Aug. 27, 1968 R. J. MADDALOZZO 3,398,730

FUEL INJECTION SYSTEM AND DISTRIBUTOR VALVE THEREFOR

Filed Oct. 31, 1966 3 Sheets-Sheet 1

Fig. 1

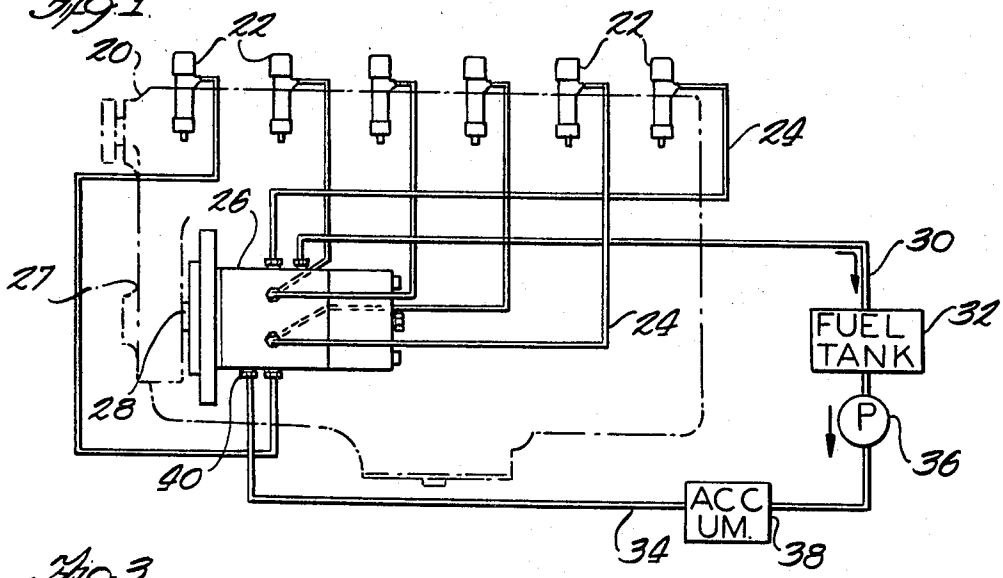


Fig. 3

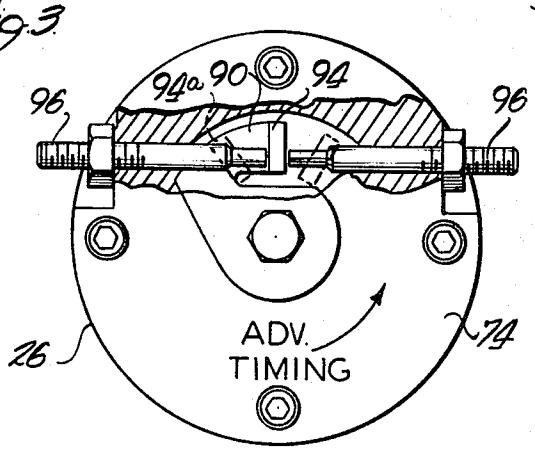


Fig. 10

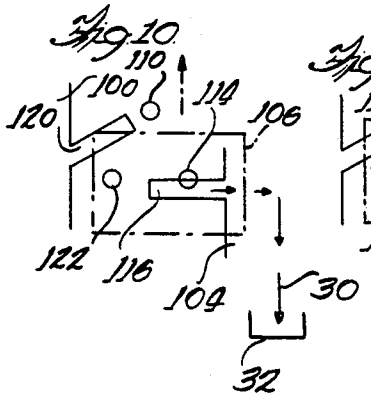


Fig. 11

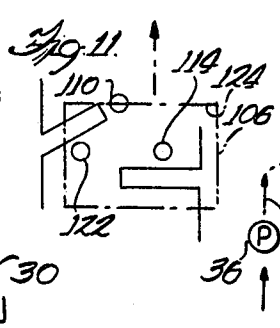


Fig. 12

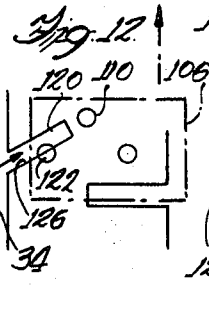
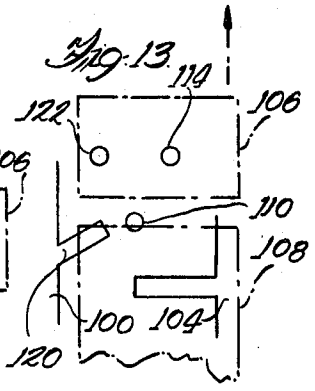


Fig. 13



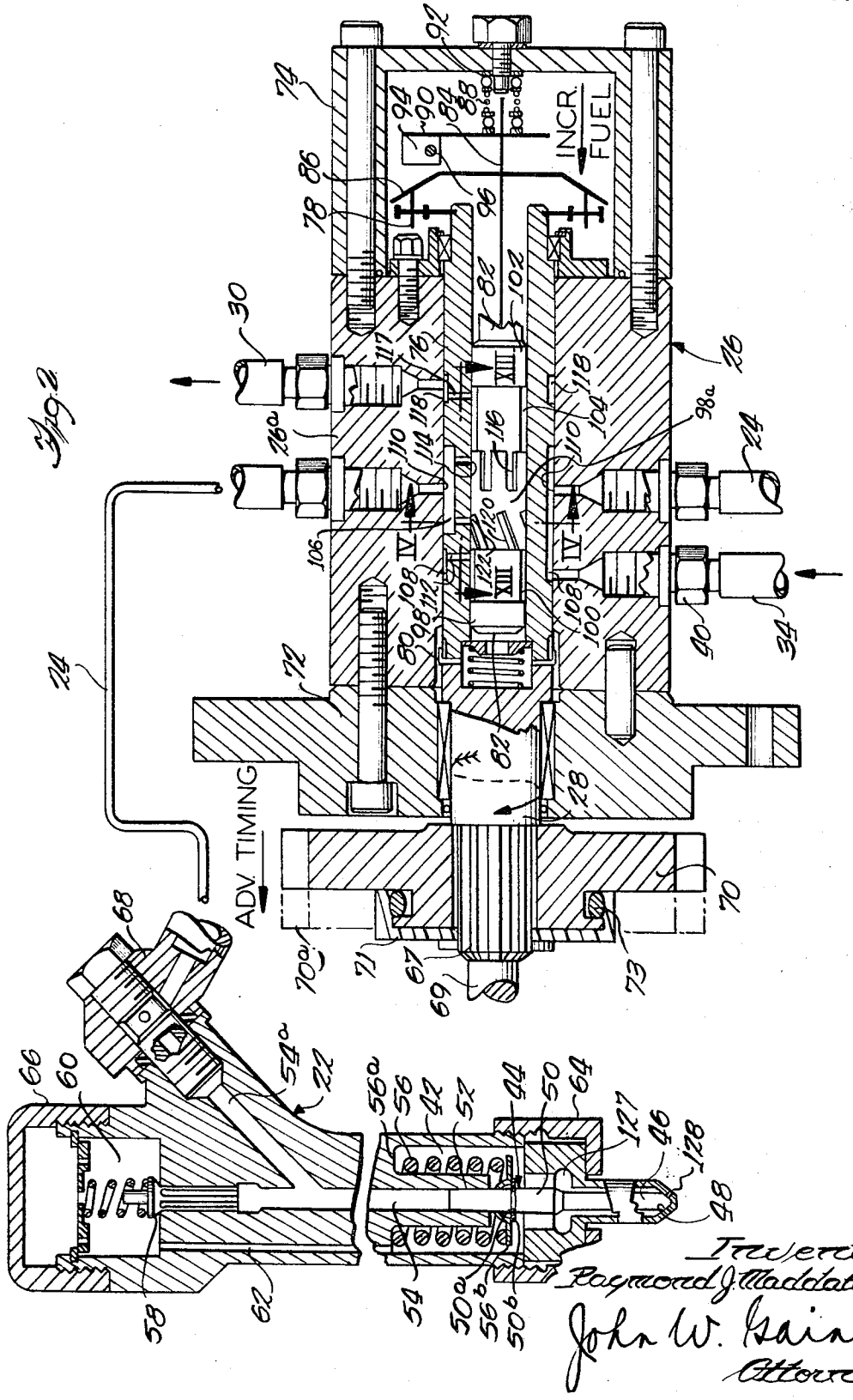
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FUEL INJECTION SYSTEM AND DISTRIBUTOR VALVE THEREFOR

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Fig. 2



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FUEL INJECTION SYSTEM AND DISTRIBUTOR VALVE THEREFOR

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Fig. 4

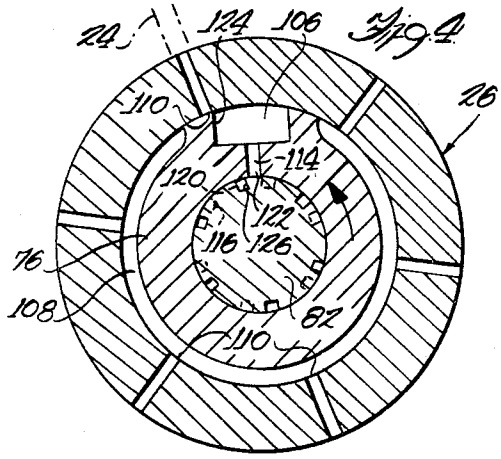


Fig. 5

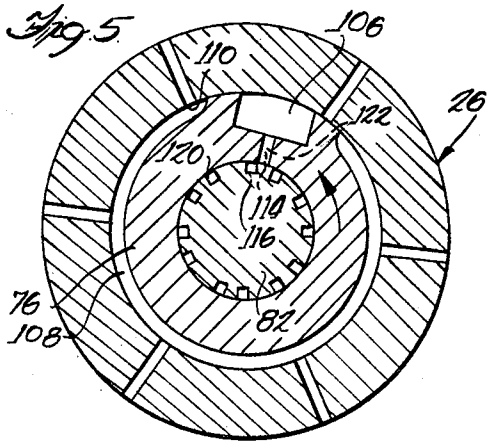


Fig. 6

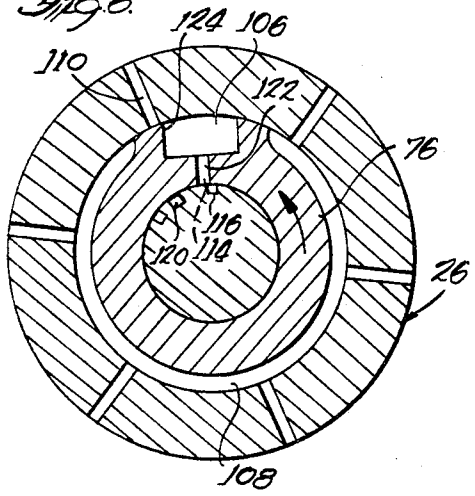


Fig. 7

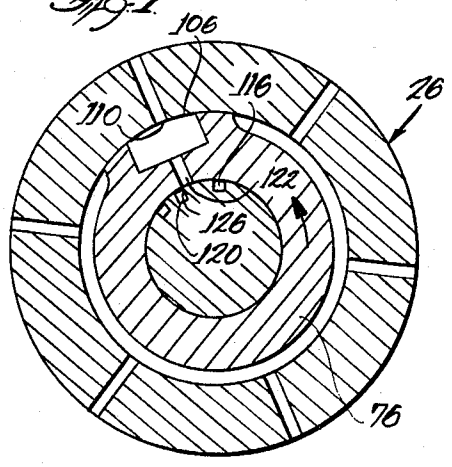


Fig. 8

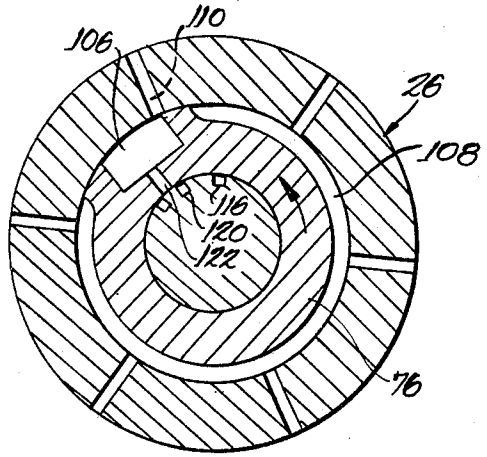
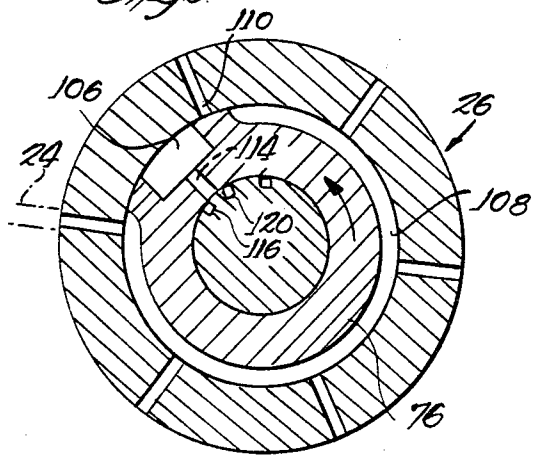


Fig. 9



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United States Patent Office 3,398,730
Patented Aug. 27, 1968

3,398,730

FUEL INJECTION SYSTEM AND DISTRIBUTOR VALVE THEREFOR

Raymond J. Maddalozzo, Chicago, Ill., assignor to International Harvester Company, Chicago, Ill., a corporation of Delaware

Filed Oct. 31, 1966, Ser. No. 590,779
5 Claims. (Cl. 123—139)

ABSTRACT OF THE DISCLOSURE

A rotary distributor valve driven in timed relation with the engine supplies a plurality of individual lines, each connected with a fuel injector nozzle from an inlet connected with a source of fuel under pressure. During operation of the valve the individual lines are sequentially subject to full pressure to charge an accumulator chamber in each nozzle, then to drain pressure to initiate injection from the nozzle and then an intermediate pressure to terminate injection. The period between initiation and termination is determined by a governor.

This application describes certain of my improvements on the fuel injection structure of Heintz U.S. Patent No. 3,187,733, the disclosure of which is incorporated in entirety herein by reference.

My invention relates to the fuel injection system of a compression ignition engine, and specifically to a central control or distributor valve which coordinates the operation of the injection system. It more specifically relates to a port-type rotary valve common to the injectors for all cylinders for supplying and operating them in sequence, each to deliver a metered quantity of fuel injected over the course of an accurately predetermined number of degrees of rotation of that valve.

An object of the present invention is the provision, in a multinozzle fuel injection system for an engine, of a rotary distributor valve connected in common to all nozzles, and hydraulically actuating them so that the nozzles inject fuel under high pressure from their accumulators for the right time and duration of injection.

A further object in line with the preceding objective is to provide a system of multinozzles having a rotary distributor valve therefor, and being hydraulically operated by the latter by a rapid pressure-wave actuation principle affording immediate response of each nozzle in initiating and stopping its injection.

Another object of the invention in line with the preceding objective is the provision of a rotary distributor valve for a plurality of accumulator type injection nozzles, the valve being internally ported in a manner for not only causing each of the nozzles in rotation to discharge precisely metered fuel from and at the high pressure of the accumulator of that nozzle, but also insuring that the accumulators of all nozzles not momentarily engaged in the fuel injection phase of the cycle freely communicate through the valve with a source of rail pressure. The latter nozzles are thus afforded a comparatively long recharging or replenishing phase in the cycle, so that fuel will be stored in the accumulator of each nozzle in about the same quantity as injected during the preceding injection phase of the cycle of that nozzle.

In an accumulator type fuel injection system employing differential type nozzles, a disadvantage exists in adapting the system to high speed, multicylinder engines wherein the time interval is very short for distributing and injecting metered quantities of fuel to and from the various nozzles in succession. A conventional distributor valve, provided in common to all nozzles, encounters

stringent conditions and difficulties in the effort to cause injection in each nozzle to start and to stop at precisely the right times, and under varying loads, and throughout the entire range of speeds of the high speed engine.

This invention materially reduces or substantially eliminates the foregoing difficulties through provision of the present injection system and distributor valve therefor, and a novel pressure-wave disseminating, internal porting arrangement in the distributor valve in the system, all as will now be explained in detail.

Features, objects, and advantages will either be specifically pointed out or become apparent when, for a better understanding of the invention, reference is made to the accompanying drawings which form a part thereof, and in which:

FIGURE 1 is a left side elevational view of a fuel injection system as applied to a compression ignition engine and embodying the present invention;

FIGURE 2, being partly schematic, is a longitudinal section view of the distributor valve and one of the associated nozzles in the system of FIGURE 1;

FIGURE 3 is a rear end view, in elevation, of the distributor valve shown in FIGURES 1 and 2;

FIGURE 4 is taken along the section lines IV—IV of FIGURE 2 and is a section showing the valve adjusted for idling the engine;

FIGURE 5 is somewhat similar to FIGURE 4, but shows all grooves;

FIGURES 6, 7, 8 and 9 follow FIGURE 5 in a sequence showing only the fuel injection phase of an entire inject-refill cycle for one engine cylinder; and

FIGURES 10, 11, 12 and 13 are schematic developed views, taken generally along the section lines, XIII—XIII of FIGURE 2 so as to show the injection phase of a cycle from another viewpoint.

In the present multiinjector system, the accumulator chamber in each injector receives fluid under rail pressure over a long period of the cycle time thus storing all of the potential energy in the accumulating fluid so collected. Opening of an injector nozzle releases considerable hydrostatic energy causing the fluid to escape under high pressure and in a metered amount which must spray out in an exceptionally short period of time. Each injector, following injection, is immediately connected on recharge by the distributor valve. A six-cylinder compression ignition engine is herein illustrated to show the adaptability of the present valve to multicylinder work, but for the sake of brevity only the cycle for one injector nozzle is specifically described.

FIGURE 1

An injection system for the engine **20** includes injector nozzles **22**, one for each cylinder which are not shown. The nozzles **22** inject according to the engine firing order and each has a single supply-control fuel delivery line **24** between it and one of a plurality of equally circumferentially spaced apart points on a rotary distributor valve **26** which includes a housing designated **26***a* in FIG. 2. A gear-driven input shaft **28** is mounted in the valve **26** so as to provide a speed which is one-half of the engine crankshaft speed and the rotational angle is accurately phased to the position of the pistons, not shown, in the engine **20**.

A small-capacity drain line **30** leading from the valve **26** returns fluid, mostly leakage fluid, to a fuel tank **32** in the system. A pump line **34**, drawing fluid from the fuel tank **32**, includes an exceptionally high pressure pump **36** driven by the engine, an accumulator **38** which collects the fluid under the high or rail pressure, and a supply fitting **40** at the end of the line **34** at which it delivers rail pressure to the distributor valve **26**.

FIGURE 2.—Nozzle structure

Internally of each nozzle **22**, there is an accumulator chamber **42** at the front surrounding a poppet valve assembly generally indicated at **44**. The poppet assembly **44** includes a needle **46** which rests on a spray jet seat **48** and which has an enlarged head **50** of which the upper portion **52** serves as a piston in a central longitudinal passage **54** within the nozzle. A semispherical member **50***a* surrounding the head **50** rests against a snap ring **50***b* on such head. A heavy, valve-seating spring **56** within the accumulator chamber **42** acts between a shoulder **56***a* in the nozzle and a spring seat **56***b* on the member **50***a* in a direction constantly to bias the needle **46** in closed position on the seat **48**.

Within an upper portion of the central passage **54**, a poppet check valve **58** establishes one-way communication between that passage and a rear section **60** of the accumulator chamber. An offset, longitudinal passage **62** within the nozzle interconnects the rear section **60** with the main portion of accumulator chamber **42**. A cap **64** at the front end of the injector **22** and a headcap **66** at the head or rear end seal the injector against leakage.

The fuel delivery line **24** for each nozzle is connected thereto by a banjo fitting **68** which establishes communication with the central passage **54** through a branch **54***a* thereof.

FIGURES 2 and 3.—Distributor structure

A bevel gear **70**, forming the last step of a stepped gear train driven by the engine, is axially adjustably carried by the valve input shaft **28** on a set of straight splines **67**. The stepped gear train further includes a wide bevel pinion **71** which is in a fixed position on a driving shaft **69** therefor and which pinion **71** is in continuous mesh with the bevel gear **70**. A timing adjustment fork **73** schematically indicated within a groove on the bevel gear **70** adjusts the rotational and axial position of the gear **70** by causing the gear to slide along the teeth of the pinion **71**. The timing is advanced or retarded depending upon whether the gear **70** is retarded pursuant to axial advancement in the direction of the arrow toward or into the advanced position shown by the broken lines **70***a*, or axially withdrawn into the retarded position as shown by the solid lines.

The valve housing **26***a* rotatively supports the shaft **28**. Said valve housing includes a supporting and attachment cap **72** at the front end adjacent the gear **70** and a rear cap **74** at the opposite end.

A sleeve-shaped rotor **76** is mounted within the valve housing and carries governor flyweight parts **78** which are at one end of the rotor and which are included in a speed control mechanism. The rotor **76** is driven by a connection **80** formed of interlocked jaws between it and the input shaft **28**. The rotor **76** supports a relatively fixed inner stator part **82**.

The inner stator part **82** cooperates with the rotor **76** in a mechanical manner, and in that connection the stator part **82** is rigid with an axially extending shaft **84**. For purposes of controlling axial adjustment of the stator part **82**, the shaft **84** carries a governor cup **86** which restrains movement of the flyweights **78** radially outward of such cup which is included in the governor mechanism in a manner to increase fuel when the flyweights **78** collapse and the governor cup **86** accordingly moves with the shaft **84** in the direction of the arrow indicated in FIGURE 2.

A governor spring **88** is schematically shown between the end cap **74** and a timing plate **90** and serves to bias the shaft **84** in the fuel increasing direction. An adjustment fork **92**, schematically shown, can be moved by the operator in a direction tending to collapse the spring **88** and thus advance the governor condition from idle setting to a full power setting.

The governor mechanism is schematically shown herein for ease in understanding. In actual practice, various

idling, full power, and overload control devices of conventional construction will be provided to avail precise control.

The rotational position of the shaft **84** is controlled by the timing plate **90**, which carries a transverse lug **94**. Two opposed bolts **96** which are threaded through the end cap **74** clamp the lug **94** in various angular positions to produce the timing desired. Timing is advanced (FIGURE 3) by releasing one of the bolts and tightening the opposed bolt, moving the lug **94** in the direction of the arrow.

A center land **98***a* on the inner stator part **82** defines, between it and an end spool **98**, a first stator chamber **100**. The center land defines, between it and another end spool **102**, a second stator chamber **104**. The outer periphery of the rotor **76** is grooved to form a cul de sac or rotor chamber **106** and is further grooved to form a pressure chamber **108** separate from chamber **106** and of which a portion is in alignment with the chamber **106** circumferentially of the stator **82**, FIGURES 2 and 4, and which portion is of sufficient circumferential extent to concurrently register communicatively with all but one of the supply-control line ports **110**.

Rail pressure is maintained in pressure chamber **108** by the pump line **34**, and this pressure is communicated from chamber **108** through a passage **112** to the first stator chamber **100** and thence into helical metering grooves **120** which successively communicate radially through a stator port **122** with the cul de sac **106**. The pressure chamber **108** which is maintained by the pump concurrently supplies the rail pressure through all but one of the passages **110** and thence through all but one of the supply-control lines **24** to all injectors except the one which is then being controlled to deliver a metered quantity of fuel to its associated engine combustion chamber, in a manner hereinafter explained in detail.

Periodically, the rotor chamber **106** is opened to drain in a path including a spill port **114**, individual straight spill grooves **116** formed on the inner stator part **82** parallel to the axis thereof, the second stator chamber **104**, a connecting radial passage **117**, an annular groove **118** formed on the rotor **76**, and the drain line **30** which is at low pressure. At other times the revolving rotor chamber **106** communicates with the successive delivery line port **110** each connected to an associated nozzle **22** by an associated delivery line **24**. During the last portion of this communication through a delivery line port **110** with each nozzle, the first stator chamber **100** also communicates with the rotor chamber **106** through a path including an associated one of the helical metering or pressure grooves **120** and the metering port **122** which then registers with such metering groove.

An examination of FIGURES 11 through 13 will make it apparent that when the governor mechanism slides the stator part **82** in the fuel increasing direction indicated by the arrow in FIGURE 2, there will be a longer interval of time between the start of registry of the revolving and then pressure-relieved rotor chamber **106** and a delivery line port **110** (negative injection-causing pressure wave thus incurred in line **24** to the associated nozzle to open the nozzle poppet) and the start of registry of the port **122** of chamber **106** with a metering or pressure groove **120** (positive poppet-reseating wave thus incurred in line **24**, terminates injection). The converse is similarly true, wherefore the metered quantity of fuel is decreased as a direct function of movement of the inner stator part **82** by the governor mechanism in a direction to the right as viewed in FIGURE 2.

FIGURE 4.—Metering and recharging

If the width of rotor chamber **106** is considered to extend in the axial direction of the sleeve or rotor **76**, then said chamber in the lengthwise sense is comparatively short, i.e., it has minor angular extent annularly, amount-

ing only to about 32.5° in the illustrative embodiment of the invention. The groove-shaped pressure chamber **108**, though shallower, is a comparatively long one in terms of angular extent, amounting to 300° in such illustrative embodiment of the invention. The significance of this is that the pressure chamber **108** is always keeping five of the delivery line ports **110** (angular spacing 60° apart) charged or charging with rail pressure, insuring that their respective nozzle accumulator chambers **42** are completely refilled prior to each injection portion of their operating cycle.

The rotor **76** is shown in FIGURE 4 axially adjusted for engine idling. In that position of adjustment, an equalizing edge **124** of the advancing rotor chamber **106** uncovers the next delivery line port **110** at a point which can occur as little as 4° ahead of the point at which the port **122** communicating with the rotor chamber **106** uncovers a poppet reseating edge **126** of the pressurized metering groove **120**. The two points referred to establish the start and termination of injection of a metered quantity of fluid, as will now be more fully explained.

FIGURES 5, 6, 7, 8 and 9.—Injection phase of the cycle

In FIGURES 5, 6, 7, 8 and 9 the inner stator part **82** is illustrated in the full load position, which is between the engine idling position of FIGURE 4 and an overload position assumed by the valve in FIGURE 2. Before the start of the injection phase of each injector cycle, the cul de sac or rotor chamber **106** undergoes three changes of internal pressure, namely from a higher pressure to a drain level or residual pressure, thence to an equalizing or high pressure, thence to a poppet reseating pressure which is above the high pressure, and thereafter it reaches the higher pressure first referred to which is a pressure between the poppet reseating pressure and rail pressure. Actually the chamber **106** never attains full rail pressure, which is the highest in the system.

On the way to the delivery line port **110** beginning in FIGURE 5, the far port **114** which communicates with the rotor chamber **106** is about to encounter one of the straight spill grooves **116**. At this time, that is, just before such encounter, the chamber **106** is at the higher pressure referred to, whereas the delivery line **24** and the delivery line port **110** continue communication with the pressure chamber **108** and with the accumulator chamber in the nozzle associated therewith to cause this accumulator chamber to be fully charged at rail pressure. When the far port **114** encounters groove **116** the pressure in chamber **106** will be diminished to said drain or residual pressure which will prevail into the FIGURE 6 position.

In FIGURE 6, the equalizing edge **124** of the rotor chamber **106** has advanced and is about to uncover the delivery line port **110**, the chamber **106** with its pressure reduced to drain pressure having at this time either entered into sequestration or started to enter into sequestration from the spill groove **116** due to rotor rotation. In the latter connection, it is sometimes desirable if not essential to provide for an overlap so that the delivery line port **110** and the spill groove **116** intercommunicate through the chamber **106** for a short overlap period to insure that the nozzle pressure drop is sufficient to start injection.

At and after the point at which the equalizing edge **124** uncovers the delivery line port **110**, the fluid in the delivery line and inside the nozzle bore **54** is diminished to an equalizing pressure common with that in the rotor chamber **106**. Depending upon the referred to port-and-groove overlap being designed into the system or not, the equalized pressure can range upwardly from slightly above the drain pressure level, to a value only slightly below the initial nozzle pressure (rail pressure) because of the small capacity of the rotor chamber. The equalized condition is illustrated in FIGURE 7. The resulting diminution of pressure provided for in the nozzle bore **54**, allowing it to reach the so-called equalizing pressure, is always sufficient to enable the pressure in nozzle accumulator chamber **42** to prevail over the pressure in said bore **54**, thereby lifting both the valve needle head **50** and the needle **46** so as to commence fuel injection.

Continued rotation of the ports in the direction of rotation indicated by the arrow in FIGURE 7 toward the FIGURE 8 position, causes the near port **122** leading into the rotor chamber **106** to uncover the poppet reseating edge **126** of the pressure or metering groove **120**. Fluid flow into the rotor chamber **106** establishes poppet reseating pressure therein, thereby terminating injection by the associated nozzle.

In FIGURE 8, it is seen that the rotor chamber **106** by progressive movement is blocked from the metering groove **120** and will be blocked from the delivery line port **110**, in that order, leaving the chamber **106** and the delivery line each sequestered at the higher pressure referred to. The trailing or sequestering edge of the chamber **106** blocks it from the port **110** as illustrated in FIGURE 9.

FIGURE 9 shows the valve parts shortly after completion of the injection phase of the cycle for the nozzle referred to, the pressure chamber **108** approaching the port **110** in the direction of the arrow in order to communicate and thereby recharge the depleted pressure in the nozzle to rail pressure, in the process of refilling the accumulator chamber therein. The far port **114** connected to the rotor chamber **106** is about to come into registry with the next spill groove **116** for reducing the chamber **106** back to drain level or to the aforesaid residual pressure somewhat below rail pressure in the nozzle accumulator **42** and bore **54**. Thus, the valve parts advance 60° in progressing from the FIGURE 5 showing to the FIGURE 9 showing, being readied to undertake the injection phase of the cycle for the next nozzle whose delivery line **24** is indicated in broken lines in FIGURE 9.

FIGURES 10, 11, 12 and 13.—Energy and fluid transfer

The volume of the rotor chamber **106** is very small and the pressure differences available for charging and for depressurizing it are very high. For that reason the time lag between successive port communications with this rotor chamber and the respective consequential establishment therein of the successive pressure values for causing discharge of metered quantities of fuel from the nozzles is so slight as to be practically instantaneous.

FIGURE 10 shows a transition period of the valve parts intervening between their relative position shown in FIGURE 5 and their relative position shown in FIGURE 6. Registration between the straight spill groove **116** and the far rotor chamber port **114** produces primarily an energy transfer, because the chamber **106** retains just about the same quantity of fuel at the higher pressure and at the drain pressure.

FIGURE 11 shows the valve parts in an intervening position from their relation shown in FIGURE 6 and their relation shown in FIGURE 7. The illustrated registration between the rotor chamber **106** and the delivery line port **110**, due to uncovering of the latter by the equalizing edge **124**, results principally in an energy transfer, because the chamber **106** represents a dead end for the delivery line and holds approximately the same quantity of fluid at the drain level pressure as at the equalizing pressure mutually reached by the delivery line and the rotor chamber **106**. This energy transfer is in the form of a negative pressure wave which is immediately communicated to the nozzle for opening the poppet and initiating injection.

FIGURE 12 illustrates the valve parts in a position intervening between their relationship as shown in FIGURE 7 and their relationship as shown in FIGURE 8. Registration between a pressure or metering groove **120** and the near port **122** connected to the rotor chamber **106** results principally in substantially only an energy transfer, because the poppet reseating pressure is restored by

substantially the amount of fluid which originally transferred from the delivery line into the rotor chamber **106** so as to equalize the pressures.

FIGURE 13 shows the valve parts in the recharge relation which occurs just subsequently from the position shown in FIGURE 9. Commencement of registration between the delivery line port **110** and the pressure chamber **108** on the rotor starts recharging of the nozzle accumulator chamber **42**, which continues throughout the intervening degrees of rotation of the value until the chamber refills completely with a replenishing amount of fluid equal to the amount rejected.

FIGURE 2.—Nozzle cycle

Each negative pressure wave which is communicated to a nozzle to initiate injection allows the pressure in the accumulator chamber **42** to partially collapse the spring **56** in the nozzle by forcing the needle head **52** and other parts of the poppet assembly **44** upwardly as viewed in FIGURE 2. The needle **46** moves from the poppet spray jet seat **48** and the internal pressure of the expanding fuel causes it to inject and, at the same time, the pressure acts against the opening needle **46** and the portion of the needle head **50** exposed in an annulus **127** in the fluid passage, so as to fully open the nozzle orifices **128** for fuel jet discharge.

The subsequently following positive pressure wave through the bore branch **54***a* increases the pressure on the upper end of the needle head **50**, allowing supplementing the force of the spring **56** to reseat the needle **46** and close the poppet seat orifices **128**.

During recharge, rail pressure existing in the passage **54** within the nozzle **22** keeps the spring-pressed check valve **58** off its seat until fluid flowing therethrough causes the accumulator chamber **42** and the rear section **60** thereof to attain rail pressure. During this period, the needle **46** remains seated.

EXAMPLE

Following is an example of values for the diesel fuel system foregoing:

Number of engine cylinders	6.
Engine operation	4 stroke cycle.
Rail pressure developed by pump **36**	20,000 p.s.i.
Number of helical slots or grooves **120**	6.
Number of straight slots or grooves **116**	6.
Injection angle range	0°–19° of valve rotation.

It is apparent from the foregoing that the parameters of injection are precisely controlled over a wide range. The precise time of injection, to occur as illustrated in FIGURE 11, is appropriately set relative to engine piston motion by angularly adjusting the gear **70** in the range illustrated in FIGURE 2, and by equally angularly adjusting the timing plate **90** in the range illustrated in FIGURE 3. The desired negative pressure wave is communicated to the nozzle instantaneously with opening of the port **110** to the rotor chamber **106** and there is no inertia problem.

The precise cut-off of injection, to occur by reapplication of pressure as illustrated in FIGURE 12, is appropriately set relative to engine piston motion by longitudinally adjusting the position of the metering grooves **120**. The desired positive pressure wave is communicated to the nozzel instantaneously with opening of the port **122** to the groove **120** and there is no inertia problem.

I claim:

1. A liquid fuel delivery distributor valve for cyclically controlling the fuel delivery between a source of fuel under high pressure and an accumulator type elastic pressure-wave actuated fuel injection nozzle of an operative compression-ignition internal combustion engine comprising:

(a) an outer stator member having a first passage communicatively connected to said source and a second passage communicatively connected to said nozzle,

(b) a rotatable valve element having a rotor chamber disposed therein and driven by said engine in phased relation, said chamber being positioned in relation with said second passage to alternately register and deregister with respect thereto,

(c) an inner stator member disposed in said valve element,

(d) first means in the inner stator member and said valve element adapted to reduce the pressure in said chamber during a cycle-initiating first operational period,

(e) said chamber registering with said second passage during a subsequent second operational period for inducing an elastic negative-pressure wave to actuate said nozzle thereby commencing injection of fuel under pressure into said engine,

(f) second means in said inner stator member and said valve element positioned to communicate said first passage with said rotor chamber while such rotor chamber is in communication with said second passage during a subsequent third operational period for inducing an elastic positive pressure-wave in said second passage to de-actuate said nozzle after a predetermined quantity of fuel has been injected into said engine,

(g) and a third means in said valve element positioned to communicate said source with said second passage following deregistration of the rotor chamber with respect to said second passage and during a final operational period of said cycle for charging fuel into the accumulator of said nozzle

whereby said distributor valve controls fuel delivery and hydraulic actuation of said injection nozzle in phased relation through said second passage for operating said engine.

2. The invention of claim **1**, the first means comprising a spill groove on the inner stator member, and a spill port in the valve element connected to said rotor chamber and positionable in registrable relation with the spill groove during said cycle initiating first operational period.

3. The invention of claim **2**, the second means comprising a pressure groove on the inner stator member, and a second port in the valve element connected to said rotor chamber and positionable in registrable relation with the pressure groove during said third operational period.

4. The invention of claim **3**, said pressure and spill grooves formed on the inner stator member comprising, respectively, a metering groove which is a helical groove and an equalizing groove which is a straight groove.

5. The invention of claim **4**, said inner stator member and said valve element having means of effecting relative rotational adjustment and making relative longitudinal adjustment therebetween to adjust injection timing and fluid metering.

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