# United States Patent [19]

**Hussey**

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[45] **Sept. 30, 1975**


[54] **HYDRAULICALLY LOADED INJECTOR NOZZLE**


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[73] Assignee: **AMBAC Industries, Inc.**, Springfield, Mass.



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[52] **U.S. Cl.** ..... **123/139 R**; 123/139 AA; 123/139 AT; 123/32 JV; 123/139 E

[51] **Int. Cl.**[2] ..... **F02M 39/00**

[58] **Field of Search** ... 123/139 AT, 32 JV, 139 AK, 123/139 AW, 139 R, 139 E, 33 J



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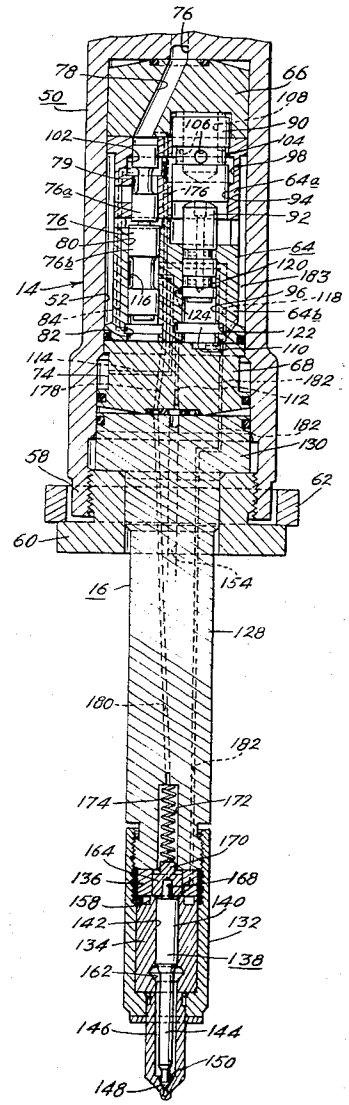
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[57] **ABSTRACT**

In a diesel engine fuel injection system wherein the fuel injection nozzle comprises a valve which is opened by the injection pressure of the fuel during the injection interval, the present invention provides a nozzle valve closing means characterized by a continuously applied hydraulic closing force which varies directly as a function of the injection pressure. The invention is particularly suited for use with a "common rail" injector system wherein the common rail pressure, which is amplified to provide the fuel injection pressure, is utilized to provide the hydraulic nozzle valve closing force. The invention provides a lower than maximum valve closing force during engine starting, idling and periods of part load operation when the injection pressure is desirably reduced.


**8 Claims, 5 Drawing Figures**



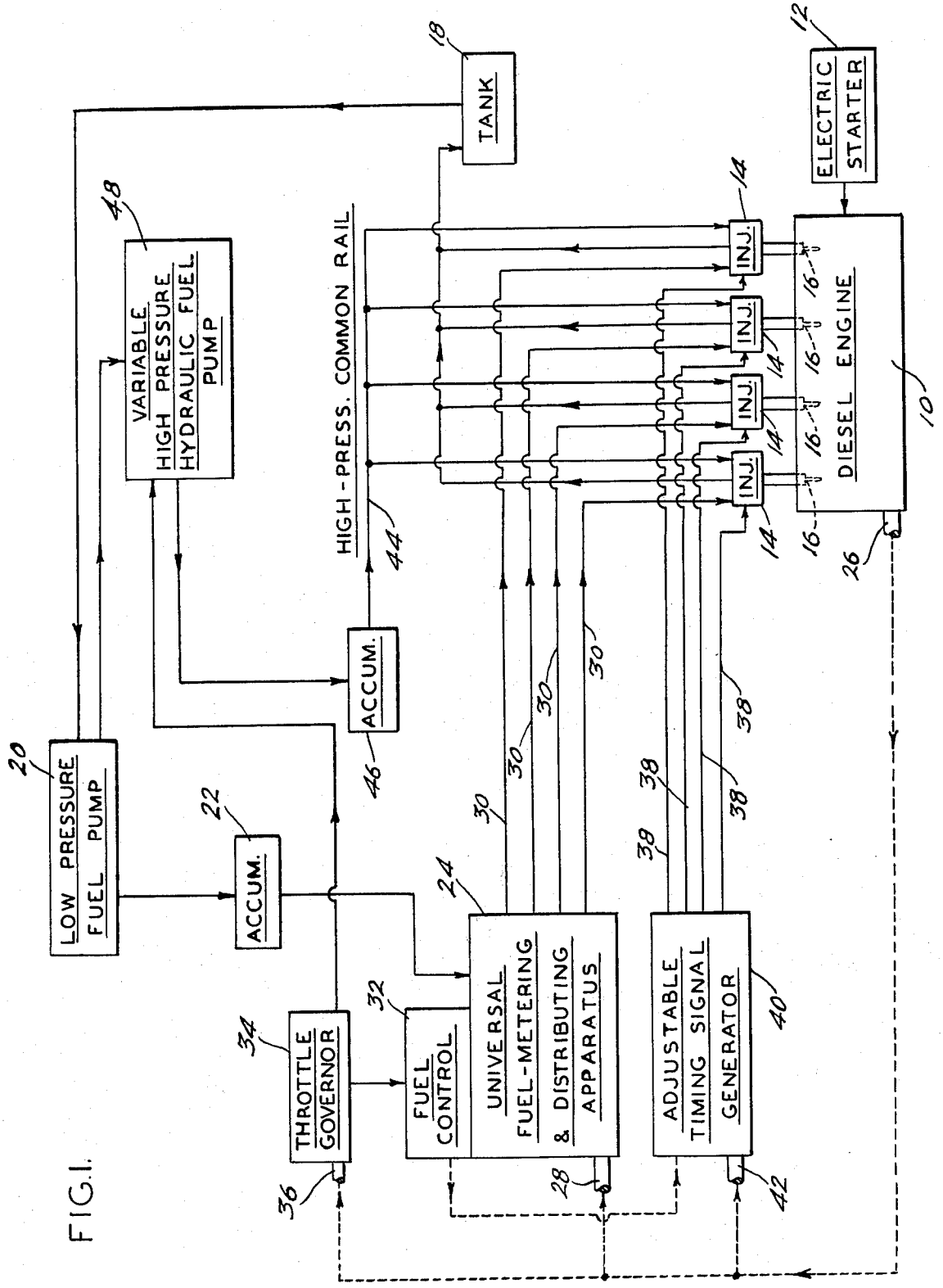

TANK
18
ELECTRIC STARTER
12
VARIABLE HIGH PRESSURE HYDRAULIC FUEL PUMP
48
HIGH-PRESS. COMMON RAIL
INJ.
14
DIESEL ENGINE
10
16
26
ACCUM.
44
46
LOW PRESSURE FUEL PUMP
20
ACCUM.
22
24
30
38
39
THROTTLE GOVERNOR
34
FUEL CONTROL
32
UNIVERSAL FUEL-METERING & DISTRIBUTING APPARATUS
ADJUSTABLE TIMING SIGNAL GENERATOR
40
28
36
42

FIG.1.

FIG. 2.

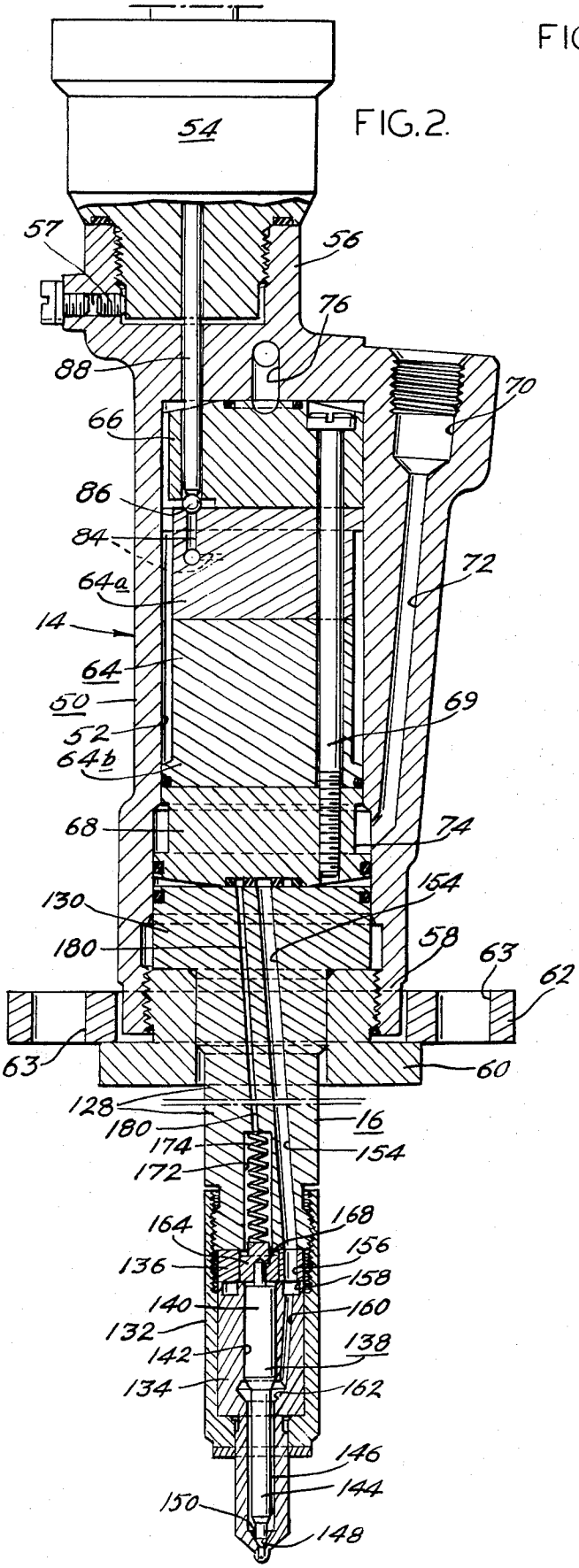

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FIG. 3.

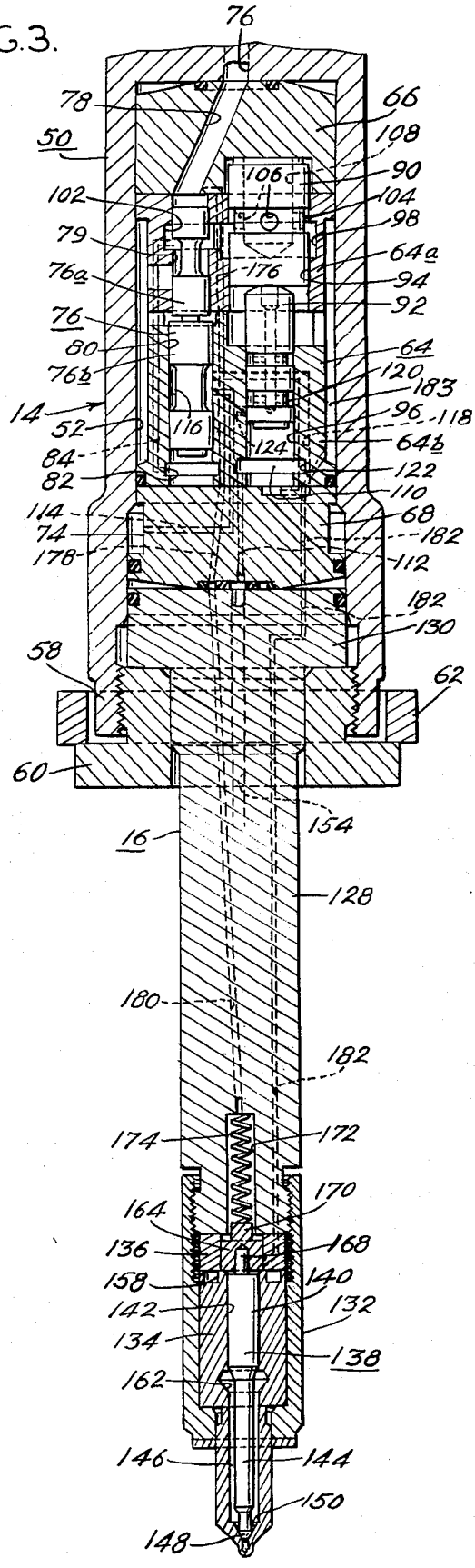

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FIG. 4.

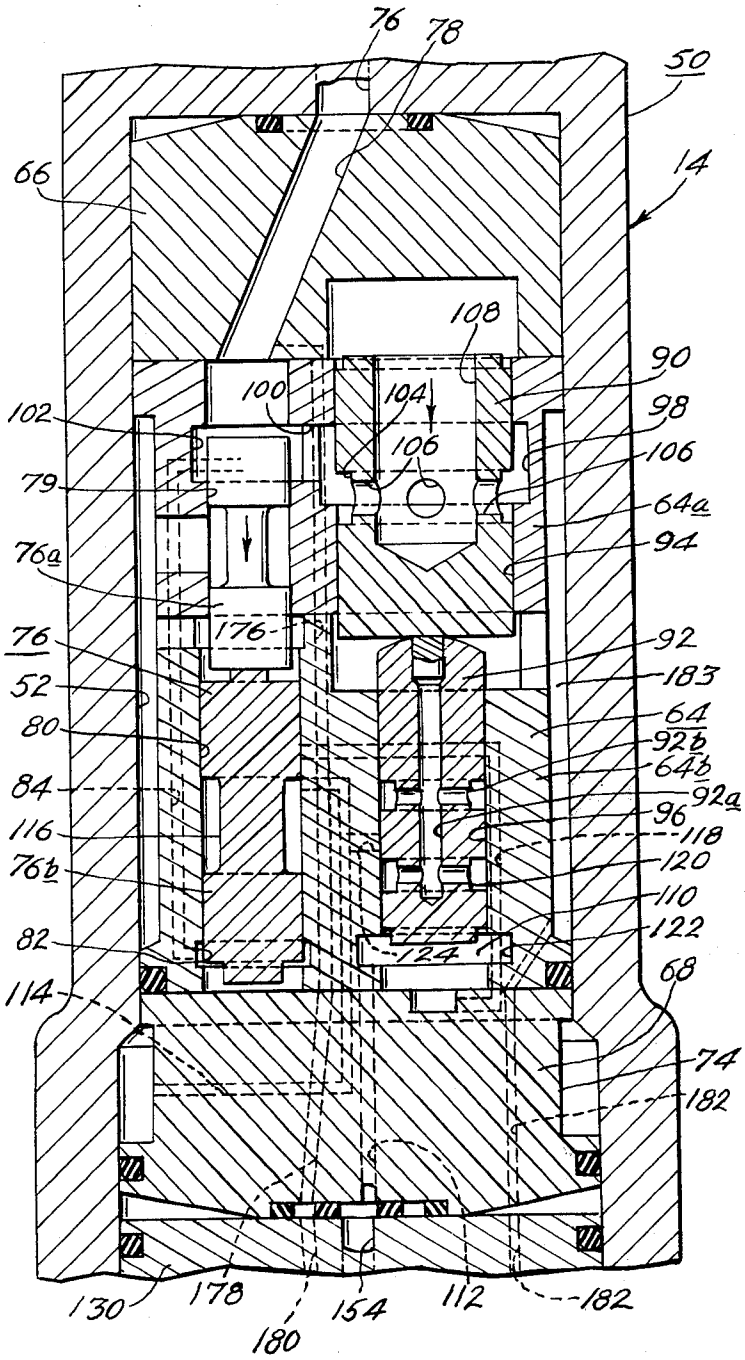
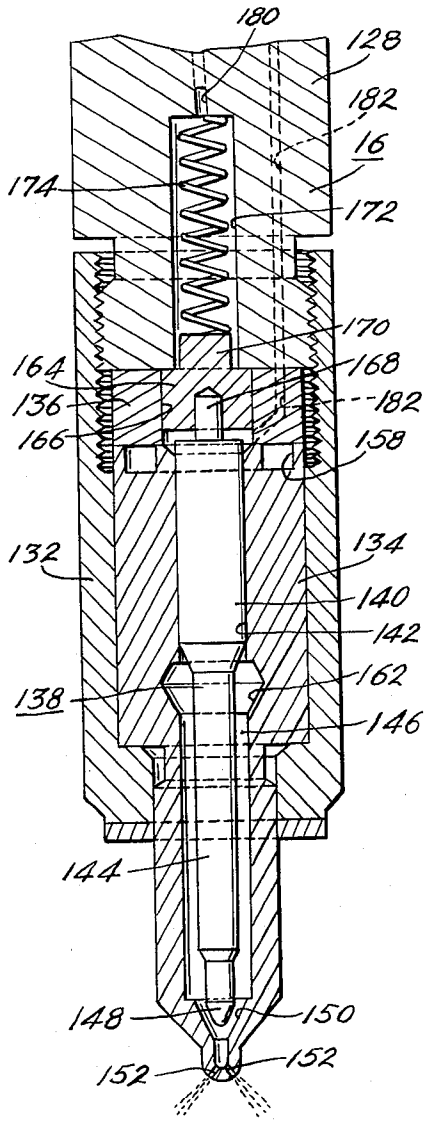

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92b
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92a
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FIG. 5.


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# HYDRAULICALLY LOADED INJECTOR NOZZLE

The present invention relates generally to fuel injection systems and more particularly to an improved fuel injection nozzle for injection systems of the type wherein the nozzle valve is opened by the force of pressurized fuel delivered thereto at timed intervals.

In a common type of diesel engine fuel delivery system, metered amounts of fuel to be injected are delivered in timed pulses to a nozzle extending into a combustion chamber of the engine. The valve in the conventional nozzle is opened by the delivery of the pressurized fuel against the closing force of a compression spring. Substantial nozzle valve closing forces are necessary, both to provide a sharply defined beginning and end of the injection interval and to prevent the valve opening under the influence of combustion chamber pressures. Because of the increasingly high combustion chamber pressures of advanced diesel engines, the conventional nozzle valve closing spring has, in some instances, become so heavy that valve opening pressures of over 4,000 psi are required.

In view of the fact that the selected force of the valve closing spring is based on the maximum combustion chamber pressure of the engine, which normally occurs at an engine full load condition, and since the valve closing spring characteristically exhibits the same closing force regardless of the engine speed or load, it is not surprising that the spring force in a conventional nozzle is not the optimum for other than full load conditions. For example, at idle and low power, a low injection pressure such as 2,500 psi is considered desirable. Under high load conditions, due in part to the higher combustion chamber pressures, a higher injection pressure, for example 9000 psi, is required. The various factors such as injection duration, spray pattern, engine smoothness, etc. which make a decreased injection pressure desirable at idling, starting and part load operation are well known and need not be discussed in detail. Suffice it to say that an engine operates best with a reduced injection pressure under other than full load conditions and that a spring loaded nozzle valve lacks the desirable flexibility of operation to permit a lower injection pressure under such circumstances.

The present invention which is particularly adapted for use with the "universal" fuel injection system disclosed in U.S. Pat. No. 3,587,547 issued June 18, 1971 and assigned with the present application to a common assignee, provides a fuel injector which utilizes a continuously applied hydraulic nozzle closing force which is variable with and directly proportional to the fuel injection pressure. In the universal system, the high pressure common rail fluid which is utilized to drive the amplifier and hence the injector piston of each injector is also employed to continuously apply a closing force to the injector nozzle valve. The common rail pressure in a preferred embodiment of the invention is introduced into the nozzle through a passage in the nozzle body and acts against a piston axially aligned with and bearing against the nozzle valve. Since the common rail pressure providing the valve closing force is the same pressure actuating the amplifier piston to drive the injector piston, the ratio of the injection pressure to the common rail pressure will be proportional to the respective diameters of the injection piston and the amplifier piston. A variation of the common rail pressure related to engine load will accordingly automatically proportionately vary the injection pressure and valve closing force, and the injection pressure can thus be regulated to suit the engine loading condition. At engine conditions other than full load, this system permits a lower injection pressure with resultant reduction of the average closing force of the nozzle valve on the nozzle seat thereby reducing the wear and stress of the valve and seat and increasing the life of the nozzle assembly.



It is accordingly a first object of the present invention to provide a fuel injection system of the type wherein the fuel injection nozzle valve is opened by the pressure of the injected fuel characterized by the hydraulic loading of the nozzle valve.

A further object of the invention is to provide a fuel injection nozzle as described wherein the hydraulic valve loading force is variable with engine loading.

A still further object of the invention is to provide a nozzle as described including means for varying the pressure of the nozzle valve loading fluid in direct proportion to the variations in fuel injection pressure.

Another object of the invention is to provide a nozzle as described which is particularly adapted for use in a universal fuel injection system characterized by an engine generated variable common rail high pressure fluid which is utilized to drive the fuel injectors in timed sequence.

A still further object of the invention is to provide a nozzle for a fuel injection system as described which permits a reduction of the fuel injection pressure during starting, idling, and part load operation to improve the operating characteristics of the engine.

A still further object of the invention is to provide a nozzle for a fuel injection system as described which permits a variation of the nozzle valve closing force commensurate with the compression chamber pressure whereby the average closing force employed will be substantially less than that of conventional nozzles, thereby decreasing the wear of the nozzle elements and increasing the effective life of the nozzle assembly.

Additional objects and advantages of the invention will be more readily apparent from the following detailed description of an embodiment thereof when taken together with the accompanying drawings wherein:

FIG. **1** is a schematic diagram of a universal fuel injection system including the novel injector nozzle construction of the present invention;

FIG. **2** is a partial axial sectional view taken through an injector constructed in accordance with the present invention showing the nozzle valve in a closed position;

FIG. **3** is a partial sectional view taken through the injector of FIG. **2** at substantially right angles to the view of FIG. **2** showing further details of the injector;

FIG. **4** is an enlarged sectional view of the upper part of the injector shown in FIGS. **2** and **3** showing the position of the injector components during fuel injection; and

FIG. **5** is an enlarged view of the lower end of the injector showing the nozzle as it appears during fuel injection.

Referring to the drawings, FIG. **1** thereof shows schematically a universal fuel injection system of a type nearly identical with that disclosed in the above referenced U.S. Pat. No. 3,587,547, which, in view of its close relation to the present invention, is hereby incorporated by reference. The system as illustrated is employed to deliver fuel to a diesel engine **10** having an

electric starter **12.** The engine is assumed to have four cylinders, and fuel is delivered into the combustion chamber of each cylinder by a fuel injector **14,** the injection nozzle **16** of which as schematically indicated extends into the cylinder combustion chamber. For the engine to operate properly, it must be supplied with successive intermittent injections of fuel through the nozzles. The operation of the engine is determined by the fuel discharges and for the engine to operate in an optimum fashion, the discharges must occur with the proper timing, duration, fuel quantity and with an appropriate pattern or wave form of injection rate during its discharge. The function of the remaining elements of the fuel injection system shown in FIG. **1** is to provide the optimum fuel injection parameters for the engine as will be briefly described herebelow.

In the system shown, a tank **18** provides a source of a suitable diesel fuel which is withdrawn from the tank by low pressure fuel pump **20** and delivered through a conventional accumulator **22** to universal fuel metering and distributing apparatus **24.** The pressure provided by the fuel pump **20** may by way of example be of the order of 200 psi.

The universal fuel metering and distributing apparatus **24** may be of the type described and claimed in U.S. Pat. No. 3,615,043, issued on Oct. 26, 1971 and assigned with the present application to a common assignee, which patent is also incorporated by reference. This apparatus is of a positive displacement shuttle piston type, wherein each discrete metered quantity of fuel is produced by the travel of the piston through a controlled distance, the stroke of the piston being synchronized with operation of the engine **10** by appropriate mechanical linkage between a shaft **26** driven by the engine and a shaft **28** which controls the timing of the displacement piston of the metering and distributing apparatus **24.**

The fuel output of the metering and distributing apparatus **24** passes to the injectors **14** through conduits **30.** The quantity of metered fuel delivered by the apparatus **24** is controlled by a fuel control **32** associated with the apparatus **24.** The fuel control **32** may comprise a control sleeve slidably longitudinally on the fuel metering piston to control the quantity of fuel delivered. A throttle governor **34** is provided to control the fuel control **32** and is of a largely conventional construction, a drive shaft **36** thereof being mechanically linked as schematically illustrated to the engine shaft **26** by appropriate gearing.

The details of the structure and operation of each of the injectors **14** will be described in detail hereinafter. Considering broadly the injector operation, each injector receives its successive discrete quantities of fuel from the fuel metering and distribution apparatus **24** and stores each quantity briefly until the injector is actuated, at which time piston means contained in the injector operates to discharge the stored fuel through the nozzle **16** into the corresponding engine cylinder. The time at which each injector is actuated to produce a fuel discharge is controlled and determined by an actuating signal supplied thereto over a corresponding one of the four electrical timing lines **38** leading to the injectors from the adjustable timing signal generator **40.** The timing signal generator is synchronized with the engine operation by appropriate mechanical linkage of the generator drive shaft **42** with the engine shaft **26** as schematically indicated so that the output electrical timing signals bear a predetermined phase relation to the phase of operation of the engine **10.** The timing signals delivered to the several injectors of course are provided according to the differences in phase of operation of the pistons in the four corresponding cylinders of the engine.

A mechanical linkage from the fuel control **32** to the timing signal generator **40** is preferably also provided to permit automatic variation of the advance or retard of the signals generated as a function of the position of the fuel control and hence as a function of the load on the engine. The timing signal generator may also include an appropriate flyweight retard-advance mechanism for automatically varying the phase of the output signals as a function of engine speed. In the present embodiment it is preferred to use the photoelectric timing signal generator apparatus described and claimed in U.S. Pat. No. 3,587,535 issued on June 28, 1971 and assigned with the present application to a common assignee, which patent is also incorporated by reference.

The force for operating the piston means in each of the injectors **14** is provided by fluid pressure from a high pressure common rail **44.** While a different fluid may be used for this purpose, in the present embodiment fuel from the tank **18** is utilized as the high pressure fluid which is delivered to the common rail **44** through a conventional accumulator **46** by a variable high pressure hydraulic fuel pump **48** connected with the low pressure pump **20.** The common rail pressure may be, for example, of the order of 800–3,000 psig and is determined by the required operating pressures of the injectors. Suitable high-pressure pumps for providing pressures of these magnitudes and enabling controlled pressure variation are well known in the art.

Since a complete description of the overall universal system as well as the individual elements thereof are set forth in the above-referenced patents, only a limited further description of the system and its operation is deemed necessary. The following detailed description is accordingly directed to the injector structure and operation and especially the novel hydraulically loaded injector nozzle.

Referring to FIGS. **2–5,** each injector **14** comprises an injector housing **50** having a hollow substantially cylindrical chamber **52** within which are disposed a number of axially arranged injector components. An electromagnetic valve actuating means **54** is threadedly connected to the internally threaded injector housing extension **56** at the upper end of the injector housing and is secured therein by the lock screw **57.** The injector housing is internally threaded at its lower end **58** to receive the nozzle holder nut **60.** A securing flange **62** includes bolt holes **63** and is adapted to overlie the flange portion of the nut **60** to secure the nut and the entire injector assembly to an engine.

As shown most clearly in the enlarged view of FIG. **4,** the interior details of the injector components within the bore **52** are in large measure similar to that of the injector of the above-referenced U.S. Pat. No. 3,587,547. The injector barrel **64** composed of upper and lower barrel sections **64a** and **64b** is disposed within the injector housing bore **52** between the upper end cap **66** and the bottom end cap **68.** A screw **69** (FIG. **2**) passes through the barrel sections and end caps to secure them in angular relation. Appropriate seal rings are employed as illustrated between the injec-

tor housing and the barrel and end caps to prevent fluid leakage along the bore **52.**

Metered fuel from the universal fuel metering and distributing apparatus **24** is introduced into the injector housing **50** through the port **70** (FIG. **2**) and inlet conduit **72** thereof which communicates with an annulus **74** of the lower end cap **68.** This metered fuel then flows through passage **114** (FIG. **3**) into an annulus **116** in the servo piston portion **76**b and thence through passage **118** into the chamber **110.** It will be noted that the flow through passage **118** is cut off by the decent of the servo piston. The flow of metered fuel into the chamber **110** thus takes place only when the servo, amplifier and injector pistons are in the raised position. The high pressure common rail fuel from the pump **48** enters the injector through passage **76** in the injector housing **50** and passes through the conduit **78** in the upper end cap **66.** The injector utilizes the high common rail pressure to inject the metered fuel stored in the injector in chamber **110** into the nozzle upon receipt of a signal by the electromagnetic valve actuating means **54** as described below.

A servo piston **76** comprising upper and lower portions **76***a* and **76***b* is disposed for slidable axial movement within the injector barrel, the portion **76***a* being disposed in the bore **79** of the barrel portion **64***a*, and the portion **76***b*, which is of a slightly larger diameter, being disposed within the bore **80** of the barrel section **64***b*. The passage **78** in the upper end cap **66** communicates with the upper end of the bore **79,** thus delivering the high pressure common rail fuel into the bore **79** to act against the upper end of the servo piston. Conduit means (not shown) are provided to introduce the high pressure common rail fuel into the chamber **82** at the lower end of the servo piston wherein the fuel acting against a larger piston area, serves to move and hold the servo piston in the raised position of FIG. **3.** A passage **84** is provided from the chamber **82** upwardly to a valve **86** (FIG. **2**) operated by the valve stem **88** of the electromagnetic valve actuating means **54,** which valve opens the passage **84** to a peripheral fuel sump and hence lowers the pressure in the chamber **82,** thus permitting the servo piston to descend to the position of FIG. **4** when a signal is received by the means **54.**

An amplifier piston **90** and an injector piston **92** of smaller diameter are disposed in abutting axially aligned relation respectively within the bores **94** and **96** of the injector barrel **64.** An annulus **98** in the bore **94** communicates by means of a port **100** with an annulus **102** of the bore **78,** thus permitting the high pressure rail fluid to enter the annulus **98** to drive the amplifier piston when the servo piston **76** is lowered to permit flow into the annulus **102.**

The amplifier piston **90** includes an annulus **104** in the cylindrical outer wall thereof from which pass spaced radial ports **106** to permit flow into an inner cylindrical bore **108.** The arrangement of the annulus **98,** annulus **104,** and the ports **106** is preferably chosen in accordance with the disclosure of U.S. patent application Ser. No. 247,333, entitled APPARATUS FOR CONTROLLING RATE OF FUEL INJECTION, filed Apr. 25, 1972 which is assigned with the present application to a common assignee.

The downward movement of the amplifier piston drives the injector piston **92** downwardly into the chamber **110** to force the metered fuel previously directed thereinto out through the passage **112** in the lower end cap **68** into the injection nozzle **16.** The injection stroke is terminated when the annulus **120** of the injector piston aligns with the annulus **122** of the chamber **110,** thereby permitting a venting of the chamber **110** to passage **114** by means of the axial passage **92***a* in the injector piston and annulus **92***b* which communicates with passage **124** which intersects passage **114.**

The nozzle assembly of the injector includes a nozzle holder **128** which, as shown most clearly in FIG. **3,** is of a generally cylindrical configuration and includes a flange portion **130** at the upper end thereof which is clamped in position within the bore of the injector body by the nozzle holder nut **60.** A cap nut **132** threadedly secured on the lower end of the holder **128** secures the nozzle body **134** and adaptor disc **136** to the holder in axial alignment therewith. As shown most clearly in the enlarged view of FIG. **5,** the nozzle assembly includes a valve element **138** having an upper cylindrical portion **140** disposed in slidable relation within a bore **142** of the body **134.** A lower cylindrical portion **144** of the valve element is of a reduced diameter, and with the bore **142** defines an annular fuel passage **146.** The valve element portion **144** terminates in a conical tip **148** which cooperates with the conical valve seat **150** at the lower end of the valve body. Spray orifices **152** in the lower end of the nozzle body direct the injected fuel into an engine combustion chamber in a predetermined spray pattern.

The injected fuel passes from the injector lower end cap passage **112** into a communicating passage **154** in the holder **128** as shown in FIG. **2,** a passage **156** in the adaptor **136,** and thence into annulus **158** of the body **134** from which it flows through body passage **160** into annulus **162,** thence through the annular passage **146** to the lower end of the body. The high injection pressure within the annulus **162** and passage **146** acting against the valve element shoulders, lifts the element from its seat to permit flow through the orifices **152** in a well known manner.

In the conventional nozzle, the valve element is downwardly loaded against its seat by a powerful spring within the nozzle holder, which spring must exert a sufficient force on the valve element to properly control the opening and closing of the valve during the highest injection and combustion chamber pressure conditions of full load operation. Since diesel engines are not operated at full load much of the time, most operating conditions do not require the high closing force characteristic of a conventional valve spring which is therefore unnecessary and, in fact, undesirable. For example, during starting, idling and part load operation, it is preferable to reduce the fuel injection pressure and to reduce the valve closing pressure in a commensurate fashion as outlined above. In the present invention this desirable function is effected by hydraulic loading of the valve element as will now be described.

A piston **164** is slidably mounted within a bore **166** of the adaptor disc **136** and is connected with the valve element **138** by means of a pin **168** extending from the upper end of the valve element into a suitable bore in the piston. A portion **170** of the piston of reduced diameter extends upwardly into a bore **172** of the holder **128,** the portion **170** of the piston being of a smaller diameter than that of the bore **172.** A light coil spring **174** is disposed in compression within the bore **172** bearing against the upper end of the bore and at its

lower end against the portion **170** of the piston, serving to bias the piston and hence the valve element downwardly.

Means are provided for introducing the high pressure common rail fuel into the bore **172** of the holder. Said latter means comprises a passage **176** in the barrel **64** of the injector which extends from the passage **78** in the upper end cap **66** downwardly through the barrel to communicate with the passage **178** in the lower end cap **68** and passage **180** in the nozzle holder **128,** the latter passage opening into the upper end of the holder bore **172.** The common rail pressure introduced into the bore **172** acts against the piston **164** and thus serves to continuously apply a closing pressure to the nozzle valve element **138.** The closing force generated by the piston and high pressure rail fluid are of course dependent on the area of the piston, and the use of a piston as illustrated of a diameter substantially larger than that of the valve element serves to amplify the valve closing force. The light spring **174** as described below, is needed only during starting of the engine to close the valve until the rail pressure in the fuel system has become established. A passage **182** from bore **166** through the adapter disc and holder extends upwardly through the lower end cap and into the sump **183.** Any fuel leakage around the valve portion **142** or the piston **164** will be carried off by the passage **182** to the sump.

The operation of the universal system will be apparent from the above-referenced patents and from the general description heretofore presented. In addition, the injector operation need be only briefly described since it is essentially the same as that described in U.S. Pat. No. 3,587,547 although some structural differences exist, such as the location of the valve actuating means **54.** Of primary importance, is the hydraulic loading of the nozzle valve to permit a proportional variation in valve closing pressure with injection pressure, and the loading of the valve element with the universal system high pressure common rail fluid.

In FIG. **3,** the amplifier piston **90** and the injector piston **92** are shown in the raised position with the injection chamber **110** being opened to the passage **118** permitting a charging of the chamber **110** with the metered fuel delivered from the universal fuel metering and distributing apparatus **24.** Upon receipt of a signal from the adjustable timing signal generator **40,** the electromagnetic valve actuating means **54** opens the valve **86** permitting the high rail pressure to drain from the chamber **82** and thus allow the servo piston to move downwardly to the position of FIG. **4,** simultaneously opening the annulus **102** and port **100** to the high rail pressure thus actuating the amplifier injection pistons and also cutting off the passage **118** which delivered the metered fuel into injection chamber **110.** As the amplifier and injector pistons move downwardly, fuel is delivered from the chamber **110** through passages **112, 154, 156, 160** into annulus **162** and thence along annular passage **146,** the injection pressure acting against the stepped nozzle valve being sufficient to overcome the valve closing force of the common rail pressure acting on piston **164** plus the small force of the light spring **174,** thereby opening the valve element as shown in FIG. **5.** The fuel passing along the valve seat is injected in a spray pattern into the engine combustion chamber by the spray orifices **152** as schematically illustrated in FIG. **5.**

When the annulus **120** of the injection piston communicates with the annulus **122** of the chamber **110,** the injection stroke ends, the valve **86** (FIG. **2**) closes and the servo piston is returned to its raised position to permit the recharging of the injection chamber **110** with the next increment of metered fuel. As the injection pressure drops upon completion of the injection stroke of the injector piston, the nozzle valve element **138** closes against its seat under the influence of the hydraulic loading force of the common rail pressure to cleanly cut off injection and hold the valve element closed against the combustion chamber pressure during the burning of the injected fuel.

It will be apparent that the injection pressure of the fuel will always be higher than the common rail pressure by the amplifier ratio established by the area difference between the amplifier piston and the injector piston. In the preferred embodiment, this ratio is 3:1. The permissible adjustability of the common rail pressure during the running of the engine provides a consequent adjustment of the injection pressure and injection duration. The injection pressure may thus be attuned to the requirements of the engine at all times.

For example, at idle and at part load, an injection pressure of as low as 2,500 psi (approximately 840 psi rail pressure) is desirable. On the other hand, at full load on injection pressure of 9,000 psi (3,000 psi rail pressure) is necessary since under these conditions, the combustion chamber pressures are high.

With the present hydraulically loaded nozzle, the closing force on the nozzle valve is automatically at a high level when high combustion chamber pressures are encountered since high injection pressures and hence a high rail pressure are necessary at high engine load. With a conventional spring loaded nozzle, however, the spring is chosen to close the nozzle valve against the maximum combustion chamber pressures encountered which force may for example require a fuel injection pressure of 3,200 psi just to open the nozzle. Since the conventional spring will always exert the same closing force on the nozzle valve, the same 3,200 psi opening pressure is required under all conditions, including starting, idle and part load. Since testing has shown that at idle the typical engine operates best at injection pressures as low as 2,500 psi, it is impossible to obtain the optimum flexibility of operation with a spring loaded type of nozzle. With the hydraulically loaded nozzle, since the nozzle closing force is directly proportional to the common rail pressure and to the injection pressure, the nozzle opening force may be varied to achieve the optimum injection pressure at the nozzle spray holes.

During engine starting, the conventional fuel injection pump is rotating very slowly with a consequent long duration pumping cycle and an opportunity for a substantial amount of leakage to occur. With a spring loaded nozzle, the leakage problem becomes quite significant during starting since a typical pressure of 3,200 psi must be generated just to open the nozzle.

With the present universal system described, however, the hydraulically loaded nozzle might typically open during starting when an injection pressure of only 500 psi is obtained. The common rail pressure pump need only generate 170 psi during this critical period, a pressure which is easily obtained even if the pump efficiency is diminished by wear or production tolerances.

It will be apparent that if starting were attempted in the absence of the small spring **174** in the bore **172** acting on the piston **164,** until a sufficient rail pressure had been established, there would be insufficient closing force on the nozzle valve to hold the nozzle valve closed against the compression pressure in the combustion chamber. The combustion chamber pressure would then push the nozzle off its seat and introduce air into the fuel passage, disrupting the metered flow of fuel. The provision of the spring effectively prevents this occurrence but requires only a small opening force. In a preferred embodiment of the invention, the spring has a force of approximately 7 lbs. which requires an injection pressure of 500 psi to open the nozzle valve in the absence of any common rail pressure. This spring force will of course be added to the closing force exerted on the nozzle valve by the common rail pressure, but during normal operation will be only a small fraction of the closing force and will not interfere with the advantageous variation of the closing force commensurate with the change in injection pressure.

Although the hydraulically loaded nozzle valve is preferably used in conjunction with the described universal fuel injection system by utilization of the common rail pressure thereof, other arrangements may be devised for hydraulically loading the nozzle valve in a variable manner to achieve the desired functional advantages.

The permissible variation of the nozzle closing force with engine load not only improves the engine operation, but in addition, by reducing the average closing force on the nozzle seat, reduces the wear of the valve element and seat and should appreciably increase the life of the nozzle.

Manifestly, changes in details of construction can be effected by those skilled in the art without departing from the spirit and scope of the invention.

I claim:

**1.** In a diesel engine fuel injection system comprising a fuel injection nozzle, a valve within said nozzle, and means for supplying metered quantities of fuel to said nozzle valve in timed relation to an engine operating cycle at a high injection pressure varying with engine load, said means comprising a source of high pressure fluid varying with engine load, said valve being adapted to open upon presentation thereto of the high pressure metered fuel, the improvement comprising means providing a variable closing force to said nozzle valve after the engine starting cycle, said latter means comprising said source of high pressure fluid, means operable to convert said high pressure fluid into a closing force acting on said nozzle valve to close said valve, and means for maintaining a substantially constant proportional relationship of the pressure of said high pressure fluid and the injection pressure of the fuel, thereby providing a variation of the nozzle closing force with engine load.

**2.** The invention claimed in claim **1** wherein said means operable to convert said high pressure fluid into a closing force acting on said nozzle valve to close said valve comprises piston-cylinder means connected with said nozzle valve.

**3.** The invention claimed in claim **1** including light spring means operatively connnected with said valve to supplement said means providing a closing force during engine start-up.

**4.** A fuel injection system for diesel engines comprising an injection nozzle for injecting a spray of fuel into an engine cylinder, said nozzle including a nozzle valve adapted to open upon introduction of a metered quantity of pressurized fuel into said nozzle, means for biasing said nozzle valve toward a closed position, an injector comprising means for receiving metered quantities of fuel from a fuel metering means and for successively delivering each metered fuel quantity to said nozzle under pressure in timed relation with the engine cylinder cycle, a source of high pressure fluid, means for varying the pressure of said high pressure fluid in accordance with engine load, means in said injector for injecting the metered fuel quantities into said nozzle at a pressure varying with engine load and exceeding the pressure of said high pressure fluid, said fuel injecting means comprising cylinder-piston means driven by said high pressure fluid, said means for biasing said nozzle valve toward a closing position comprising cylinder-piston means connected with said valve, and passage means for introducing said high pressure fluid into said cylinder-piston means whereby the closing force on said nozzle valve provided by said cylinder-piston means as well as the valve opening force of said injected fuel will vary proportionately with engine load after the engine starting cycle.

**5.** The invention claimed in claim **4** wherein said means in said injector for injecting the metered fuel quantities into said nozzle at a pressure exceeding the pressure of said high pressure fluid comprises an amplifier cylinder-piston means.

**6.** The invention claimed in claim **5** wherein said amplifier cylinder-piston means serves to deliver the metered fuel into the nozzle at substantially three times the pressure of said high pressure fluid.

**7.** The invention as claimed in claim **1** wherein said high pressure fluid comprises the engine fuel.

**8.** The invention as claimed in claim **4** wherein said high pressure fluid comprises the engine fuel.

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