chamber through the barrel port on the upstroke of the plunger. On the cam actuated downstroke the plunger closes the port, and the fuel is forced through the dual flat seat check valves and six orifices of the spray tip into the combustion chamber. Transverse helix slots in the plunger connecting with axial and radial holes control the end of injection. A rack and gear are used to rotate the plunger for varying the fuel quantity. Overhead cams are used to actuate the plunger.

Cummins PT system[7]

This system derives its name from the fact that the fuel metered through an orifice in- to the injector depends on the fuel pressure and the absolute time that the orifice is open, as fixed by the cam-actuated plunger. The functions of metering and injecting the fuel are per- formed separately, which is possible since the metered fuel is forced through the spray holes into the combustion chamber by the injector plunger.

Fig. 47 shows the fuel flow through the lower portion of the injector during various phases of the injection cycle.

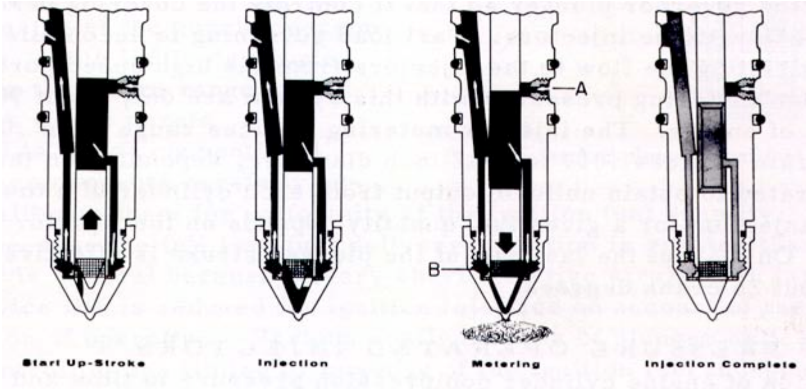


Fig. 47 Fuel injection cycle (Cummins Engine Co.)

Fuel at controlled pump pressure enters the injector and flows down the passage at the right. On the engine intake stroke the injector plunger starts to lift, uncovering inlet orifice A, so that fuel circulates through the injector and up the drain pas- sage at the left. This cools the injector and vents it of air. As the plunger continues to rise metering orifice B is uncovered, and metered fuel flows into the cup between the tapered end of the plunger and the spray holes. Obviously, the fuel metered through B varies as the square root of the fuel pressure and inversely as the engine rpm, so that the fuel pres- sure has to be adjusted accordingly.

On the downstroke of the injector plunger the deposited metered fuel is discharged through the spray holes with peak injection pressures as high as 17,000 psi. At the bottom of its stroke the end of the plunger practically contacts the injector cup. The plunger then remains seated until the next engine intake stroke.

The pump, Fig. 48, consists of a gear pump, governor controlled pressure regulator, and manually operated throttle. The pressure delivered by the gear pump is dependent on the forces acting to close the gap between the axially drilled end of the governor plunger and the recessed face of the pres- sure control button. Since the centrifugal force of the governor fly weights varies as the square of the speed and the pressure delivered by the pump also varies in like manner, this results in practically constant engine torque throughout the speed range.

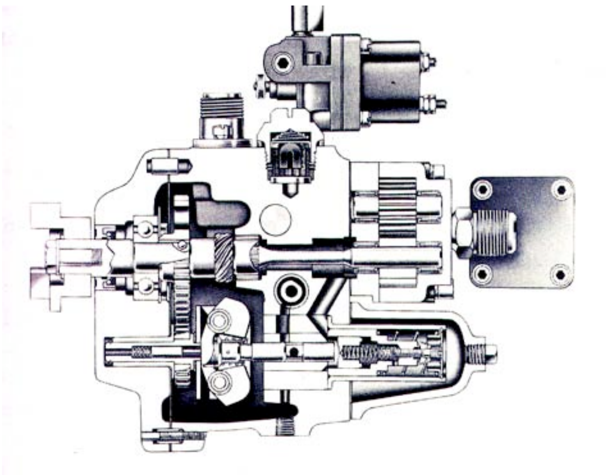


Fig. 48. Controlled pressure pump (Cummins Engine Co.)

Several means are available for modifying the torque curve. Maximum torque is determined by the preload of the governor spring. When the plunger starts to move at higher speeds, fly weight force is added to the fuel pressure to counterbalance the spring force. Thus, as the speed increases the fuel pressure decreases, and the engine torque also de- creases. The torque is also influenced by the lever ratios of the flyweights. For increased torque backup, an additional spring can be mounted on the governor sleeve and a smaller recessed area pressure button used. For increased fuel at low speeds and starting, another spring can be installed inside the hollow governor drive shaft to load the governor plunger at low speeds. This results in higher fuel pressures at low speeds.

At low idle the throttle is closed, and the fuel pressure to the injectors is regulated by a light idle spring acting against the pressure button and by movement of the governor plunger. A groove in the plunger controls the opening to the idle port. For maximum speed, the flyweights position the governor plunger so that it controls the covering of the high speed port to decrease the fuel flow to the injectors. Part load governing is accomplished by means of a throttle valve restricting the flow to the injectors from the high speed port.

The maximum metering pressures with this system are only about 90 to 200 psi, depending on the size of engine. The injector metering orifices range from .017 to .024 inch diameter and the drain orifices .037 to .042 inch diameter, depending on the application. The injectors are calibrated to obtain uniform output from each cylinder of a multi~cylinder engine. Duration of injection for a given fuel quantity depends on the injector cam and injector plunger diameter. Only about the last half of the plunger stroke is effective at full load, and this amounts to about 25 crank degrees.

COMPRESSION PRESSURE OPERATED INJECTORS

The utilization of engine cylinder compression pressure to time and pressurize the fu- el for injection was patented as early as 1918 by the Russian professor M. Vadime Archaouloff. It was originally developed as a simple method for converting an air injection engine to solid injection without the complication of adding timed fuel pump cams. Even the original fu- el metering pumps of the air injection system were retained.

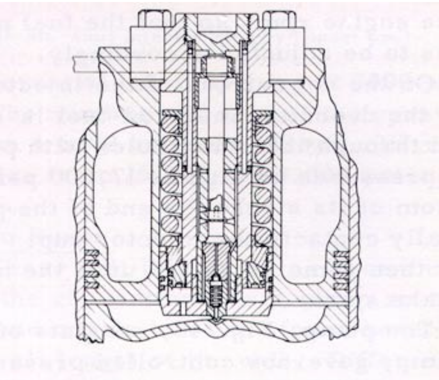


Fig. 49. Compression actuated injector. (After Patent No. 2,572,118)

In the Archaouloff system metered fuel was delivered to a pressure intensifier mount- ed on the cylinder head cover adjacent to a conventional spring loaded, differential valve nozzle. The intensifier consisted of a lower piston, which communicated by a pipe with an engine cylinder, and connected above at an injection plunger. The ratio of areas was about 12 to 1, so that with 400 psi compression pressure the initial injection pressure would be 4800 psi. The injection timing was controlled by adjustment of the nozzle opening pressure. For starting the nozzle opening pressure was reduced. This system was especially suitable for marine engines as no fuel cam reversing mechanism was required when changing the direction of engine rotation. Although the intensifiers were cooled by circulation of en- gine cooling water through them, some trouble was experienced with pistons seizing and carboning up. This was attributed to fuel leakage past the plungers.

The Archaouloff system was first used on large, slow speed engines. General Motors have applied the method experimentally for actuating unit injectors in their small two-cycle engines at speeds up to 2800 rpm. Fig. 49 shows one concept for a cylinder pressure operated unit injector in which the plunger barrel and spray tip are attached to the gas piston and reciprocate with it. One difficulty is control of injection timing at various speeds.

DUAL FUEL PUMPS

Dual fuel engines are able to operate completely on fuel oil or predominantly on gaseous fuel with oil ignition, and they are fully convertible during operation from one to the other. The injection pumps must therefore be capable of providing suitable injection characteristics from full load to ignition quantity (5 to 10 percent of full load). At first separate pumps and nozzles were used for delivering the ignition and main injections but troubles were experienced with carbonization or valve sticking of the nozzles not continuously operating. This led to developing pumps for metering the entire range of fuel quantities through a single nozzle.

One method is to use conventional pumps with all the components carefully selected, and then calibrate them for uniformity at the ignition fuel quantity. This arrangement is a compromise because the full load fuel delivery variation is greater than normal, control of ignition quantity is critical because of very short effective stroke with large diameter plungers, and service life is reduced for ignition injection on account of the short plunger seal at this condition of operation.

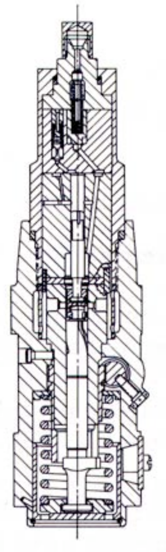


Fig. 50 Stacked plunger, dual fuel pump (American Bosch)

Various configurations of plunger helices were used in this method for controlling the timing or quantity of the ignition fuel independently of the main injection.

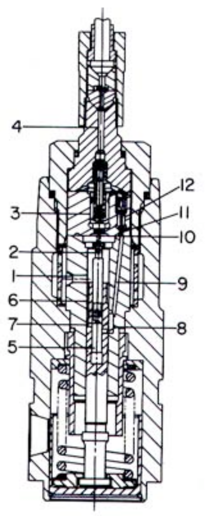


Fig. 51. Telescopic plunger dual fuel pump (American Bosch)

" Stacked Plungers "

The pump shown in Fig. 50 is the result of a joint development of American Bosch and Worthington Corporation. It consists of two interconnected, coaxial plungers operating simultaneously in one pump and controlled by the same rack. The large diameter lower plunger delivers fuel when the engine is operating mostly or entirely on fuel oil, and the small upper plunger delivers the ignition fuel quantity. A separate delivery valve is provided for each plunger. Thus, the larger valve has retraction for optimum results at full load, whereas the smaller valve has no retraction to maintain the high residual pressures required for regular injection at ignition quantities. The ignition quantity plunger helices can be arranged to give advanced, retarded, or any combination of variable timing of the ignition quantity in relation to the main injection plunger.

Telescopic Plungers

This pump (Fig. 51) is similar in operation and injection characteristics to the previous design. By containing the ignition quantity plunger within the larger plunger, the height of the pump is reduced nearly to that of a conventional pump. For full load operation fuel enters the barrel through port (1) and is delivered through a cross hole and axial passage (2) in the ignition plunger to the delivery valve (3) and outlet (4).

For the ignition quantity fuel enters through ports (1) passes into the main plunger bore (5) through grooves (6) and port (7). Metered fuel is delivered through port (8), passage (9), annular groove (lo), duct (1l), and delivery valve (12) to the outlet (4).

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