

March 25, 1952

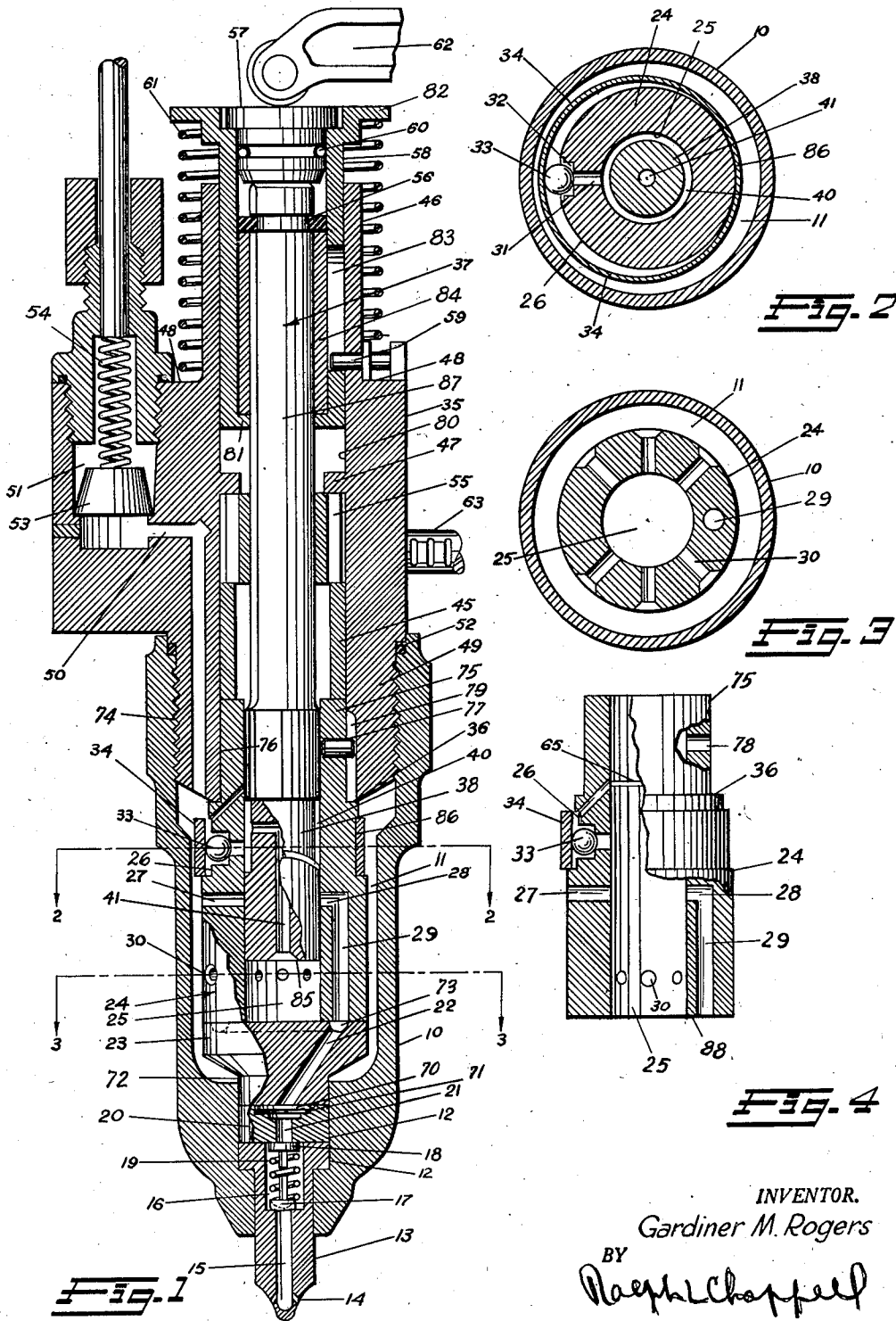
G. M. ROGERS

2,590,575

FUEL INJECTOR

Filed Aug. 2, 1945

2 SHEETS—SHEET 1



INVENTOR.

Gardiner M. Rogers

BY

Ralph Chappel

ATTORNEY

March 25, 1952

G. M. ROGERS

2,590,575

FUEL INJECTOR

Filed Aug. 2, 1945

2 SHEETS—SHEET 2

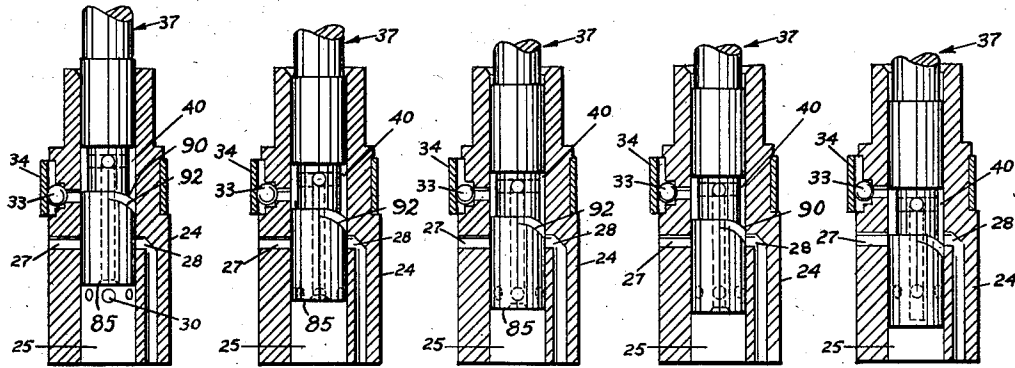


Fig. 10

Fig. 11

Fig. 12

Fig. 13

Fig. 14

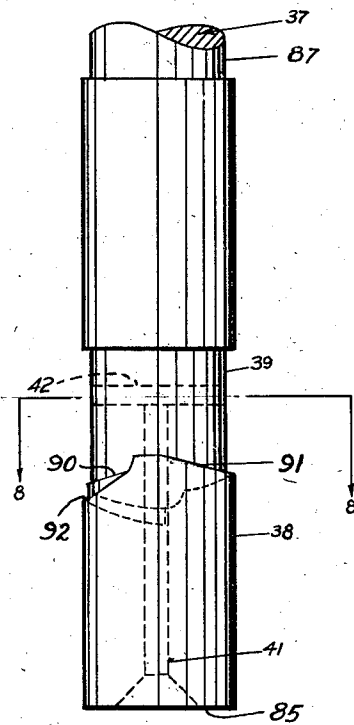


Fig. 7

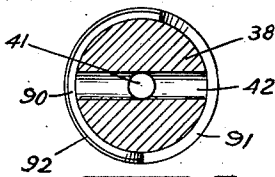


Fig. 8

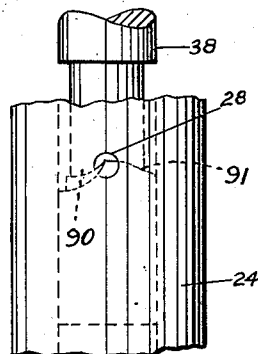


Fig. 9

Fig. 5

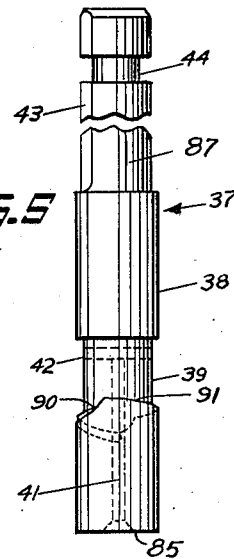
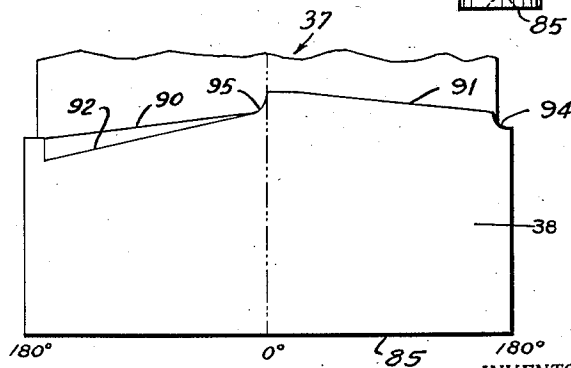


Fig. 6

INVENTOR.
Gardiner M. Rogers

BY

Ralph L. Chappell

ATTORNEY

UNITED STATES PATENT OFFICE

2,590,575

FUEL INJECTOR

Gardiner M. Rogers, Ithaca, N. Y.

Application August 2, 1945, Serial No. 608,588

13 Claims. (Cl. 103-41)

1

This invention pertains to the art of fuel injectors for internal-combustion engines, and is specifically directed to improvements in unitary fuel injectors of the jerk pump type such as are employed, for example, on "General Motors type 71" engines.

An object of this invention is to improve the operating characteristics of fuel injectors for internal-combustion engines, and especially to improve injector operation consistently throughout the range of conditions encountered in use, whereby efficient combustion and engine operation is obtained at cranking and idling speeds as also under load conditions.

Another object is directed to a fuel injector structure of the pump type wherein the pressure of the fuel injected is controlled at an established pressure that provides good atomization for efficient combustion over a wide range of engine speeds.

A further object is to improve the operating characteristics of fuel injectors whereby a more efficient fuel distribution to the engine is obtained.

Still another object is an improved structure whereby good atomization of the fuel injected is obtained at both high and low operating speeds.

It is a further object to provide a fuel injector in which fuel leakage does not materially affect the quantity of fuel metered per stroke, thereby permitting greater manufacturing tolerances with consequent increase in interchangeability of parts.

Further objects and advantages of this invention, as well as its construction, arrangement and operation, will be apparent from the following description with the accompanying drawings, and from the accompanying claims. In the drawings

Fig. 1 is a longitudinal sectional view of one form of an injector that embodies the invention, Fig. 2 is a section taken on line 2-2 of Fig. 1, Fig. 3 is a sectional view on line 3-3 of Fig. 1, Fig. 4 is a side elevation partly in section of a detail of the injector in Fig. 1,

Fig. 5 is a detail elevation of the plunger of the injector in Fig. 1,

Fig. 6 is an enlarged developed profile view of the piston surface of the plunger in Fig. 5,

Fig. 7 is an enlarged detailed elevation of the piston embodied in the plunger in Fig. 5,

Fig. 8 is a sectional view, taken on line 8-8 of Fig. 7,

Fig. 9 is a diagrammatic view that illustrates relationship between the plunger and the working barrel, and

2

Figs. 10 to 14 illustrate the sequence of operation of the piston in the pumping chamber during one stroke of the plunger.

The injector of the present invention is constructed to utilize the full capacity of its pump only at cranking and idling speeds, so that sufficient pressure for proper fuel atomization is maintained even at low operating speeds. As the speed of the engine is increased, surplus fuel is vented or spilled back into a reservoir through the use of a pressure-control valve, which operates to main adequate pressure on the fuel to insure proper atomization over the desired range of operating speeds. Thus, good fuel atomization is maintained at the various operating speeds of the engine. Means are also provided to divide the quantity of fuel injected into a preliminary and main charge, which are fed to the engine in sequence and timed to insure efficient and smooth combustion.

Fig. 1 shows a section taken on the longitudinal centerplane of an injector that embodies the invention. The general form of this type of injector is the so-called jerk pump type similar to that employed, for example, in the "General Motors series 71" diesel engine.

The disclosed injector comprises an outer housing or shell 10, provided with a cavity 11, which terminates at its end in a shouldered bore 12 forming an opening to the exterior of the shell 10 at its end. A nozzle 13 fits in the bore 12 and seats against the shoulder thereof as shown, the nozzle being extended through the bore 12 to project out of the end of shell 10. The nozzle 13 is provided with an orifice 14 preferably consisting of a plurality of orifice openings which spray the fuel that is forced through them under pressure. When the injector is mounted on an internal-combustion engine positioned for use in the manner of prior art practice, the exposed tip of the nozzle 13 including the orifice 14 projects into a cylinder of the engine to feed or deliver fuel thereto.

The nozzle 13 is provided with the bore 15 that serves as a passage for fuel to the orifice 14, and the nozzle is provided with a counterbore that constitutes the chamber 16 which serves as a housing for a spring loaded valve consisting of the spring guide and stop pin 17, closure member 18 and the spring 19.

A spacer member 20 also fits in the bore 12 against the nozzle 13 and has a central aperture or duct 21 terminating in an enlarged check-valve chamber 70. A check valve 71, in the form of a clover-leaf shaped disc, and with its

edges notched accordingly, seats in the chamber 70. Check valve 71 permits the passage of fuel downward to the spray nozzle 13, but prevents reverse flow of the fuel in the event the closure member 18 leaks.

A base or abutment piece 23 is provided with a cylindrical plug 72 that fits in the bore 12 of the shell 10, and seats against the spacer member 20. The remainder of the abutment piece 23 projects into the cavity 11 of the shell 10. The piece 23 is pierced lengthwise by a duct 22 that extends from an annular chamber or groove 73 in one end face of piece 23 to its opposite end where the duct 22 communicates with the check valve chamber 70 of the spacer piece 20. Sleeve member 24 seats or abuts against the end face of the base piece 23 that embodies the annular groove 73, and is disposed centrally of the cavity 11 in the shell 10. The sleeve 24 is provided with a lengthwise duct 29 having the same radial disposition as the annular groove 73 with which it communicates.

The body or base 35 of the injector comprises the nipple 49 onto which the shell 10 is screw-threaded at 74 to close the cavity 11, the gasket 52 being provided to make the closure fluid-tight. As will be seen more fully hereinafter, the various components inside the shell 10 are held in proper operating positions relative to each other, in the manner illustrated in Fig. 1, by the shell 10 being attached to the nipple 49 of the body 35.

The sleeve member 24 forms the body of the pump, and as shown in Fig. 4 is generally cylindrical in shape, comprising a central cylindrical working barrel or pumping chamber 25. The plunger 37, Fig. 1, provides the piston 38 of the pump. The body 35 of the injector comprises the bore 76 which is directed upwardly into nipple 49 coaxially with the threads 74, and accordingly the bore 76 is disposed coaxially with the bore 12 when the shell 10 is attached to the body 35. One end of the sleeve 24 is reduced to provide a cylindrical plug 75 concentric with the working barrel 25. The plug 75 fits into the bore 76 and extends therein to the shoulder 36 which abuts against the end of the nipple 49 when the sleeve 24 is positioned in the housing of the shell 10 attached to the body 35. This construction positions the working barrel 25 centrally in the shell 10 concentrically with the bore 76, and it positions duct 29 in communication with the annular groove 73 of abutment piece 23. Lengthwise thrust of the abutment piece 23 against the end face 88, Fig. 4, of the sleeve 24, and thrust of the abutment shoulder 36 against the end of the nipple 49, operates to hold the pump of the sleeve 24 rigidly positioned in the housing of shell 10 screwed onto nipple 49. A dowel hole 78, Fig. 4, is formed in the wall of the plug 75, and receives the dowel or keying pin 77 that engages the keyway 79 in the bore 76 of nipple 49, thereby preventing rotation of the sleeve 24 in the housing of shell 10.

The cavity 11 constitutes a reservoir of fuel around the sleeve 24, and it is supplied from the chamber 51 in a manner similar to prior art practice. A set of ducts 50 extend from the chamber 51 to the reservoir 11 for circulation of fuel from the chamber 51 through the ducts 50 to and around the reservoir 11, and by this means the reservoir 11 is constantly supplied with fuel. Filters 53 prevent foreign particles that are non-fluid from entering the reservoir 11, each filter

53 being backed by a compression spring 54 to hold it seated in place in the chamber 51.

The piston 38 fits in the working barrel 25, and is operated therein by the lengthwise reciprocation of the plunger 37. The shank 87 of the plunger 37 extends away from the piston 38, and is projected out of the end of sleeve 24 that embodies the plug 75 and through the bore 76 of body 35 into position for its actuation by the pump operating mechanism.

The bore 80 extends into the body 35 from the direction opposite to and coaxial with the bore 76, both bores extending to the internal annular shoulder 47 between them. The shank 87 of the plunger 37 passes through the shoulder 47 and through the bore 80. The body 35 is provided with the external cylinder or collar 46, coaxial with the bore 80, and extending to the shoulder 48.

The sleeve 58 fits in the bore 80, and at its one end has the inwardly directed shoulder 81 that fits around the shank 87 of the plunger 37. At its other end the sleeve 58 is provided with the outwardly directed flange or shoulder 82, which serves as a seat for one end of the compression spring 61, the spring being disposed around the collar 46 of body 35 and bearing at its other end against the shoulder 48 of collar 46. The spring 61 thereby tends to actuate the sleeve 58 out of the bore 80 by pressing against flange 82. Sleeve 58 is provided with the longitudinal slot 83. Stop pin 59 is projected through the wall of collar 46 into the bore 80 of body 35 and into engagement with the slot 83 of sleeve 58. The stop pin 59 holds the sleeve 58 in the bore 80 against the action of the spring 61 tending to push the sleeve 58 out of bore.

The annular groove 44, Fig. 5, near the end of the shank 87 of plunger 37, is adapted to receive the horseshoe washer 56, Fig. 1, that fits inside the sleeve 58. The spacer sleeve 84 fits around the shank 87 of the plunger 37 and inside the sleeve 58, and it extends from the internal shoulder 81 of the sleeve 58 on which it rests to the horseshoe washer 56 against which it bears. Under action of the spring 61, shoulder 81 of sleeve 58 is pressed against the horseshoe washer 56 with the spacer sleeve 84 between them, whereby the sleeve 58 is attached to the plunger 37 and the two of them move together.

To assemble the plunger 37 and the sleeve 58, spacer sleeve 84 is positioned around the shank 87 of plunger 37 and against the shoulder 81 of the sleeve 58, which is actuated along the shank 87 of the plunger 37 towards the piston 38 until the annular slot 44 projects out of its end far enough for the horseshoe washer 56 to be positioned in slot 44. Now the sleeve 58 is pressed into the bore 80 against the tension of spring 61 until the pin 59 can be projected into the slot 83 of sleeve 58. When the spring 61 is now released to act, it presses the end of slot 83 against the pin 59, and presses the shoulder 81 of sleeve 58 against the horseshoe washer 56, thereby holding the elements assembled in the body 35 of the injector.

The abutment head 57 fits in the end of the sleeve 58 that embodies the flange 82, and it engages the end of the plunger shank 87. Abutment head 57 is held in position in sleeve 58 by the spring clip 60. The rocker arm 62 presses against the abutment head 57 and thereby actuates the plunger 37 opposed to spring 61 in the downward direction of Fig. 1, and by this means the piston 38 moves downwardly in working

5

barrel 25 to perform its working or pumping stroke. The spring 61 actuates the plunger 37 in the opposite upwardly direction during the upward stroke of the rocker arm 62, and by this means the piston 38 moves upwardly in the working barrel 25 to perform its fuel intake stroke. The slot 83 is elongated to permit lengthwise reciprocation of the plunger and sleeve assembly 37—58 relative to pin 59 when the plunger 37 is reciprocated down and up in the manner described.

The pinion 55 fits in the cavity of the bore 76 of body 35, and is keyed to the shank 87 of the plunger 37 by means of the flat 43 thereof. See Fig. 5. The spacer sleeve 45, Fig. 1, fits in the bore 76 and is positioned between the end of sleeve 24 and the pinion 55, whereby the pinion 55 is held adjacent to the internal shoulder 47 and is thus properly located within the body 35. The pinion 55 is properly positioned in the bore 76 when it and the internal components of the injector are housed in the manner described hereinbefore by the shell 10 being attached to the nipple 49. The rack 63 is geared to the pinion 55 and is operable in a manner similar to prior art practice to rotate the pinion, thereby adjustably rotating the plunger 37 to control the quantity of fuel supplied to the engine.

The sleeve 24 is transversely bored to provide a plurality of radially disposed holes that constitute the intake port 30, Fig. 3, through which fuel flows from the reservoir of cavity 11 into the working barrel 25. The bore 27 is directed radially through the sleeve 24, and is extended beyond the bore of working barrel 25 to include the bore 28 on the opposite side. The bore 28 connects with the lengthwise duct 29 to provide a fuel passage out of the working barrel 25 that extends to and through the end face 88 of the sleeve 24. Bore 28 thus constitutes the feed port of the pump. Bore 27 comprises a by-pass port for the passage of fuel from the working barrel 25 into the reservoir 11.

Fuel that flows out of the working barrel 25 through port 28 passes through lengthwise duct 29 and annular groove 73 into duct 22, from which it enters the check-valve chamber 70. The fuel is free to pass from chamber 70 into the duct 21 through the notches around the edge of the check-valve disc 71. To enter the chamber 16 from duct 21 the fuel must be under sufficient pressure to overcome the spring 19 and displace the closure member 18. The stop pin 17 is provided with a transverse notch, shown in dotted lines in Fig. 1, through which fuel is enabled to pass from the chamber 16 into the passage 15, and from passage 15 the fuel is driven under pressure through orifice 14 into the engine.

A valve passage or duct 31 leads from the pumping chamber 25 to a ball valve recess 32. The duct 31 is a bore through the wall of the working barrel 25. The recess 32 is a counter-bore exteriorly of the pump body 24 and is large enough to receive the ball 33, the recess 32 extending inwardly to a depth that is predetermined for the ball 33 to project out of the recess a predetermined amount. See Figs. 2 and 4. The port 31 constitutes a vent for the flow of fluid out of the working barrel 25 and the ball 33 constitutes a closure therefore.

The ring or band 34 holds the ball 33 in closed position seated on the port 31. A cylindrical groove 26 is turned in the exterior surface of the pump body 24 at the recess 32 to form a seat for the band 34, the groove 26 of the dis-

6

closed embodiment being concentric with the working barrel 25. The flow of fluid out of the working barrel 25 through the vent 31 is resisted by the band 34 tending to hold the ball 33 seated. The fuel in the working barrel 25 is stressed under pressure by operation of the pump 38—25, and when this pressure reaches a predetermined magnitude sufficient to overcome resistance of the band 34, the closure 33 is lifted from the port 31 and fuel is thus vented out of the working barrel 25. The port 31, ball closure 33 and band 34 therefore constitute a valve which operates to control the operating pressure in the working barrel 25.

The bore 25 is considerably larger, for an engine of commensurate size, than the bore of prior art positive-displacement pumps which are constructed for the entire pump displacement to be injected. Applicant's pump is constructed for a displacement in excess of power requirements of the engine, and it is that portion of the pump displacement which is not injected into the engine that is by-passed back to the reservoir 11 through the vent 31. The orifice opening of the vent 31 is varied by the closure 33 being lifted to different heights, and the closure 33 is lifted under pressure of the working barrel 25 to the height that affords the magnitude of orifice opening large enough to accommodate whatever portion of the pump displacement is not injected into the engine. The pressure in the working barrel 25, and accordingly the operating pressure of the pump 38—25, is balanced against the stress of the band 34 tending to seat the ball 33.

The band 34 is constructed of any suitable material that affords the desired degree of resiliency, steel, and more particularly a suitable spring steel, having been used and found satisfactory. Homogeneous composition circumferentially is important, and the ring 34 therefore constitutes a continuous or closed band that is preferably constructed of seamless tubing or rod stock.

Band 34 is preferably manufactured circular, with predetermined inside and outside diameters that are concentric. These dimensions are precisely established to determine and control the operating pressures of the pump 25—38, and to control and vary the operating characteristics of the pump, the injector and the engine under circumstances that will be described more fully hereinafter.

The annular groove or seat 26 provides a rest at 86, Fig. 2, which is opposite the recess 32. The inside surface of the band 34 bears against the rest 86 at one point in its circumference, and at an opposite point the band is pressed over the ball 33 positioned in the recess 32. This operates to distort the ring 34 from its normal circular contour to an ovate contour, the extent of ovate distortion depending upon the inside diameter of the ring with reference to the greater dimension measured from the surface of the rest at 86 to the remote surface of the ball 33.

The ring 34 has become spring-tensioned by its distortion to ovate contour, and the inside diameter of the ring is precisely predetermined to produce the exact extent of ovate distortion that affords the desired character of spring tension in the valve of ball 33. When the ball 33 is lifted from port 31 by the pressure in the working barrel 25 attaining a magnitude sufficient to overcome the tension of spring 34, the magnitude of ovate distortion of the ring is in-

creased accordingly, and the tension of the spring 34 is thereby increased. When the valve of ball 33 is in open condition, the pressure within the working barrel 25 continues to be balanced in opposition to the spring tension of band 34, and the operating pressure of the pump 38—25 is accordingly predetermined by the degree of ovate distortion of the band 34.

The wall thickness of the band 34 in addition to its inside diameter is a factor which, considering the width of the band also, determines the magnitude of resilient resistance that the band 34 exerts. The width of the band 34 being standard to fit the seat 26, the outside diameter of the band is precisely established for determining the resiliency characteristics of the band. Thus, the various dimensions of the circular band 34 are established to provide resiliency characteristics that provide the desired pressure characteristics in the pump 38—25.

The band 34 being manufactured circular, it is capable of being machined more easily to accurate dimensions within close manufacturing tolerances. This is an important factor of merit in the pressure relief valve 33—34 of the present invention. It enables the pressure characteristics of the pump to be controlled precisely, thereby providing operating characteristics in the engine that are precisely best suited to meet the demands of the service expected from the contemplated engine installation. It also enables the operating characteristics of the engine to be variably controlled easily and at low cost by replacing its set of rings 34 with a set having different dimensions providing different desired resiliency characteristics. The attribute of the dimensions of the rings 34 being determinable precisely accurately also functions to establish uniformity in the operating characteristics of different cylinders of a multiple-cylinder engine, and this results in smoother performance of any engine embodying the invention.

Spring operation of the ring 34 extends from its contact with the ball 33 in opposite directions annularly towards the rest 86. Surface contact between the rest 86 and the inside surface of band 34 extends from the mid-point of contact at 86 away therefrom in opposite directions annularly. The portion of the band 34 that affords resiliency is that portion which is not in contact with the seat 26 at rest 86. Any increase in the extent of contact at rest 86 produces a corresponding diminution of the extent of the resilient portions of the band 34, and increased ovate distortion of the band caused by the valve of vent 31 being opened to a greater extent increases the extent of surface contact between the ring 34 and seat 26 at rest 86.

When the ball 33 is lifted, the two portions of the ring 34 that extend away from the ball 33 in opposite directions annularly to the points where contact is made with seat 26 at rest 86, in addition to becoming shortened, become progressively straighter, and they thus become increasingly stressed under tension at the same time losing quality of flexure. It is possible to dimension the spring 34 for its legs extending away from ball 33 to approach straightness when valve 33—34 is opened to its fullest extent, and then the spring becomes virtually rigid, i. e., its resistance against further distortion approaches infinity. In practice it is desirable that the spring 34 operate resiliently at any extent to which the valve 33—34 becomes opened.

The inherent resiliency characteristic of the

band spring 34 is that its tension increments increase progressively with unit increments of spring displacement. The curvature of the rest 86 with reference to the inside diameter of the band 34 is an additional factor which contributes to this characteristic of resiliency. Resistance to ovate distortion of the spring 34 accordingly increases in increments that increase progressively as the valve 33—34 is opened to progressively greater extents. When the fuel bypassed through the valve of ball 33 constitutes an increased percentage of the pump displacement, the percentage of the pump displacement that is injected into the engine is correspondingly reduced. Then the progressively increasing increments of increased spring tension resulting from the by-pass valve 33—34 being opened to a greater extent operates to compensate for the reduced quantity of fuel injected through the nozzle 14 which has a fixed orifice opening. Increased operating pressure of the pump 38—25 is thereby provided, which compensates to produce atomization that enables efficient combustion of the reduced charge of fuel.

In the practice of the invention, a spring 34 is selected with dimensions which afford desired resiliency persistently over the contemplated range of orifice openings of the valve 33—34, and which establishes operating pressures in the working barrel 25 that are suitably high for good atomization under all contemplated conditions of engine operation either under load or when idling. Thus, efficient combustion is attained throughout the range of speeds at which the engine is constructed to operate. This is an important advance in the modern trend towards higher engine speeds. In addition, efficient combustion at idling speed inhibits engine fouling. Also, applicant attains high pump pressure while starting, and easier engine starting than under existing prior art practice is attained by practice of the present invention.

The diameter of the cylindrical surface of the seat 26 is important. It is constructed to a diameter which provides a radius of curvature less than the radius of curvature of the inside diameter of the band 34 by the predetermined amount that produces the desired extent of surface contact at rest 86 when the band is assembled in the valve 33—34. The depth of the recess 32, a ball 33 of given diameter being employed, is an additional factor of importance. These dimensions of elements of the pressure control device that are embodied in the pump body 24 are established, with reference to the inside and outside diameters of the band 34, pursuant to the purpose of attaining the hereinbefore described efficient engine performance throughout the range of operating conditions contemplated in use.

The piston 38 has a reduced waist or annular groove 39, Fig. 5, which breaks the lengthwise continuity of the cylindrical surface of piston 38 and forms an annular pressure chamber 40 in the working barrel 25 when the plunger 37 is assembled in the sleeve 24 as shown in Fig. 1. A transverse bore 42, Figs. 7 and 8, extends across waist portion 39 of the piston 38 and connects with an axial fuel passage 41 leading to the end face 85 of the piston 38. The ducts 41 and 42 provide communication between the two portions of the pumping chamber comprising the annular chamber 40 formed by groove 39 and the space within the working barrel 25 between the member 23 and the end face 85 of the piston 38. See Fig. 1.

The end face 85 of the piston 38 constitutes a valving shoulder for the intake port 30. One of the edges formed by the waist portion 39 of the piston 38 is contoured to provide valving shoulders 90 and 91 for the feed port 28 and the by-pass port 27 respectively. Fig. 6 is a developed view of a portion of the piston 38 of the plunger 37, illustrating the profile of the valving shoulders 90 and 91 for the respective ports 28 and 27, the shoulder 90 extending to the left, and the shoulder 91 to the right, of the center line "0°."

The valving shoulder 90 is positioned relative to valving shoulder 85 lengthwise of the piston 38 to open the feed port 28 at a predetermined time interval during the downward stroke of the plunger 37 after the intake port 30 has been closed by its valving shoulder 85. Closure of intake port 30 operates to trap fuel in the working barrel 25, which becomes stressed under pressure by the continued downward travel of plunger 37. The magnitude of this pressure is invariably sufficient to overcome the spring 19 and therefore to inject fuel into the cylinder of the engine through the orifice 14 of the nozzle 13. As previously explained, the pressure in the working barrel also becomes sufficient to overcome the tension of spring 34, and the entire pump displacement spills through the vent 31 until the port 28 is opened. The pumping pressure under control of the spring 34 is sufficient for good atomization, and this pressure is available for injection when the port 28 is opened.

The valving shoulder 91 is located at a position as shown in Fig. 6 that is upwardly from valving shoulder 90, and therefore valving shoulder 91 opens the port 27 later during the downward stroke of the plunger 37 than valving shoulder 90 opens port 28. The operation of the by-pass valve 27 being opened causes the pressure in the working barrel 25 and in the chamber 40 to be relieved, and it drops off to a magnitude that is not sufficient to sustain injection. The spring 19 therefore actuates the closure member 18 to close the duct 21, and the flow of fuel through the duct 28 and orifice 14 stops. The fuel that is displaced by the continued downward movement of the plunger 37 after the port 27 is opened becomes by-passed into the reservoir 11 through the passage of duct 27.

The valving shoulder 90 is preferably contoured helically as indicated by its downward inclination to the left away from the center line "0°" in the developed view of Fig. 6. Valving shoulder 91 also is preferably contoured helically, and is inclined downwardly in the direction to the right away from center line "0°" in Fig. 6.

Rotation of the plunger 37, for example, by means of the rack 63, Fig. 1, operates to regulate the time during the downward stroke of piston 38 when the valving shoulder 90 passes and opens the port 28, and the time when the valving shoulder 91 passes and opens the port 27 is also regulated thereby. Rotation of the piston 38 in the counterclockwise direction in Fig. 3 corresponds with moving the valving shoulders 90 and 91 relative to their respective ports 28 and 27 to the right in Fig. 6, the effect being relative movement of the ports 28 and 27 to the left in Fig. 6. The port 28 is thereby opened at an earlier time during the downward stroke of plunger 37 than previous to the adjustment, and the port 27 is opened later than its previous opening. By the port 28 being opened earlier and the port 27 later,

the time interval in the pumping stroke is prolonged during which injection takes place.

Clockwise movement of the plunger 37 in Fig. 8 has the opposite effect. The valving shoulders 90 and 91 move relative to their respective ports 28 and 27 to the left in Fig. 6 causing the port 28 to be opened later and the port 27 to be opened earlier, the time interval of injection being reduced accordingly. This adjustment of the valving shoulders 90 and 91 to the left in Fig. 6 can be extended to a condition of the by-pass valve 27 being opened simultaneously with feed port 28, or slightly earlier. In this event, no injection takes place, and this adjustment is available to stop the engine. The arcuate portion 94 at the extreme right of valving shoulder 91 in Fig. 6 is provided for the port 27 to register therewith when the arcuate portion 95 at the extreme right of valving shoulder 90 registers with port 28. See also Fig. 9. It will be recalled that the ports 27 and 28 are diametrically opposite each other, or 180° apart, in the working barrel 25.

The valving shoulder 90 for the feed port 28 is stepped as indicated at 92 in Fig. 6, and as shown exaggerated in Figs. 5, 7, 8 and 9. This serves to inject a preliminary charge into the engine in advance of the main charge that supplies engine power requirements, the preliminary charge operating to enhance combustion of the main charge.

In practice, the step 92 is formed by grinding a recess into the cylindrical surface of the piston 38 adjacent to the valving shoulder 90. The depth of the recess is predetermined for a preliminary charge of predetermined magnitude, and in practice a very slight depth serves the purpose, about one thousandth of an inch (0.001") being sufficient.

The recess of step 92 extends away from the valving shoulder 90 a predetermined distance lengthwise of the piston 38 for the step 92 to open the port 28 a predetermined time interval before the valving shoulder 90 reaches the port 28 to open it fully. This times the preliminary charge with reference to injection of the main charge.

The preliminary charge is diminutive and is fed to a quantity of air that is in excess of what is required for its combustion. The preliminary charge therefore ignites readily, and is burning before the main charge enters the cylinder. When the main charge enters the cylinder it contains a flame. Ignition of the main charge is accelerated thereby, and this improves the combustion characteristics of the engine to a great extent.

The preliminary charge is a small percentage of the total displacement of pump 38—25, and therefore almost the total pump displacement is by-passed through the vent 31 when the preliminary charge is being injected. When the main charge is being injected, the orifice opening of the valve 33—34 is reduced to accommodate the reduced percentage of the pump displacement that is by-passed therethrough. Ovate distortion of the band 34 is greater when the preliminary charge is being pumped than when the main charge is being injected, thereby providing increased operating pressure of the pump 38—25 that compensates for the diminutive rate of flow through the nozzle orifice 14 and producing good atomization of the preliminary charge.

The cost of the injector of the present invention, as also its maintenance costs, are greatly reduced by practice of the present invention. In

the prior art practice of positive-displacement pumps, the quantity of fuel pumped to the engine to constitute a charge is diminished by any leakage past the piston, and to prevent leakage the plunger and working barrel are manufactured with as little clearance as possible. The costly lapping-in process is conventionally employed, and this produces a piston and working barrel that fit only each other. Leakage past the piston of the pump in the injector of the present invention, so long as it does not constitute all the fuel otherwise by-passed through the vent 31, does not detract from the quantity of fuel fed to the engine. Therefore, the piston 38 and the working barrel 25 can be manufactured to greater permissible production tolerances, and accordingly less costly manufacturing practices are suitable and available. The production cost is reduced thereby, and the parts are made interchangeable.

The annular collector groove 65, Fig. 4, is provided in the wall of the working barrel above the valve ports 28 and 31, the by-pass duct 64 communicates with the groove 65 for the passage of fuel therefrom to the reservoir 11. Leakage between the piston 38 and the working barrel 25 is arrested in the groove 65, and is directed to the reservoir 11 through the duct 64.

The operation of the injector follows, reference being had particularly to Fig. 1 and the sequence diagrams comprising Figs. 10 to 14.

Fuel oil from a suitable low pressure source of supply is fed to filter chamber 51 in pump block 35 from whence it flows through the oil ducts 50 into the reservoir 11 formed by the annular space around the sleeve 24 in shell 10. The reservoir 11 provides a source of fuel for the pump 38-25 and supplies its needs continuously during all stages of its operation. The fuel from the filter cavities 51 circulates in reservoir 11 and serves to cool the injector.

During the suction stroke, as the piston 38 moves upwardly from its position shown in Fig. 14 towards its position in Figs. 10 and 1, the valving shoulder 85 at the end of piston 38 opens the intake port 30. Thereafter during continued upward movement of piston 38, oil from the reservoir 11 is drawn through the intake port 30 into the pumping chamber 25. Figs. 1 and 10 show the plunger piston 38 at the end of the suction stroke.

In this position it will be seen that the feed and by-pass ports 28 and 27 respectively of the pumping chamber 25 are covered by the cylindrical portion of the plunger piston 38. As the injection stroke of the plunger 37 starts, fuel in the lower end of the pumping chamber 25 is forced upwardly through the ducts 41 and 42 into the annular pressure chamber 40.

After the beginning of the downward stroke of the piston 38, the fuel in the working barrel 25 is displaced through the intake port 30 into the reservoir 11 until the intake port 30 is closed by passage of the valving shoulder 85. See Fig. 11. The feed and by-pass ports 28 and 27 respectively continue to be closed. Now the fuel contained in the working barrel 25 and the chamber 40 is trapped, and it is stressed under pressure therein against resistance of the spring 34. When the pressure in the working barrel 25 builds up to a magnitude sufficient to overcome the resistance of spring 34, the ball 33 is lifted from the port 31 in opposition to the tension of the spring 34. This takes place at a time during the pumping stroke of the plunger 37 before the

port 28 is opened. Previous to fuel being fed to the engine through the feed port 28, the total displacement of the pump 38-25 is by-passed into the reservoir 11 through the valve of ball 33. This is so excepting for any leakage past the piston 38, which is also by-passed into the reservoir 11 through the wiper groove 65 and duct 64, Fig. 4.

The port 28 is opened during the downward stroke of the plunger 37 first by the stepped shoulder 92. See Fig. 12. A diminutive preliminary charge is thereby injected into the engine preceding the main or power charge by a predetermined time interval that enables the preliminary charge to become ignited before the main charge enters the cylinder. The main charge is injected when the valving shoulder 90 opens the port 28. See Fig. 13. The flame of the preliminary charge, which is present in the cylinder when the main charge enters, operates to assist ignition of the main charge, the main or power charge being thereby ignited smoothly without detonation or shock which detracts from smooth engine performance. The operating pressure of the pump 38-25 during injection of the power charge is established to produce good atomization and an improved spray pattern, the operating pressure being determined by the pressure in the working barrel 25 being balanced against the tension of spring 34. Components of the valve 33-34 are dimensioned accordingly as hereinbefore explained.

Injection continues after port 28 is opened by valving shoulder 90 until the valving shoulder 91 reaches and uncovers the by-pass port 27, by-passing the oil under pressure in the chamber 40 back into the reservoir 11 which is under relatively low pressure. See Fig. 14. The pressure in the chamber 25 is thereby relieved, and the valve of ball 33 is closed. The spring 19, Fig. 1, also operates to close the valve 18. The injection of the fuel charge is thus completed while the plunger 37 continues to travel to its extreme position in the downward direction. After the vent 31 closed in the manner described, the total displacement of the pump 38-25 is by-passed into the reservoir 11 through the port 27. The cycle is repeated beginning with the suction stroke as shown in Fig. 10.

The plunger 37 is rotated by the pinion 55 in response to a particular setting of the rack 63 which may be actuated manually or by governor action in the usual manner common in the operation of internal-combustion engines. The effect of rotating the plunger 37 is to present different aspects of the helical valving shoulders 90 and 91 of piston 38 to the feed and by-pass ports 28 and 27 respectively. This has the effect of varying the distance lengthwise of piston 38 between those positions along the respective valving shoulders 90 and 91 circumferentially that register with the respective ports 28 and 27, and this operates to vary the interval of time between the respective feed and by-pass ports 28 and 27 being opened. The quantity of fuel that constitutes a charge to meet the power requirements of the engine is thereby determined.

The spring-loaded valve 33-34 controls the pressure of the fuel in the chamber 40 since it acts as a pressure limiting valve. The spring band 34, because of its resiliency, exerts pressure on the valve ball 33, thereby determining the operating pressure in the chamber 40. The spring 34 places an initial loading on the ball 33 sufficient to maintain the necessary pressure for good

atomization at cranking and low speeds. As the engine speed is increased, the rate of pump displacement increases. The rate of fuel flow through the port 31 is increased accordingly, producing increased ovate distortion of the spring band 34, and increased operating pressure in the working barrel 25. Increased rate of flow through port 28 is resisted, the size of the orifice 14 being fixed, and accordingly the effect of increased engine speed is to by-pass a greater percentage of the pump displacement through the valve 33—34, thereby increasing the operating pressure still more. Accordingly, at higher speeds relative to lower speeds, the quantity of fuel fed to the engine is less, and at the same time quality of atomization is enhanced by the increased velocity of fluid flow through orifice 14 as also by increased operating pressure of pump 38—25 generated because of increased ovate distortion of the band 34. This characteristic of the invention is not employed to eliminate use of a governor, but it functions to improve the stability of engine performance under varying load conditions, and over a wide range of engine speeds including idling speed.

Referring to Fig. 2 it can be seen that, by varying the internal diameter of the spring ring 34, the amount of initial tension in the spring when assembled over the valve ball 33 can be changed. A ring which is put on relatively loosely, that is with a small initial tension, will be springy, and will control the valve ball 33 to give a low pressure response at low speeds, the pressure response increasing as the speed increases. That is, the ring 34 will have a slowly increasing spring constant. Similarly a ring of a size that requires a greater degree of initial tension in positioning it over the valve ball 33 generally will be relatively stiff, and will result in a high pressure response at low speed, the amount of lift of the valve ball 33 being progressively resisted in relatively higher increments. Such type of spring can be said to have a rapidly increasing spring constant.

The ring spring 34 also offers means for producing uniformity of pressure level in the pressure chamber 40, Fig. 1, among a series of injectors. This is important in multi-cylinder engine construction where uniformity of operation among cylinders is vital.

For accomplishing the advantages noted hereinbefore, the pressure control valve 33—34 is preferably constructed and assembled in the manner now to be described, reference being made particularly to Fig. 2 of the drawings. During the process of manufacture, the valve ball 33 is inserted into and valve ball recess 32 and is pressed against port 31 to form a seat, and it is further lapped into place until the measurement from the rest 86 of spring seat 26 to the remote surface of the seated ball 33 gages to the predetermined desired value. This dimension can be made readily identical for all injectors of a set for a given engine, and can be established correctly for the desired operating characteristics of the injectors.

The ring spring 34 in turn is ground to predetermined inside diameter and thickness, so that any spring 34 made to specification when assembled on any sleeve 24 and valve ball 33 manufactured to predetermined dimensions as above described, will have a predetermined initial tension as also spring constant values that are predetermined for any magnitude of valve opening of vent 31. By this means the pressure response characteristics of the injectors of a set for a given engine are easily and reliably controlled for a

range of operating conditions they are expected to encounter in their intended use.

In the ordinary jerk pump, the pressure on the fuel at the spray nozzle will vary with the speed of the engine, increasing as the engine speeds up and vice versa. Therefore the pressure of the oil sprayed through the orifice of the spray nozzle will be relatively low at low engine speeds. Ordinarily, small orifice openings are required to secure sufficient atomization of the fuel at such relatively low pressure. The use of small spray orifice openings, however, limits the fuel flow at higher operating speeds, when the velocity of oil flow becomes high necessitating the use of high fuel pressures. This in turn requires almost microscopic fits between parts of the injector in order to minimize the high fuel leakage which tends to occur at such high pressures.

From the above it can be seen that the size of the orifice openings and the range of fuel pressures employed in the nozzles of the prior art impose limitations on engine design. Whereas a prior art injector may be designed for satisfactory operation at a given operating speed it will not operate efficiently over a wide range of speeds and an engine equipped with an injector constructed for full load conditions will be difficult to start at cranking speeds and will cause fouling and poor combustion when idling. Such limitations of the operating characteristics of a fuel injector over a wide range of operating speeds has acted as limiting factor in the field of high speed engine design for the above presented reasons.

The injector comprising the present invention overcomes the prior art difficulty by utilizing a plunger piston 38 of larger size than one normally employed in a jerk pump type of injector unit of equivalent capacity, together with orifice openings 14 of sufficient size to accommodate the quantity of fuel injected. The larger sized piston 38 enables a larger fuel displacement to be had at low engine speeds, the relatively large displacement causing sufficient pressure even at low speeds to cause good atomization of the fuel sprayed through the orifice 14. This construction has been found to result in a marked improvement in the operating characteristics of a standard Diesel engine equipped with injectors embodying the present invention because it permits the engine to be operated for long periods at idling speed without fouling the engine. As the engine speed increases for a given rack setting, and the pressure builds up, the spring-loaded ball 33 is raised from its seat thereby controlling the pressure in chamber 40 at a predetermined level and spilling the excess fuel back into the reservoir 11, so that, at higher engine speeds the proper amount of fuel under sufficient pressure for good atomization will be sprayed through the orifice 14. A rising fuel pressure characteristic with speed can be obtained in chamber 40, if desired, by positioning the upper contour edge of waist section 39 of plunger 37 so that port 31, Fig. 4, is partially covered during the injection interval.

Thus the injector comprising the invention permits satisfactory engine operation over a wide range of operating speeds. The injector, by functioning satisfactorily at cranking speeds to permit easy starting, by operating efficiently at idling speeds over long periods without fouling, and and by delivering power efficiently at full load conditions makes higher engine speeds feasible, in addition to improving the operation of present engines.

From the preceding description of operation, it becomes clear that the amount of fuel metered through the spray nozzle 13 of the injector comprising the present invention depends on the time interval between the opening of the feed port 28 by the valving shoulder 90 and the opening of the by-pass port 27 by valving shoulder 91. Moreover as the engine speed increases the pressure is maintained at a predetermined level in the pressure chamber 40 by the pressure regulating valve 33—34. Thus the time interval between the opening of the feed port 28 and the by-pass port 27 varies directly as the speed of the engine, and, as the amount of fuel injected depends on this time interval, it follows that for a given setting of the rack 63 the amount of fuel dispensed to the engine decreases with increase of engine speed and vice versa, a condition which gives the engine inherent speed stability without dependence on governor or manual control. This is of marked contrast with the action of existing injectors wherein the amount of fuel injected per stroke varies but little with change of speed.

It will be noted that the operating characteristics of the injector of the present invention can be readily modified as desired merely by selecting a spring ring 34 dimensioned to provide the desired resiliency characteristic. Since the pressure in the chamber 40 is regulated by the spring 34, it is obvious that the discharge pressure at which the fuel is sprayed through the spray orifice 14 is controlled by the characteristic of the spring 34. By this means the pressure of the fuel injected can always be regulated at an established value so as not to exceed any predetermined maximum, for a wide range of engine speeds.

From the above description it is clear that the full displacement of the injector pump 38—25 is utilized only at low engine speeds to maintain adequate spray orifice pressure. At low speeds, the fuel leakage is insignificant due to the relatively low pressure of the fuel. Since, at higher speeds, the fuel displacement is not utilized because of the by-passing through the pressure control valve 33—34 as above described, it follows that a certain amount of fuel leakage, as long as it does not exceed the amount to be spilled through valve 33—34, is permissible.

In prior art constructions, the amount metered to the orifice by the plunger depends on the plunger displacement for all operating speeds, and any leakage is critical. That is, each increment of plunger displacement has to be utilized to pump the required amount of fuel to be dispensed through the spray orifice and any fuel leakage in the injector is vital in that it robs from the required amount of fuel to be injected. Microscopic fits between plunger and pumping chamber are required with resulting expense in manufacturing cost and impracticability of interchangeability of parts.

In the present invention such close tolerances between pumping parts are not so critical for reasons presented hereinbefore, and as a consequence, manufacturing tolerances are greater. Interchangeability of parts is thus made possible and mass production possibilities increased.

Use of the stepped shoulder 91 for delivery of a preliminary charge results in an efficient fuel distribution to the engine. By injecting a small quantity of fuel as a preliminary charge, followed sequentially by injection of the main charge on the already burning preliminary charge, a relatively slow, expansive combustion

takes place in the cylinder instead of the detonation resulting when the main charge is injected en masse. As is familiar to those skilled in the art, a relatively slow, expansive combustion results in a desirable smoothness of operation and greater operating efficiency. The relative size and position of the stepped shoulder 91, Figs. 5 to 8, depends on the desired characteristics of the engine and the type of fuel employed, and is shown in Figs. 5 to 8 by way of illustration only.

The injector of the invention has been described and illustrated as a self contained unit in which the pumping mechanism and spray nozzle are combined into a unitary structure ready for application to an engine. It is obvious however, that the novel pumping mechanism and injection control means comprising the present invention may be incorporated into a central pumping unit adapted to be connected by fuel distribution lines to separate discharge nozzles provided in the engine cylinders in the manner well known in the art.

It is to be understood that various modifications and changes may be made in this invention without departing from the spirit and scope thereof as set forth in the appended claims.

The invention described herein may be manufactured and used by or for the Government of the United States of America for governmental purposes without the payment of any royalties thereon or therefor.

What is claimed is:

1. In a fuel injector for an internal-combustion engine, an injector pump comprising a pumping chamber, apparatus for determining the operating pressure of the pump comprising a vent in the pumping chamber and a closure therefore, a spring holding the closure in closed condition and permitting the vent to be opened resiliently, the spring comprising a continuous band around the pump engaging the closure and the exterior surface of the pumping chamber at points that are opposite each other on its inside surface.

2. In a fuel injector for an internal-combustion engine, an injector pump comprising a sleeve embodying a cylindrical pumping chamber, apparatus for determining the operating pressure of the pump comprising a vent from the pumping chamber to the exterior of the sleeve and a closure for the vent exteriorly of the sleeve, a spring holding the closure in closed position and permitting the vent to be opened resiliently, a seat for the spring comprising a cylindrical exterior surface of the sleeve at the closure disposed concentric with the pumping chamber, the closure projecting exteriorly of the sleeve above the cylindrical surface of the seat, the spring comprising a normally circular continuous band engaging the closure and the seat at opposite points on its inside surface.

3. In a fuel injector for an internal-combustion engine, a pump with a capacity in excess of power requirements of the engine and comprising a working barrel and a piston operable therein to pump fuel to the engine, a feed port in the working barrel and the piston comprising a valving shoulder positioned to open the feed port at a predetermined time in the pumping stroke, a pressure control device for the pump comprising a vent in the working barrel, a spring operable resiliently opposed to the operating pressure of the pump to hold the vent closed against the flow of fuel therethrough, the vent comprising an orifice opening that is variable under variations of

17

the operating pressure of the pump opposing the tension of the spring, the valving shoulder being positioned in the piston to open the feed port after the tension of the spring has been overcome to open the vent.

4. In a fuel injector for an internal-combustion engine, an injector pump with a capacity in excess of the power requirements of the engine and comprising a pump body embodying a cylindrical working barrel, a port from the working barrel through the wall thereof to the exterior of the pump body, a closure for the port exteriorly of the pump body, a recess into the exterior of the pump body and extending to the port, the recess being of a size to receive the closure positioned against the port, a spring holding the closure in closed position against the port and permitting the closure to lift away from the port resiliently, a seat for the spring comprising a cylindrical exterior surface of the pump body at the recess disposed concentrically with the working barrel and including a rest opposite the recess, the recess being of a depth for the closure to project out of the pump body above the cylindrical surface of the seat by a predetermined amount, the spring comprising a continuous band positioned in the spring seat and engaging the closure and the rest at opposite points on its inside surface, the band being normally circular and being tensioned to hold the closure seated by being stressed to ovate contour, the resiliency characteristics of the spring depending upon the inside diameter of the band being less by a predetermined amount than the measurement from the closure to the rest.

5. In a fuel injector for an internal-combustion engine, an injector pump with a capacity in excess of the power requirements of the engine and comprising a pump body embodying a cylindrical working barrel, a port from the working barrel through the wall thereof to the exterior of the pump body, a closure for the port exteriorly of the pump body, a recess into the exterior of the pump body and extending to the port, the recess being of a size to receive the closure positioned against the port, a spring holding the closure in closed position against the port and permitting the closure to lift away from the port resiliently, a seat for the spring comprising a cylindrical exterior surface of the pump body at the recess disposed concentrically with the working barrel and including a rest opposite the recess, the recess being of a depth for the closure to project out of the pump body above the cylindrical surface of the seat by a predetermined amount, the spring comprising a continuous band positioned in the spring seat and engaging the closure and the rest at opposite points on its inside surface, the band being normally circular and being tensioned to hold the closure seated by being stressed to ovate contour, the resiliency characteristics of the spring depending upon the thickness of the band and upon the inside diameter of the band being less by a predetermined amount than the measurement from the closure to the rest.

6. In a pump as defined in claim 4, the diameter of the cylindrical surface of the spring seat being of a predetermined magnitude less than the inside diameter of the band for the desired extent of surface contact between the band and rest when the band is seated.

7. In a fuel injector for an internal-combustion engine, an injection pump with a capacity in excess of engine power requirements and com-

18

prising a pumping chamber, a device for controlling the operating pressure of the pump comprising a vent through the wall of the pumping chamber to by-pass the portion of the pump displacement in excess of the injected fuel, a closure for the vent seated against it exteriorly of the pumping chamber, a spring holding the closure seated and being operable to hold the vent open resiliently in balance with and opposed to the operating pressure of the pump to control the orifice area of the vent variably, the spring comprising a continuous band around the pumping chamber and engaging the closure and the pumping chamber exteriorly at opposite points on its inside surface, the band being normally circular and exerting its spring tension by being distorted to ovate contour, ovate distortion of the band varying according to various orifice areas of the vent to control the balancing operating pressure of the pump variably accordingly.

8. In a device for controlling the operating pressure of a pressure fluid apparatus, a vent and a spring operating opposed to the operating pressure of the apparatus to control the flow of fluid through the vent, the spring comprising a normally circular continuous band that is tensioned by being distorted to ovate contour.

9. In a set of fuel injectors for a multiple-cylinder internal-combustion engine, a pump for each injector to feed fuel to its cylinder, a device for controlling the operating pressures of the several pumps uniformly comprising a vent for each pump and a spring for each vent operable resiliently opposed to the operating pressure of its pump to control the flow of fluid through the vent, each spring comprising a normally circular continuous band that is tensioned by being distorted to ovate contour, the dimensions of the several bands being alike for uniform operating pressures in the several pumps of the respective cylinders.

10. For a set of injectors as defined in claim 36, selectively interchangeable sets of spring bands, the dimensions of the several bands of each set being alike and the dimensions of the bands of different sets being different to provide different resiliency characteristics.

11. In a fuel injector for an internal-combustion engine, a pump with a displacement in excess of power requirements of the engine and embodying a working barrel and a plunger comprising a piston operable therein, a feed port in the working barrel, the piston comprising a valving shoulder positioned lengthwise thereof to open the feed port to inject a main charge that supplies engine power requirements at a predetermined time in the pumping stroke of the plunger, the piston comprising a step ground into its face at the valving shoulder to a predetermined depth to inject a diminutive preliminary charge of predetermined quantity, a by-pass port for the passage of that portion of the pump displacement in excess of the injected fuel from inside to outside the working barrel and a closure therefore projecting above the exterior surface of the working barrel, a spring resiliently holding the closure in closed position on the port opposed to the operating pressure of the pump, the spring comprising a continuous band around the working barrel engaging the closure and the exterior surface of the working barrel at opposite points on its inside surface, the feed and by-pass ports being positioned relative to each other lengthwise of the working barrel for the by-pass port to be opened under pressure of the pump before the step of the

19

piston reaches the feed port to inject the preliminary charge.

12. In a fuel injector for an internal-combustion engine, a pump comprising a working barrel and a piston operable therein to inject fuel to the engine, the working barrel comprising a port for fuel passage to the engine, the piston comprising a valving shoulder to open the port for injecting a main charge that supplies engine power requirements, a step adjacent the valving shoulder ground into the piston face to a predetermined depth to inject a diminutive preliminary charge of predetermined magnitude, the step extending a predetermined distance away from the valving shoulder lengthwise of the piston to time the preliminary charge with reference to the main charge.

13. For the injector pump of an internal-combustion engine, a piston comprising a cylindrical surface, an edge of the cylindrical surface extending circumferentially comprising a valving shoulder for injecting the main charge to supply engine power requirements, a recess adjacent the valving shoulder and ground to a depth in the cylindrical surface predetermined to inject a diminutive preliminary charge of predetermined

20

magnitude, the recess extending a predetermined distance away from the valving shoulder lengthwise to time the preliminary charge with reference to the main charge.

GARDINER M. ROGERS.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
1,588,046	Nelson	June 8, 1926
1,596,194	Lang	Aug. 17, 1926
1,732,918	Sutton	Oct. 22, 1929
2,096,711	Fielden	Oct. 26, 1937
2,140,956	Hall	Dec. 20, 1938
2,144,861	Truxell, Jr.	Jan. 24, 1939
2,144,862	Truxell, Jr.	Jan. 24, 1939
2,174,898	Scott	Oct. 3, 1939
2,378,165	Waeber	June 12, 1945
2,380,148	Camner	July 10, 1945
2,441,167	Raspet	May 11, 1948
2,462,363	Cook	Feb. 22, 1949
2,464,288	Belt	Mar. 15, 1949
2,496,804	Meitzler	Feb. 7, 1950