

Oct. 15, 1968

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3,405,700

FUEL INJECTION PUMP

Filed June 1, 1966

4 Sheets-Sheet 1

Fig.1.

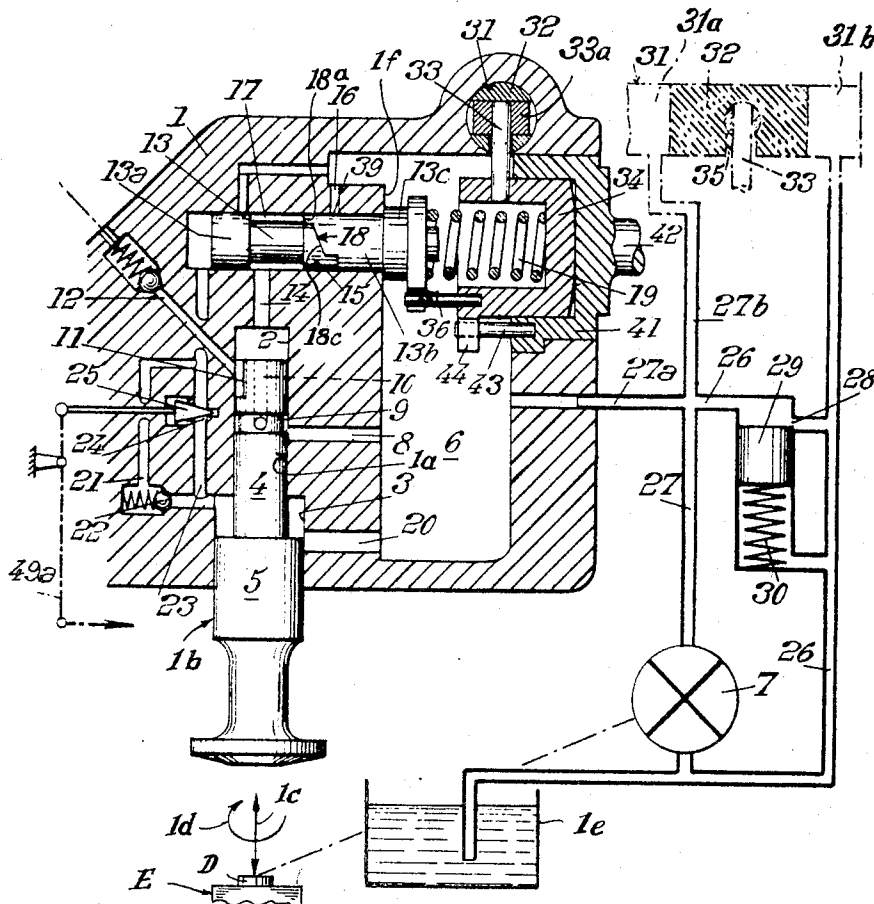
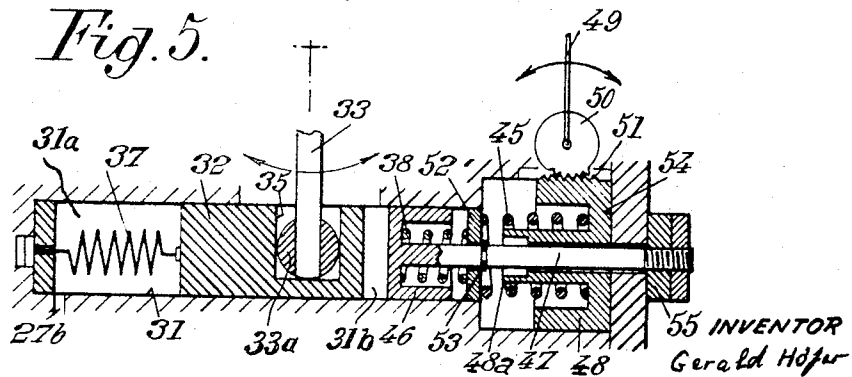


Fig. 5.



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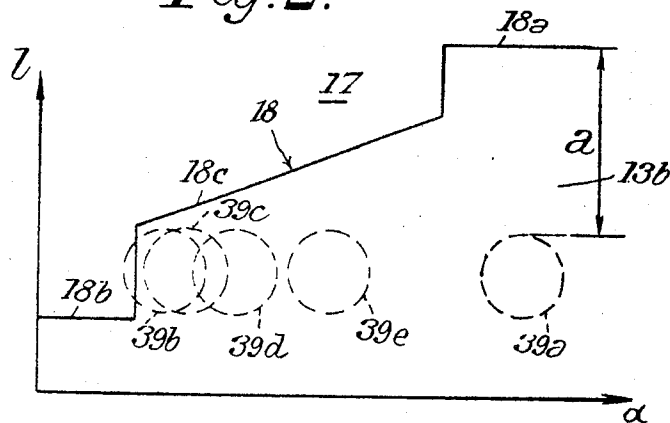
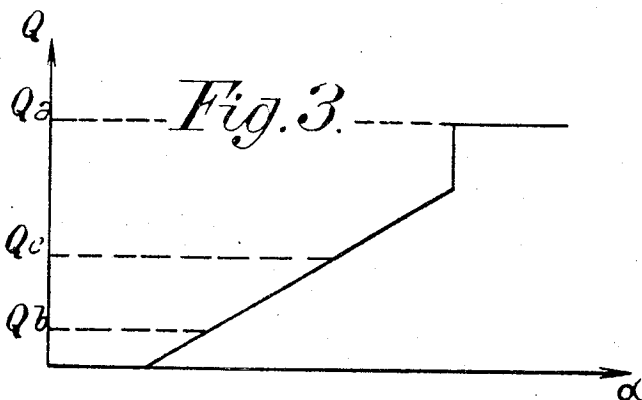
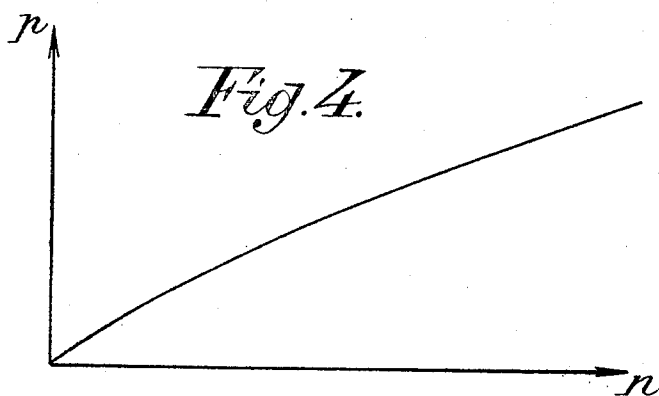
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Fig. 2.



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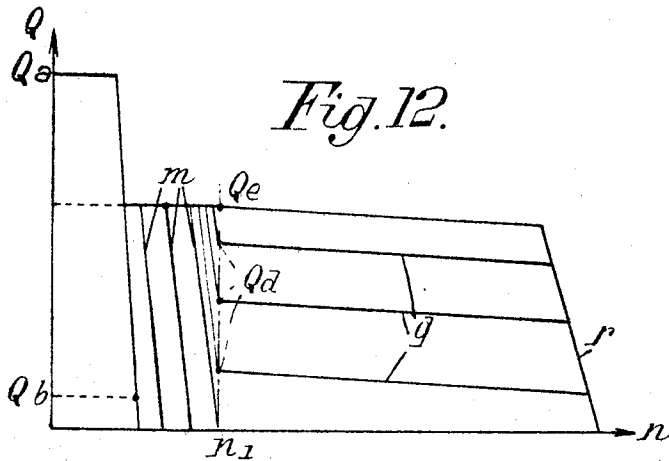
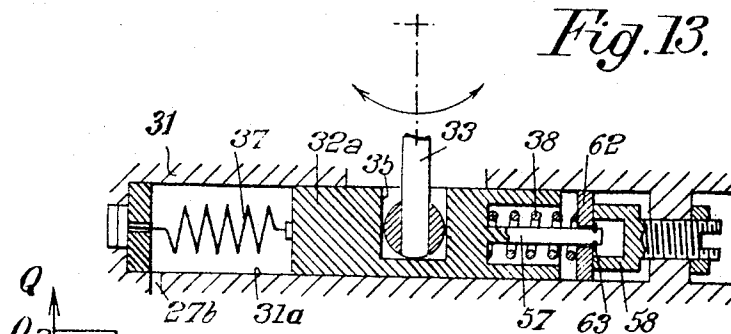
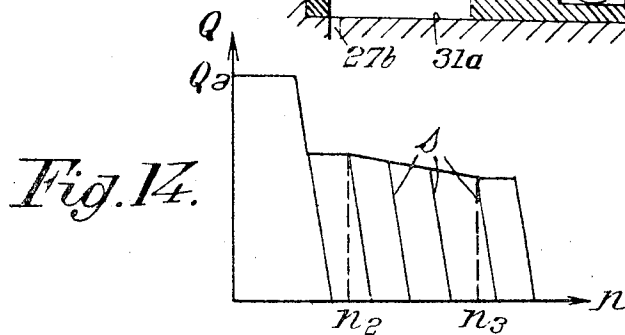
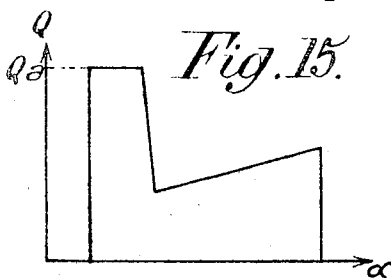
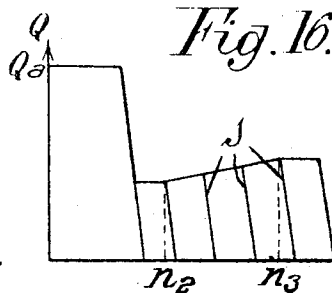
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FUEL INJECTION PUMP

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FUEL INJECTION PUMP

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19,127

6 Claims. (Cl. 123—140)

ABSTRACT OF THE DISCLOSURE

A fuel injection pump for internal combustion engines in which injection of fuel to the cylinders of the engine is interrupted at least after the engine has reached its maximum speed by a regulating valve reciprocable in axial direction and turnable about its axis and which is provided with a peripheral controlling face extending in part inclined to its axis and controlling, depending on the axial and angular position of the valve, a relief channel communicating with the working chamber of the pump, in which the valve is moved in one direction by fuel displaced by an auxiliary pump operating in synchronism with the piston of the injection pump and in the opposite direction by elastic means, and including means operatively connected to the valve for changing the angular position of the latter as a function of the rotational speed of the engine.

The present invention relates to fuel injection pumps, and more particularly to improvements in fuel injection pumps of the type disclosed in Patents Nos. 3,044,404 and 3,122,100 to Bessiere. Still more particularly, the invention relates to improvements in the construction and operation of devices which can regulate the quantities of fuel admitted by a fuel injection pump to the cylinder or cylinders of an internal combustion engine.

The aforementioned patents to Bessiere disclose a fuel injection pump wherein a main piston performs alternating working and suction strokes to respectively expel fuel from and to draw fuel into a working chamber. The main piston not only reciprocates but also rotates at a speed which is synchronized with the rotational speed of the internal combustion engine, and its angular movement is utilized to distribute metered quantities of fuel to the cylinders of the engine. The Bessiere fuel injection pump further comprises a regulating valve which is movable back and forth in response to the action of a return spring and in response to the pressure of fuel supplied by an auxiliary pump whose fuel displacing element operates in synchronism with the main piston. The valve is utilized to regulate the quantities of fuel which are injected into the cylinders during starting and at different rotational speeds of the engine. The principle underlying the operation of such regulating valve is known as a liquid abutment or liquid stop and can be defined as including such adjustment in the strokes of the valve in response to different rotational speeds of the engine that the valve will open an overflow channel from the working chamber at an earlier or later stage while the main piston performs a working stroke whereby the fuel which is expelled from the working chamber finds the path of least resistance and escapes through the overflow channel rather than opening a check valve in the fuel line which connects the working chamber with the injection nozzle of a cylinder in the internal combustion engine.

It was found that the just described fuel injection pump requires further important improvements, especially as regards the selection of various stages in operation of the engine when the regulating valve permits escape of fuel

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through the overflow channel in order to insure that, for example, when the engine operates at full load; its cylinders will invariably receive such quantities of fuel as are needed, at any desired rotational speed, to produce a fuel-air mixture which burns without the generation of smoke. The Bessiere pump must be provided with an adjustable throttle which controls the return flow of fuel from the chamber of the regulating valve, and the throttle must be adjusted for each rotational speed. Such continuous adjustment is undesirable when the engine is built into an automotive vehicle. Furthermore, the regulating action during certain stages of the operation should not be dependent on the rotational speed of the engine and such mode of operation of heretofore known fuel injection pumps can be achieved only by resorting to relatively complicated and sensitive auxiliaries.

Accordingly, it is an important object of my invention to provide an improved fuel injection pump of the type wherein the quantity of fuel which is expelled from the working chamber during a working stroke of the main piston may be regulated by a valve operating on the principle of liquid abutment.

Another object of the invention is to provide a fuel injection pump of the just outlined characteristics wherein the regulating valve is adjustable in response to changes in rotational speed of or in response to changes in load upon the engine in which the pump is being used and wherein the regulating valve may perform a composite movement which includes axial movement and an additional movement, both such movements being utilized to determine the exact quantity of fuel which can be expelled from the working chamber of the cylinders of an internal combustion engine.

A further object of the invention is to provide a novel control unit which can regulate the additional movement of the regulating valve.

Another object of the instant invention is to provide a fuel injection pump which occupies very little room, which can be built into or combined with existing internal combustion engines, which can distribute fuel to engines having one or more cylinders, and which can be readily adjusted so that it can be built into different types of internal combustion engines.

A concomitant object of the present invention is to provide a fuel injection pump wherein the regulating valve may be adjusted in a plurality of ways and wherein at least two such adjustments are fully automatic.

A further object of my invention is to provide a fuel injection pump which operates on the principle of liquid abutment and wherein the regulating valve is not only operated but is also constructed in a novel way.

Briefly stated, one feature of my invention resides in the provision of a fuel injection pump for supplying metered but variable quantities of fuel to one or more cylinders of an internal combustion engine. The fuel injection pump comprises a source of fuel which preferably includes a fuel tank and a supply conduit provided with a fuel pump which is driven by the engine to produce a fuel pressure proportional to the rotational speed of the engine, housing means having a main chamber or working chamber and a relief channel which is connected to the working chamber and may be arranged to return surplus fuel back to the source or into a suction space of the housing means, a regulating chamber provided in the relief channel, a supply conduit adapted to connect the source (for example, the aforementioned suction chamber) with the working chamber in the housing means, at least one discharge conduit adapted to connect the working chamber with a cylinder of the engine, a main piston movable in the housing means to perform alternating suction and working strokes to thereby respectively draw fuel from the source into the

working chamber through the supply conduit and to expel fuel from the working chamber through the discharge conduit or conduits and/or through the relief channel, drive means for rotating and reciprocating the main piston at a rate proportional to the rotational speed of the engine whereby the reciprocatory movement of the main piston results in admission and expulsion of fuel from the working chamber while the angular movement brings about distribution of fuel to one or more discharge conduits, a regulating valve installed in the regulating chamber for angular and axial movement therein to thereby control the flow of fuel from the working chamber through the relief channel, an auxiliary pump having a fuel displacing element operating in synchronism with the main piston to feed fuel into the regulating chamber during working strokes of the main piston and to thereby effect axial movement of the valve against the opposition of a suitable return spring tending to maintain the valve in a starting axial position, and control means including a device for changing the angular position of the valve as a function of the rotational speed of the engine, preferably as a function of the pressure generated by the aforementioned fuel pump. Of course, and since the pressure of this pump is function of the rotational speed of the engine, such rotational speed indirectly controls angular movements of the valve in the regulating chamber.

The valve is provided with a peripheral regulating face which can regulate the flow of fuel between two portions of the relief channel in response to axial and/or angular displacement of the valve with reference to the housing means, the two portions of the relief channel being separated from each other by the regulating chamber.

The valve seals the relief channel during starting of the engine so that each cylinder of the engine then receives a maximum quantity of fuel. On the other hand, at least when the engine is driven at a maximum rotational speed, the regulating chamber receives fuel from the auxiliary pump at such a rate that the valve cannot return to starting position during each suction stroke of the main piston whereby the relief channel invariably allows escape of at least some fuel from the working chamber while the main piston performs a working stroke. Such action of the regulating valve constitutes the aforesaid liquid abutment.

The control means may further include a manually adjustable actuating member which can bring about changes in angular position of the regulating valve. The aforementioned device of the control means preferably includes a control cylinder one chamber of which is connected with the pressure side of the fuel pump, a control piston which is reciprocable in the control cylinder and tends to move in one direction in response to rising fuel pressure, an operative connection between the control piston and the regulating valve to change the angular position of the valve in response to axial movement of the control piston, and resilient means including at least two springs which oppose the fuel pressure acting against the control piston with differences forces. The control means also includes means for adjusting or changing the force of at least one such spring.

The novel features which are considered as characteristic of the invention are set forth in particular in the appended claims. The improved fuel injection pump itself, however, both as to its construction and its mode of operation, together with additional features and advantages thereof, will be best understood upon perusal of the following detailed description of certain specific embodiments with reference to the accompanying drawings, in which:

FIG. 1 is a central section through a fuel injection pump which embodies one form of my invention;

FIG. 2 is a diagram showing relative positions of the regulating valve and the inlet of a portion of the relief channel in different angular positions of the valve;

FIG. 3 is a diagram showing the relationship between the quantities of fuel which are injected into the cylinders of an internal combustion engine and the angular position of the regulating valve;

FIG. 4 is a diagram whose curve illustrates variations in fuel pressure as a function of the rotational speed of the engine;

FIG. 5 illustrates in section certain components of the control unit which adjusts the angular position of the regulating valve;

FIG. 6 illustrates the structure of FIG. 5 with its components in positions they assume when the engine is at a standstill;

FIG. 7 shows the parts of FIG. 5 or 6 in positions they assume when the engine is idling;

FIG. 8 shows the structure of FIG. 7 but with the parts in positions they assume when the engine is running under partial load;

FIG. 9 shows the same structure with the parts in positions they assume when the engine operates at full load;

FIG. 10 again shows the same structure with the parts in positions they assume when the engine operates at full load and at maximum rotational speed;

FIG. 11 illustrates the same structure as FIGS. 5 to 10 but showing the parts in positions they assume when the operator decides to arrest the engine;

FIG. 12 is a diagram showing different stages in operation of the fuel injection pump which is illustrated in FIGS. 1 and 5 to 11;

FIG. 13 is a section through a modified control unit for the regulating valve of the fuel injection pump shown in FIG. 1;

FIG. 14 is a diagram showing the relationship between the quantity of injected fuel and the rotational speed of the engine which embodies the structure of FIG. 13;

FIG. 15 is a similar diagram showing the relationship between the quantity of injected fuel and the angular position of the regulating valve which is controlled by the unit of FIG. 13; and

FIG. 16 is a diagram similar to that of FIG. 14 but showing a different relationship between the rotational speed and the quantity of injected fuel.

Referring first to FIG. 1, there is shown a fuel injection pump which comprises a housing 1 having a composite blind bore 1a receiving a multi-stage piston 1b. The piston 1b comprises a smaller-diameter first stage 4 (hereinafter called main piston) which is slidable and rotatable in the housing 1 and defines therewith a working chamber 2 constituting the innermost part of the bore 1a. The larger-diameter second stage 5 of the multi-stage piston 1b constitutes the liquid displacing element of an auxiliary pump and defines with the housing 1 an auxiliary pump chamber 3 occupying that portion of the bore 1a which is adjacent to its open end. The multi-stage piston 1b is driven by the drive D of an internal combustion engine E so that it performs a composite movement including a reciprocatory movement (arrow 1c) and a rotary movement (arrow 1d). The drive D which moves the multi-stage piston 1b preferably includes a cam and follower unit (not shown) which may be of the type as disclosed in the copending application Ser. No. 538,664 of Eheim, now Patent No. 3,333,542. The main piston 4 serves to distribute metered quantities of fuel to the cylinders of the engine E.

The housing 1 further defines a suction space 6 which is connected with a tank 1e or another suitable source of fuel. The feed conduit 27 which connects the tank 1e with the suction space 6 contains a fuel pump 7. The pressure produced by the pump 7 is a function of the r.p.m. of the engine E. The operative connection between the drive D and the fuel pump 7 is indicated in FIG. 1 by a phantom line.

The supply conduit which can deliver fuel from the suction space 6 to the working chamber 2 comprises a bore 8 connecting the suction space 6 with the smaller-

diameter portion of the blind bore 1a, a circumferential groove 9 in the periphery of the main piston 4, and an axially extending bore 10 provided in the main piston 4 and communicating with the groove 9. The outlet of the bore 8 is positioned in such a way that it can be sealed by the main piston 4 when the latter performs a working stroke, i.e., while the main piston moves upwardly as viewed in FIG. 1 to expel fuel from the working chamber 2. The inner end portion of the main piston 4 is formed with an axially extending peripheral groove 11 which forms part of the discharge conduit and is in permanent communication with the working chamber 2. This axial groove 11 terminates short of the circumferential groove 9 and communicates seriatim with a plurality of fuel lines 12 (only one shown in FIG. 1) leading to the fuel injection nozzles (not shown) of the individual cylinders in the engine E. The intake ends of the fuel lines 12 are distributed uniformly around the circumference of the main piston 4 and each thereof contains a one-way valve here shown as a ball check valve 12a. During each of its working strokes, the main piston 4 places its axial groove 11 into communication with a different fuel line 12. The fuel lines 12 form with the axial groove 11 the aforementioned discharge conduit which serves to convey fuel expelled from the working chamber 2 into consecutive cylinders of the engine E.

The working chamber 2 communicates with a relief channel which evacuates from this working chamber any surplus fuel that should not be admitted to the cylinders of the engine E. The relief channel comprises two bores 14, 16 which are separated from each other by a regulating chamber 15. The outlet of the relief bore 14 communicates with the regulating chamber 15 which latter accommodates a rotary slide valve 13 serving to regulate the quantity of fuel that is actually admitted into the cylinders of the engine E. The valve 13 (hereinafter called regulating valve) is formed with a circumferential groove 17 extending between two cylindrical portions or plungers 13a, 13b. The relief bore 16 connects the regulating chamber 15 with the suction space 6 but its inlet 39 can be sealed from the chamber 15 by the right-hand plunger 13b. A bypass bore 40 also connects the suction space 6 with the regulating chamber 15 but its intake end can be sealed by the left-hand plunger 13a. The circumferential groove 17 is in permanent communication with the relief bore 14 and its right-hand axial end is bounded by a specially configured regulating face 18 of the plunger 13b. The configuration of the regulating face 18 will be described in connection with FIG. 2. The plungers 13a, 13b are fluid-tightly but slidably fitted into the regulating chamber 15 and the regulating valve 13 is permanently biased by a strong return spring 19 which tends to maintain a collar 13c of the valve 13 in abutment with the adjoining internal surface 1f of the pump housing 1. When the collar 13c abuts against the surface 1f, the inlet 39 of the relief bore 16 is sealed by the plunger 13b.

The fuel displacing element 5 and the adjoining portion of the housing 1 together form the aforementioned auxiliary pump whose purpose is to shift the regulating valve 13 against the bias of the return spring 19. Such displacement of the regulating valve 13 will take place in response to expulsion of fuel from the auxiliary chamber 3. The connection between the chambers 3 and 15 comprises a duct 21 which is a bore machined into the housing 1 and contains a check valve 22. The main piston 4 will cease to deliver fuel into one of the fuel lines 12 when the fuel expelled from the auxiliary chamber 3 via duct 21 and entering the left-hand end of the regulating chamber 15 shifts the regulating valve 13 to such an extent that the relief bore 14 is free to communicate with the bore 16, i.e., when the regulating face of the plunger 13b exposes at least a portion of the inlet 39. The fuel escaping from the working chamber 2 then follows the path of least resistance and, instead of overcoming the resistance of a check valve 12a, flows through the relief bore 14, groove 17, inlet 39, the remainder of the bore 16, and back into the suction space 6.

The auxiliary chamber 3 can receive fuel through a suction port 20 which connects to the suction space 6. The fuel displacing element 5 allows the suction port 20 to communicate with the chamber 3 when the multi-stage piston 1b reaches the end of its suction stroke.

During each interval between two successive working strokes of the fuel displacing element 5, the spring 19 tends to return the regulating valve 13 to the starting position of FIG. 1 in which the collar 13c abuts against the internal surface 1f. The plunger 13a then expels fuel from regulating chamber 15 and such fuel flows through a duct 23 which bypasses the one-way valve 22 in the duct 21. Return flow of fuel through the duct 23 can be regulated by a throttle 24 having a needle like throttling member 25 which is adjustable by a linkage 49a in a manner to be fully described hereinafter. The throttling member 25 can impede the return flow of fuel from the regulating chamber 15 to such an extent that the fuel displacing element 5 performs the next working stroke before the valve 13 returns all the way to the starting position of FIG. 1. In other words, in a certain range of rotational speeds of the engine E, the regulating valve 13 cannot return to starting position after each successive working stroke of the main piston 5. This means that the fuel lines 12 will receive lesser quantities of fuel because the valve 13 will establish a connection between the relief bores 14 and 16 in response to admission of a relatively small quantity of fuel into the chamber 15. This so-called liquid stop or liquid abutment is fully disclosed in the aforementioned Patent No. 3,044,404 to Bessiere. In other words, and when the rotational speed of the engine E reaches a predetermined value, the spring 19 effects only partial return movement of the regulating valve 13 to starting position to thereby reduce the length of effective working strokes of the main piston 4.

In accordance with an important feature of the present invention, the regulating valve 13 is movable axially of the chamber 15 in response to bias of the return spring 19 or in response to admission of fuel through the duct 21, and this valve 13 is also rotatable to thereby further select the exact moment when the relief bore 14 begins to communicate with the bore 16. This specific configuration of the regulating face 18 on the plunger 13b insures that the exact moment when the bores 14, 16 of the relief channel 14-16 begin to communicate with each other during a working stroke of the main piston 4 can be changed in response to angular displacement of the valve 13. The purpose of the regulating face 18 on the plunger 13b is to prevent escape of fuel via bore 16 when the engine E is started and its cylinders should receive greater quantities of fuel, as well as to determine the exact quantities of fuel which will be admitted to the engine cylinders while the engine operates normally.

The regulating valve 13 can be caused to change its angular position in response to changes in rotational speed of the engine. For example, the angular position of the valve 13 could be changed by resorting to a conventional centrifugal governor. However, in the embodiment of FIG. 1, the angular position of the valve 13 is changed in response to changes in pressure produced by the fuel pump 7. This is tantamount to changes in rotational speed of the engine E because the latter includes the drive D which operates the fuel pump 7. In order to prevent too rapid rise in pressure produced by the fuel pump 7, the structure of FIG. 1 preferably comprises a pressure regulating unit including a pipe 26 which connects to the tank 1e and bypasses the fuel pump 7. As clearly shown in FIG. 1, the intake and discharge ends of the pipe 26 are respectively connected with the fuel conduit 27 upstream and downstream of the fuel pump 7 and the flow of fuel through the pipe 26 may be throttled at 28 by means of a throttling device including a plunger 29 which is permanently biased by a helical expansion spring 30. The upper ends face of the plunger 29 is subjected to the pressure of fuel so that such pressure opposes the bias of the spring 30. As shown in FIG. 4, the just described pressure regulating unit 26, 28-30 insures that the pressure p in the first outlet 27a of

the fuel conduit 27 is a function of the rotational speed n of the engine E.

A second outlet 27b of the fuel conduit 27 is connected to one chamber 31a of a control cylinder 31 which accommodates a control piston 32, the latter serving to change the angular position of the regulating valve 13. The operative connection between the control piston 32 and the regulating valve 13 comprises a mechanism which can convert axial movement of the piston 32 into angular movement of the valve 13. In the embodiment of FIG. 1, the just outlined mechanism comprises a pin 33 which is rigid with a cupped spring retainer 34 rotatable in the suction space 7 and arranged coaxially with the valve 13. As best shown in FIG. 5, the pin 33 passes through a radial slot 35 of the control piston 32 and its outer end extends into a cylindrical head 33a which is turnable and movable radially in the slot 35 of the control piston. This pin 33 can rock the spring retainer 34 in response to axial displacement of the control piston 32. The coupling between the spring retainer 34 and the valve 13 comprises an axially parallel pin 36 which allows for axial movement of the valve with reference to the spring retainer. As clearly shown in FIG. 1, the spring retainer 34 accommodates the major part of the return spring 19.

In order to facilitate understanding of the operation of the motion transmitting mechanism 33-36, the control cylinder 31 and its piston 32 are shown in FIG. 1 first by solid lines and also by phantom lines. The other chamber 31b of the control cylinder 31 is connected with the pipe 26. The cylinder 31 is shown as forming an integral part of the pump housing 1.

Referring to FIG. 5, the cylinder chamber 31a accommodates a helical spring 37 which biases the control piston 32 in a direction to the left so that it opposes the pressure of fuel admitted through the outlet 27b of the fuel conduit 27. The spring 37 yields when the fuel pressure in the chamber 31a rises from zero (while the engine E is at a standstill) to a first value corresponding to a rotational speed during idling of the engine. In other words, the spring 37 will yield when the regulating valve 13 can begin to establish a connection between the bores 14 and 16 during each working stroke of the main piston 4. The other chamber 31b of the control cylinder 31 accommodates at least one additional spring 38 and/or 45 which yields when the rotational speed of the engine E and hence the fuel pressure exceeds a second value which is variable and which is always higher than the aforementioned first value.

Referring now to FIG. 2, the regulating face 18 of the valve 13 is illustrated in developed view. This regulating face comprises a section 18a which is located in a plane making right angles with the axis of the valve 13. When the collar 13c abuts against the internal surface 1f, the distance a between the section 18a and the inlet 39 of the relief bore 16 at least equals the maximum axial stroke of the valve 13. FIG. 2 shows that the section 18a is located at a considerable axial distance from the inlet 39, i.e., the axial position of the regulating valve 13 corresponds to a position which the valve assumes when the collar 13a abuts against the surface 1f. The angular position of the valve 13 corresponds to that which the spring retainer 34 assumes in response to a minimum fuel pressure in the cylinder chamber 31a. The abscissa of the diagram shown in FIG. 2 indicates various angular positions α of the regulating valve 13 and the ordinate indicates the axial distances 1 covered by the valve 13 in the chamber 15. For the sake of simplicity, the regulating face 18 of FIG. 2 is assumed to be stationary and it is therefore assumed that the inlet 39 of the relief bore 16 changes its angular positions with reference to the valve 13. In other words, while observing FIG. 2, one should bear in mind that it is the regulating face 18 which turns about the axis of the regulating valve 13, not the inlet 39.

When the relative position of the inlet 39 with reference to the section 18a is such as that indicated at 39a, 75

the engine E is being started whereby the section 18a cannot permit flow of fuel from the circumferential groove 17 into the relief bore 16. Therefore, all of the fuel expelled from the working chamber 2 by the main piston 4 will enter one of the fuel lines 12 and will be admitted to the corresponding injection nozzles.

The regulating valve 13 may be turned by means of the control piston 32 and also by hand. Such manual adjustment will be made to arrest the engine E. As illustrated in FIG. 1, the spring retainer 34 is received in a cupped socket 41 which is coaxial with the regulating valve 13 and is rotatable in the pump housing 1. The socket 41 is rigid with an actuating shaft 42 which may be rotated by the driver of the vehicle, either directly or by remote control. The coupling between the spring retainer 34 and the socket 41 comprises an axially parallel pin 43 which engages a radial projection 44 of the spring retainer. In order to enable the driver to arrest the engine E, the regulating face 18 of the valve 13 comprises a section 18b which is parallel with and is located opposite the section 18a. The section 18b exposes at least a portion of the inlet 39 (see the latter's position 39b) when the driver turns the actuating shaft 42 in order to change the angular position of the valve 13. In this way, the valve 13 allows all of the fuel which is expelled from the working chamber 2 to flow through the relief bore 14, through the groove 17, inlet 39, the remainder of the relief bore 16, and back into the suction space 6. In other words, the engine E does not receive any fuel.

The aforementioned liquid abutment or liquid stop is effective at least at such times when the engine operates at a maximum rotational speed. The exact rotational speed at which the liquid abutment becomes effective can be determined by the throttle 24. In the embodiment of FIGS. 1 and 5 to 11 the liquid abutment will become effective only when the engine E operates at a maximum rotational speed. FIG. 13 illustrates a modified fuel injection pump wherein the liquid abutment will become effective in a plurality of speed ranges, i.e., not only when the engine is operated at a maximum speed.

In the embodiment of FIGS. 7 and 5 to 11, the throttle 24 is adjusted only once in such a way that it prevents full expansion of the return spring 19 only at such times when the rotational speed of the engine E reaches a maximum value. The control piston 32 in the control cylinder 31 is acted upon by three springs 37, 38 and 45. The bias of the spring 45 may be selected by the driver, and the bias of this spring is weaker than that of the spring 38, i.e., the spring 45 will yield before the spring 38. As shown in FIG. 5, the control cylinder 31 accommodates a pusher or plunger 46 which is slidable in the chamber 31b and has a centrally located rod 47 extending through a spring retainer 48. The latter is coaxial with the pusher 46 and its axial position may be adjusted to thereby change the bias of the spring 45. Such adjustment may be carried out in response to angular displacement of a pinion 50 which meshes with a rack 51 provided on the spring retainer 48. The pinion 50 is rotatable about a fixed axis by means of a lever 49.

The spring 45 is inserted between the bottom wall of the spring retainer 48 and a disk 52 which latter is traversed by the rod 47. The spring 38 is installed between the disk 52 and the bottom wall of the pusher 46. The rod 47 carries a stop 53 (for example, a split ring fitted into a circumferential groove of the rod 47) which prevents excessive displacement of the disk 52 under the bias of the spring 38. The rightward stroke of the spring retainer 48 is limited by a second stop 54.

As shown in FIG. 6, the leftmost convolution of the spring 45 can move at a maximum distance b from the disk 52. It is assumed that the right-hand end convolution of the spring 45 is attached to the bottom wall of the spring retainer 48. If the spring 37 is contracted, the pusher 46 is spaced from the control piston 32 by a distance c . When the disk 52 abuts against the stop 53, this

disk is separated from the pusher 46 by a distance d . The spring retainer 48 has an annular central portion or hub 48a having a recess dimensioned to accommodate the stop 53 so that the disk 52 can move into direct abutment with the hub 48a. The stroke of the pusher 46 in a direction toward the control piston 32 is limited by nuts 55, shown in FIG. 5, which mesh with the right-hand end of the rod 47, see FIG. 5.

The fuel-injection pump shown in FIGS. 1 and 5 to 11 operates as follows:

As stated before, the liquid abutment of this pump becomes effective only when the engine E operates at a maximum rotational speed, i.e., when such rotational speed exceeds a predetermined value which is selected for each engine embodying the fuel injection pump. In other words, the position of the throttling member 25 is adjusted in advance, but such position can be changed by the linkage 49a so that the same pump may be used to inject fuel into different types of internal combustion engines.

The length and inclination of the slanting section 18c of the regulating face 18 on the valve 13 are selected in such a way that, depending on the angular and axial position of such inclined section, the valve 13 will admit a given quantity of fuel to the relief bore 16 whereby such quantity varies between zero and maximum quantity, depending upon whether the engine E is idling or operates at full load. An equalization will take place only when the inlet 39 of the bore 16 is controlled by that section of the regulating face 18 which is selected to regulate the flow of fuel at maximum load.

As stated before, the rotational speed (with the exception of maximum speed) of the engine E is selected by rotating the regulating valve 13 about its own axis. Such rotation can be brought about by the control piston 32 or by the actuating shaft 42, i.e., in response to changes in pressure produced by the fuel pump 7 or by manual adjustment of the socket 41 through the intermediary of the actuating shaft 42. The pressure of fuel admitted into the chamber 31a of the control cylinder 31 is opposed by one, two or all three springs 37, 38, 45. The range of rotational speeds below the aforementioned maximum rotational speed includes a certain range in which no regulation of the rotational speed takes place excepting by means of the shaft 42. This will be described hereinafter.

When the engine E is at a standstill, the control unit including the cylinder 31, piston 32, pusher 46 and springs 37, 38, 45 assumes the position shown in FIG. 6. The spring 37 is contracted and maintains the control piston 32 in the left-hand end position whereby the relative position between the inlet 39 of the relief bore 16 and the regulating face 18 of the valve 13 corresponds to that shown for the inlet 39a and section 18a of FIG. 2. The spring retainer 48 then abuts against the stop 54.

When the driver thereupon starts the engine E, the rotational speed n (see FIG. 4) does not immediately increase to such a value that the pressure p of fuel discharged from the pump 7 and acting in the cylinder chamber 31a could expand the spring 37. Therefore, the inlet 39 remains sealed by the regulating valve 13 and the liquid abutment is not effective. In other words, each of the fuel lines 12 receives a maximum quantity of fuel expelled from the working chamber 2 by the main piston 4. In FIG. 3, such maximum quantity of fuel is indicated by the line Qa. In the diagram of FIG. 3, the quantity of injected fuel is plotted on the ordinate as a function of the angular position (angle α) of the regulating valve 13. The diagram of FIG. 12 is analogous; in this illustration, the quantity of injected fuel is shown as a function of the rotational speed n of the engine.

When the engine E is running, the pressure p produced by the fuel pump 7 rises and assumes a value which suffices to overcome the bias of the spring 37. This spring 37 determines the quantity of injected fuel during starting. Once the bias of the spring 37 is overcome, the components of the control unit assume the positions shown in

FIG. 7. This is the position of the control unit during idling of the engine E. The clearances c and b are eliminated because the fuel flowing through the inlet 27b acts against the left-hand face of the control piston 32 and displaces the latter in a direction to the right so that the piston 32 abuts against the pusher 46 and the disk 52 abuts against the spring 45. The springs 38 and 45 oppose further rightward movement of the pusher 46 whereby the position of the inlet 39 with reference to the regulating face 18 corresponds to that shown in FIG. 2 at 39c. The spring retainer 48 continues to abut against the stop 54. Once the inlet 39 assumes the position 39c of FIG. 2, only small axial displacement of the regulating valve 13 suffices to allow for escape of fuel through the relief bore 16 in response to a working stroke of the main piston 4 and fuel displacing element 5. Therefore, the cylinders of the engine E receive very little fuel. This is indicated in FIG. 3 or 12 by the horizontal line Qb.

If the operator turns the lever 49 to shift the spring retainer 48 in a direction to the left, as viewed in FIG. 7, he compresses the spring 45 because the spring 38 is stiffer and, during such adjustment of the spring retainer 48, the spring 38 acts not unlike a rigid body. It is assumed that the lever 49 has been rocked to select a satisfactory quantity of fuel for partial load at low rotational speed and that the spring retainer 48 then takes the position shown in FIG. 8. The spring 45 is biased and displaces the control piston 32 against the action of fluid pressure in the chamber 31a. This takes place when the rotational speed of the engine is relatively low. Of course, the control piston 32 changes the angular position of the regulating valve 13 so that the inlet 39 of the relief bore 16 assumes the relative position 39d of FIG. 2. The quantity of fuel which is injected into the cylinders of the engine E then corresponds only to a fraction of the maximum fuel quantity. Such position of the control piston 32 corresponds to a given rotational speed which again causes the pump 7 to produce a predetermined fuel pressure acting against the left-hand end face of the control piston 32. If the rotational speed rises above that which is desired by the operator of the vehicle, the control piston 32 moves in a direction to the right, as viewed in FIG. 8, because the fuel pressure in the chamber 31a rises. The disk 52 moves nearer to but does not as yet abut against the hub 48a of the spring retainer 48. In other words, the spring 45 yields but is not fully compressed so that the disk 52 assumes an intermediate position between the positions shown in FIGS. 8 and 9. Such axial displacement of the control piston 32 toward the spring retainer 48 causes the regulating valve 13 to change its angular position and to reduce the quantities of fuel which are injected into the engine cylinders during successive working strokes of the main piston 4. Under certain circumstances, the quantity of injected fuel can be reduced to match that during idling. The downwardly inclined lines m shown in FIG. 12 indicate such type of regulation which is not due to the liquid abutment but rather to the regulating action of the face 18 on the valve 13.

FIG. 12 shows a rotational speed n_1 starting from which the pressure of fuel against the left-hand end face of the control piston 32 rises sufficiently to effect such compression of the spring 45 that the disk 52 comes into actual contact with the hub 48a of the spring retainer 48. The spring 45 is then fully compressed. If the operator thereupon rocks the lever 49 to such an extent that the spring retainer 48 assumes its left-hand end position shown in FIG. 9, the control piston 32 assumes a position in which the relative angular position of the inlet 39 with reference to the regulating face 18 of the valve 13 is such as shown in FIG. 2 by the broken-line circle 39e. This is a position corresponding to maximum load. The quantity of injected fuel in such relative position of the inlet 39 and face 18 is indicated in FIGS. 3 and 12 by horizontal lines Qe.

It is also possible that, when the spring 45 is fully

compressed, the spring retainer 48 assumes the position shown in FIG. 8 which corresponds to partial load. In such situation, the quantity of injected fuel is as indicated by the points Qd in FIG. 12. Each point Qd corresponds to a different position of the spring retainer 48, i.e., to a different position of the lever 49. Such positions of the lever 49 are selected by the operator of the vehicle. In each instance, the spring 45 is fully compressed when the rotational speed of the engine E reaches the value n_1 .

If the rotational speed exceeds the value n_1 , the control piston 32 can be displaced only by the small distance d , namely, until the disk 52 comes into actual abutment with the pusher 46 and effects full compression of the spring 38. The piston 32 then takes the position shown in FIG. 10, it being assumed that the lever 49 maintains the spring retainer 48 in the left-hand end position. This short distance d is represented in FIG. 12 by the sloping lines q .

Aside from such minor adjustment in response to reduction of the distance d to zero, there is no other adjustment in the quantity of admitted fuel when the rotational speed rises above the value n_1 but is less than such rotational speed at which the liquid abutment becomes effective to prevent undesirable increase in the rotational speed. The effect of the liquid abutment is illustrated in FIG. 12 by the slanting line r .

In order to terminate the injection of fuel, the operator returns the spring retainer 48 to the right-hand end position, namely, into abutment with the stop 54 as shown in FIG. 11. This is the starting position of the member 48, see also FIG. 6. At the same time, the operator turns the shaft 42 in order to shift the control piston 32 and pusher 46 to the right-hand end position whereby the spring 37 expands. The position of the inlet 39 with reference to the regulating face 18 of the valve 13 is then such as indicated in FIG. 2 by the broken-line circle 39b. This prevents admission of any fuel into the cylinders of the engine E because the fuel which is expelled from the working chamber 2 can invariably flow through the relief bore 14 and back to the suction space 6.

The control piston 32 can assume the position of FIG. 11 when the rotational speed of the engine E increases while the spring retainer 48 remains in the starting position of abutment against the stop 54. This can happen when the vehicle travels downhill. The pressure p at the downstream side of the fuel pump 7 then rises as shown in FIG. 4 and the piston 32 yields to such pressure to move to the position shown in FIG. 11. In other words, the injection of fuel is terminated in a fully automatic way.

In the embodiment of FIG. 13, the position of the throttling member 25 (see FIG. 1) is adjustable by the linkage 49a. Thus, the operator can select that rotational speed n of the engine E at which the liquid abutment becomes effective. In this embodiment, too, the spring 38 determines the maximum quantity of injected fuel in dependency on the rotational speed of the engine.

The control piston 32a of FIG. 13 has an axially extending stud 57 which carries a slidable disk 62. The outward movement of the disk 62 is limited by a stop 63 at the free end of the stud 57. The spring 38 is installed between the disk 62 and an internal surface of the piston 32a. An adjustable stop 58 can arrest the disk 62 but allows for further rightward movement of the piston 32a.

The operation of the fuel injection pump which embodies the control unit of FIG. 13 is as follows:

Each position of the throttling member 25 corresponds to a different range of rotational speeds n . Each such range is controlled by the liquid abutment. In other words, the drive can select the rate of fuel flow through the return duct 23 by changing the position of the linkage 49a shown in FIG. 1, and each position of the linkage 49a corresponds to a different range of rotational speeds. The total quantity of fuel delivered by the main piston 4 will be injected into the cylinders of the engine E until the

liquid stop becomes effective. The face 18 of the regulating valve 13 corrects only the maximum amount of fuel in dependency on the rotational speed n . The effect of the regulating face 18 is relatively small and corresponds to the inclination of the curve which indicates in FIG. 14 or 16 the quantity Q of injected fuel. This takes place between the rotational speed n_2 (when the starting operation is completed and the cylinders of the engine need not receive a maximum amount of fuel) when the compression of the spring 38 shown in FIG. 13 begins, and the rotational speed n_3 at which the piston 32a abuts against the disk 62 and the latter engages the stop 58. In the diagrams of FIGS. 14 and 16, the quantity Q of the injected fuel is plotted along the ordinate and the rotational speed n of the engine is plotted along the abscissa, the same as in FIG. 12. It will be seen that the quantity Q may be varied with the type of engine in such a way (FIG. 16) that the quantity Q increases with increasing speed n or that the quantity Q decreases (FIG. 14) when the rotational speed n increases. The diagram of FIG. 15 is similar to that of FIG. 16 with the distinction that the angle α is measured along the abscissa. In FIGS. 14 and 16, the action of the liquid abutment is indicated by the inclined lines s .

Referring again to FIG. 13, the disk 62 abuts against the stop 58 but the control piston 32a is illustrated in an axial position which corresponds to that when the injection of larger quantity of fuel for starting is already terminated. Before the admission of such larger quantities of fuel for starting is terminated, and particularly when the engine is idle, the disk 62 is separated from the stop 58 by a gap corresponding to the gap c shown in FIG. 6.

It will be seen that an important feature of my invention resides in the provision of a fuel injection pump whose regulating valve 13 may change its angular position in response to changes in rotational speed n of the engine E, namely, in response to changes in axial position of the control piston 32 or 32a as a function of fuel pressure generated by the auxiliary pump 7, such fuel pressure being a function of the rotational speed of the engine. While the rotational speed n of the engine rises from zero speed to idling speed (first range of rotational speeds), the valve 13 completely seals the relief conduit 14-16 which includes the bores 14, 16 so that all of the fuel expelled from the working chamber 2 while the main piston 4 performs a working stroke enters one of the fuel lines 12, i.e., the discharge conduit including the axial groove 11 of the main piston 4 and one of the fuel lines 12. In other words, when the rotational speed n reaches the idling speed, the first spring 37 of the resilient means 37, 38, 45 or 37, 38 yields to fuel pressure to allow for axial displacement of the control piston 32 or 32a. The additional spring 45 and/or 38 yields when the rotational speed n rises to a higher second value, and the force (bias) of at least one of these additional springs 38, 45 is adjustable by the lever 49 or by turning of the adjustable stop 58 to effect axial movement of this stop.

In the embodiment of FIGS. 1 and 5 to 11, the bias of the spring 38 is stronger than that of the spring 45, and the bias of the spring 45 is adjustable by the lever 49 to effect change in the rate of fuel flow to the cylinders of the engine when the latter is operated under partial load. This insures that the quantity of fuel which is injected into the cylinders of the engine when the latter operates under partial load need not be regulated by means of the throttle 24 (linkage 49a). In other words, in the partial load range, the rate of fuel flow to the cylinders of the engine if effected in response to changes in angular position of the regulating valve 13 rather than in response to changes in the position of the throttling member 25.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others can, by applying current knowledge readily adapt it for various applications without omitting features which fairly constitute essential characteristics of the generic and spe-

cific aspects of my contribution to the art and, therefore, such adaptations should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be protected by Letters Patent is:

1. In a fuel injection pump for supplying metered quantities of fuel to the cylinders of an internal combustion engine, a source of fuel; housing means having a working chamber and a relief channel connected with said working chamber and including a regulating chamber; a supply conduit adapted to connect said source with said working chamber; at least one discharge conduit adapted to connect said working chamber with a cylinder of the engine; a main piston movable in said housing means to perform alternating suction and working strokes to thereby respectively draw fuel from said source into said working chamber via said supply conduit and expel fuel from said working chamber; drive means for rotating and reciprocating said piston at a rate proportional with the rotational speed of the engine; a regulating valve installed in said regulating chamber for axial and angular movement to thereby control the flow of fuel from said working chamber via said channel; an auxiliary pump operating in synchronism with said piston to feed fuel into said regulating chamber during working strokes of said piston and to thereby effect axial movement of said valve; and control means comprising a fuel pump driven by said engine and arranged to raise the fuel pressure in response to increasing rotational speed of the engine, and a device for changing the annular position of the valve in response to change in said fuel pressure, said device comprising a control cylinder, a control piston, reciprocably received in said control cylinder, an operative connection between said control piston and said valve for rotating the valve in response to reciprocation of said control piston, an outlet connecting the pressure side of said fuel pump with said control cylinder at one side of said control piston so that the pressurized fuel tends to move said control piston in one direction, and resilient

means for biasing said control piston in the other direction and including a first spring which opposes the fuel pressure against said control piston within a first range of relatively low rotational speeds of the engine, and at least one additional spring which opposes the fuel pressure against said control piston within a second range of relatively high rotational speeds of the engine.

2. A structure as set forth in claim 1, wherein said first range covers rotational speeds during idling of the engine.

3. A structure as set forth in claim 1, further comprising means for changing the opposition of said additional spring to thereby change said second range of rotational speeds.

4. A structure as set forth in claim 1, wherein said resilient means includes two additional springs each arranged to oppose the fuel pressure against said control piston with a different force higher than the force of said first spring.

5. A structure as set forth in claim 4, wherein one of said additional springs is weaker than the other additional spring and further comprising means for changing the force of said one additional spring, said valve being arranged to change the rate at which the fuel escapes from said working chamber via said relief channel in response to rotation by said control piston while said control piston overcome the opposition of said one additional spring.

6. A structure as set forth in claim 1, wherein said regulating valve comprises a peripheral controlling face controlling flow of fuel from said working chamber via said channel, said controlling face extending in part inclined to the axis of said valve.

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