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FUEL INJECTION PUMP

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Fig. 1.

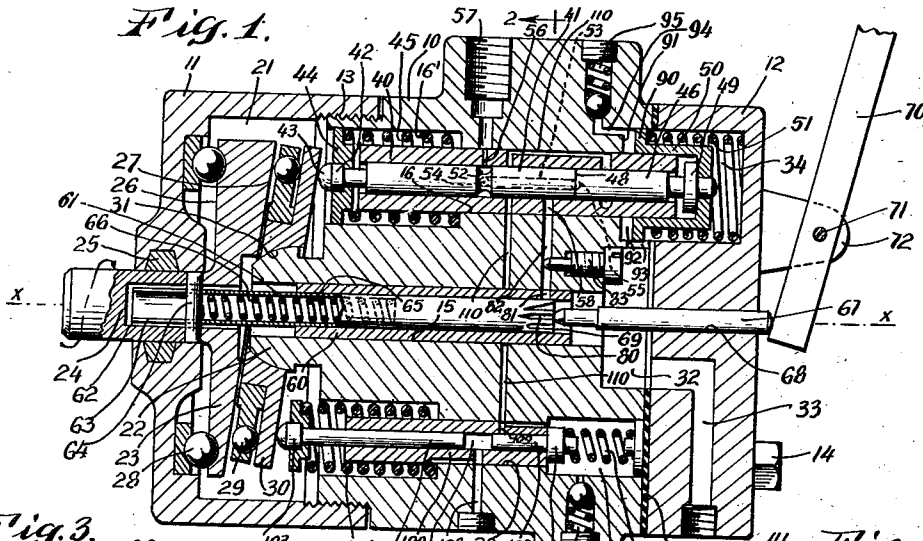
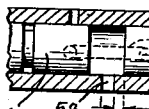
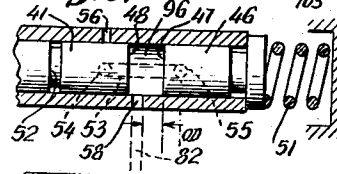


Fig. 3.



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FUEL INJECTION PUMP

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The invention relates to fuel injection pumps. Injection pumps for high speed internal combustion engines in which the control of the fuel quantity may be coupled with the control of the combustion air, have to satisfy certain requirements in respect to mechanical features, controllability, kinematics and physical properties. Mechanically, it is necessary that the drive of the pump plungers is safely operative at very high speeds and that the mass forces are small. As to controllability, it is required that the quantity of fuel delivered by the pumps is adjustable and can be adapted to the air consumption of the engine at any time throughout the entire ranges of speed and output. Kinematically, the injection process must be controllable, depending on the prevailing injection law. In order to attain a stable control of the fuel quantities, it is also necessary to prevent or at least greatly to reduce the harmful physical effects of the fuel, such as the formation of steam, gas or air bubbles as well as of cavitations during the processes of the fuel inlet and outlet.

Injection pumps of conventional type are suited to fulfill the one or the other of the basic requirements. However, none of them is satisfactory either from the view point of operation or from the view point of simplicity of structure.

It is an object of the invention to avoid the drawbacks of the conventional devices and to provide an injection pump which fulfils all or at least most of the mentioned requirements in a compact structure with simple means and by applying a single control organ.

The invention particularly contemplates the fulfilment of the requirements by providing any or all of the following means, viz.: an antifric-tion swash plate drive in a fuel injection pump with one or several cylinders, to operate the plungers and, if so desired, also ancillary pumps, such as a lubrication pump, low pressure fuel pump, etc., thereby avoiding the conventional tappets and rocking levers; an axially shiftable slide valve coupled with the pump drive for common rotation and controlling an inlet duct to the pressure chamber of the pump, in order to adjust the fuel output, in dependence of the R. P. M., by a predeterminable and variable control of a throttle, of the inlet time, and of a maximum fuel quantity; one or several stepwise arranged grooves provided on the outer surface of the slide valve and which are located in the area of the inlet duct along the line of the selected adjustment path and may be axially straight or slanting, wedge-shaped or arcuate ac-

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cording to the required adjustment characteristic, whereby, the inlet angle and the inlet moment can be changed dependent on the axial shift of the slide valve, and the fuel quantity to be injected by the pump will be adjustable dependent on the R. P. M. of the slide valve.

The invention further aims to provide means to adjust the volumetric capacity of the pressure chamber of the pump at the moment of closing or opening of the inlet duct by either the plunger of the pump or by the slide valve, whereby, the adjustment of the actual highest amount of fuel quantity to be ejected by the pump, i. e. injected into the engine, may be rendered approximately constant or variable for the purpose of adapting that quantity to the ratio of air supplied to the engine in the ranges of low R. P. M. and output.

Another object is the provision of a plunger system whereby the pressure of the admeasured quantity of fuel is raised to the injection pressure, and of fuel ducts from the pump to the injection pipe, which ducts are located outside of the pressure chamber in order to avoid axial or transverse bores in the plunger system which might be the cause of harmful cavities in the pressure chamber.

The invention also contemplates to control the moment of ejection by a ring groove or bore which is provided in the body of the plunger but not directly connected to the pressure chamber, and to predetermine the ejection angle by a throttle at the opening into the injection pipe of the engine.

Another object is the provision of means to prevent or reduce the formation of gas-, steam-, or air bubbles and their harmful effect on the adjustment of the quantity of the fuel by creating a strong fuel circulation and rinsing of the suction chamber, and reducing the capacity of the pressure chamber to practically zero at the end of the ejection period.

The invention finally aims to provide a plunger system comprising, for each injection pump cylinder, a pair of cooperative plungers wherein one of the plungers is simultaneously operative with one portion in the injection pump cylinder and with another portion in the cylinder of a circulation pump connected with the injection pump.

Further objects and details of the invention will be apparent from the description given hereinafter and the accompanying drawing illustrating an embodiment thereof by way of example. In the drawing

Fig. 1 is a longitudinal cross-section, along line 1—1 of Fig. 2, of a fuel pump according to the invention;

Fig. 2 is a cross-section along line 2—2 of Fig. 1;

Figs. 3 and 4 are diagrammatical illustrations of the cooperation of the plunger system of the pump with the slide valve in controlling the opening and closing of the fuel inlet duct, Fig. 3 showing part of the pump cylinder in cross-section with plungers therein, and Fig. 4 showing a development of the grooved portion of the slide valve;

Figs. 5 and 6 and Figs. 7 and 8 are views similar to Figs. 3 and 4 showing modified slide valve grooves, and illustrate changes of the fuel control in relation to the volumetric capacity of the pressure chamber, and

Fig. 9 is an elevation partly in section of a modified slide valve within its bushing approximately in its position of the fuel control for engine idling.

Referring now to the drawing, Figs. 1 and 2 illustrate a device according to the invention in which four fuel pumps are combined in one for charging an engine with four cylinders. However, any other number of pumps may be similarly combined. The unit comprises a central housing part 10 of substantially circular cross-section. End pieces 11 and 12 are secured thereto by any suitable means, such as for instance the thread 13 to connect the piece 11, and the screw bolts 14 to connect the piece 12. A gasket 12' is provided between parts 10 and 12. The central housing part 10 is provided with an axial bore 15 and five bores parallel to the housing axis $x-x$, of which four bores 16, 17, 18 and 19 are equally spaced from one another, whereas the fifth bore 20 is located between the bores 18 and 19. The housing parts 10 and 11 form a compartment 21 for the accommodation of the driving elements, and into which the part 10 projects with a spherical head 22. The drive is of the swash-plate type and comprises the plate 23 integral with stub axle 24 which is journaled in housing part 11 at 25 and may be coupled by means not shown to a rotating part of the engine for which the pump is destined. The plate 23 has one face 26 in a plane at right angles to the axis $x-x$ and another face 27 in a plane aslant thereto. An anti-friction thrust bearing 28 is arranged between the housing part 11 and the face 26 and another similar bearing 29 is provided between the face 27 and another plate 30 which has a central bore 31 with a spherical inner face seated on the spherical head 22.

The housing part 12 closes an axial chamber 32 in the part 10, and contains a conduit 33 which opens into chamber 32 and may be connected to a fuel tank (not shown). Chamber 32 communicates with bore 15. The part 12 is furthermore provided on its inner side with four blind bores which are co-axial with the bores 16 to 19 respectively and of which only the bore 34 is visible in the drawing.

The bores 16 to 19 contain the cylinders and plungers of the four pumps which are alike in every respect. Their detailed description may be limited therefore to the parts accommodated in bore 16 and their associated elements. A bushing 40 is firmly pressed into the bore 16 and extends with its end into a widening 16' of the bore 16 and with its other end into the bore 34. A main plunger 41 is axially shiftable in bushing 40. The outer end of the plunger is formed

with a collar 42 and a head 43 the latter bearing against the pressure plate 30 of the swash plate drive. A washer 44 engages the collar 42, and a spring 45 interpositioned between the washer 44 and the shoulder at the inner end of the widening 16' urges the plunger 40 against the pressure plate 30. A second plunger 46 which in order to distinguish it from the pump plunger 41 will be hereinafter denoted as the follower plunger, is movable in the same bore 16 and has an end face 47 adjacent the end face 48 of the pump plunger 41. The other end of the follower plunger 46 which projects from the bushing 40, has a collar 49 against which a cup-shaped washer 50 bears. A spring 51 is inserted between the washer 50 and the bottom of the bore 34. Now it will be clear that upon rotation of the axle 24 with swash plate 23, plunger 41 will first be axially shifted against the restraint of spring 45, and then bearing with its face 48 against face 47 of the follower plunger 46, will also shift the latter against the restraint of the spring 51. On the return stroke, the springs 45 and 51 will tend to urge the plungers to follow the pressure plate 30 which swivels about the center of the ball head 31. However, whereas plunger 41 can follow pressure plate 30 through the whole length of the return stroke, plunger 46 will be stopped when flange 50 of cup washer 50 bears against the bottom 52 of a recess 52 provided in housing 10 in continuation of the bore 34 in housing part 12, or when the collar 49 abuts against the end of the bushing 40.

For a purpose to be explained hereinafter, plunger 41 is provided with an annular groove 52 spaced a substantial distance from its front face 48. Bushing 40 is provided with a longitudinal groove 53 communicating at its one end with ring groove 52 through a transverse bore 54 and having its other end communicating with another transverse bore 55 which ends at the point where the front face 48 of plunger 41 is located when in its outer dead center position illustrated in Fig. 1. Bore 55 forms the outlet port of the pressure chamber of the pump as it will be described hereinafter. Another transverse bore 56 in bushing 40 also communicates with ring groove 52 in the plunger position of Fig. 1 and extends at right angles with respect to bore 54. The bore 56 which is narrow to form a throttle communicates with a conduit 57 in the housing part 10, which constitutes the discharge passage of the pump and may be connected to the injection pipe of the engine (not shown). There is still another transverse bore 58 in bushing 40, which is located intermediate the ends of groove 53 but does not communicate with the latter. Bore 58 forms the inlet port of the pump.

The major part of the axial bore 15 of housing portion 10 is lined with a bushing 60 within which a cylindrical slide 61 is rotatable and axially shiftable. The one end of slide 61 extends into an axial bore 62 of the axle 24 and is longitudinally slotted at 63 engages a pin 64 which traverses the bore 62 of axle 24. Slide 61 is also provided with an axial bore 65 extending a substantial length from the slotted end. Within the bore 65, a spring 66 is located and bears with its one end against the aforementioned pin 64, thus tending to shift the slide 61 towards the right hand side in Fig. 1. A rod 67 co-axial with slide 61 is shiftable in a bore 68 provided in the housing 12 and bears with its one end against the front face 69 of slide 61 and with its other end against a control lever 70 pivoted at 71 on a bracket

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extension 72 of the housing part 12. The outer surface of the slide 61 is provided with a groove or grooves in general denoted by 80 and extending from the front face 69 a distance towards the other end of the slide. Thus, the groove or grooves 80 are in permanent communication with chamber 32. In the portion of bushing 60 in which the grooved end of the slide is movable, a transverse bore 31 is provided which communicates through a duct 82 in housing portion 10 with the aforementioned bore 53 of bushing 40. An adjustable throttle of the duct 82 is provided by means of a screw 83 the head of which extends in the space 32, and the pointed end of which can penetrate into the duct 82 so as to open or close the latter more or less depending on the setting of the screw 83.

The pump so far described operates in the following manner: Duct 33 is to be connected to a fuel container and the duct 57 to the injection port of the engine to be charged by the pump. Control lever 70 may be either directly operated by the user or it may be connected with the air control of the engine in order automatically to adjust the ratio of air and fuel charge. Provided the apparatus is in the position of Fig. 1 and the drive is started to rotate in the direction of the arrow, fuel will enter through the duct 33 and fill the suction chamber 32 of the pump and the control groove 80 of the slide 61. In the outer dead center position of plunger 41, both plungers engage each other with their front faces 47 and 48, as shown in Fig. 1. When, now, the swash plate 23 rotates both plungers 41 and 48 will move together towards the inner dead center position under the action of springs 45 and 51. The common movement continues until flange 90 bears against face 91 of the housing and stops the follower plunger 48 whereas plunger 41 continues its movement. The separation of the plungers opens the pressure chamber 96 (Fig. 4) of the pump between the front faces of the plungers. During this part of the stroke the face 48 of plunger 41 passes the bore or port 53 and opens the inlet duct 82. As soon now as groove 90 of the rotating slide 61 registers with the port 31 fuel will flow from chamber 32 through grooves 80, port 31, duct 82 and port 53 into the pressure chamber 96. The chamber will be filled partly or entirely depending on the duration the connection with chamber 32 is established. This duration is determined by the shape of the flanks of groove 90 and the position and R. P. M. of the slide 61. On the following stroke of plunger 41 towards the outer dead center the fuel confined in the pressure chamber 96 will be urged against the front face of 47 of plunger 48 and some of the fuel may be forced back to the suction chamber 32. As soon however, as port 53 has been closed by the advancing plunger 41, plunger 48 will be caused to take part in the movement against the restraint of spring 51. When, then, front face 47 of plunger 48 passes the port 55, the fuel confined under a pressure which is higher than the ejection pressure will enter the longitudinal groove 53 and the ring groove 52 of plunger 41 through bore 54. From there the fuel passes through the aforementioned throttle bore 56 and can charge the engine through the passage 57. The ejection continues as long as time is required to force the fuel through the throttle 56 under the pressure of spring 51 bearing on plunger 46 via the cup member 50. The ejection period is ended when both plungers engage each other with

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their front faces and all fuel has been expelled from the pressure chamber. The length of the ejection period in degrees of a cycle and the pressure can be predetermined by a correct selection of pressure of spring 51 in relation to the cross-section of the throttle bore 56.

The fact that the ducts 55, 53, 54 leading the fuel to the injection passage 57 are arranged outside the pressure chamber 96, and that the ejection moment is controlled by means of the annular groove 52 of plunger 41 has essential advantages. On the one hand, the principal effect of the system of the cooperating pump and follower plungers can be fully utilized while simultaneously the formation of harmful cavities and of gas- and steam bubbles in the pressure chamber 96 is avoided. On the other hand the throttle 56 ensures gradually increasing and abruptly stopping ejection kinematics independent from the drive of plunger 41, and also helps towards a thorough atomization of the fuel.

Means are provided in order to prevent the formation of bubbles of gas, air, steam or foam also in the suction chamber of the pump by thoroughly flushing that chamber with fuel through circulation between the fuel tank and the pump. This is accomplished by designing flange 93 of the spring washer 50 which is in any way required, as a plunger operative in the cylinder space 92, 34 so as to constitute a suction and return pump. For this purpose a passage 93 is provided between the space 92 and the suction chamber 32, and another passage 94 connects the space 92 to a check valve 95 from where a return pipe (not shown) may lead to the fuel tank. It will be clear, that the plunger 46 imparts to flange or plunger 90 its reciprocating motion so that fuel will be constantly pumped from the tank through chamber 32, pump space 92, valve 95 and back to the tank.

Means are also provided to lubricate the movable parts of the apparatus. Whereas, in the illustrated embodiment four fuel pumps are combined in one unit, only one oil pump is shown. However, it will be clear that more than one such pump, may be present. The oil pump is arranged in the aforementioned bore 20 of the housing part 10 and is of a design similar to that of the fuel pumps. Thus, in a bushing 100, two plungers 101 and 102 are reciprocable. Plunger 101 bears with its head 103 against the pressure plate 30 of the drive under the action of a spring 104. Plunger 102 has a head 105 which can abut against the front face of bushing 100, and another spring 106 is interpositioned between head 105 and the wall of housing portion 12. The space 107 intermediate the plungers when they are in the inner dead center position or close to it as shown in Fig. 1, is the pump chamber provided with an inlet duct 108 which may be connected to an oil tank (not shown), and with an outlet 109 which is connected through ducts such as 110 to the parts to be lubricated. A check valve 111 is in communication with the space 112 housing the spring 106, from where the oil may be returned to the tank. The pressure against which the valve 111 opens is preferably selected somewhat higher than the pressure against which the fuel pumps operate. The output of the oil pump is constant and is predetermined by the distance of the front face of the plunger 102 from the inlet 108 in the inner dead center position. The oil pump may also be used to lubricate the engine for which the fuel pump is destined.

As stated hereinbefore, it is another object of

the invention to avoid harmful disturbances of the fuel quantity control and to provide for a great stability of the control. This can be attained according to the invention by keeping the flow velocity of the fuel in the inlet to the pressure chamber 96 as low as possible in order to prevent a break of the flow to the pressure chamber particularly if the fuel is heated and also to prevent the formation of cavitations and of strong back pressure waves. For this purpose the groove 80 is so shaped and arranged that in order to adjust the fuel quantity in the pressure chamber 96 throughout the entire control range that angle of a cycle of operation is as large as possible during which the inlet occurs. Furthermore, for the same purpose, the fuel inlet duct 82 is provided with an adjustable throttle in the form of the screw 83 although according to experience, it is possible to supply the maximum fuel quantity required at highest R. P. M. over a much smaller inlet angle.

The control of the fuel quantity preferably coupled by means of a cam or the like (not shown) with the air control of the engine is accomplished by varying the axial position of the cylindrical slide 61, that means by a shift of lever 70 engaging rod 68 against which the slide is urged by spring 66. More or less fuel flows into the pressure chamber 96, depending on the time during which the inlet duct 82 is open which time in turn depends on the varying width of the control groove 80 as well as on the position and R. P. M. of the slide 61. In each axial position of the slide the fuel quantity decreases with an increase of the R. P. M. of the pump and slide, and increases with a decrease of the R. P. M. In order to prevent the fuel quantity from exceeding the maximum amount when the R. P. M. decrease due to a higher load of the internal combustion engine, and with a view to suiting the quantity of the fuel to that of the air charged into the engine in the range of the lower R. P. M., the volumetric capacity of the pressure chamber between the inlet port 58 and the front face 41 of the plunger 45 will be predetermined according to the required fuel-air-ratio. According to the invention, the effective volumetric capacity of the pressure chamber can be rendered approximately constant or variable at the moment of the closing or opening of the inlet duct 82 by the plunger 41 or by the slide 61.

Figs. 3-8 illustrate diagrammatically three different embodiments for determining the volumetric capacity of the pressure chamber in relation to the moments of opening and closing the inlet duct 82, wherein Figs. 3, 5, and 7 show the pump plungers and Figs. 4, 6 and 8 are developments of the surfaces of the respectively coordinate slides.

In Figs. 4, 6 and 8, the moments of the opening and of the closing of the inlet duct 82 are entered on the developed slide surfaces by dash lines of which the line o indicates the opening and the line s the closing of the duct by plunger 41. In Figs. 3, 5 and 7, the plunger 41 is shown in its inner dead center position and the dash lines ID in Figs. 4, 6 and 8 indicate the points of the slide which register with the center of the pore 81, i. e. the entrance end of the inlet duct 82, when the plunger is in the inner dead center position. The location of the front face 48 of plunger 41 in its outer dead center position is indicated by line OD in Fig. 3.

The parts shown in Figs. 3 and 4 fully correspond to those of Fig. 1. As clearly appearing in

Fig. 4, the control groove 80 of the slide is tripartite, the parts being denoted by 801, 802 and 803 respectively. The tripartite groove is equivalent in its effect to one single groove as the lands between the part grooves are actually overlapped by the diameter of the inlet port 81. This particular form renders it possible to design part grooves with a rather pointed taper, which gradually register with the port 81 upon a shift of the slide, and thus to locate the range of idling and small loads in the neighborhood of the inner dead center position of the plunger 41, whereby the highest regulating effect in dependence of the R. P. M. will be attained at the highest vacuum in the pressure chamber 96.

In the control position for full load, the groove 801 opens port 81 of the inlet duct 82 shortly after the moment (line o) at which plunger 41 opens port 58. By an axial displacement of the slide, towards the right in Fig. 4 the beginning of the fuel inlet will be shifted more and more towards the inner dead center owing to the inclination of the controlling edge a. When moving towards the outer dead center, plunger 41 will close the inlet duct before the axially directed trailing control edge b of the groove 803 slides across the port 81. Owing to this structure, if the inlet period is long enough, which is possible provided the R. P. M. of the slide is small, the entire pressure chamber 96 will be filled with fuel. When, then, the plunger moves towards its outer dead center, excess fuel will be returned through the inlet duct 82 and groove 80 to the suction chamber 32 until the plunger has passed the inlet port 58. The quantity of fuel enclosed between the plungers 41 and 46 constitutes the approximately constant, maximum output quantity which can be attained. However, this quantity is not available owing to the throttling and the short inlet angle in the range of the point of the control groove 802, just as it should be at approximately idling R. P. M.

The embodiment of Figs. 5 and 6 differs from the foregoing in that there is an uninterrupted groove 800 having a leading control edge c of S-curve form and a straight inclined trailing edge d. In this embodiment the maximum fuel output is variable and is determined by the inclination of the edge d closing the port 81 at an earlier or later moment depending on the axial position of the slide. The quantity of fuel which can be entrapped between the plunger 41 and plunger 46 at the moment (line s) when plunger 41 closes the port 58 is approximately one half of the maximum fuel quantity required in the full load position. The regulation of the quantity for idling is accomplished in this embodiment shortly before the moment of closing (line s) of the port 58 by plunger 41, and the controlling effect in dependence of the R. P. M. is less high.

In the embodiment of Figs. 7 and 8, means are provided whereby the plunger 41 is prevented to follow the drive fully to the inner dead center position. In other words there is an interval in which the plunger does not move after a stroke from the outer towards the inner dead center. For this purpose, an abutment 97 may be provided against which the plunger can bear, so that it separates from the receding pressure plate 30 diagrammatically shown in Fig. 7. The groove 80 of the slide in Fig. 8 comprises two substantially triangular part grooves 804 and 805 with inclined trailing edges.

In this embodiment, the range of control of the maximum fuel quantity is also variable. Con-

trary to the other embodiments the control is accomplished by the inclined control edge *e* of groove 804 or edge *e'* of groove 805 closing port 81 at an earlier or later moment figured from the opening of port 52 during the inward stroke of plunger 41. The volumetric capacity of the pressure chamber during the rest period of plunger 41 corresponds to the required maximum fuel quantity.

Only a short stroke of plunger 41 is needed in order to obtain the required capacity of the pressure chamber, but it is desirable that the inlet angle of the combined grooves 804, 805 is as large as possible. The rest period of the plunger is helpful to that end, as it renders it possible to apply a sufficiently wide groove, considering the angle *f* through which the slide turns during the period of rest of the plunger.

Now it will be clear that according to the invention, any control characteristic of the maximum fuel quantity can be attained for all load ranges of an internal combustion engine by a suitable formation and arrangement of the control edges *d* or *e* of the slide groove in relation to the opening or closing moment of the inlet duct.

It has been stated that in order to be able to apply a large inlet angle, it is advisable to throttle the inlet passage of the fuel flow to the pressure chamber of the pump, for which purpose screw 83 has been provided in Fig. 1.

Another means for the same purpose is shown in Fig. 9 where the slide 161 which may be substituted for slide 61 in Fig. 1 has a front extension 162 of a somewhat reduced diameter. The extension is provided with a restricted radial bore 163 communicating with an axial bore 164 which opens through another radial bore in the groove 180. Thus, fuel can enter from the chamber 32 through the throttling bore 163 and can pass through groove 180 into inlet duct 32. In this embodiment it is advisable to apply an undivided groove 180. The trailing edge *g* of the latter is so located that it controls the port 81 after the moment of closing of the inlet duct by the plunger. Edge *g* may be axially directed or inclined as the edge *d* in Fig. 6.

It will be apparent to those skilled in the art that many modifications and alterations of the illustrated structure can be made without departure from the essence and spirit of the invention which for that reason shall not be limited but by the scope of the appended claims.

I claim:

1. In a fuel injection pump, a housing part having an axis and being formed with a ball head at one of its ends, a pressure plate having a central spherical cavity engaging said ball head, a swash plate including a stub axle co-axial with said housing portion, an antifriction bearing between said swash plate and said pressure plate, a plurality of cylinders formed in said housing, parallel to said axis and equally spaced from one another, a first plunger in each of said cylinders, a first spring in engagement with the outer end of said first plunger so as to urge it against said pressure plate, a second plunger reciprocable in said same cylinder, a second spring connected to said second plunger and tending to urge the latter with its front face in engagement with the front face of said first plunger, an abutment for said second plunger to prevent the latter from following the first plunger the total length of the stroke of said first plunger under the action of said second spring so as to form a pump cham-

ber in said cylinder when said plungers separate, a control organ rotatable in an axial bore of said housing portion, a fuel inlet duct for each cylinder forming a port in said pump chamber under the control of the coordinate first plunger, the other ends of said ducts being under the control of said control organ, said control organ communicating with a fuel source so as to admit fuel to said inlet ducts, one after the other, upon rotation of said axle.

2. A fuel injection pump as claimed in claim 1, further comprising a lubricating pump arranged in said housing between two adjacent ones of said cylinders and including a plunger in operative connection with said pressure plate, and oil ducts between said lubricating pump and said cylinders.

3. In a fuel injection pump the combination of a cylinder, a first plunger reciprocable therein, a rotary drive including means to reciprocate said plunger, a second plunger reciprocable in said cylinder and adapted to engage said first plunger in a face to face relationship, a spring bearing on said second plunger and urging it towards said first plunger, an abutment for said second plunger to prevent the latter from following said first plunger the total length of the stroke of said first plunger in the direction of the force of said spring so as to form a pump chamber in said cylinder when said plungers separate, a fuel inlet duct forming a port in said chamber under the control of said first plunger, a rotary control organ coupled with said drive and controlling flow through said duct at a point of the latter other than said port, and a two part discharge duct provided in the cylinder wall, the first part being arranged substantially parallel to the cylinder and forming with its one end a discharge port of said chamber under the control of said second plunger, the other end of said first part of said discharge duct forming a third port in said cylinder between said chamber and said drive and being under the control of the edges of an annular groove provided on said first plunger, the other part of said discharge duct opening with one of its ends towards the outside and forming with its other end a fourth port peripherally spaced from said third port but at substantially the same axial distance from said chamber as said third port.

4. A fuel injection pump as claimed in claim 3, wherein said fourth port constitutes a throttle.

5. In a fuel injection pump a housing including a cylinder, a first positively driven plunger reciprocable in said cylinder, a second plunger in said cylinder in a face to face relationship to the first plunger and extending with its end from said cylinder, said plunger end embodying a third plunger reciprocable in a cylindrical space provided in said housing coaxial with said cylinder, a spring between said third plunger and the bottom of said space to urge said second plunger into engagement with said first plunger, said third plunger being adapted to abut against a portion of said housing to prevent said second plunger from following said first plunger the total length of the stroke of the latter thereby to open a pressure chamber between the first and the second plunger, a suction chamber in said housing and adapted to be connected to a fuel source, an inlet duct from said suction chamber to said pressure chamber, a discharge duct connected to said pressure chamber for delivery of fuel to an engine, both said ducts being under the control of said first plunger, an

inlet connection between said suction chamber and said cylindrical space under the control of said third piston, an outlet duct from said space, and a check valve in said outlet duct.

6. In a fuel injection pump, the combination of
 5 a housing, a first cylinder and a second cylinder in said housing, a first plunger reciprocable in said first cylinder, a rotary drive including means to reciprocate said plunger, a second plunger reciprocable in said first cylinder and adapted to en-
 10 gage said first plunger in a face to face relationship, a spring bearing on said second plunger and urging it towards said first plunger, an abutment for said second plunger to prevent the latter from following said first plunger the total length of the stroke of said first plunger in the direction of the force of said spring so as to form a pump cham-
 15 ber in said first cylinder when said plungers separate, a fuel inlet duct forming with its one end in an inlet port in said chamber under the control of said first plunger, and having its other end in another port of said second cylinder, a cylindrical slide coupled with one of its ends to said drive
 20 so as to be rotatable and axially shiftable in said second cylinder, at least one groove provided in the surface of said slide and extending from the free end towards the other end of said slide a distance substantially the distance said slide is axially shiftable, the edges of said groove being in controlling relationship to said other port, means
 25 to establish communication of said groove with a fuel source, and operative means to shift said slide in the axial direction.

7. A fuel injection pump as claimed in claim 6, wherein the groove edge leading upon rotation of said slide is rearwardly inclined from the free end of the slide, and the trailing edge is axially directed, the drive being so connected to said first plunger and said slide and the parts being so arranged that the leading edge opens said other port after the first plunger opens said inlet port, and that said trailing edge closes said other port after the closing of said inlet port.

8. A fuel pump as claimed in claim 6, wherein said leading groove edge is rearwardly curved in an S-shape, and said trailing edge is rearwardly inclined.

9. A fuel pump as claimed in claim 6, wherein the leading edge of said groove is axially directed and the trailing edge is forwardly inclined from its open towards its closed end.

10. A fuel pump as claimed in claim 6, wherein another spring is included in said means to reciprocate said first plunger and connected to the latter so as to urge said plunger into engagement with said drive, a second abutment means connected with said first plunger to prevent said first
 15 plunger from following said drive under the action of said other spring before the end of the retreating stroke of said drive, so as to cause an interval during which said first plunger does not move, the edge of said slide groove leading when said slide rotates being axially directed, and the trailing edge being forwardly inclined from the free end of said slide.

11. A fuel injection pump as claimed in claim 6, said groove being divided in several parts, the lands between the groove parts extending in a direction from the free end of said slide towards its other end and being of a width smaller than the diameter of said other port, and at least one of said part grooves being sharply pointed.

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