March 14, 1967 R. J. MADDALOZZO 3,309,024

FUEL DELIVERY VALVE AND INJECTION APPARATUS EMPLOYING SAME

Filed Jan. 7, 1965 3 Sheets-Sheet 1

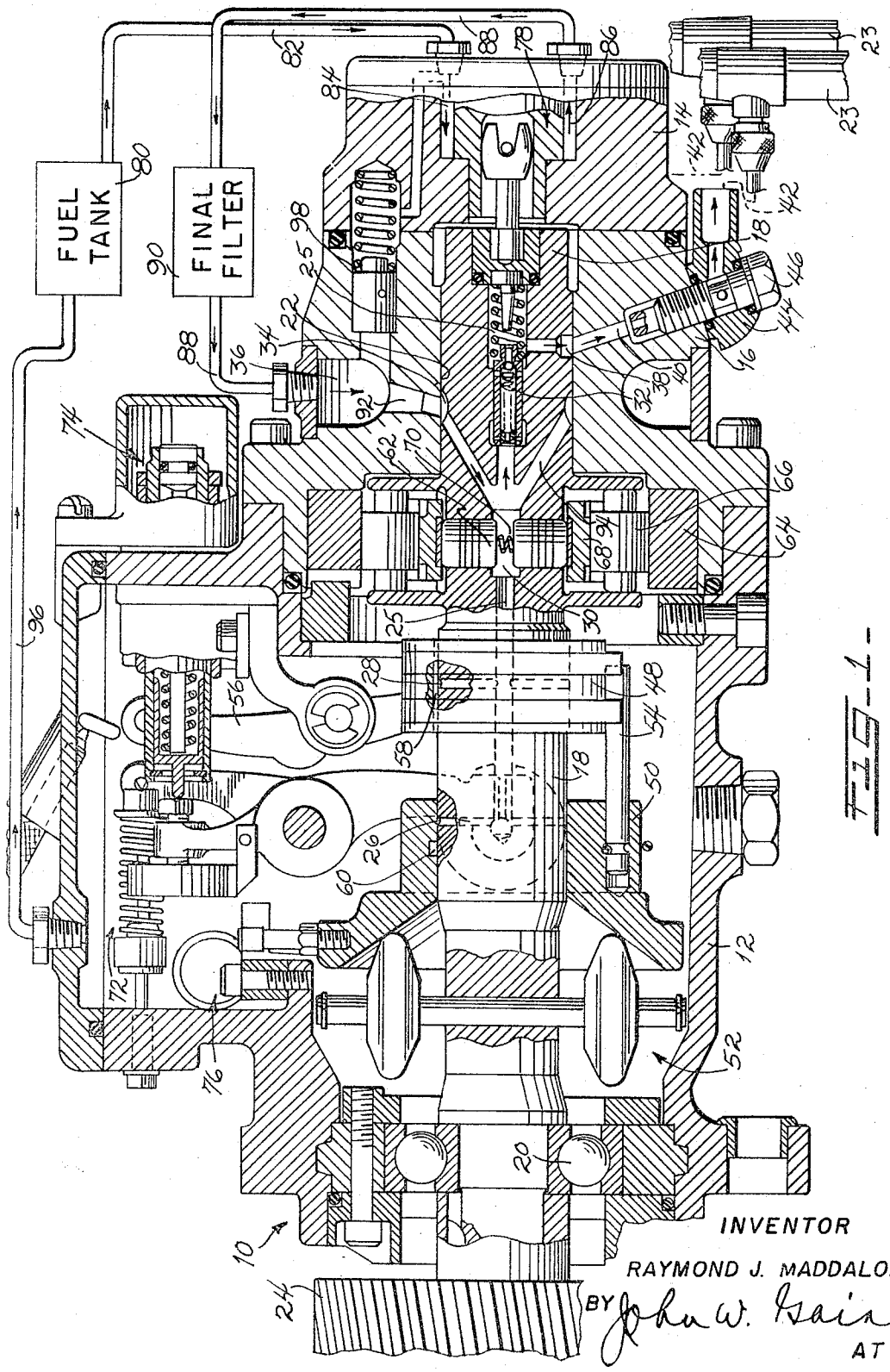


FIG. 1

INVENTOR
RAYMOND J. MADDALOZZO
BY John W. Gaines
ATT'Y.

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Filed Jan. 7, 1965 3 Sheets-Sheet 2

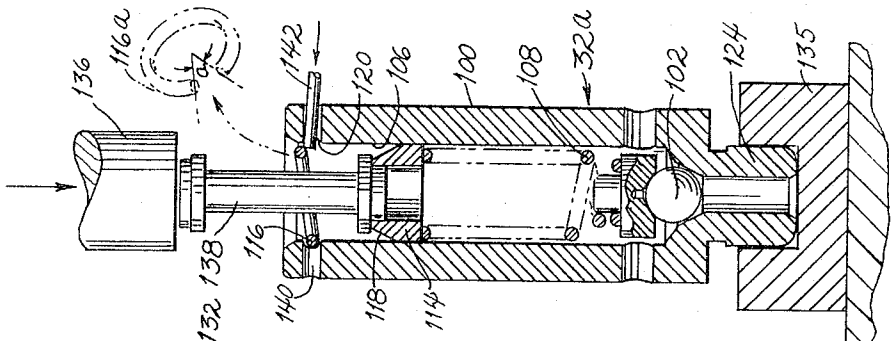


FIG. 4.

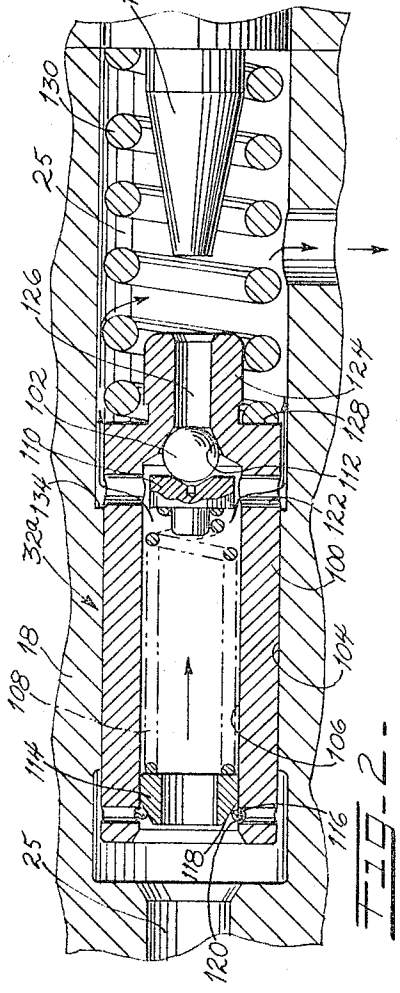


FIG. 2.

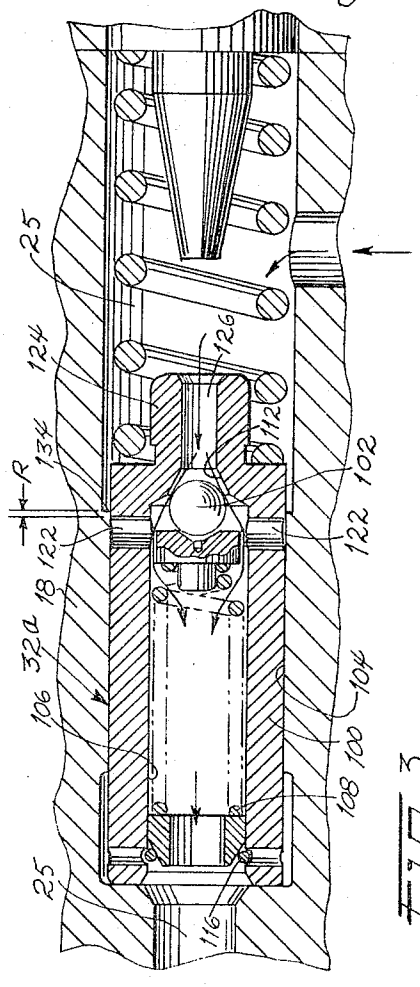


FIG. 3.

INVENTOR

RAYMOND J. MADDALOZZO

BY John W. Haines

ATT'Y.

March 14, 1967

R. J. MADDALOZZO

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3 Sheets-Sheet 3

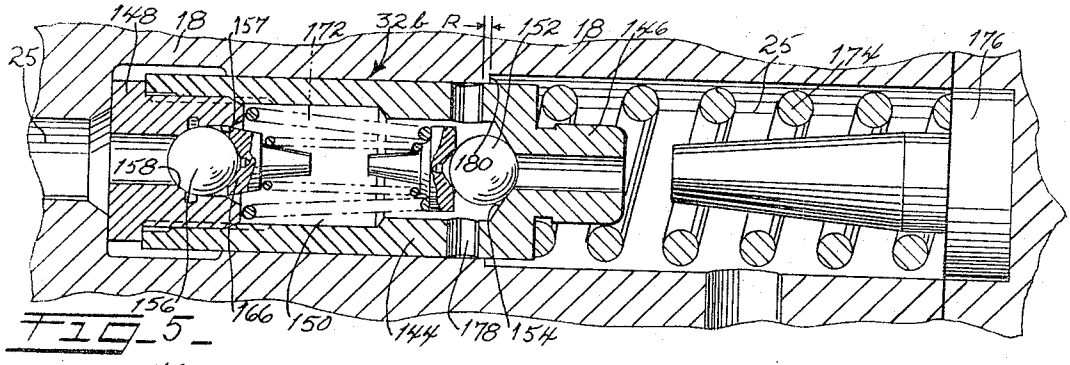


FIG. 5

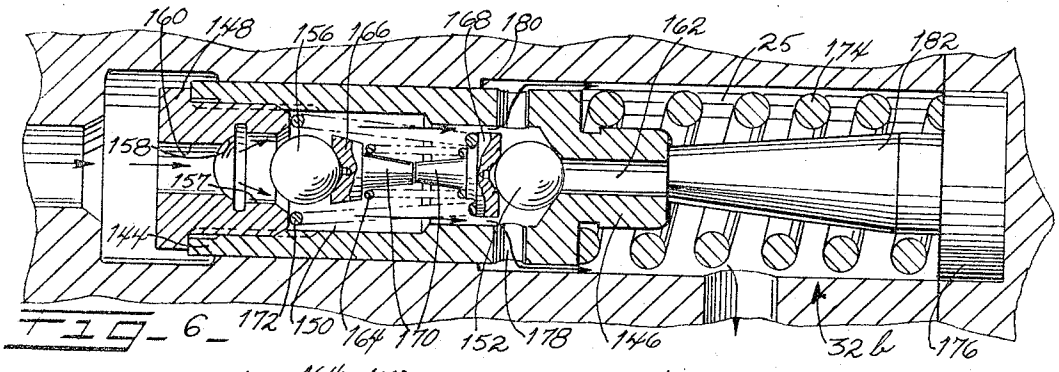


FIG. 6

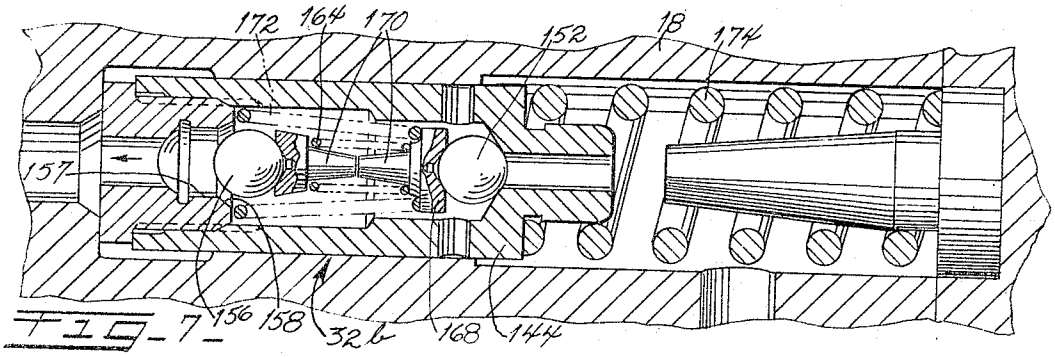


FIG. 7

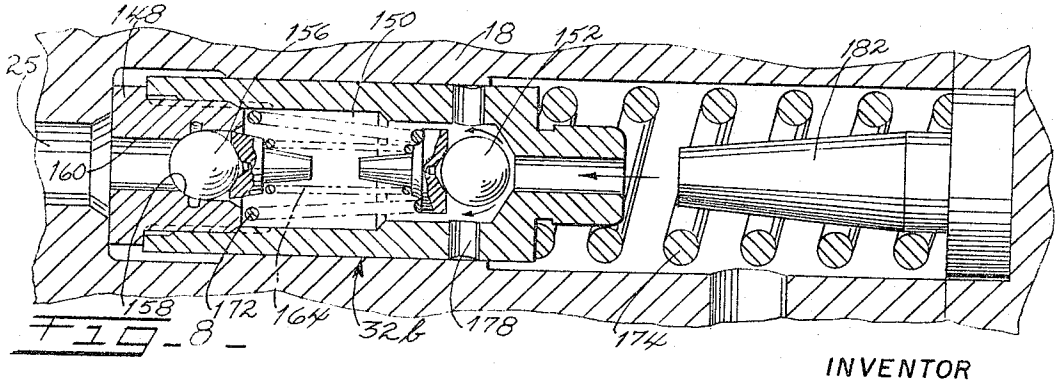


FIG. 8

INVENTOR

RAYMOND J. MADDALOZZO

BY John W. Haines

ATT'Y.

United States Patent Office

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FUEL DELIVERY VALVE AND INJECTION APPARATUS EMPLOYING SAME

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

Filed Jan. 7, 1965, Ser. No. 423,916
8 Claims. (Cl. 239—5)

This application is a continuation-in-part of copending Maddalozzo U.S. patent application Ser. No. 411,999 filed Nov. 18, 1964, and my description of the fuel pump delivery line structure and of other common matter disclosed therein is incorporated herein by reference.

The present invention relates to engine fuel injection apparatus. It more particularly relates to an improved form of the basic spring loaded retraction valve or delivery valve employed in such apparatus. The basic valve is included in an upstream position in a fuel delivery line leading to the injection nozzle valve or nozzle plunger provided on an engine cylinder, and functions to maintain valves after each injection takes place.

According to practice in the past, the fuel delivery line in a diesel engine handles the controlled intermittent flow of liquid fuel one way between the working chamber of a charging pump and a combustion chamber or the like. It is therefore the practice to have the delivery valve in the line function as a check valve in the output of the pump working chamber so as to prevent flow reversal. The lack of bidirectional flow gives rise to a serious difficulty in the event of a plugged nozzle in an engine cylinder, and the check valve operation of the delivery valve allows the delivery line to be pumped up to unrelieved pressures of an unwanted high value. Permanent damage can result due, among other things, to swelling and unbalanced hydrostatic loading in the distributor of the injection pump.

My invention, in accordance with a preferred embodiment thereof, materially reduces or substantialy eliminates the foregoing difficulty. Moreover in one modification of the invention, fluid under a transient excessive pressure from another cause, is released by being sequestered in the delivery line for employment on the next pumping stroke, without being allowed to recirculate to the fuel tank and thereafter to be re-introduced by the pump into the flow passage to the delivery line, all as will now be explained. Also, certain 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 following written description taken in conjunction with the accompanying drawings, which show certain preferred embodiments thereof and in which:

FIGURE 1 is a longitudinal sectional view of a port metering, opposed plunger, rotary distributor, diesel fuel injection pump embodying the present invention;

FIGURES 2 and 3 are fragmentary enlargement views of the retraction valve appearing in the longitudinal fuel passage of FIGURE 1, the valve being shown with the parts in sequential operating positions;

FIGURE 4 is a longitudinal sectional view of the valve being disassembled in a fixture; and

FIGURES 5, 6, 7 and 8 are longitudinal sectional views of a modification of the valve shown in sequence of operating positions.

More particularly in FIGURE 1 of the drawings, a housing or casing **10** of the present device is shown having three bolted together sections consisting of a pump **12** at one end, a primary pump housing **14** at the opposite end, and a so-called distributor head housing **16** assembled between the end sections or housings. An engine connected pump shaft **18** extends longitudinally through the device and is journalled for rotation in spaced apart roller type and sleeve type bearings **20** and **22**, respectively, fixed in the pump housing **12** and in the distributor head housing **16**. The particular fuel injection device illustrated is intended for a six cylinder, four stroke cycle, solid injection diesel engine, in which case the pump shaft **18** is driven in time to the engine and at a speed proportionate to crank shaft speed, specifically, at one half of the crank shaft speed.

The shaft **18** is the heart of the mechanism of several operating components within the housing **10**, all contributing in properly timed relation to supply metered amounts of fuel to the individual nozzles **23** which communicate with combustion chambers in the six engine cylinders, not shown. The power input to the pump is the torque applied to a gear **24** fast to the shaft **18**, the gear being connected to the engine.

The pump shaft **18** has a longitudinally drilled passage **25** which is intersected at one end by first transverse passages terminating in metering ports **26**, and by second transverse passages terminating in timing ports **28**, there being two metering ports and two timing ports.

A centrally located charging pump component within the so-called distributor head housing **16** is shown having a transversely disposed, cylindrically shaped pump chamber **30** formed in the shaft **18** and communicating with the passage **25** approximately midway between its ends.

A distributor component comprises a retraction or delivery valve **32** connected in the longitudinal passage **25** so as to unseat or open in a direction away from the pump component, a set of suction ports **34** which intersect the passage **25** on the pump side of the valve **32** and which periodically communicate with an annular chamber **36** which holds the so-called transfer fuel in the housing **16**, a distributor port **38** which communicates with the passage **25** on the opposite side of the valve **32** and which, during two revolutions of the engine crank shaft, registers at uniformly spaced apart intervals with each of six housing passages **40**, and a set of fuel delivery lines **42** leading from the six housing passages to the nozzles **23**. So-called banjo fittings **44** connect the lines liquid tight to the distributor head housing **16**, being secured to the latter by individual hollow connecting screws **46**.

A dual sleeve component which establishes the time of actuation of the retraction valve **32** includes a timing sleeve **48** which rotatably receives the pump shaft **18**, a governor sleeve **50** which rotatably receives the shaft **18** at a point between a governor component **52** and the timing sleeve **48**, and an alignment pin **54** connected between the sleeves **48** and **50** and holding them so as to be nonrotatable but relatively axially movable to one another. The timing sleeve **48** has a connection, not shown, to a pivoted arm termed a governor spring arm **56**.

Six straight cut slots **58** in the bore of the timing sleeve **48** are parallel to the pump shaft axis and cooperate with the timing ports **28** to spill fuel therefrom during short, equally spaced apart intervals. Six helically cut slots **60** in the bore of the governor sleeve **50** cooperate with the metering ports **26** to allow fuel to escape therefrom during

short, equally spaced apart intervals in the cycles. The straight timing sleeve slots **58** keep the beginning of delivery constant whereas the helically cut slots **60**, when the governor sleeve **50** is adjusted by the governor component **52**, vary the end of fuel delivery.

The charging pump component referred to provides the pressure for opening the retraction valve **32**, and comprises the transverse, cylindrical chamber **30**, a pair of rotary and reciprocatory plungers **62** slidably received in the opposite outer ends of the chamber **30**, a cam ring **64** fixed in the housing **16** in the plane of the plungers **62**, a pair of rotary and reciprocatory cam followers or rollers **66** which ride along a set of six inwardly protruding cams on the ring **64**, and an interposed tappet **68** connecting each roller **66** to a different one of the plungers **62** causing the plungers to periodically compress and then release an interposed return spring **70** upon respective pumping and suction strokes.

Further components in the device, indicated by general reference numerals are a control component **72**, a maximum torque component **74**, a timing plate component **76**, and a transfer or primary pump component **78**.

As an outline preliminary to an understanding of the structure, operating principles, port inter-relationships, and overall theory of the presently improved retraction valve **32**, diesel fuel from a tank **80**, which fuel is ultimately drawn into the pump chamber **30**, flows into a path including a fuel tank outlet conduit **82**, a transfer intake port **84** which is in the housing **14** and which communicates with the suction side of the pump mechanism of the primary pump **78**, a transfer outlet port **86** which communicates with the pressure side of the pump **78**, a transfer conduit **88** leading from the outlet port **86** and including therein a final filter **90**, the annular chamber **36**, a communicating charging passage **92** in the housing **16** which is connected during shaft rotation to the suction ports **34** in periodically timed relation, a set of diagonal intake passages **94** formed in the shaft **18**, and thence into the pump chamber **30**.

On the discharge stroke of the pump plungers, the fuel follows a sequence flowing in different directions through a three-way split path. During predetermined initial collapsing movement of the plungers **62** toward one another, fuel escapes from the shaft **18** into the pump housing **12** through the registering straight slots **58** until the spill is cut off by the sleeve **48** covering the timing ports **28**. During further collapsing movement of the plungers **62**, fuel pressure increases so as to open the retraction valve **32**, and fuel is forced through the registering one of the fuel nozzle delivery lines **42**. During final, radially inward collapsing movement of the plungers **62**, two of the helical slots **60** register with the two metering ports **26** so as to spill the balance of the pumped fluid into the pump housing **12**.

A drain conduit **96** returns a portion of the spilled fuel from the pump housing **12** to the fuel tank **80**. A spring loaded recirculation valve, not shown, connected between the housing **12** and the suction side of the primary pump component **78** returns another portion of the spilled fuel. Finally, a pressure regulating valve **98** connected between the annular chamber **36** and the suction side of the primary pump component **78** unseats and returns fuel to the latter whenever pressure becomes excessive on the pressure side of the pump component **78**.

The injector on an engine cylinder is supplied with fuel toward the end point of each compression stroke. When the injector on the cylinder sticks down, at that point at each successive compression stroke a metered quantity of fuel is pumped into the fuel delivery line, and is trapped there by closure of the retraction valve attendant with opening of the spill ports in the fuel pump. The pressure build-up (8,000 to 10,000 p.s.i.) internally expands the pump shaft on the downstream side thereof, closing the clearance between the shaft and sleeve type bearing bore as well as forcing the shaft to one side in the bearing. In some instances the lubrication film breaks down, causing pump failure due to shaft seizure. This difficulty is avoided in the embodiment of my invention next described.

In the embodiment of the retraction valve **32***a* illustrated in FIGURES 2, 3 and 4, the assembly includes a main or primary valve **100** having a generally cylindrical shape and a secondary ball valve **102** therein. The primary valve **100** fits within a guiding bore **104** forming an enlargement of the general longitudinal passage **25** within the shaft **18**. A lengthwise extending passage **106** within the valve **100** communicates at one end through the passage **25** with the fluid inside the pump chamber **30**, not shown. The valve **32***a* controls the flow passage leading from the pump chamber and including a portion of the passage **25**, the distributing port **38**, not shown, the individual housing passage **40**, and the fuel delivery lines **42**, not shown. The fluid transferred through the ball valve **102** escapes from the flow passage and is returned similarly to the spilled fluid of the charging pump so as to enter the suction side of either the charging pump or primary pump.

The mechanism of the ball valve **102** is self-contained within the primary valve **100**, and includes a helical compressing spring **108**, and a backing pad **110** which is between one end of the spring **108** and the ball **102** at that end and which cooperates therewith to keep the ball biased against a conical seat **112** formed in the passage **106**. At the opposite end the spring **108** thrusts against the flat end of a spring seat **114**. The seat **114** traps a circular snap ring **116** between the bevelled end **118** of the seat and an inside annular groove **120** that is provided in an end of the valve chamber which is formed by the lengthwise extending passage **106**. The diameter of the seat **114** decreases in the axially outward direction due to the bevel.

The seat **112** for the ball valve **102** is offset downstream from a set of side ports **122** extending radially through the wall of the primary valve **100**. The valve **100** has a reduced end portion **124** defining a one-way flow passage **126** and complementing the body of the valve **100** to present a spring seat shoulder **128**. A primary valve spring **130** has a fixed end engaging a combined valve stop and plug **132**, and a free end engaging the seat shoulder **128** for urging the primary valve **100** to move toward the pump into the closed position illustrated in FIGURE 3.

Although the retraction distance illustrated at R between the edges of the ports **122** and covering shoulder **134** at the end of the guiding bore **104** is shown with exaggeration so as to be perceptible at all, in an actual physical embodiment of the invention the primary valve **100** was a zero retraction valve. In other words, in the valve closed position the edges of the ports **122** were at or just barely past the plane of the shoulder **134**. Increase in the delivery line volume following cut-off was not only unnecessary but undesirbale, because the nozzles **23**, not shown, were cam driven, mechanical injectors in which mechanical force rather than injection pump pressure caused injection (20,000 p.s.i.) to occur. Hence there was no possibility of secondary injection, and reducing the delivery line pressure any further would have been wasted effort.

If the location of the ports **122** is charged, the valve **32***a* can be used with injection apparatus of a different type. That is to say, the ports **122** can be relocated by positioning them more to the left than is shown in FIGURE 3, in which case the retraction distance R can have a finite value such as several millimeters. In that case the valve **32***a* is readily adaptable for use with the so-called open or check valve type of injector nozzles, or the closed type nozzles in which injection is controlled by a spring biased valve between the pump and injection nozzle spray orifices and which are actuated to the open position by the fuel pressure.

The zero retraction valve just referred to, from an initial injector-filling, open position illustrated in FIGURE 2, affords a novel method of controlling the flow of fluid. The respective steps of such method comprise inducing a pressure drop in the delivery line while cutting off communication between the pump chamber and the line, which drop will normally result in approximately a predetermined residual pressure after each injection, and immediately thereafter, in reacting as illustrated in FIGURE 3 to each plugged nozzle situation, affording through the one way passage **126** and into the pump chamber an unlimited escape, from the line, of excess fluid down to a pressure higher than said predetermined pressure but preferably only slightly higher. For use with the cam driven, mechanical injectors referred to, the ball valve **102** unseats in the direction of the pump chamber, and the pressure in one instance was at or above 1600 p.s.i.

Obviously, if the fuel flow were maintained undiminished through the ports **122** as the primary valve **100** shifts into fully closed position, the portion of the volume of the delivery line vacated by the valve would be taken up by a like volume of incoming fuel. But an inherency of the valve **100** is not to maintain unrestricted flow, and the valve actually induces a pressure drop in the line while progressively cutting off communication between the chamber **30** and the fuel delivery line. It does so because it is physically enlarging the line while moving toward cut off. The pressure drop thus induced is more at the higher running speeds than at low load, low speeds for reasons which, at least in theory are explained hereinafter, and hence the predetermined residual pressure in the line is higher at the low load, low speeds.

In case the valve ports **122** are drilled at points giving the retraction distance R some finite value, for use with closed type or open type injection nozzle valves for example, the flow is controlled thereby in steps comprising: inducing a pressure drop in the delivery line while cutting off communication between a pump chamber and the line; enlarging the internal volume of said line a predetermined fixed amount, after cutting off communication between the line and chamber, under substantially all normal operating conditions so that a further drop occurs normally resulting in approximately a predetermined residual pressure; and immediately thereafter, in each plugged nozzle situation, affording unlimited escape from the line of excess fuel down to a pressure higher than the predetermined pressure, the latter pressure being at least a major fractional proportion of the regular nozzle injection pressure. In this situation, the predetermined residual pressure which is sought when the nozzles operate normally is a major fractional proportion of the normal nozzle injection pressure.

In FIGURE 4, the valve **32***a* during assembly and disassembly is placed with its reduced end **124** downwardly and is received in the socketed fixture **135** of an arbor press **136**. An adapter **138** is inserted and pressed downwardly, forcing the seat **114** inwardly away from the groove **120**. During disassembly, radial tool admitting holes **140** in the valve **100** receive a tool **142**, the holes intersecting the groove **120** and allowing the tool **142** to spring loose the snap ring **116** into its withdrawn position indicated by the broken lines **116***a*. The snap ring is a mutilated spring wire ring, of which the opposite ends are deliberately spaced apart by angular spacing *a* which in one instance was 60°. In this way, sufficient clearance is provided for the spring ring to circumferentially collapse, both for disassembly and for assembly of the valve **32***a*.

Complete disassembly comprises the foregoing steps, followed by opening the arbor press **136**, withdrawing and inverting the valve **100**, and axially outwardly removing the subassembly therein including the ball valve **102**.

The embodiment of my valve next described relieves the line from the transient, rebounding pressure waves resulting from injection cut off in open type and closed type injection nozzles.

In the embodiment of FIGURES 5, 6, 7 and 8, the primary valve **144** shown has a reduced end **146** at the right as viewed in the figures, and at the opposite end carries an auxiliary end part **148** threaded thereto and defining therewith an internal, through cavity **150** in the valve. A secondary ball valve **152** which opens toward the pump chamber is seated on a bevelled seat **154**. Another secondary ball valve **156** is received in a close fitting cylindrical bore **157**, and unseats away from the pump chamber and from its semi-spherical seat **158**. A one-way flow passage **160** is within the auxiliary part **148** and a coaxial one way flow passage **162** is in the reduced end **146** of the primary valve.

A common helical spring **164** biases both valves **156** and **152** upon their seats, the spring being a compressing spring and having an interposed backing pad **166** at one end engaging the ball **156** at that end and having another interposed backing pad **168** at the opposite end engaging the ball **152** at that end. A pair of confronting stops **170** connected to the respective backing pads **166** and **168** limits opening movement of the ball valve **156**. Another helical spring **172** acts to seat only the ball valve **152**, the spring thrusting at one end against the end part **148** and thrusting at the opposite end against the backing pad **168**. The primary valve **144** has a closing spring **174** which thrusts at one end against a plug **176** and which thrusts at the other end against a shoulder on the valve adjacent the reduced end **146** thereof.

The valve **32***b* includes one-way primary ports **178**. FIGURE 5 shows the valve with the ports **178** in closed position, but the showing is exaggerated so that the retraction distance R of the ports past a cut off edge or shoulder **180** is perceptible. Actually in practice the primary valve **144** has zero retraction, i.e., when the valve **144** halts, the edge of the primary passages **178** registers with or is just barely past the shoulder **180** so that the valve **144** does not enlarge the delivery line volume after cutting off injection flow of fuel.

The retraction valve **32***b* of FIGURE 5 retains the full closed position illustrated, during the suction phase of a pumping cycle and during both spill phases of the pumping cycle.

Beginning with a fully open, injection phase illustrated in FIGURE 6, the valve **32***b* controls the flow in three steps. The predominant closing spring **174** is the first to act as injection terminates. Such spring action on the valve **144** causes the one-way ports **178** to move past the shoulder **180**, progressively restricting the flow passage from the valve cavity **150** to the delivery line while enlarging the internal volume of the delivery line. Because of a drop in the pressure due to the valve, the pressure decreases to an initial residual pressure in the line at the point of cutting off of communication.

The latter point is illustrated in FIGURE 7, following which the spring action of the expanding spring **164** forces the ball valve **156** into and through the closely fitting bore **157**. This retraction motion reduces the pressure to a low pressure in the cavity **150** by enlarging the internal volume of the cavity and leaving a void therein. The valve **156** then comes to rest on its seat **158**.

The ball **156** is readily provided on a select-fit basis for reception within the bore **157**, and can be economically supplied because highly accurate, graded metal balls are cheap and in plentiful supply. This feature of my valve gives it a decided cost advantage over volume displacement valves of the cylindrical retraction type and top sealing retraction type, both of which are in common use for fuel delivery line work. The valve **156** is in effect closed when it first contacts the downstream end of the bore **157**, and the continued upstream movement thereof until it settles on the seat **158** is pure movement directly enlarging the volume of the cavity **150**.

With the valve **156** seated as illustrated in FIGURE 8, the parallel springs **164** and **172** thereafter accommodate by yielding with opening movement of the ball valve **152**, controllably releasing the rebounding pressure wave from the delivery line by allowing it to spill into, and the delivery line transient pressure to dissipate itself to a safe operating level in, the low pressure of the void-containing cavity **150**.

The opening value for nozzle valves varies, being of the order of 750 p.s.i. for a precup engine, and of the order of 2300 to 3000 p.s.i. for direct injection nozzle. In the instance of a direct injection nozzle requiring an opening pressure of 2300 p.s.i., the force of the spring **174** would be something less than 2300 p.s.i., e.g., 2100 p.s.i. The combined force of the parallel acting springs **164** and **172** is such that they allow the ball valve **152** to open at a major fractional proportion of the nozzle opening pressure, e.g., 1600 p.s.i. which in the above instance would amount to 70% of nozzle opening pressure. The more general range for opening is 1600–2800 p.s.i.

A stop **182** on the plug **176** limits the opening travel of the primary valve **144** by engaging the reduced end **146**.

It is apparent in FIGURE 5 that the internal volume of the fuel delivery line can be further increased by offsetting the one-way primary ports **178** so that the retraction distance R has a finite value such as several millimeters. In any case, when idling or at the lower speeds of the engine, the rebounding pressure wave resulting from closure of the injection nozzle valve is relatively mild. Therefore, the pressure pulsations in the line are small and little or no fuel is admitted by unseating the valve **152** and entering the cavity **150**.

At the higher speeds, particularly during full load operations when large quantities of fuel are being pumped, the rebounding pressure wave from the closing nozzle valve is comparatively great or severe. This wave not only tends to accelerate closure of the primary valve **144** of FIGURE 6, but has two further noticeable effects. Less importantly, it causes the valve **144** to close so fast that inadequate fuel escapes through the ports **178** within the shortened time period to properly replenish the delivery line volume being vacated by the closing valve **144**. The wave strikes the ball valve **152** with a peak pressure causing the valve to unseat and to spill a substantial quantity of fuel into the cavity **150**. However, the fuel so transferred is not bypassed to the input side of the charging pump or the primary pump, but merely trapped right within the delivery valve **32***b* for immediate re-entry into the delivery line at the start of the next injection stroke of the charging pump plungers.

From the foregoing it is seen that the valve **32***a* in accordance with FIGURES 2 and 3 provides a novel action preventing damage when an injector nozzle plugs, and allowing an engine to operate on all cylinders which remain active, with no harm to the injector pump. The retraction valve **32***b* in accordance with FIGURES 5, 6, 7 and 8 provides a novel action preventing damage from pressure fluctuations and also obviating secondary injection or nozzle dribble due to such fluctuations.

As herein disclosed, the invention is shown applied to the nozzles for a six cylinder engine. It is evident that it applies equally well to four cylinder and eight cylinder engines.

So also the bidirectional pressure relief type delivery valves herein are shown to embody cylindrical type primary valves, but self-evidently, my invention can be applied to the poppet or so called top seating type delivery valves.

What is claimed is:

1. In a fuel injection system subject to a rebounding pressure wave following injection, and including nozzle means to deliver the output and a pump chamber to supply the input of fuel thereto, and together having an interconnecting cavity with opening means therein, the method of controlling post-injection fuel flow beginning when at least one but not all opening means are open comprising:

closing the open opening means in said cavity;

drawing a void in said cavity by enlarging the internal volume thereof at the location of one closed opening means, while maintaining, at constant volume, the fuel which is immediately available as input to the nozzle means and which is in the path of the rebound wave; and

at least partially refilling the void by temporarily opening one opening means and controllably communicating thereinto the peak pressure of said rebounding wave so as to dissipate same to the maximum operating level desired.

2. In a fuel injection system, the method of controlling the flow of fluid in a delivery valve through-cavity and in a delivery line, connected in series between the pump chamber of a pump and an injection nozzle following each injection of fuel, which comprises:

progressively restricting the flow passage from said chamber and cavity to the delivery line while enlarging the internal volume of the delivery line, so that the pressure drop will result in approximately a predetermined residual pressure in said line after each injection of fuel at the time of the cutting off of communication from said chamber and cavity to the delivery line;

reducing the pressure to a low pressure in said cavity by enlarging the internal volume of said cavity; and

thereafter controllably releasing the rebounding pressure wave from the delivery line by directing it to spill into, and the delivery line transient pressure to dissipate itself to a safe operating level in, the low pressure of said cavity.

3. A fluid injection system including an injection nozzle for fueling an internal combustion engine, said system further including:

pumping means with a pump chamber for supplying fluid to the injection nozzle;

means forming a fluid delivery line interconnecting the pump chamber and the nozzle; and

an assembly of primary and secondary valves connected in the fluid line, the primary valve having guiding bore means therefor in the line wherein the valve is guided to open the bore and to shut off the bore while concurrently enlarging the internal volume of the line, said primary valve having a concentric flow cavity therein presenting separate valve seats at opposite ends;

one coaxial secondary valve within, and on the valve seat at one end of, the flow cavity in the primary valve, said secondary valve unseating in the direction away from the pump chamber or retracting in the direction thereof as to enlarge the internal volume of the flow cavity; and

one coaxial secondary valve within, and on the valve seat at the opposite end of, the flow cavity in the primary valve, and unseating in the direction of the pump chamber.

4. The invention of claim **3**, including spring means to bias a plurality of said valves toward the closed position, including a common seating spring mechanically interconnecting the secondary valves.

5. The invention of claim **4**, the first-named secondary valve of the invention being at the end of the flow cavity adjacent the pump chamber and the second-named secondary valve being at a remote end of the cavity, and each opening in the direction of the other.

6. The invention of claim **3**, the primary valve consisting of a cylinder type delivery valve and the secondary valves consisting of ball valves.

7. The invention of claim **5**, including two spaced apart, confronting stop members which back the respective secondary valves and which engage to limit relative

movement between the secondary valves during opening of at least one of the valves.

8. The invention of claim 7, characterized wherein said common seating spring surrounds the stop members.

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M. HENSON WOOD, JR., *Primary Examiner.*

V. C. WILKS, *Assistant Examiner.*

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,309,024 March 14, 1967

Raymond J. Maddalozzo

It is certified that error appears in the above identified patent and that said Letters Patent are hereby corrected as shown below:

Column 1, line 21, after "maintain" insert -- a desired residual pressure between the retraction and nozzle --.

Signed and sealed this 21st day of April 1970.

(SEAL)
Attest:

Edward M. Fletcher, Jr.
Attesting Officer

WILLIAM E. SCHUYLER, JR.
Commissioner of Patents