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[11] 3,538,897

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[33] **Germany**
[31] **1,601,397**

[54] **FUEL INJECTION PUMP FOR INTERNAL COMBUSTION ENGINES**
5 Claims, 3 Drawing Figs.

[52] **U.S. Cl.** **123/140,** 103/2, 103/41
[51] **Int. Cl.** **F04b 13/02,** F04b 49/00, F02d 1/04
[50] **Field of Search** ..... 123/140, 139, 139.11(A); 103/2, 2.1, 41, 41(S)

[56] **References Cited**

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ABSTRACT: In a fuel injection pump for internal combustion engines there is provided a throttle plunger which is operable by the driver in unison with the accelerator of the engine and which simultaneously performs two functions:

1. it varies a first flow passage section for controlling the flow of liquid between an auxiliary pump and a control plunger which is displaceable by a liquid pressure to interrupt fuel delivery during pressure strokes of the pump by detouring varying parts of the fuel into discharge means, and

2. it varies in the same sense a second flow passage section of a bypass channel for detouring, to a greater or lesser extent, liquid into discharge means thus bypassing said first flow passage section and said control plunger.

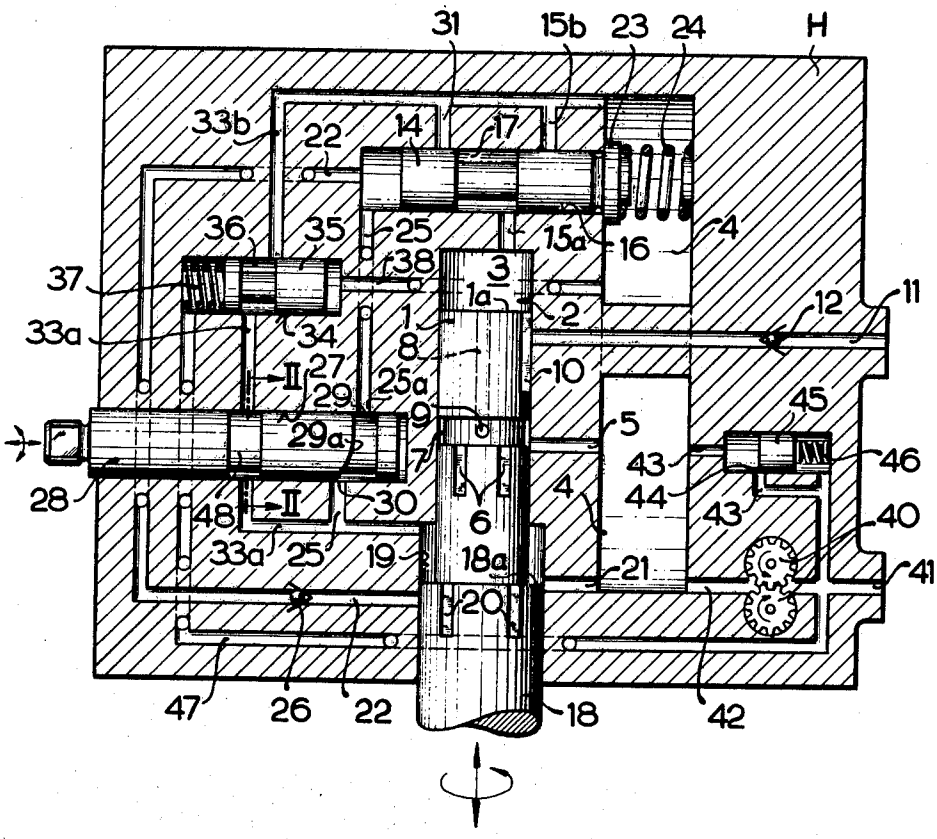


FIG.1

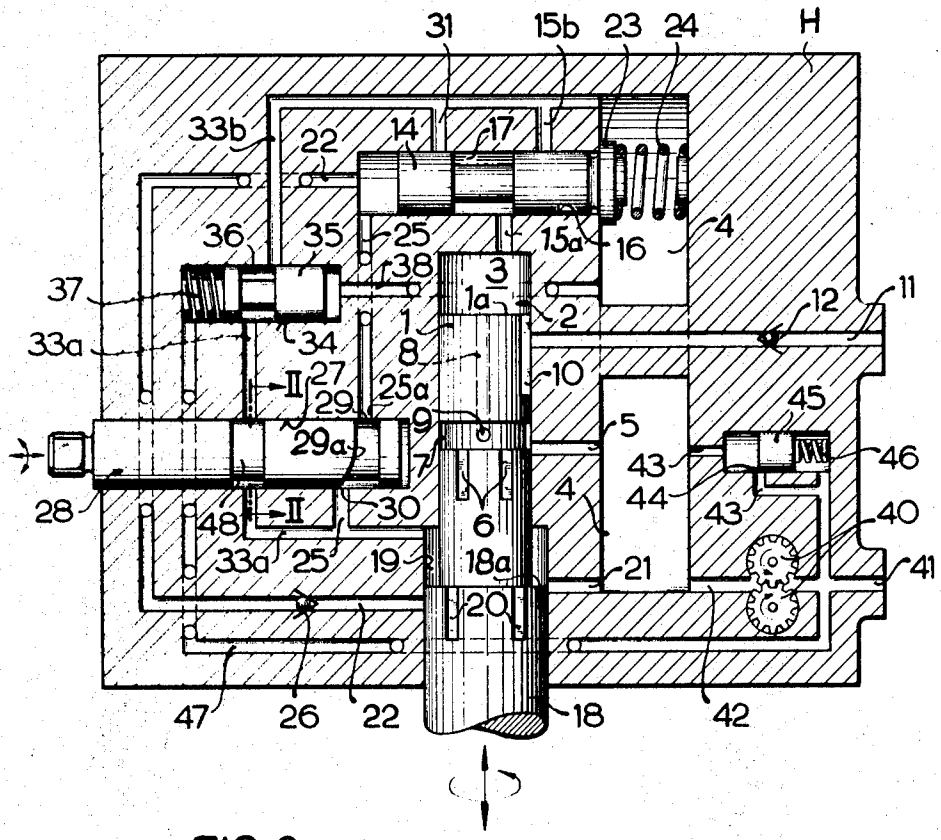


FIG.2

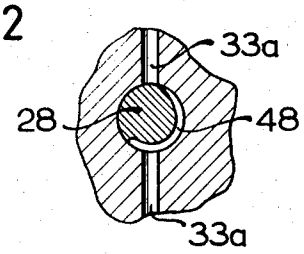
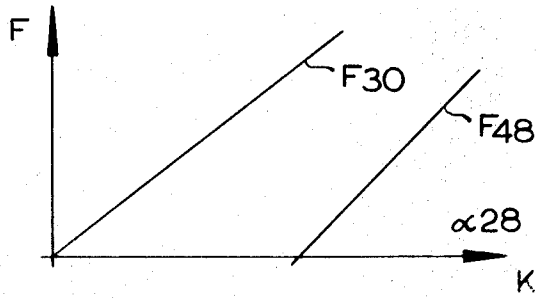


FIG.3



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FUEL INJECTION PUMP FOR INTERNAL COMBUSTION ENGINES

BACKGROUND OF THE INVENTION

This invention relates to a fuel injection pump of the type that is adapted to vary the fuel delivery dependent upon the r.p.m. of the internal combustion engine with which it is associated. This control of quantity is achieved by interrupting the fuel delivery, at least at the moment when a maximum r.p.m. is obtained, by means of opening a discharge channel of the pump work chamber during the delivery stroke of the pump piston. The opening of the discharge channel is effectuated by a reciprocating control member which is caused to perform a forward travel from its position of rest by the liquid pressure generated by the pressure stroke of an auxiliary pump operating synchronously with the main fuel injection pump. During the suction stroke, the control member is displaced towards its position of rest by a resetting means and tends to displace the liquid which caused its forward travel during the pressure or delivery stroke of the pump. The return motion of the control member is dampened to a greater or lesser extent by said liquid, since at least one part of the latter, urged by the returning control member, is forced through a flow passage section varied by a control throttle. For a given flow passage section and at a determined r.p.m., the control member, due to the appearance of the so-called fluid abutment, does no longer return to its original position of rest (determined by a solid abutment), and thus causes interruption of the fuel delivery to the engine at an earlier moment during each pressure stroke.

The control member, in order to ensure the delivery of a supplemental fuel quantity when the engine begins to run, is prevented from opening the discharge channel during the starting stage. For this purpose, at least one part of the pressurized liquid delivered by the auxiliary pump may leave through a bypass channel which contains a throttle (bypass throttle) and which is controlled by a closing member urged into its opening position by a spring and into its closing position by an r.p.m.-dependent liquid pressure.

A fuel injection pump of the afore-described structure is disclosed, for example, in U.S. Pat. No. 3,339,534. There, the pressurized liquid, delivered by the auxiliary pump, flows unutilized in the bypass channel until, by virtue of increasing r.p.m., the pressure of the liquid which operates the closing member, becomes sufficiently large to displace the latter and thus closes the bypass channel so that the control member may start its operation. Towards the end of each forward travel, the control member opens the discharge channel of the pump work chamber so that no more supplemental fuel quantity is delivered. In view of the fact that the closing of the bypass channel is dependent merely on the r.p.m. and not on the moment when a smooth operation of the engine is obtained and that this r.p.m. (which may be reached with a still cold engine) is below the idling r.p.m., the fuel quantities now delivered may be too small. As a consequence, the engine r.p.m. decreases to a value for which then the bypass channel is again opened and supplemental fuel quantities are again delivered. Due to the hysteresis of the closing member, the r.p.m. at which the closing member blocks the bypass channel, is somewhat higher than the r.p.m. at which the same is shifted, by virtue of spring means, into its initial, open position. This r.p.m. is lower than the idling r.p.m. for the purpose to prevent delivery of supplemental fuel quantities during idling of a hot engine. The engine may stall during these r.p.m. fluctuations which, in any case, cause undesired vibrations of the vehicle.

In some embodiments of known fuel injection pumps disadvantages are experienced even in case of a warm, smoothly idling engine. Thus, when the control throttle is closed for shutoff, so that the control member is displaced into its extreme forward position and the entire fuel delivered by the fuel injection pump flows unutilized through the discharge channel, the danger exists that the closing member, as soon as a correspondingly low r.p.m. is reached, opens the bypass channel and as a result the control member falls back into its initial position with the consequence that the engine again is supplied with fuel. This may lead to highly undesirable starting shocks.

Since during the starting even of a hot engine, supplemental fuel quantities are delivered, strong smoke generation occurs. Besides, it is to be taken into consideration that the driver at the cold start of the engine customarily fully depresses the accelerator (control throttle fully open) and, on the other hand, for hot starting of the engine does not actuate the accelerator at all (control throttle in idling position).

OBJECT AND SUMMARY OF THE INVENTION

It is an object of the invention to provide an improved fuel injection pump of the type described hereinbefore in which the named disadvantages are eliminated and where the bypass channel of the auxiliary pump is opened only when the flow passage section of the control throttle corresponds to the medium to maximum r.p.m. of the engine so that the r.p.m. necessary for the actuation of the closing member to terminate the delivery of supplemental (starting) fuel quantity, may be higher then the idling r.p.m.

Briefly stated, according to the invention, the flow passage section of the bypass throttle is variable by means of a mechanism, such as a throttle plunger, which also varies the flow passage section of the control throttle. This dual control is performed in such a manner that the bypass section decreases with decreasing control throttle section and is entirely closed at least when the control throttle section corresponds to the idling r.p.m. range.

The invention will be better understood as well as further objects and advantages will become more apparent from the ensuing detailed specification of a preferred although exemplary embodiment where the bypass throttle is disposed in the bypass channel between the auxiliary pump and the closing member.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial sectional view of the entire fuel injection pump;

FIG. 2 is a cross-sectional view of the bypass throttle taken along line II–II of FIG. 1; and

FIG. 3 is a diagram showing the throttle sections F as a function of angular settings α of the throttle plunger.

DESCRIPTION OF THE EMBODIMENT

Turning now to FIG. 1, there is shown a pump housing block H receiving in a cylinder 2 a reciprocating and rotating piston 1 of the fuel injection pump. The terminal face 1*a* of piston 1 and the upper portion of cylinder 2 define a pump work chamber 3. The piston 1, also functioning as a distributor, directs fuel in sequence to the cylinders of a multicylinder internal combustion engine. Piston 1 is caused, by means not shown, to reciprocate along, and simultaneously rotate about, its axis as indicated by the arrows shown immediately below the structure. With each suction stroke of the piston 1 the pump work chamber 3 is supplied with fuel which is drawn from a suction chamber 4 through a control channel 5, one of longitudinal grooves 6, a circumferential annular groove 7, and an axial bore 8 which merges into the pump work chamber 3 and which is in communication with the annular groove 7 by means of a radial bore 9.

The lateral face of piston 1 is provided with a longitudinal distributor groove 10 opening into pump work chamber 3 and cooperating in sequence, as the piston 1 rotates, with one of a plurality of delivery channels 11 (only one shown) extending from the cylinder 2 and continuing, outside the housing H, in pressure conduits (not shown). Each such pressure conduit leads from a delivery channel 11 to a fuel injection valve (not shown). The annular groove 7 as well as the longitudinal grooves 6 are, similarly to the longitudinal distributor groove

10, disposed in the lateral face of piston 1. The longitudinal grooves 6 and the delivery channels 11 are equidistantly disposed about the circumference of the piston 1 and the cylinder 2, respectively, and their number is identical to the number of cylinders in the combustion engine which is served by the fuel injection pump.

During each pressure or delivery stroke of piston 1, one of the delivery channels 11 communicates with the pump work chamber 3 through longitudinal distributor groove 10. During one complete revolution, the piston 1 executes as many pressure or delivery strokes as there are delivery channels 11 and connects these delivery channels in sequence with the pump work chamber 3 to sequentially supply the fuel injection valves with fuel. Each delivery channel 11 is provided with a check valve 12 to prevent the fuel forced to the fuel injection valves from returning into the fuel injection pump.

The quantity control of the fuel delivered to the fuel injection valves is performed in the pump with the aid of a control plunger 14 slidably disposed between a discharge channel portion 15*a* merging into the pump work chamber 3 and a discharge channel portion 15*b* communicating with the suction chamber 4. Depending upon the position of the control plunger 14, communication is established or interrupted between channel portions 15*a* and 15*b*. If the channel portions 15*a*, 15*b* communicate with one another, delivery of fuel to the fuel injection valves is interrupted since, urged by piston 1, the fuel, rather than flowing through delivery channels 11, will be discharged into suction chamber 4 through discharge channel portions 15*a*, 15*b*.

Both channel portions 15*a* and 15*b* merge into a cylinder 16 in which the control plunger 14 is slidably disposed. The latter is provided with a circumferential annular groove 17 which is in continuous communication with the channel portion 15*a* but which is separated from the channel portion 15*b* when the control plunger 14 is in its position of rest as shown in FIG. 1.

A diametrically enlarged portion of piston 1 constitutes a piston 18 which, together with a cylinder 19 in which it operates, forms an auxiliary pump. As it will become apparent hereinafter, the liquid displaced by the piston 18 shifts the control plunger 14 in the direction of its forward travel. The lateral surface of the piston 18 is provided with a plurality of longitudinal grooves 20 which terminate on the frontal face 18*a* of the piston 18 and which are the same in number as the longitudinal grooves 6. During the suction stroke of the auxiliary piston 18, which operates synchronously with main pump piston 1, fuel is drawn from the suction chamber 4 through a control channel 21 and the longitudinal grooves 20 into the auxiliary pump work chamber defined by the frontal face 18*a* of the piston 18 and cylinder 19. During its pressure stroke, the auxiliary piston 18 forces fuel, mostly through a channel 22, into the cylinder 16 adjacent the end of control plunger 14.

Should the control plunger 14 be displaced towards the right (forward travel) from its position of rest by the fuel forced into the cylinder 16 by piston 18, it establishes communication, shortly before the termination of its possible path of travel, between the discharge channel portions 15*a* and 15*b*. The position of rest of the control plunger 14 is determined by its terminal flange 23 which is adapted to abut against a wall of the suction chamber 4. During the suction strokes of auxiliary pump 18, 19, the pressure from the liquid in cylinder 16 is removed, so that spring 24 is now capable of moving the control plunger 14 back towards its position of rest. During its return travel, the control plunger 14 forces at least part of the liquid which caused its displacement from the position of rest, back into cylinder 19 of the auxiliary pump through a channel 25. During the return motion of the control plunger 14, the channel 22 is closed by a check valve 26 so that the entire liquid quantity displaced by the control plunger 14 is constrained to pass through the channel 25.

The channel 25 is intersected by a cylindrical bore 27 in which there is axially slidably and rotatably disposed a throttle plunger 28. In the lateral face of the throttle plunger 28 there is provided a control groove 29 which, at one side, is delimited by an oblique control edge 29*a*. While section 25*a* of channel 25 merging into bore 27 freely opens into the control groove 29, another section of channel 25, also merging into bore 27, is varied by the oblique edge or control throttle 29*a* to a greater or lesser extent, determining thereby a variable flow passage section 30. Depending upon the open area of this section, the return of the control plunger 14 is dampened to a greater or lesser extent. Beyond an engine r.p.m. corresponding to a determined area of flow passage section 30, the delivery stroke of the auxiliary piston 18 begins prior to the return of the control plunger 14 into its position of rest. Thus, a so-called liquid abutment appears that prevents the control plunger 14 from returning into its original position of rest during the suction stroke of pistons 1, 18. Consequently, the control plunger 14 begins its forward travel (*i.e.* its displacement towards the right as viewed in FIG. 1) from another initial position, so that during the pressure stroke of piston 18, communication is established between the discharged channel portions 15*a* and 15*b* at an earlier moment than when the control plunger 14 begins its forward travel from its initial position of rest (no liquid abutment). As a result, a decrease of the injected fuel quantities is obtained which, in turn, causes a drop in the r.p.m. of the engine.

The cylinder 16 is also connected with the suction chamber 4 by means of a safety channel 31 which is closed during the normal operation of the control plunger 14. Should the liquid abutment in cylinder 16 become so large as to cause the control plunger 14 to clear safety channel 31, the pressurized liquid will be vented therethrough into suction chamber 4. Thus, the safety channel 31 determines the extreme forward position the control plunger 14 may assume during its displacement towards the right when, by virtue of the communication between the discharge channel portions 15*a* and 15*b* no more fuel injection takes place for the remainder of the pressure stroke.

A bypass channel portion 33*a* branches off from channel 25 between the flow passage section 30 and the auxiliary pump 18, 19, and intersecting the cylinder 27, merges into a cylinder 34. In cylinder 34 there is slidably disposed a closing member 35, the lateral face of which is provided with a circumferential annular groove 36 with which the bypass channel portion 33*a* is in continuous communication. As long as the closing member 35 is in its initial position, the annular groove 36 connects the bypass channel portion 33*a* with a second bypass channel portion 33*b* which, in turn, communicates with the suction chamber 4. To impart motion to the closing member 35 for the purpose of establishing or interrupting communication between bypass channel portions 33*a* and 33*b*, there is provided, on the one hand, liquid under r.p.m.-dependent pressure from suction chamber 4 and, on the other hand, a return spring 37. The cylinder 34 is in communication with the suction chamber 4 by means of a channel 38, so that the liquid under pressure may be applied to the closing member 35 at its terminal face opposed from the return spring 37.

A delivery pump 40 draws the fuel through a suction channel 41 from a tank not shown and forces the fuel through a delivery channel 42 into the suction chamber 4. From the suction chamber 4 there extends a return channel 43, through which one part of the fuel drawn by the pump 40 may return to the suction channel 41. In the return channel 43 there is disposed a throttle 44, the flow passage section of which is preferably variable by means of a piston 45 exposed on one side to the pressure in suction chamber 4 and on the other side to the opposing force of a compression spring 46. The greater the pressure in the suction chamber 4, the more piston 45 is displaced against spring 46, thus increasing the flow passage section 44. The latter, however, increases only to such an extent as to ensure a continuous, desired increase of pressure in suction chamber 4 as the engine-r.p.m. increases.

If, after starting the engine, a predetermined pressure is obtained in the suction chamber 4 corresponding to a predetermined r.p.m., the closing member 35 is displaced against the

force of spring **37** thus interrupting communication between bypass channel portions **33***a* and **33***b*. That portion of cylinder **34** which houses spring **37** is connected to the suction channel **41** by means of a channel **47**.

The flow passage section of bypass channel portion **33***a* is controlled, at its intersection with cylinder **27**, by the throttle plunger **28**. For this purpose the lateral surface of the throttle plunger **28** is provided with a crescent-shaped groove **48** (FIG. 2) which, when the throttle plunger **28** is rotated, opens, to a greater or lesser extent, a flow passage section of the channel **33***a*, or entirely closes the same. It is thus seen that the flow passage suction of bypass channel portion **33***a* depends solely upon the angular position of the throttle plunger **28** and is unaffected by its axial displacement. The flow passage section **30** in the channel **25**, on the other hand, depends upon both the angular position and the axial displacement of the throttle plunger **28**. By virtue of additionally controlling the section **30** by means of a longitudinal displacement of throttle plunger **28**, the two flow passage sections controlled by the throttle plunger **28**, may be adjusted with respect to one another in such a manner that the opening of the bypass channel **33***a* by the bypass throttle **48** occurs at a moment when the flow passage section **30** varied by control throttle **29***a* attains a desired magnitude. This adjustment may be achieved by simply varying the axial position of the throttle plunger **28**.

Turning now to FIG. **3**, the flow passage section F_{30} of the channel **25** (varied by control throttle **29***a*) and the flow passage section F_{48} of bypass channel portion **33***a* (varied by bypass throttle **48**) are shown as a function of the angular position α_{28} of the throttle plunger **28**. It is seen from this diagram that as the throttle plunger **28** is turned, the crescent-shaped groove or bypass throttle **48** opens the flow passage section of the bypass channel portion **33***a* (represented by curve F_{48}) only at a moment when the flow passage section **30** of the channel **25** (represented by the curve F_{30}), already has attained a magnitude which corresponds to the medium to maximum engine-r.p.m. It was found in practice that the flow passage section **30** is then approximately three times larger than its magnitude for an idling r.p.m.

OPERATION OF THE EMBODIMENT

To start the cold engine, the driver customarily entirely depresses the accelerator. Due to this "full acceleration", the throttle plunger **28** is rotated to such an extent that the control throttle **29***a* opens the flow passage section **30** for a maximum r.p.m. and further, the bypass channels **33***a* and **33***b* are fully opened by the bypass throttle **48**. The annular groove **36** of the closing member **35** is in a position in which bypass channel portions **33***a* and **33***b* are in communication with one another. Due to this fully open condition, the entire fuel delivered by the auxiliary pump **18**, **19** flows through the bypass channels **33***a*, **33***b* into the suction chamber **4**. As soon as a predetermined r.p.m. is reached (which may be greater than the idling r.p.m.), the closing member **35**, responding to the r.p.m.-dependent fuel pressure in suction chamber **4**, is displaced against the force of spring **37**, whereby communication between bypass channel portion **33***a* and **33***b* is interrupted. Thus, beyond said predetermined r.p.m. no fuel may "bypass" from the pump work chamber of the auxiliary pump **18**, **19**. As a result, the entire fuel displaced by the auxiliary pump **18** is delivered through the channel **22** and also partially through the channel **25** into the cylinder **16**. The pressure in the latter is now sufficient for causing control plunger **14** to start its operation. Since the throttle plunger **28** is still set to the highest engine speed, the r.p.m. increases until the driver, by adjusting the accelerator, brings the throttle plunger **28** into a different angular position decreasing thereby the flow passage section **30**. Generally, such an adjustment is performed by the driver when the engine runs smoothly. The throttle plunger **28**, when brought into an angular position causing decrease of the flow passage section **30** by control throttle **29***a*, simultaneously blocks bypass channel portion **33***a* by means of bypass throttle **48**. Consequently, even if the closing member **35** causes, at an r.p.m. higher than the idling r.p.m., reopening of bypass channel portions **33***a*, **33***b*, there can be no fuel flow therein. Since the r.p.m. necessary for actuating the closing member **35** is higher than the idling r.p.m., there is in suction chamber **4** a sufficiently high pressure to securely shift the closing member **35** into its closed position.

The largest possible flow passage section of the bypass channel portion **33***a* is set by bypass throttle **48** to such a limited value that in case the closing member **35** is jammed, there appears, beyond a predetermined high r.p.m., a damping effect, by virtue of which the control plunger **14** begins to operate for decreasing the r.p.m. to a maximum permissible value.

In general, hot starting of the engine is effected without the driver depressing the accelerator. Thus, no supplemental fuel quantities are delivered since the bypass channel portion **33***a* remains closed by the throttle plunger **28**. Consequently, a smoke-free combustion may take place. For stopping the engine, no additional mechanism is necessary. For this purpose the throttle plunger **28** is sufficient which, by virtue of a suitable rotary motion, closes the channel **25** so that the control plunger **14** remains in its position in which it maintains communication between channel portions **15***a* and **15***b*.

I claim:

1. In a fuel injection pump of the known type that includes:

A. a reciprocating main piston executing alternate delivery strokes and suction strokes;

B. a control member displaceable by a liquid pressure during said delivery strokes to open a discharge means at least when reaching a maximum engine-r.p.m. to interrupt delivery of fuel to said engine during a terminal portion of said delivery strokes;

C. an auxiliary piston operating synchronously with said main piston and generating said liquid pressure;

D. resetting means to cause a return movement of said control member towards its position of rest during each of said suction strokes;

E. a first channel extending from said control member for receiving liquid displaced by said control member during said return movement thereof;

F. a control throttle varying a flow passage section in said first channel for braking said control member to a greater or lesser extent during said return movement thereof, said control member, in a predetermined setting of said control throttle and at a predetermined r.p.m., being prevented from returning to its position of rest at the end of the suction strokes due to the appearance of a fluid abutment;

G. a second bypass channel extending from said auxiliary piston and adapted to vent at least part of the liquid delivered by said auxiliary piston to prevent sufficient displacement of said control member and thus prevent opening of said discharge means during starting of said engine for the purpose of delivering a supplemental fuel quantity thereto;

H. a bypass throttle disposed in said bypass channel;

I. a closing member controlling said bypass channel and disposed therein, said closing member having a face adapted to be exposed to an r.p.m.-dependent liquid pressure urging said closing member into a position shutting off said bypass channel; and

J. spring means urging said closing member into a position maintaining said bypass channel open; the improvement comprising a throttle member including both said control throttle and said bypass throttle to simultaneously vary in the same sense said flow passage section of said first channel and said flow passage section of said second bypass channel, said control throttle and said bypass throttle being arranged on said throttle member in such a manner that at least in a position of said control throttle corresponding to the idling r.p.m. range, said bypass throttle entirely closes said bypass channel.

2. The improvement as defined in claim 1, wherein said bypass throttle is disposed in that portion of said bypass channel that extends between said auxiliary piston and said closing member.

3. The improvement as defined in claim 1, wherein said flow passage section of said first channel is variable by both a rotary and a linear motion of said throttle member, whereas said flow passage section of said bypass channel is variable solely by a rotary motion of said throttle member.

4. The improvement as defined in claim 1, wherein the maximum opening of said flow passage section of said bypass channel allowed by said bypass throttle on said throttle member is dimensioned in such a manner that at a predetermined high r.p.m. said control member is caused to start its operation.

5. The improvement as defined in claim 3, including a cylinder intersecting both said first and said second channels, a cylindrical plunger slidably and rotatably disposed in said cylinder and forming said throttle member, a first circumferential groove in said plunger, said first circumferential groove has a crescent-shaped cross section and constitutes said bypass throttle, a second circumferential groove in said plunger, said second circumferential groove has an oblique control edge constituting said control throttle.