United States Patent

Links

[15] **3,689,205**

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[54] **PUMP-AND-NOZZLE ASSEMBLY FOR INJECTING FUEL INTO INTERNAL COMBUSTION ENGINES**

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[30] **Foreign Application Priority Data**

March 14, 1970 Germany..........P 20 12 202.5

[52] **U.S. Cl.**...............................**417/401,** 123/139 R

[51] **Int. Cl.**......**F04b 17/00,** F04b 35/00, F02n 39/00

[58] **Field of Search**........417/326, 399, 461; 91/279, 91/459; 123/139 R, 139 AA

[56] **References Cited**

UNITED STATES PATENTS

2,537,748	1/1951	Evans et al.	417/401
2,598,528	5/1952	French	417/399

Primary Examiner—Robert M. Walker

Attorney—Edwin E. Greigg

[57] **ABSTRACT**

A pump - and - nozzle assembly for injecting fuel into internal combustion engines includes a servo piston which operates a smaller pump piston in response to pressurized fuel periodically admitted to and removed from said servo piston by virtue of a respective first and second position of a solenoid valve also forming part of said assembly. In said first position the fuel injection is triggered, while in said second position the return stroke of the servo piston is initiated and the injection is rapidly terminated. For the latter purpose a continuous communication exists between the work chamber of the pump piston and the injection nozzle.

5 Claims, 4 Drawing Figures

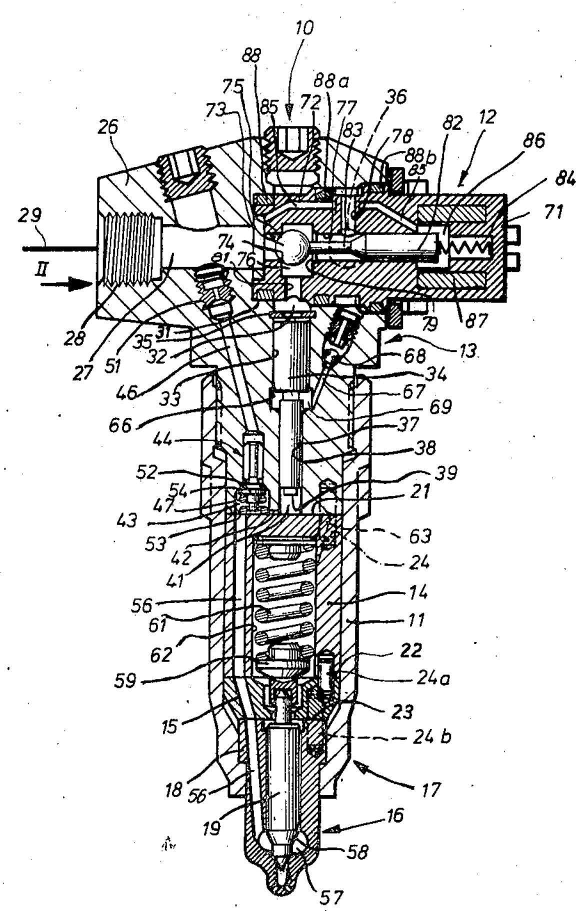

Fig. 1

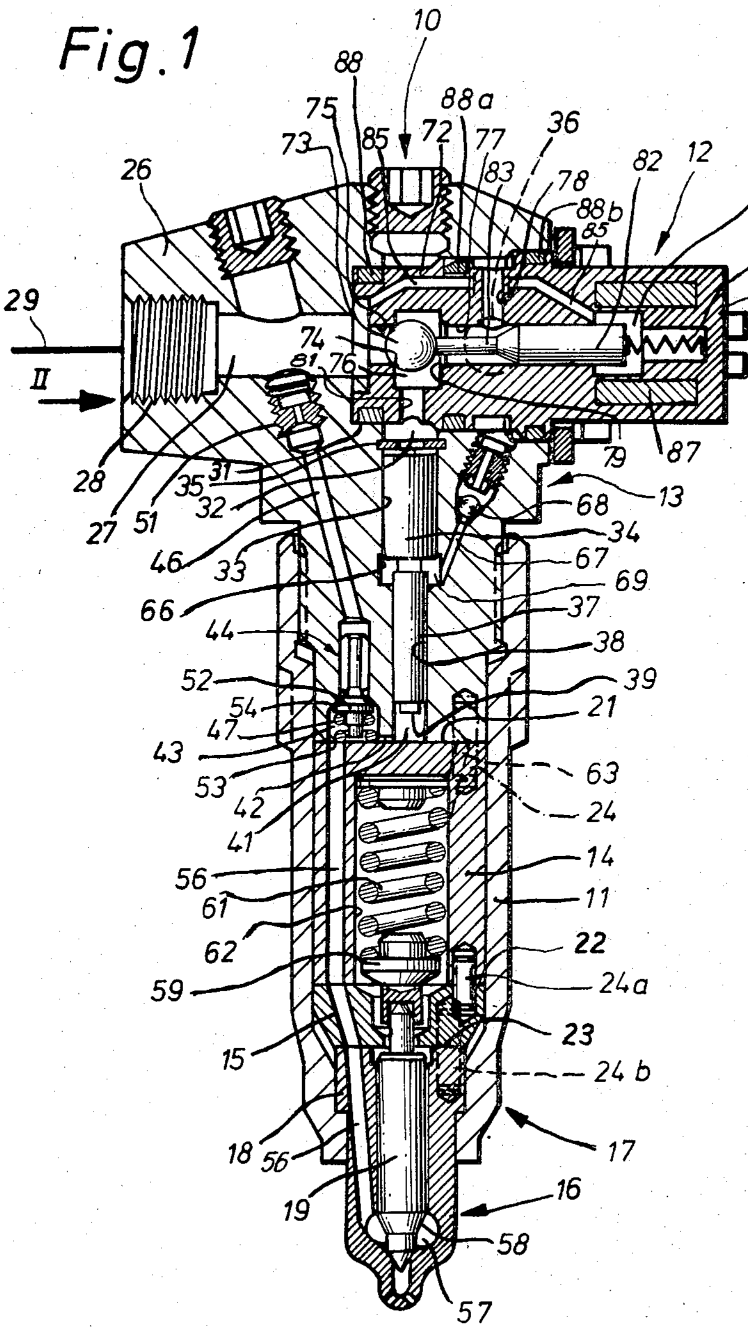
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Fig. 2

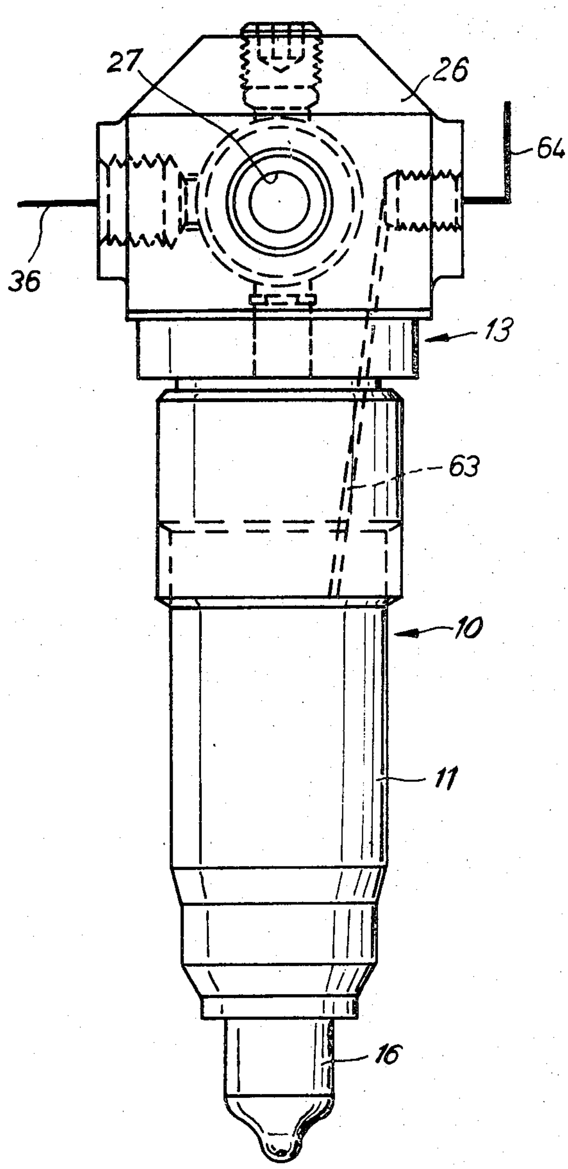

Fig. 4

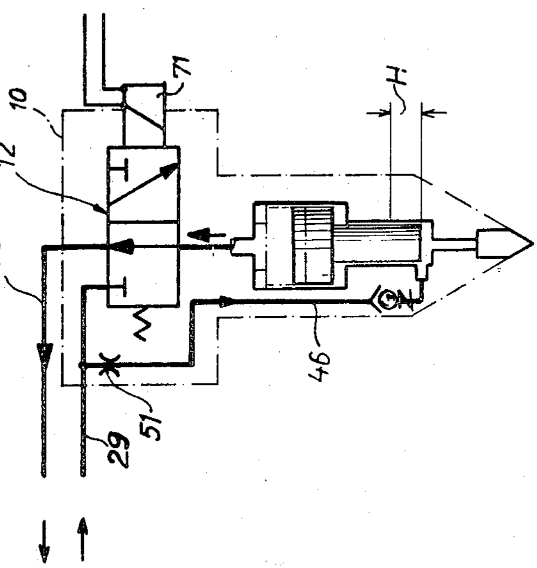

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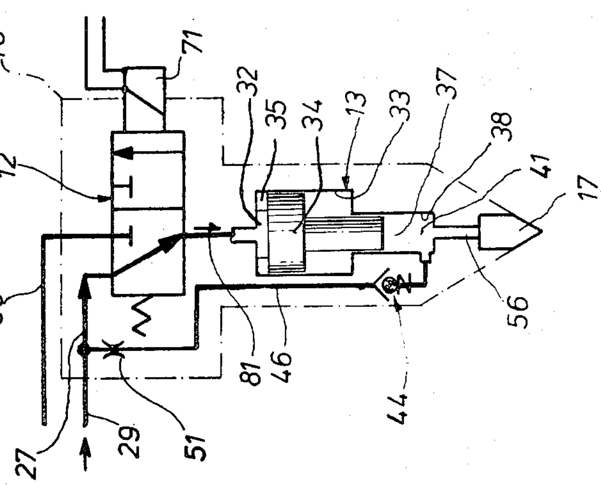
Fig. 3



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PUMP-AND-NOZZLE ASSEMBLY FOR INJECTING FUEL INTO INTERNAL COMBUSTION ENGINES

BACKGROUND OF THE INVENTION

This invention relates to a fuel injection pump - and - nozzle assembly associated with internal combustion engines, particularly diesel engines operating on directly injected fuel. The assembly is of the type which has a pump piston driven by a servo piston having a larger diameter than the pump piston and which is connected to a pressure source. The latter supplies fuel under pressure to the pump piston through a supply valve and to the servo piston. The assembly further includes a solenoid valve which is operated synchronously with the engine operation and the movable valve member of which may assume two positions. In the first position the flow of fuel is directed to the servo piston and in the second position the flow of fuel is directed from the servo piston to a return conduit. In the first position the solenoid valve triggers the beginning of injection while in the second position it controls the start of the return stroke.

In a known pump - and - nozzle assembly of this type (as shown, for example, in FIG. 1 of U.S. Pat. No. 2,598,528) and electromagnetically operated control plunger in its first position directs the fuel into a pressure chamber to that side of a two-way servo piston which is remote from the pump piston and initiates thereby the injection of fuel. The termination of the injection is controlled by means of an oblique control edge on the servo piston at its side adjacent the pump piston. The control edge closes an exit port of a fuel return conduit so that a hydraulic abutment is formed.

The aforeoutlined pump - and - nozzle assembly has the disadvantage that due to the hydraulic abutment affecting the servo piston, the pump piston and the servo piston remain in their end position until the beginning of the return stroke. As a result, an effective timely depressurization of the pump work chamber necessary for closing the injection nozzle does not take place.

In another embodiment of this type of pump - and - nozzle assembly (as shown, for example, in FIG. 5 of U.S. Pat. No. 2,598,528), the terminal moment of the injection is controlled by an oblique control edge which is provided on the pump piston and which cooperates with a bypass bore in the pump cylinder in a known manner. The servo piston and the pump piston both execute their entire stroke; the effective delivery stroke which is variable by the oblique control edge, and thus determined by the angular position of the piston, determines the injected fuel quantities In the second position of the control plunger associated with the electromagnet, the fuel is admitted to the servo piston at its side adjacent the pump piston, whereupon the servo piston is returned into its initial position. During this return stroke, the fuel admitted to the servo piston in the first position of the control plunger, is forced into a return conduit.

This last-outlined pump - and - nozzle assembly, to be sure, does not have the disadvantage of the first described structure, but, because of the charging of the entire pump work chamber required for the always uniform stroke even in case of fuel injection for partial load and because of the always uniform servo fuel quantity, the pressure source has to deliver even during idling or partial load operation the maximum fuel quantities for each work cycle. This renders the entire fuel injection system expensive and complicated.

OBJECT, SUMMARY AND ADVANTAGES OF THE INVENTION

It is an object of the invention to provide an improved, solenoid valve-controlled pump - and - nozzle assembly for fuel injection in which the aforenoted disadvantages are eliminated and which, particularly in high-rpm diesel engines, ensures a rapid depressurization of the pump work chamber and the pressure conduit between the pump work chamber and the fuel injection nozzle at the end of each injection while securely avoiding the generation of vacuum in the pressure conduit and in the pump work chamber.

Briefly stated, according to the invention, the terminal moment of the injection is controllable by means of the second position of the solenoid valve simultaneously with the beginning of the return stroke of the servo piston, whereupon the charging stroke of the pump piston begins and further, between the pump work chamber and the injection nozzle there is provided a continuously open connection. This results in a sudden depressurization of the pump work chamber and the pressure conduit and thus results in a desired rapid termination of the injection. The pump nozzle, according to the invention, has further the advantage that the pressure prevailing prior to the beginning of each injection in the channel leading to the injection nozzle is equal to the servo pressure.

The invention will be better understood as well as further objects and advantages of the invention will become more apparent from the ensuing detailed specification of a preferred, although exemplary embodiment, taken in conjunction with the drawing.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is an axial sectional view of a pump - and - nozzle assembly representing the preferred embodiment of the invention;

FIG. 2 is a side elevational view of the same embodiment in the direction of arrow II in FIG. 1;

FIG. 3 is a simplified schematic view depicting the preferred embodiment prior to the beginning of the fuel injection; and

FIG. 4 is a simplified schematic view depicting the preferred embodiment upon termination of the fuel injection.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Turning now to FIGS. **1** and **2**, the pump - and - nozzle assembly **10** composed of two structural groups is held together as a single unit by means of a threaded clamping sleeve **11**. The first structural group forming part of the assembly **10** comprises a hydraulically driven pump device **13** controlled by a solenoid valve **12**, while the second structural group is formed of an injection nozzle device **17** comprising a spring housing **14**, an intermediate disc **15** and a nozzle member **16**. The latter has a nozzle body **18** and includes a valve needle **19** disposed therein for axial reciprocal motion. The contact face **21** between the pump device **13** and the spring housing **14**, as well as the contact face **22** between the intermediate disc 15 and the spring housing 14 and the contact face 23 between the nozzle body 18 and the intermediate disc 15 are machined planar and, by means of the clamping force of the sleeve 11, provide fluid-tight seals for the channels and chambers passing through said contact faces. Further, the structural group 13 and the components 14, 15 and 16 are conventionally secured against rotation by means of securing pins 24, 24*a* and 24*b*.

The housing 26 of the pump device 13 has a supply bore 27 which extends transversely to the longitudinal axis of the assembly 10 and which is provided with a coupling thread 28 connected to a supply conduit 29 which, in turn, delivers fuel under a servo pressure p_s to the assembly 10 from a pressure source, not shown. This pressure source may be formed, for example, by an engine driven gear pump, the delivery pressure of which may be maintained at the desired servo pressure (for example, 100 kg/cm^2) by means of a pressure regulating valve.

The solenoid valve 12, to be described in greater detail hereinbelow, is inserted into a stepped continuation 31 of the supply bore 27 and controls the fuel flow from or to a pressure chamber 32 situated above a servo piston 34 reciprocating in a bore 33 of the pump housing 26. In its position shown in FIGS. 1 and 3, the servo piston 34 engages an upper abutment 35. The fuel returns from the pressure chamber 32 into a fuel tank, not shown, through a return conduit 36. Adjoining the end of the servo piston 34 remote from the pressure chamber 32 there is positioned a pump piston 37 of smaller diameter. The pump piston 37 is slidably held in a cylinder bore 38 of the pump housing 26; it has a frontal radial face 39 which delimits the pump work chamber 41 from one side.

The pump work chamber 41 is connected through a port 42 with a chamber 43 of the check valve 44 functioning as a supply valve to control the admission of fuel to the pump work chamber 41 through a charging channel 46. The chamber 43 also serves as a housing for a valve spring 47 urging the check valve 44 into its closed position. The valve spring 47 is preloaded to a charging pressure which is substantially below the servo pressure and which may have a magnitude of approximately 10 kg/cm^2. If the pressure in the pump work chamber 41 is approximately equal to the servo pressure in the charging channel 46, the spring 47 closes the valve 44. The charging channel 46 branches off from the supply bore 27 between the coupling thread 28 and the solenoid valve 12 and has in the vicinity of the branch-off a throttle 51. The supply valve 44 is located at the downstream end of the charging channel 46. In valve 44 the periphery of its movable valve member 52 and wall 53 of the spring chamber 43 define a gap 54 which may be so designed that it forms a constricted flow passage section and functions as an additional throttle besides the throttle 51.

From the spring chamber 43 there extends a pressure channel 56 to an annular chamber 57 disposed in the nozzle member 16 of the injection nozzle device 17. Thus, the annular chamber 57 and the pump work chamber 41 are in continuous communication. The pressurized fuel in the annular chamber 57 affects a frustoconical differential face 58 of the valve needle 19 in a known manner. A coil spring 61 disposed in the spring chamber 62 of the spring housing 14 engages, through a spring seat disc 59, the valve needle 19 and urges it into its closed position as shown in FIG. 1.

The fuel which leaks into the spring chamber 62 from the annular chamber 57 through the valve needle 19, is returned to the fuel tank through a leakage port 63 extending from the spring chamber 62 and indicated in broken lines in FIGS. 1 and 2 and through a leakage conduit 64 (FIG. 2) adjoining the leakage port 63.

The fuel leaking through servo piston 34 and pump piston 37 is collected in an annular groove 66 which is located at the end of the cylinder bore 33 adjacent the end of the servo piston 34 remote from pressure chamber 32. The annular groove 66 communicates with the return conduit 36 by means of a channel 67 containing a check valve 68. The purpose of the latter is to prevent fuel from being drawn from the return conduit 36 into the chamber 69 delimited by the annular groove 66 and the pump piston 37 during the return stroke of the servo piston 34. The occurrence of an aforenoted drawing of fuel would prevent a subsequent downward motion of the servo piston 34 during the delivery stroke.

The solenoid valve 12 inserted into the enlarged extension 31 of the supply bore 27 is formed of a pressure-equalized 3/2-way valve operated by an electromagnet 71, a valve housing 72 and a sphere 73 constituting the movable valve member. The sphere 73, in its position shown in FIG. 1, closes the valve seat 74 at the mouth of a bore 75 communicating with the supply bore 27 and thus prevents the admission of fuel from the supply bore 27 to a control chamber 76 in the valve housing 72. The control chamber 76 is connected by means of a bore 77 and a channel 78 with the return conduit 36. The mouth of the bore 77 at the control chamber 76 is formed as a second valve seat 79 for the sphere 73. The control chamber 76 and the pressure chamber 32 of the servo piston 34 are interconnected by means of a bore 81.

The electromagnet 71 is provided with an armature 82 which is movably held in an extension of the bore 77 in the valve housing 72 and which, by means of a pin 83, urges the sphere 73 against valve seat 74 under the effect of spring 84 when the electromagnet 71 is in a de-energized condition. In order to ensure that the spring 84 does not have to work against the high servo pressure (p_s = 100 kg/cm^2) and thus may have a closing force within practical limits, the solenoid valve 12 is pressure-equalized by communicating the pressure prevailing in the supply bore 27 through a channel 85 to the chamber 86 which receives the spring 84 and which is located behind the armature 82. The surfaces exposed to the pressure of the fuel on the sphere 73 and on the armature 82 are identical in magnitude so that the hydraulic forces exerted on the sphere 73 in the opening and in the closing direction are also identical. Therefore, no substantial force of spring 84 is needed to keep the sphere 73 at its seat 74 in a closed position.

The electromagnet 71 is provided with a sole control winding 87. As soon as the latter is energized, for example, by means of an electronic control apparatus (not shown), the force of the spring 84 is overcome and the armature 82 is drawn thereagainst. Thus, the sphere 73 is lifted from valve seat 74. This results in a flow of pressurized fuel from supply bore 27 through bore 75 into the control chamber 76. The force of this fuel flow presses the sphere 73 to the second valve seat 79 closing off the bore 77 leading to the return conduit 36. Thus the fuel flows through bores 27, 75 and 81 into the pressure chamber 32. Portions of the valve housing 72 which are under different pressures, are, in the stepped extension 31 of the supply bore 27, sealed from one another by means of packing rings 88, 88a and 88b. Because of the small moving masses of the solenoid valve 12, its switching occurs practically without delay and its switching period is very short.

Referring now to FIG. 3, the servo piston 34 and the pump piston 37 are in their upper position of rest (also shown in FIG. 1), but the solenoid valve 12 is shown in its position in which it allows the flow of fuel from the pressure source to the pressure chamber 32. The delivery stroke of the pump device 13 begins in this position of the solenoid valve 12.

Turning now to FIG. 4, the servo piston 34 and the pump piston 37 are shown in a position in which they have terminated their delivery stroke "H" and the solenoid valve 12 is deenergized, whereupon the armature 82 and the valve sphere 73 reassume their position shown in FIG. 1. As a result, the pressure chamber 32 is depressurized through the return conduit 36.

OPERATION OF THE PREFERRED EMBODIMENT

In the description that follows a full operating cycle of the pump - and - nozzle assembly will be described.

When the solenoid valve 12 is in its position shown in FIG. 1, in the pump work chamber 41 and in the pressure channel 56 there prevails the servo pressure p_s the magnitude of which may be, for example, 100 kg/cm^2. In this position, the supply valve 44 is closed. When the solenoid valve 12 switches over to a position shown in FIG. 3 under the effect of a control signal, the fuel, which is under servo pressure, is admitted from the supply bore 27 into the pressure chamber 32 above the servo piston 34. The force now exerted on the servo piston 34 moves the servo piston 34 and the pump piston 37 downwardly, whereupon in the pump work chamber 41 there is generated an injection pressure p_E, of, , for example, 300 kg/cm^2, which, corresponding to the ratio between the servo piston 34 and pump piston 37 is larger than the servo pressure.

The closing spring 61 of the injection nozzle 17 is in the present example preloaded to 150 kg/cm^2 nozzle opening pressure. The fuel which is under the injection pressure p_E affects the differential face 58 of the valve needle 19, lifts the latter and, as a result, fuel is injected into the engine cylinder.

The injection stroke "H" is terminated when, due to the discontinuation of the control signal to the solenoid valve 12, the latter is switched over into its position shown in FIG. 4. At that moment, the pressure chamber 32 is suddenly depressurized, the fuel which is under injection pressure and which prevails in the pump work chamber 41 and in the pressure channel 56 moves the pistons 34 and 37 slightly upwardly, whereby the pressure in the channel 56 falls under the nozzle opening pressure. As a result, the valve needle 19 returns into its closed position and the fuel injection is terminated.

As soon as the pressure in the pump work chamber 41, together with the pressure corresponding to the force of the spring 47 falls below the pressure prevailing in the charging bore 46, the supply valve 44 opens and fuel flows into the charging bore 46 and, through port 42, into the pump work chamber 41 and moves the pump piston 34 and the servo piston 37 upwardly. By virtue of the effect of throttle 51, a pressure drop of such a magnitude results, that the servo piston 34 is dampened in its motion as it reaches the upper abutment 35. Such a braking of the servo piston is advantageous since it extends the life expectancy of the pump - and - nozzle assembly and diminishes the operating noise. By coordinating the supply valve 44 and the throttle 51, the charging stroke may be extended until shortly before the beginning of the subsequent injection stroke. The charging stroke which occurs immediately after depressurization of the pressure channel 56, prevents a highly undesirable cavitation which normally would occur at rapid depressurization and vacuum.

The pressure drop caused by throttle 51 in the charging bore 46 and in the pump work chamber 41 lasts only as long as the pump piston 37 and the servo piston 34 move upwardly. As soon as these pistons arrive at their upper abutment 35, the pressure again increases to the servo pressure in the charging bore 46, in the pump work chamber 41, as well as in the pressure conduit 56. This feature has the advantage that the residual pressure prevailing between the pump work chamber 41 and the annular chamber 57 before the beginning of each injection, may be determined exactly by setting the servo pressure to a definite value. If the servo pressure is set in such a manner that it is only slightly lower than the nozzle opening pressure, then the opening of the nozzle occurs very rapidly because of the short delay in injection.

A further advantage of the above-described pump - and - nozzle assembly resides in the fact that the change in difference between the actually injected fuel quantities and the desired injected fuel quantities (quantity scattering) of subsequent injection steps is substantially decreased because of the down-stepped surface ratio between the servo piston 34 and the pump piston 37. It is known that solenoid valves operate with "time scatter," , that is, the opening moments and the closing moments are not always exactly the same in case of identical control periods. Such a "time scatter" results in a quantity scattering of the output of the pump - and - nozzle assembly. Since, according to the invention, the fuel quantities controlled by the solenoid valve are admitted to the servo piston, the quantity scattering in the pump piston is decreased by the surface ratio between the servo piston and the pump piston.

What is claimed is:

1. In a pump - and - nozzle assembly for injecting fuel into internal combustion engines, wherein said assembly is of the known type that includes (a) a reciprocating pump piston associated with a pump work chamber, (b) a servo piston driving said pump piston and having a diameter larger than that of said pump piston, (c) means for supplying fuel under servo pressure for driving said servo piston, (d) means for supplying fuel under said servo pressure to the pump work chamber of said pump piston, (e) a solenoid valve actuated synchronously with the operation of said engine and adapted to assume a first position in which it

directs said fuel to said servo piston causing a forward stroke thereof, thereby triggering the start of fuel injection into said engine and a second position in which it causes said servo piston to be relieved of the servo pressure of said fuel to permit its return stroke to begin and (f) a nozzle through which fuel is injected into said engine, the improvement comprising,

A. means for terminating said fuel injection simultaneously with the start of said return stroke of said servo piston upon assumption of said second position by said solenoid valve and

B. means providing a continuous communication between said pump work chamber of said pump piston and said nozzle.

2. An improvement as defined in claim 1, wherein said means defined in (d) includes a fuel supply channel; said improvement comprises a throttle means to extend the period of fuel supply to said pump work

chamber until the start of the fuel injection of a successive work cycle.

3. An improvement as defined in claim 2, including a fuel supply valve formed of a check valve disposed between said throttle means and said pump work chamber, said check valve includes means for biasing the same to a fuel charging pressure which is below said servo pressure.

4. An improvement as defined in claim 3, including a restriction at said check valve to additionally throttle the fuel supply to said pump work chamber.

5. An improvement as defined in claim 1, wherein said solenoid valve is formed of a pressure-equalized, 3/2-way valve including a displaceable sphere as a movable valve member and an electromagnet including a sole control winding and a movable armature operatively connected to said sphere.

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