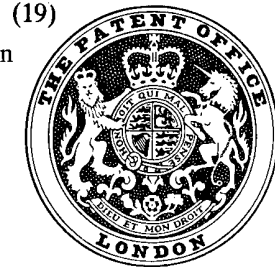


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(54) FUEL INJECTION PUMP FOR INTERNAL COMBUSTION ENGINES

(71) I, GUNTER STEIN, a German citizen, of Weihengaier Strasse 1, 8882 Lauingen, Federal Republic of Germany, do hereby declare the invention, for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:

The invention relates to a fuel injection pump for use in internal combustion engines.

Injection pumps are known which include a fuel distributor driven in rotation at a speed synchronous with the engine speed, and have a valve slide surrounding the fuel distribution member and displaceable axially by a hydraulic control mechanism. Control edges, surfaces and recesses internal to the valve slide cooperate with radial bores in the fuel distribution member of the pump for controlling the overall injected fuel quantity. The pump is supplied with fuel by a fuel supply pump which is driven at a speed proportional to the engine speed. In a known fuel injection system of this type, for example as described in U.S. Patent No. 2,828,727, the possibilities for adaptation to the present day requirements made of engine manufacturers are very limited. In addition, the hydraulic regulator is separate from the actual injection pump.

It is one object of the invention to provide a fuel injection pump which permits a multiple adaptation of its regulating characteristics to the requirements of any particular engine.

According to the present invention there is provided a fuel injection pump for an internal combustion engine, the pump comprising a distributor member which, in use, is driven synchronously with the engine, an axially displaceable annular valve slide disposed around the distributor member and operative to control the fuel injection quantity by means of control edges and recesses

which cooperate with radial bores in the distributor member, a hydraulic regulator producing a regulating pressure in a regulating pressure line for axially displacing the valve slide, a feed pump adapted to be driven at a speed proportional to engine speed and operative to supply the hydraulic regulator with a quantity of fuel which increases with increasing engine speed, an arbitrarily settable throttle disposed in a conduit between the feed pump and the hydraulic regulator for producing a speed dependent pressure drop for the hydraulic regulator, and a spring-loaded pressure control valve for causing a control pressure in a control pressure line downstream of the throttle to increase with increasing fuel flow, the pressures prevailing upstream and downstream of said throttle acting to influence the regulating pressure which in turn influences displacement of the annular valve slide.

A particular advantage of the invention is that the control pressure and the regulating pressure are substantially independent of one another.

A preferred embodiment of the invention includes regulating means for the direct adjustment of the axial position of the annular valve. The preferred embodiment of the invention also has means for changing the timing of the injection within the pumping cycle. Provision may be made for generation of the engine-starting excess fuel quantity. Provision may also be made for cooling the fuel injection pump.

The invention will hereinafter be further described by way of example with reference to the accompanying drawings in which:-

Figure 1 is a longitudinal section through a fuel injection pump according to the present invention;

Figure 2 shows an alternative form of a pressure regulator of a fuel injection pump according to the present invention;

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Figure 3 is a cross section through the injection pump of Figure 1 along the line III-III; and

5 Figure 4 shows the control piston of Figure 1 in its starting position.

10 The illustrated exemplary embodiments of the invention are radial piston type distributor injection pumps having a hydraulic regulator. However, fuel injection pumps in which the fuel distributor and the pump element are separated also would fall within the scope of the invention.

15 The fuel injection pump and its regulator include the following main constituents viz. a hydraulic pressure regulator 1, an annular valve control mechanism 2 for controlling the injected fuel quantity, an injection timing adjustment mechanism 3, an engine starting excess control 4 and a control mechanism 5 for controlling the pump temperature.

20 The hydraulic regulator includes a pressure control valve 7 and a pressure regulating valve 8. The pressure control valve 7 provides a pressure which changes with engine speed, in particular, in the exemplary embodiment shown, the control pressure increases with increasing engine speed. By contrast, the pressure regulating valve 8 produces a regulating pressure which is changed as a function of load and which also changes as a function of engine speed. The regulating pressure serves for actuating a control member affecting the quantity of injected fuel. The fuel injection pump also includes a positive displacement fuel feed pump 9 driven at a speed equal to the engine speed and mounted rotatably within the housing 10 of the fuel injection pump. The feed pump 9 is driven by the drive shaft 11 of the injection pump. The feed pump 9 supplies fuel at low pressure from the interior chamber 12 of the housing 10 and delivers it via a feed pump pressure line 13 to the hydraulic pressure regulator 1. The flow cross section of the feed line 13 is controlled at least at a throttle point 14 by an arbitrarily settable throttle element 15. The amount of fuel delivered by the fuel feed pump 9 increases uniformly with engine speed and its static pressure upstream of the throttle point 14 depends on the position of the throttle element 15. In motor vehicles, the arbitrarily settable throttle element 15 may be connected to the accelerator pedal, for example.

55 The feed line 13 terminates downstream of the throttle point 14 at a control pressure line 16 which, in turn, terminates in a control pressure cylinder 17. The control pressure cylinder 17 includes a control pressure piston 18 which can be slidably displaced in opposition to the force of a control spring 19 and which opens a port 20 to varying degrees. If the fuel quantity

70 supplied by the feed pump 9 via the control pressure line 16 increases, the control pressure piston 18 is moved against the force of the spring 19 in the sense of opening a larger flow cross-sectional area at the port 20. The characteristics of the spring 19 and the shape of the port 20 together provide for a characteristic curve of control pressure versus engine speed. Depending on the requirements, this characteristic curve can be linear, shallow or steep, or even curved. The control pressure is not influenced directly by the throttle location 14 because the total fuel quantity from the feed pump 9 must pass through the throttle location 14. The pressure regulating valve 8 belonging to the hydraulic pressure regulator 1 operates in a different manner. Inasmuch as the regulating pressure directly defines the injected fuel quantity, it must change in a load-dependent manner. For example, if the vehicle is climbing an incline so that the load on the engine increases and the engine speed tends to drop, the driver attempts to increase the engine speed by depressing the accelerator pedal further. Thus in a sense, the arbitrarily settable throttle element 15 is adjusted in a load-dependent manner. Any such adjustment must therefore have an immediate influence on the pressure determined by the pressure regulating valve.

90 The pressure regulating valve has a regulating piston 21 containing an annular groove 22 which controls a port 23 communicating with the control pressure line 16. The regulating piston 21 is slidably disposed within a regulating cylinder 24 and moves in opposition to the force of a regulating spring 25. The regulating piston 21 divides the cylinder 24 into two chambers 26 and 27. The chamber 27 in which the spring 25 is disposed is connected via a line 28 to the interior chamber 12 of the pump experiencing low pressure, whereas the chamber 26 in which the regulating pressure prevails communicates via a bore 29 in the piston 21 with the annular groove 22. Depending on the regulating pressure prevailing in the chamber 26, the regulating piston 21 is displaced in opposition to the spring 25 and changes the flow cross section at the port 23 by means of the annular groove 22. The manner of changing the flow cross section is such that any constriction of the flow cross section results in a diminution of the pressure in the cylinder chamber 26, and an augmentation of the flow cross section at the port 23 results in an increase of that pressure. It is assumed throughout that fluid may flow out of the cylinder chamber 26 to a member which determines the injected fuel quantity. If a substantial amount of fuel flows out, the pressure in the cylinder chamber 26 also changes. Thus, the flow cross section at the port 23 depends on the

force of the spring 25 as well as on the pressure in the cylinder chamber 26 so that a pressure change results in a flow cross section change. In order to change the regulating pressure in a load-dependent manner, one of the variables which determines that pressure must be changed. In the illustrated embodiment, this variable is the pre-compression of the regulating spring 25. For this purpose, a spring stressing piston 30, embodied as a stepped piston, acts on the end of the spring 25 remote from the regulating piston 21. The annular surface of the stepped piston 30 is subjected to the control pressure in line 16 and its large face 32 is subjected to the pressure prevailing in the feed line 13 upstream of the throttle point 14. A spring 33 further acts on the stepped piston 30 in the direction to assist the control pressure. The face 34 of the piston section 35 having the smaller diameter extends into the chamber 27 of the regulating cylinder 24 in which low pressure prevails. Thus, if the throttle element 15 is turned in a direction to diminish the size of the flow cross section at the throttle point 14, and if the engine speed remains constant, the pressure in the feed line 13 upstream of the throttle point 14 will increase. Due to this pressure increase, the spring stressing piston 30 is displaced in opposition to the spring 33, and the regulating spring 25 is loaded more heavily. The increased loading of the regulating spring 25 causes the regulating piston 21 to be pushed further into the chamber 26, i.e., the flow cross section at the port 23 is opened to a greater extent. Thus, the pressure drop across the port 23 is diminished and, if the engine speed remains constant, the regulating pressure in the cylinder chamber 26 will increase. Such a pressure change must then result in a corresponding reduction of the injected fuel quantity.

As soon as the amount of fuel flowing out of the control pressure line 16 through the port 23 changes, the control pressure in the line 16 also changes with respect to the speed of operation of the feed pump 9. In the present exemplary embodiment, the amount of fuel which is used for injection also flows through this port 23 and changes in both load and speed-dependent manner. This means that the control pressure also changes, not only speed-dependently but also, within certain limits, load-dependently. Such an effect may be desirable, as will be explained further below. If it is not desirable, then the amount of fuel injected must be taken from the housing chamber 12 of the pump instead of from the control pressure line 16 or the fuel which is not required for injection must be discharged into the housing chamber 12 of the pump via grooves 61a and 61b (yet to be

described). For that purpose, the grooves 61a and 61b must be closed at their ends adjacent a pressure chamber 64 (also to be described) and must be open at their ends facing the housing chamber 12 of the pump. The location and the width of the groove 61a must be such that, when the grooves 61a and 61b are open, no fuel can flow from the pressure chamber 64 through supply bores 47 into the chamber 12 of the pump. The above-described hydraulic pressure regulator is by no means limited to the described example but may be used as a hydraulic regulator for fuel injection pumps of different construction.

A particular advantage of the pressure regulator 1 is that pressure changes can be achieved rapidly and reliably and that any pressure attained will be constant if the related variables no longer change. In order to limit the stroke of the spring stressing piston 30, the face 32 thereof can open an overflow port 36 leading to the control pressure line 16 after traversing a maximum stroke corresponding to a maximum static pressure. In that position of the spring stressing piston 30, the regulating spring 25 is loaded to its maximum, i.e., the regulating piston 21 lies adjacent the right-hand end face of the regulating cylinder 24. In that position, the flow cross section at the port 23 experiences the least amount of throttling and thus results in the highest regulating pressure for a particular engine speed. In the present exemplary embodiment this means that, when the regulating pressure is high, the tendency is to admit less fuel, whereas when the regulating pressure is low, the tendency is to increase the amount of fuel from whatever its instantaneous value happens to be. The magnitude of the regulating pressure within the cylinder chamber 26 is limited by an overflow port 37 which lends to the line 28 in which low pressure prevails and which is opened by the corresponding end face of the regulating piston 21 as soon as the latter has traversed its maximum stroke in opposition to the force of the regulating spring 25.

The hydraulic pressure regulator illustrated in Figure 1 is a so-called servo-regulator, i.e., any position of the throttle element 15 results in an automatic regulation of a particular engine speed, especially during load changes. Such a servo-regulator requires a greater regulating time than would be required by a direct mechanical linkage. In principle however, for practical reasons and for safety reasons, the idling and the maximum speed must be fully regulated. Any mechanical adjustment during idling might result in stalling the engine and at maximum speed might result in running the engine at excessive speed.

Figure 2 illustrates a hydraulic pressure

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regulator which is identical to that shown in Figure 1 except for the throttling element 15. This so-called idling regulator really operates in the described manner only at idling and at maximum speed whereas, in the intermediate speed regions, a differently embodied throttle element 38 having a cam 39 directly engages the spring stressing piston 30. Since the spring stressing piston 30 directly changes the injected fuel quantity, the fuel quantity change as a function of speed is eliminated by this idling regulator in the intermediate speed region. Only at idling and at maximum speed, i.e., when the cam 39 does not engage the spring stressing piston 30 does there take place a purely hydraulic pressure regulation.

The housing 10 of the injection pump includes a bushing 41 which is closed by a plug 42. A distribution member 43 is located within the bushing 41 and is capable of axial and rotating motion. The distribution member 43 has a central bore 44 terminating at one end in a pump working chamber 45 and at the other end communicating with a distribution bore 46 leading radially outward. Radial fuel inlet bores 47 are disposed between the pump chamber 45 and the distribution bore 46. Disposed in the central bore 44 between the terminus of the inlet bore 47 and the distribution bore 46 is a central pressure valve 48 which is displaceable against the force of a spring 49. The pump chamber 45 is defined between two opposed radial pistons 50. The radial pistons are driven by a cam ring 51 working via rollers 52. The rollers 52 are disposed within bearings 53 rotating with the distributor. The region of the distribution member 43 in which the radial pistons are disposed extends into an inner bore 40 of an enlarged end of the drive shaft 11. An approximately deep end groove in the enlarged end of the drive shaft 11 carries the roller bearings 53, whereby the distribution member 43 does not experience any of the driving forces acting on the pistons 50. A dog clutch 54 couples the drive shaft 11 to the distribution member 43. This clutch permits axial displacement of the distribution member 43 against the force of a return spring 55. The distribution bore 46 terminates in a distribution groove 56 which is disposed on the peripheral surface of the distribution member 43 and which communicates excessively with a number of bores 57 located circumferentially about the bushing 41 and connected to pressure lines 58 in the housing 10. The number of bores 57 and of pressure lines 58 corresponds to the number of cylinders of the engine. Each of the pressure lines 58 is connected via further pressure lines (not shown) to an injection valve in a respective one of the engine cylinders.

Surrounding the distribution member 43

is an annular slide 60 which has interior axial grooves 61a and 61b that control the mouths of the fuel supply bores 47. The external surface of the annular slide 60 sealingly slidably received in a counter-bore 62 of the bushing 41 and is displaceable axially against the force of at least one return spring 63. An annular regulating pressure chamber 64, defined in the counter-bore 62 of the bushing 41 by the distribution member 43 and the annular slide 60, communicates through a regulating pressure line 65 with the cylinder chamber 26 in the pressure regulating valve 8. Depending on the magnitude of the regulating pressure, the annular slide 60 is displaced to a greater or lesser extent against the force of the return spring 63.

As may be seen in Figure 3, the limiting edges of the longitudinal grooves 61a and 61b are not parallel, so that, depending on the axial position of the annular slide 60 with respect to the distribution member 43, the regions of the grooves 61 registering with the supply bores 47 (denoted in Figure 3 as bores 47a, 47b, 47c and 47d) during the rotation of the distribution member 43 is of different magnitude. In the exemplary embodiment illustrated, the supply bores 47 serve as inlet ports and as spill ports for any fuel displaced by the pump pistons 50 but not used for injection. Thus, a fuel supply via the pressure valve 48 to the engine can take place only if the supply bores 47 are blocked. Controlled closing of these bores during the pressurized fuel delivery of the pump pistons 50 will thus define the onset of fuel delivery whereas an opening of these bores will define the termination of fuel delivery. Depending on the particular disposition of the limiting edges of the control grooves, the fuel quantity may be controlled with respect to the onset of delivery or the termination of delivery. It will be understood that at least one of the supply bores 47 must be opened by at least one of the control grooves 61a and 61b during at least a portion of the suction stroke of the pump pistons 50. The grooves 61a and 61b become broader in the direction of the regulating pressure chamber 64 so that displacement of the annular slide 60 against the force of the spring 63 decreases the injected fuel quantity per working stroke. In the exemplary embodiment illustrated in Figure 3, the cam ring 51 has four cam lobes 66 and would thus be associated with a 4-cylinder internal combustion engine. As shown in Figure 3, the roller 52 is just ahead of the cam lobe 66, i.e., the groove 61b is in a position just prior to its isolation from the supply bore 47b. As soon as this isolation is complete, the cam-induced motion of the piston can initiate fuel injection. During further rotation of the distribution member 43, the supply bore

47a overlaps the groove 61a and the injection process is terminated. In the subsequent suction stroke of the piston 50, either the supply bore 47a still overlaps the groove 61a or else the supply bore 47c already overlaps the groove 61b. The edge 61c indicates the narrowest portion of the groove 61a. For example, if this edge 61c cooperates with the bore 47a, the mutual overlap of bore and groove is especially short, i.e., the injected fuel quantity is relatively large at some particular engine speed. Furthermore, the bore 47 is only opened at a later time and the termination of the fuel supply is delayed so that the delivered fuel quantity is further enlarged if the onset of delivery remains constant, for example.

When the control of the annular slide 60 is considered together with the operation of the hydraulic pressure regulator 1, it is observed that constriction of the throttle point 14 results in an increase of the static pressure and hence an increase of the regulating pressure which, in turn, causes a displacement of the annular slide into a position which is associated with a reduced injected fuel quantity. When the load remains constant, the reduced fuel quantity results in a decrease of speed which decreases the static pressure and hence also decreases the regulating pressure whereupon the annular slide 60 is displaced into a position corresponding to a somewhat greater injected fuel quantity. In this manner, for any adjustment of the throttle element 15, there is automatically regulated an associated engine speed. Since the cam 66 does not have a straight line contour but rather a sinusoidal contour, the velocity of the piston at any given engine speed is different for different points of the track of the cam lobe. Fuel is delivered by a cam lobe only during the rising part of the cam track. However, since the portion of the rising part of the cam lobe which causes fuel injection is different for different amounts of fuel, the piston velocity is also variable as a function of the injection timing. The velocity of the piston has a certain influence on the quiet operation of the engine. The earlier the onset of delivery occurs with respect to the rising cam lobe curve, the flatter is the slope of the curve, i.e., the lower the injection velocity. It turns out that, especially for small delivered fuel quantities, the injection velocity should be as low as possible to achieve quiet operation at idling. This can only be achieved by defining the amount of fuel via control of the end of the injection process or, if the injection quantity is controlled by grooves 61 which influence both the onset and the termination of fuel delivery, both the onset and termination must be shifted in the

direction of the flatter curving extent of the cam lobe 66 in the cam ring 51 without rotating the cam ring 51 with respect to the drive shaft because such rotation would shift the entire injection process in the direction of early ignition which would result in noisy operation at low engine speeds. The injection timing adjustment mechanism 3 may be given the independent task of advancing the onset of injection for high rpm so as to extend the otherwise relatively short preparation time at high rpm. For this reason, an rpm-dependent fuel injection time adjustment is especially desirable and, in some cases, even with a load-dependent influence.

An adjustment of the onset of injection is achieved in the apparatus by a rotation of the cam ring 51 within the housing 10. This rotation is actuated by an injection timing adjustment piston 68 which is shown in Figure 1 of the drawings parallel to the axis of rotation but is actually at right-angles to the plane of Figure 1 as can be perceived from Figure 3. The adjustment piston 68 engages the cam ring 51 via a bolt 69. The injection timing adjustment piston 68 is axially slidable in a cylindrical cavity 70 and is radially sealed, defining a pressure chamber 71 and a spring chamber 72. The pressure chamber 71 communicates via a conduit 73 with the control pressure line 16 whereby the control pressure acts on the injection timing adjustment piston 68 against the force of a spring 74. Preferably, a throttle 75 is located in the line 73 just prior to its connection with the pressure chamber 71. Inasmuch as the control pressure increases with engine speed, a displacement of the piston 68 against the force of the spring 74 implies a displacement of the onset of injection to an earlier time (an advance of the injection timing). Conversely, the normal or quiescent position of the adjustment piston 68 is associated with a late onset of injection (retarded injection). Each time the rollers 52 pass a cam lobe 66, they exert forces tending to rotate the cam ring 51 and thus to displace the piston 68 and it is the purpose of the throttle 75 to damp the effect of these forces. The time during which such an effect takes place during the rotation is relatively short so that relatively little fuel flows from the pressure chamber 71 through the throttle 75 back into the line 73, whereas a relatively long time is available for adjusting the position of the injection timing adjustment piston 68. In order to perform a second load-dependent control effort, the spring chamber 72 is connected through a line 76 and grooves 77 with the regulating pressure chamber 64 at one side of the annular slide 60. Preferably, both during reduced fuel supply as well as during full-load operation of the injection pump, the

regulating pressure chamber 64 receives a fuel pressure from the pressure control valve 8 which, at full load, is just large enough to permit the spring 63 to pull the annular slide 60 securely against the cam 82 and, thus, the spring chamber 72 of the injection timing adjustment mechanism 3 also receives regulating pressure via the line 76. The presence of regulating pressure in the spring chamber 72 makes it possible to use the regulating pressure to change the pressure gradient between the pressure chamber 71 and the spring chamber 72 and thus also to change the onset of injection. Furthermore, the provision of the axial grooves 77 in the periphery of the distribution member 43 causes the hydraulic communication between the pressure chamber 64 and the spring chamber 72 to be interrupted during the time of fuel injection by the pistons 50 so that the piston 68 is hydraulically locked and the undesirable oscillations of the injection timing adjustment piston 68 are thereby further shortened.

In the vicinity of the counter-bore 62, the bushing 41 has bores 78 which are opened by the external surface of the annular slide 60 after the latter has traversed a certain stroke. Fuel may flow out of the regulating pressure chamber 64 through these bores 78 so that the control pressure in the line 16 is changed via the line 65. This change in the control pressure causes a displacement of the injection timing adjustment piston 68 in the direction of the pressure chamber 71 so that the onset of injection also changes in the direction of a later injection beginning at a certain engine speed.

As already mentioned above, it may be desirable to keep the driving velocity of the pump piston 50 as low as possible at low engine speeds. This object is attained by providing an axial pin 80 on the cam ring 51 engaging in a notch 81 in the annular slide 60 by cooperation with a groove 81 therein. By disposing the springs 63 in a skew manner, a good contact is obtained between the pin 80 and one wall of the wall of the notch 81. Thus, if the cam ring 51 is angularly displaced, the pin 80 also angularly displaces the annular slide 60. Such angular displacement of the annular slide 60 does not result in any change of the association of the grooves 61 and the fuel supply bores 47 but, if the pin 80 were to be skewed relative to the rotary axis, then the angular position of the annular slide 60 relative to the cam ring 51 will be different for different axial positions. Even though the relative angular displacement by the cam ring 51 is the same for each axial position of the annular slide, that relative angular displacement changes during a relative axial displacement. For certain conditions it may be

desirable to change the full-load fuel quantity, i.e., the initial position of the annular slide, depending on engine speed. For this purpose, there is disposed on the housing 10 at least one cam 82 which is attached by means of shims 83 and a screw 84 and which cooperates with a cam surface 85 on the annular slide 60. Depending on the angular position of the annular slide 60, the initial position of the slide is easily changed. The basic initial position of the slide 60 may also be changed by choosing different shims 83. Another adjustment possibility is given by a screw 86 which is located in the plug 42 and which can change the axial position of the distribution member 43 against the force of the spring 55. The manifold possibilities of regulation and control offered by the present invention make it possible to adapt the fuel injection quantity to any operational conditions of speed and load and not merely at maximum speed, namely by means of the cam 82 and the cam track 85 as well as by the freely selectable association and formation of the control edges of the grooves 61 and by the manner of generating the regulation pressure.

One problem occurring in ordinary fuel injection systems is the generation of a starting excess quantity during starting of the engine. The starting excess quantities intend to achieve a rapid run-up from standstill to just beyond the idling speed and should be shut off thereafter. There should be no influence of the starting excess control process on the other regulatory aspects or pressure control mechanisms of the fuel injection system after the event of engine starting.

Figure 4 is an illustration of the pressure control valve 7 in a position assumed when the pump is standing still. The control pressure piston 18 is seen to be placed in its initial position by the control pressure spring 19. The control pressure cylinder 17 also includes a starting piston 88 which supports one end of the control pressure spring 19 on one face, whereas the other face supports a starting spring 89. The chambers of the control pressure cylinder 17 defined by this initial position of the control pressure piston 18 are interconnected via a bypass 90 which includes a throttle 91. The region of the cylinder 17 which includes the spring 19 is connected via a starting line 92 with a chamber 93 (see Figure 1) which is limited by the distribution member 43, the bushing 41 and the plug 42. Thus, prior to starting the engine, the control pressure line 16 is directly connected to the chamber 93 via the bypass 90 and the starting line 92. As soon as even a relatively low pressure is generated in the control pressure line 16 by the supply pump 9, the distribution member 43 is displaced against the force of the spring

55 until an overflow channel 67 is opened so as to limit the stroke. The distribution member 43 causes the fuel supply bores 47 to be partially overlapped by the surface 59 in the interior bore of the annular slide so that, at the beginning of the pressure stroke of the pump pistons 50, no fuel may return into the regulating pressure chamber 64. In this axial position, fuel may flow into the pump working chamber during the suction stroke only via an extended region of the groove 61b which serves for supplying fuel to the pump working chamber 45. Thus the entire amount of fuel which is deliverable by the pump working chamber 45 is used for injection.

When the engine reaches approximately idling speed, the throttle 91 in the bypass 90 (Figure 4) causes a pressure to build up upstream of this throttle and this pressure build-up tends to displace the pressure control piston 18 against the force of the spring 19 and thus close the bypass 90. The increased compression of the control pressure spring 19, displaces the starting piston 88 against the force of the starting spring 89 and thus opens a relief port 94. The relief port 94 leads to the line 28 which communicates with the housing chamber 12 of the injection pump which is at low pressure. Thus, as soon as the starting piston 88 opens the relief port 94, there is a communication between the chamber 93 and the low pressure line 98 via the starting line 92 so that the distribution member 43 is pushed back to its stop 86 and the process of delivering a starting excess fuel quantity is thereby terminated.

In order to achieve a smooth and uniform functioning of the fuel injection pump, it is desirable to have that pump attain its operating temperature as rapidly as possible yet to prevent over-heating even for extended periods of operation. For this purpose, the pump includes the temperature control mechanism 5 which employs the fuel supplied from a tank (not shown) to the pump by a pre-feed pump (not shown) and in part returned to the fuel tank since the pre-feed pump always delivers as much fuel as would be required by the injection pump under extreme and maximum conditions. As illustrated in Figure 1, fuel is supplied by the pre-feed pump via line 96 to the housing chamber 12 of the injection pump. Fuel then flows from the housing chamber 12 through a bore 97 to the suction side of the feed pump 9 and hence to the individual regulating and control mechanisms as well as to the inlet of the actual fuel injection pump. Any unused fuel flows through a pressure sustaining valve 98, which defines the pressure in the housing chamber 12 of the pump, back to the fuel tank. The pressure sustaining valve 98 is associated with an upstream

thermo-responsive valve 99 which either connects the sustaining valve 98 directly to the suction line 96 via a by-pass channel 100 or connects the sustaining valve 98 with the line 28 leading from the housing chamber 12. When the pump is cold, the larger part of the excess fuel flows off directly via the channel 100 so that fuel in the housing chamber 12 has a chance to warm up before being aspirated by the fuel feed pump 9. When the fuel pump temperature increases, the thermo-responsive valve 99 directs an increasing amount of fuel through the line 28 from the housing chamber 12 to the sustaining valve 98 while the passage through the channel 100 is correspondingly constricted. Beginning with a certain pump temperature, virtually the entire excess fuel flows through the housing chamber 12 and the line 28 to the sustaining valve 98 and is then returned to the fuel tank. The described mechanism and process insure a rapid heating of the pump to its operational temperature while preventing over-heating during extended operation.

WHAT I CLAIM IS:-

1. A fuel injection pump for an internal combustion engine, the pump comprising a distributor member which, in use, is driven synchronously with the engine, an axially displaceable annular valve slide disposed around the distributor member and operative to control the fuel injection quantity by means of control edges and recesses which cooperate with radial bores in the distributor member, a hydraulic regulator producing a regulating pressure in a regulating pressure line for axially displacing the valve slide, a feed pump adapted to be driven at a speed proportional to engine speed and operative to supply the hydraulic regulator with a quantity of fuel which increases with increasing engine speed, an arbitrarily settable throttle disposed in a conduit between the feed pump and the hydraulic regulator for producing a speed dependent pressure drop for the hydraulic regulator, and a spring-loaded pressure control valve for causing a control pressure in a control pressure line downstream of the throttle to increase with increasing fuel flow, the pressures prevailing upstream and downstream of said throttle acting to influence the regulating pressure which in turn influences displacement of the annular valve slide.

2. A fuel injection pump as claimed in claim 1, in which said regulator contains a pressure regulator valve including a sliding piston for defining the spring force of an associated regulator spring, such piston being itself displaced by the fluid pressures prevailing upstream and downstream of said throttle and further including a regulator piston, urged by said regulator spring, for controlling a port through which hydraulic

fluid flows from the control pressure line to the regulating pressure line.

3. A fuel injection pump as claimed in claim 2, in which said throttle is operatively associated with an actuating member for actuating said pressure regulator valve such that said actuating member displaces said sliding piston against a restoring force for an arbitrary actuation of the regulated pressure in the medium speed range of the engine.

4. A fuel injection pump as claimed in claim 2 or 3, in which said sliding piston opens a fluid connection between said feed pump and a location downstream of said throttle when a maximum feed pump pressure as determined by said throttle is exceeded.

5. A fuel injection pump as claimed in any preceding claim in which said annular valve slide moves in the manner of a piston within a counter-bore of a housing also guiding said distribution member and wherein said distribution member, said annular valve slide and said housing together define a regulating pressure chamber to which is admitted the regulating pressure actuating said annular valve slide.

6. A fuel injection pump as claimed in claim 5, in which said regulating pressure chamber has a relief bore which is controlled by the movements of said annular valve slide.

7. A fuel injection pump as claimed in claim 5 or 6 in which the interior bore of said annular valve slide has at least one longitudinal groove terminating in an end face of said annular slide subjected to said regulating pressure for the purpose of cooperating with said radial bores in said distribution member.

8. A fuel injection pump as claimed in claim 7, in which said annular valve slide is axially guided and capable of limited rotation and wherein at least one limiting edge of said longitudinal grooves forming one of said control edges is non-parallel with respect to the longitudinal axis of said annular slide.

9. A fuel injection pump as claimed in claim 8, in which said radial bores in said distribution member which are controlled by said longitudinal groove serve both as suction ports and as spill ports for controlling, respectively, the onset of fuel delivery and the termination of fuel delivery.

10. A fuel injection pump as claimed in any preceding claim, further including pump pistons and a cam driven actuator mechanism for causing reciprocating motion of said pistons, a drive shaft extending through the housing of said fuel injection pump for operating said cam mechanism and injection timing adjusting means for changing the relative disposition of said cam mechanism and said pump pistons for a given angular

position of the drive shaft for the purpose of changing the onset of pumping and the onset of fuel delivery with respect to the angular position of said drive shaft and wherein said adjusting means includes an injection timing adjustment piston which is displaceable against a spring by the control pressure determined by said pressure control valve.

11. A fuel injection pump as claimed in claim 10, further comprising a throttle in the pressure line leading from said control pressure line to said injection timing adjusting means.

12. A fuel injection pump as claimed in claim 10 or 11, in which the end face of said injection timing adjustment piston not experiencing control pressure is subjected to said regulating pressure.

13. A fuel injection pump as claimed in claim 12, in which said distribution member controls communication between said regulating pressure line and said injection timing adjusting means.

14. A fuel injection pump as claimed in claim 10, 11, 12 or 13, in which said injection timing adjustment piston is adapted to angularly adjust said annular valve slide.

15. A fuel injection pump as claimed in claim 14, in which said annular valve slide adopts an initial position corresponding to full fuel delivery through inter-action of at least one cam track with the annular valve slide during angular displacement thereof.

16. A fuel injection pump as claimed in any preceding claim in which said distribution member may be moved relative to said annular valve slide for the purpose of producing an engine starting excess quantity.

17. A fuel injection pump as claimed in claim 16, in which said distribution member is movable hydraulically against a restoring force and wherein said pump further includes a stop member for defining the initial position of said distribution member corresponding to a full load operation.

18. A fuel injection pump as claimed in claim 16 or 17 further including a starting control valve for adjusting the pressure applied to said distribution member.

19. A fuel injection pump as claimed in any of claims 16 to 18 further comprising a bypass causing communication between two separate chambers in said pressure control valve when a piston within said pressure control valve is in its initial condition, said bypass including a throttle and communicating with the end face of said distribution member; whereby, in the normal operational position of said piston in said pressure control valve, said bypass is closed thereby.

20. A fuel injection pump as claimed in claim 19, in which said pressure control

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valve includes a spring, one end of which acts on said pressure control piston, whereas the other end of said pressure control spring is supported on a starting piston which permits communication between the end face of said distribution member and a pressure relief channel when said pressure control piston is displaced and obturates said bypass.

21. A fuel injection pump as claimed in any preceding claim, in which a thermo-responsive valve is disposed in a pump by-pass between the fuel inlet to said fuel injection pump and a return line from said fuel injection pump, and closes said by-pass when said injection pump is at normal operating temperature.

22. A fuel injection pump as claimed in any preceding claim, which employs radially disposed pumping pistons whose displacement takes place in directions normal to the longitudinal axis of said distributor member.

23. A fuel injection pump constructed and adapted to operate substantially as hereinbefore particularly described with reference to and as illustrated in Figures 1, 3 and 4 of the accompanying drawings.

24. A fuel injection pump constructed and adapted to operate substantially as hereinbefore particularly described with reference to Figure 2 of the accompanying drawings.

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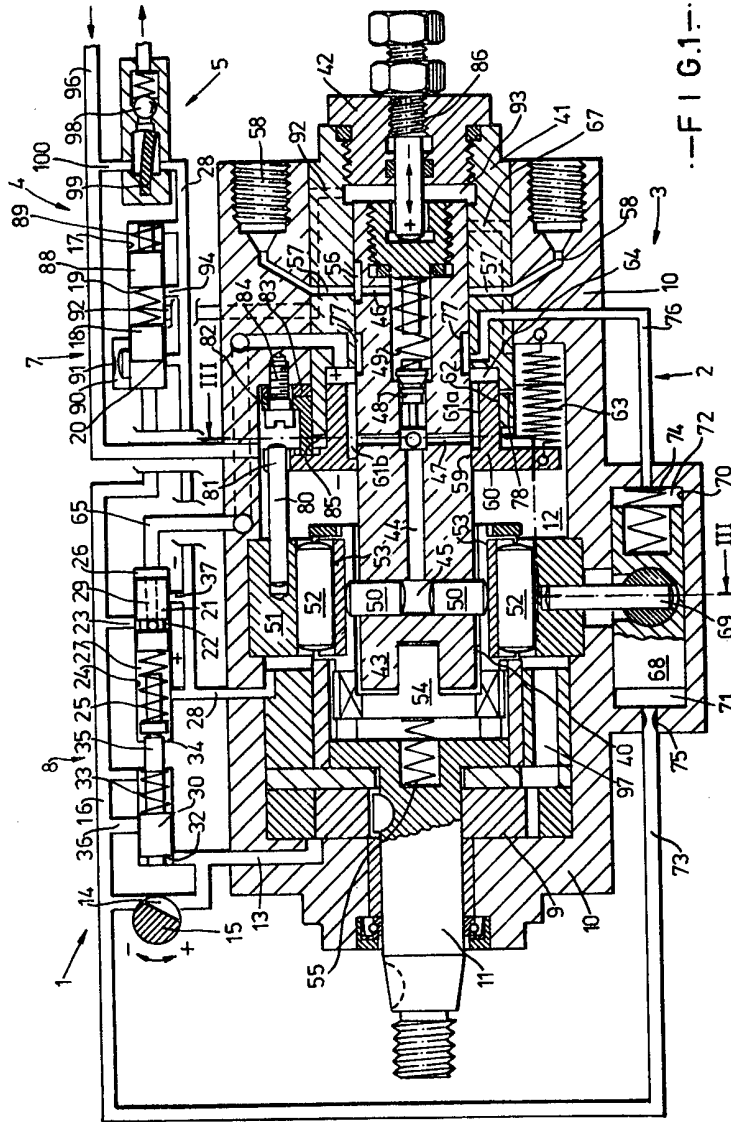


FIG. 1

