Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

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Fig. 1.

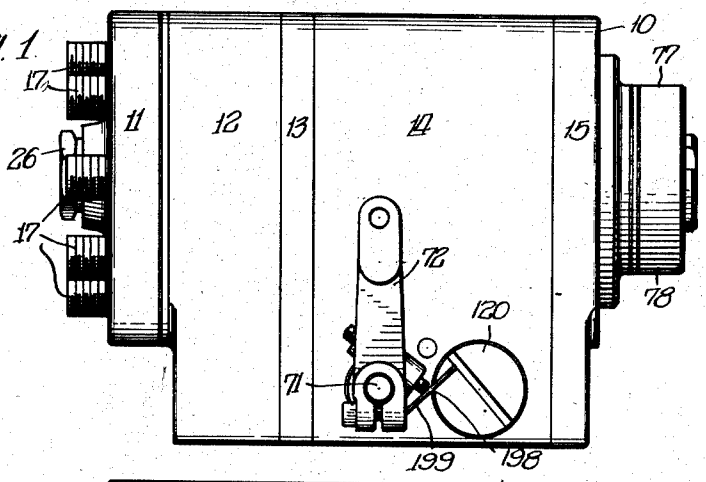


Fig. 2.

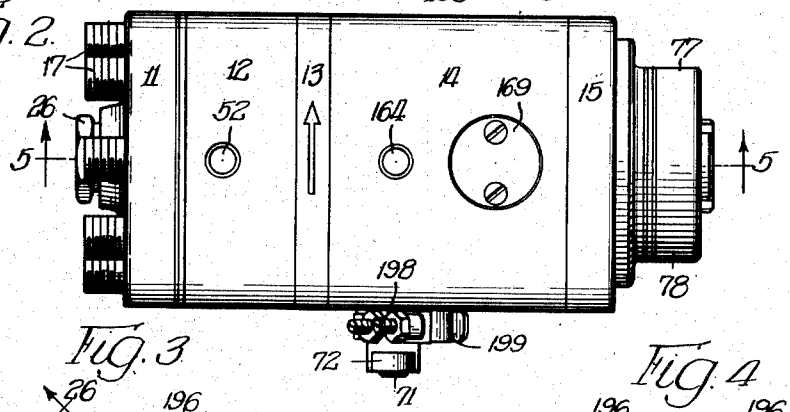


Fig. 3

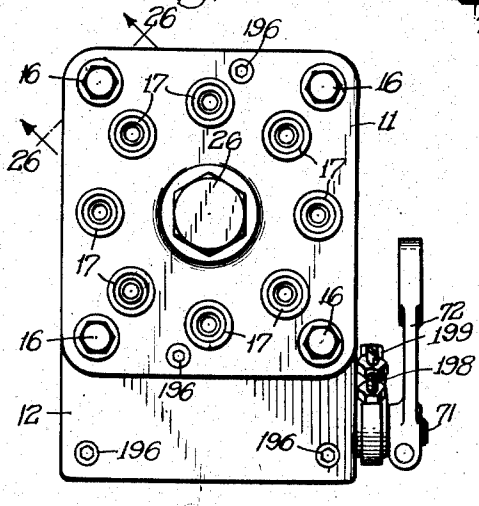
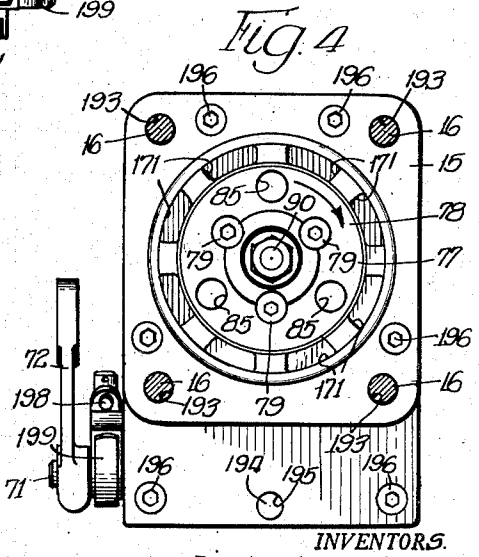


Fig. 4



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Bruest & Warden
attys.

Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

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Fig. 5

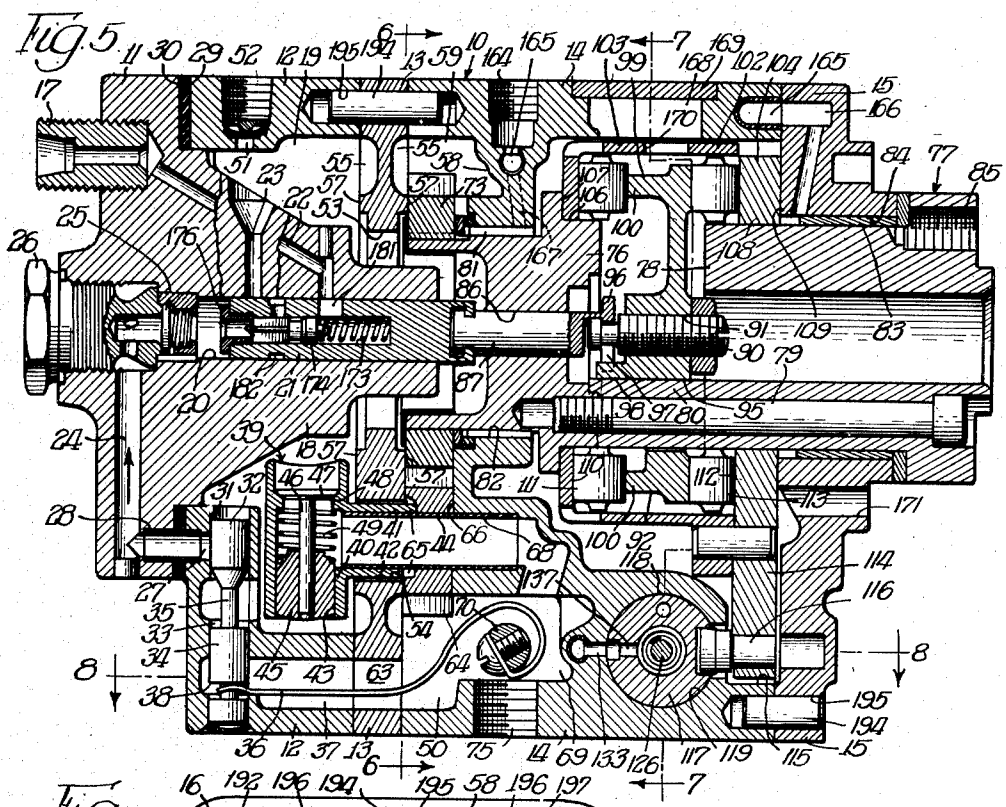
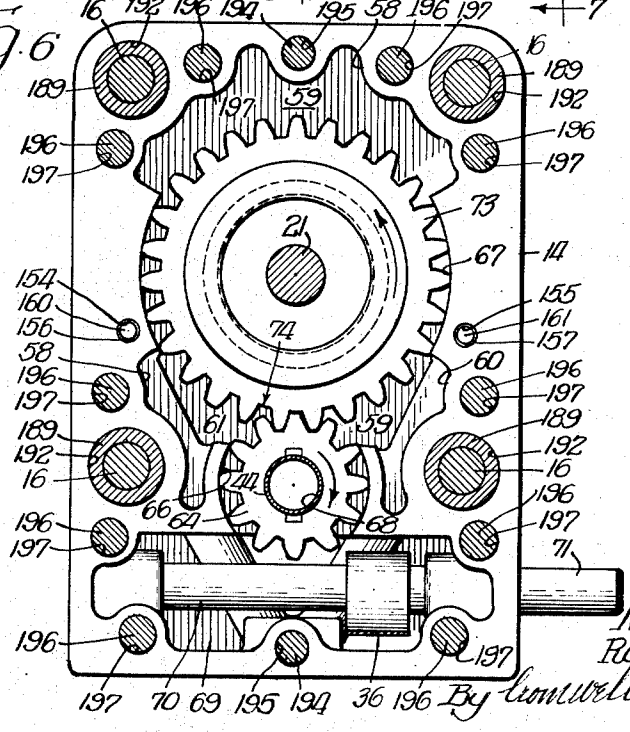


Fig. 6



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cornwell, Quist & Wardun
Attys

Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

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Fig. 8

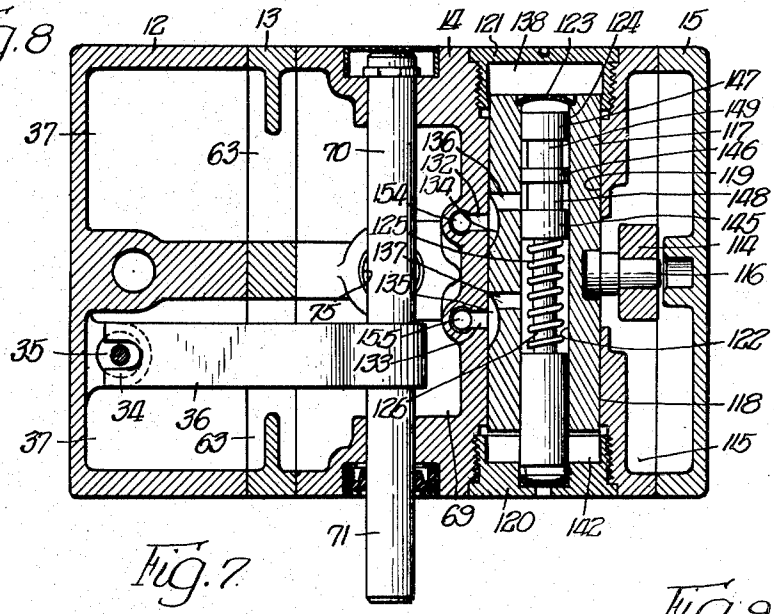


Fig. 7

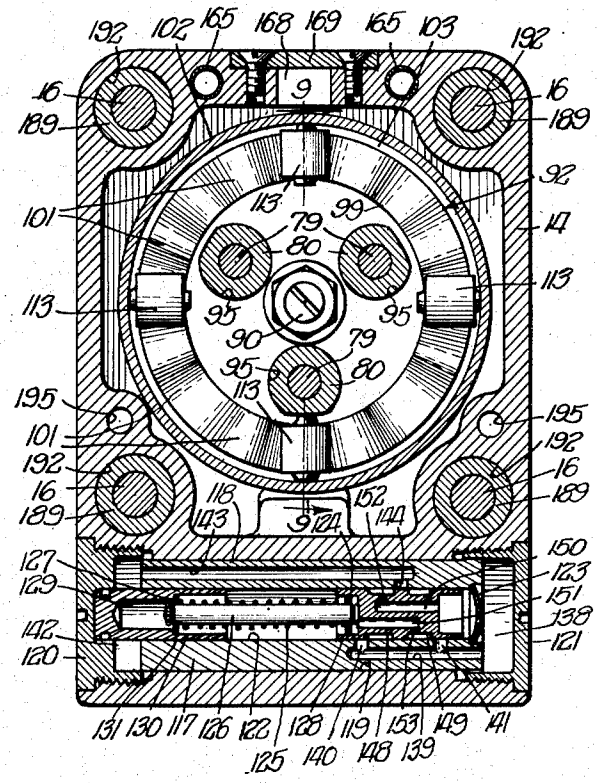
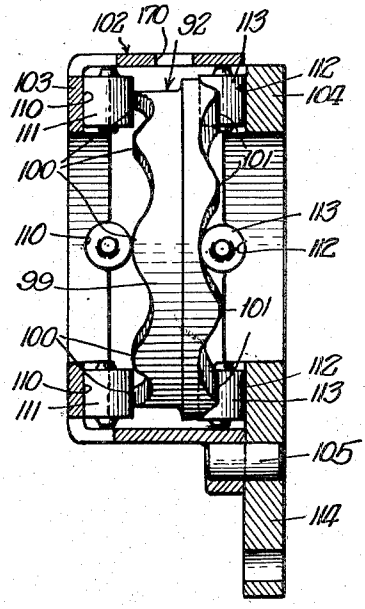


Fig. 9



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Trust + Warden
Attys.

Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

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Fig. 10.

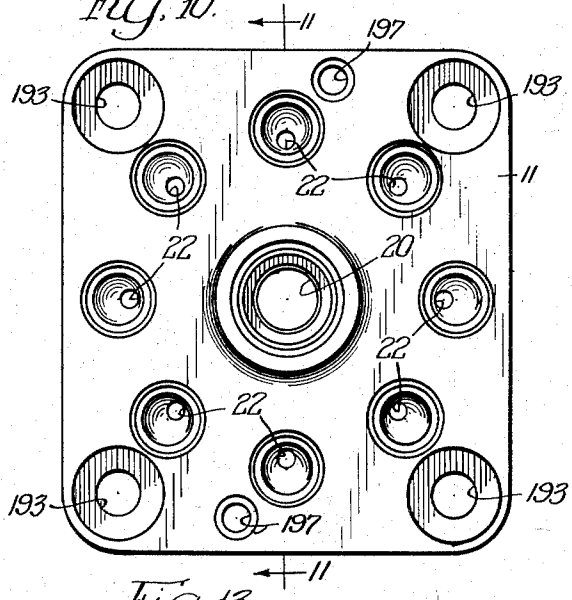


Fig. 11.

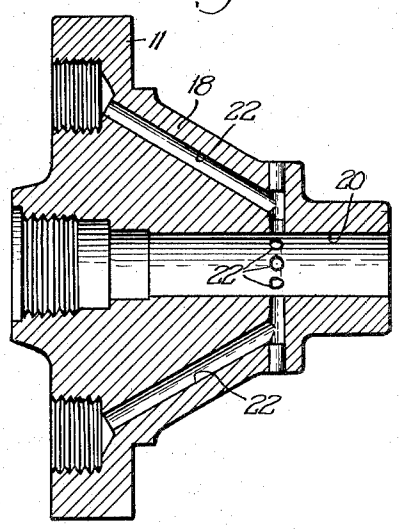


Fig. 13.

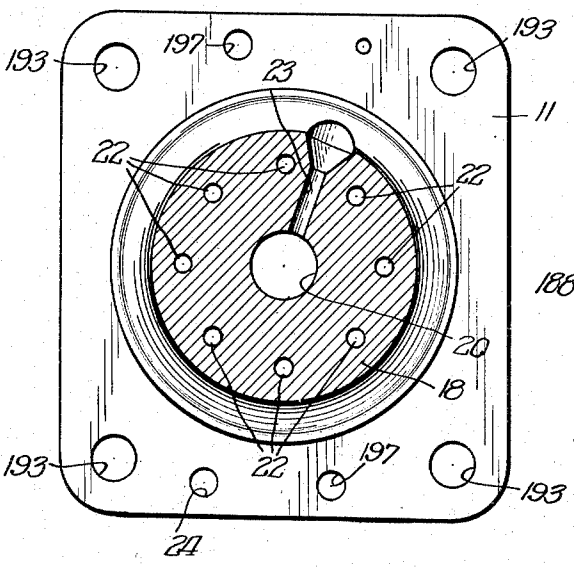
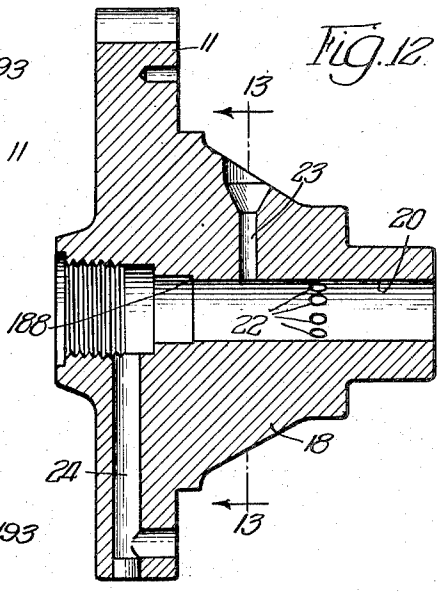


Fig. 12.



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Druce & Wardin
Attys.

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FUEL INJECTION PUMP

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Fig. 14

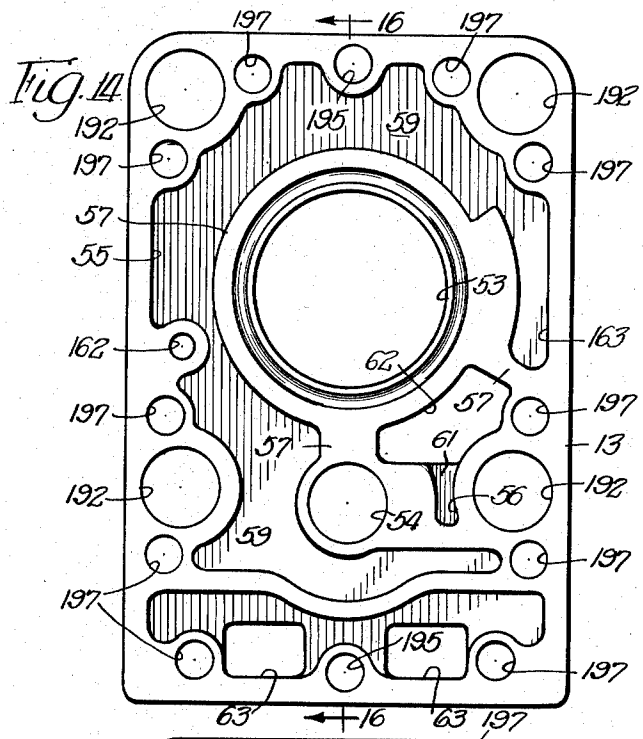


Fig. 15

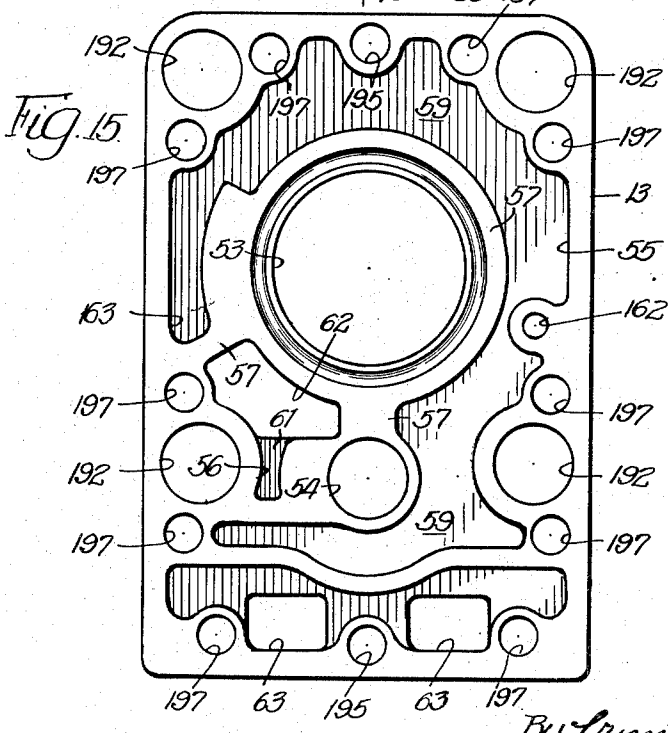
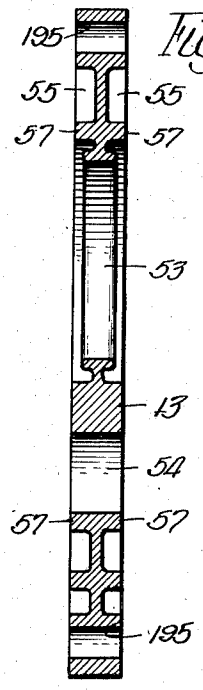


Fig. 16



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Frank & Warden
attys.

Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

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Fig. 17

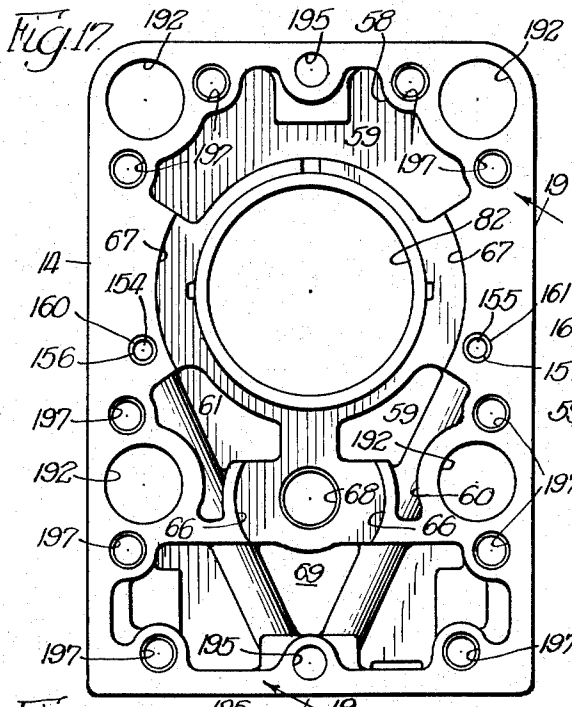


Fig. 18

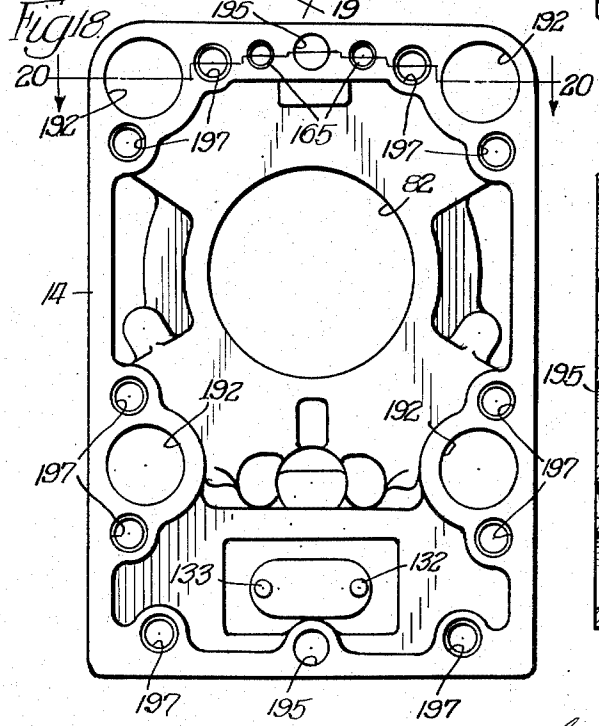


Fig. 19

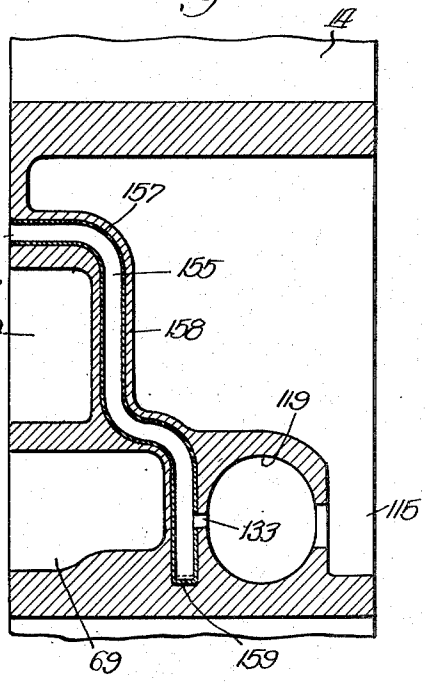
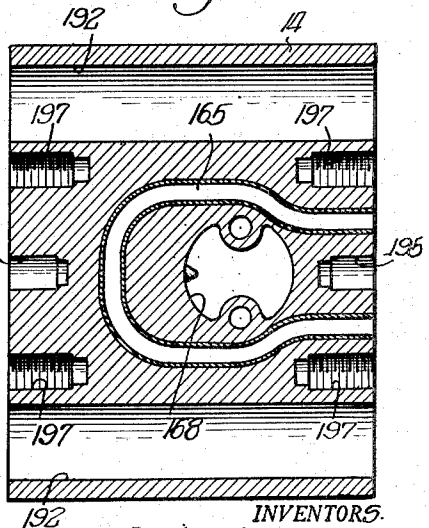


Fig. 20



INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Frost + Warden
attys.

Jan. 18, 1955 N. FODOR ET AL 2,699,766

FUEL INJECTION PUMP

Filed Nov. 21, 1946 7 Sheets-Sheet 7

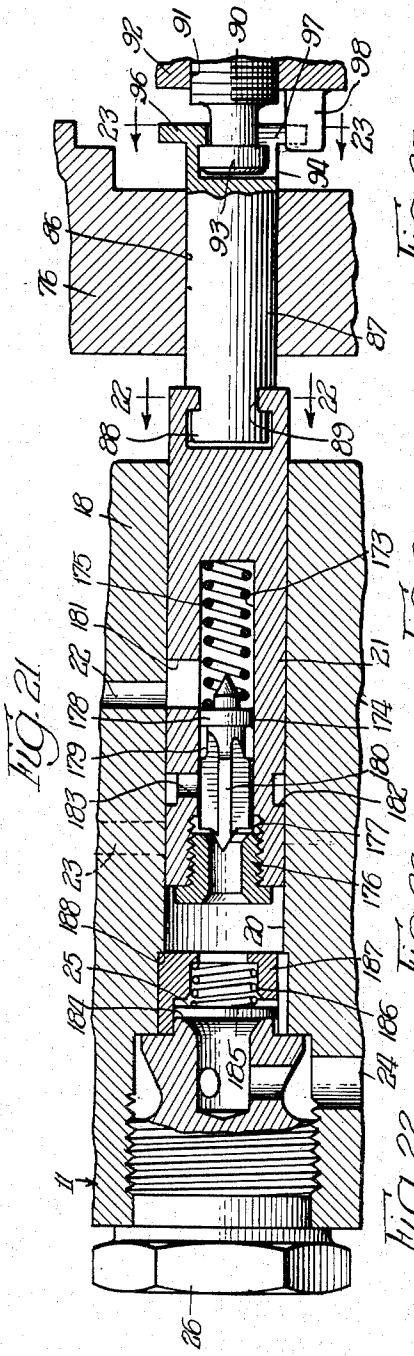


Fig. 21

Fig. 25

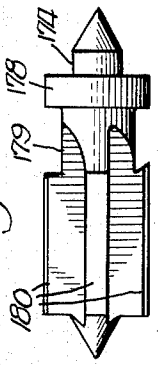


Fig. 24

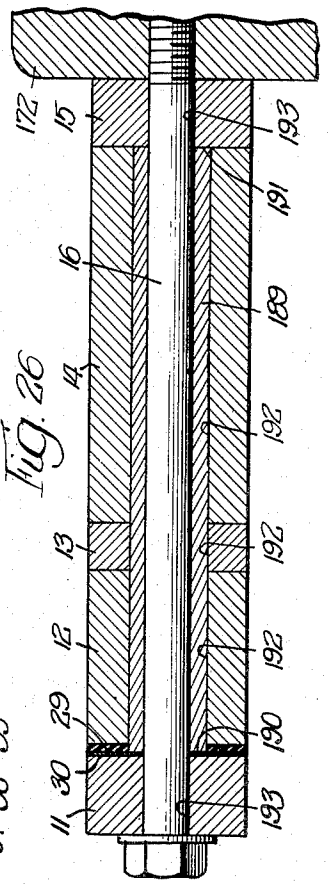


Fig. 26

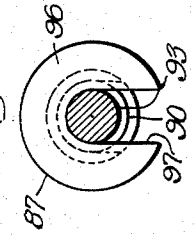


Fig. 23

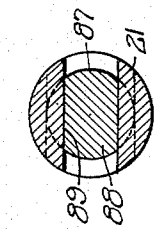


Fig. 22

INVENTORS.
Nicholas Fodor,
Robert S. Johnson,
By Cromwell, Thrust + Warden
Attys

United States Patent Office 2,699,766 Patented Jan. 18, 1955

2,699,766

FUEL INJECTION PUMP

Nicholas Fodor, Wilmette, and Robert S. Johnson, Evanston, Ill., assignors to Micro-Precision Inc., a corporation of Illinois

Application November 21, 1946, Serial No. 711,350

13 Claims. (Cl. 123—140)

The present invention has to do with multiple fuel injection pumps for internal combustion engines.

The purpose of the invention, generally stated, is to provide a multiple fuel injection pump of new and improved construction which will operate efficiently at all speeds, and which is simple, compact and inexpensive to manufacture.

More specifically, one of the objects of the invention is to provide an improved pump of the character described which will deliver fuel effectively in the desired quantities to each of the cylinders in turn of an engine at all operating speeds.

Another object is to provide such a pump in which all of the parts are of simple construction and are compactly arranged.

Another object is to provide such a pump in which certain of the parts can be made very inexpensively of die-cast aluminum without affecting the accuracy and operating efficiency of the pump in any particular.

Another object is to provide such a pump in which the pressure of the fuel supplied to the injection section of the pump by the primary section is automatically varied in accordance with the speed of the pump and in which such pressure is utilized to automatically vary the timing of each injection.

Another object is to provide such a pump in which the quantity of fuel delivered on each injection is controlled by the rate at which the fuel is admitted from the primary section of the pump, without varying the stroke of the injection plunger.

Another object of the invention is to provide such a pump which can be quickly converted to reverse operation by the simple expedient of reversing the position in the pump of certain readily accessible parts.

Still another object is to provide such a pump in which all of the parts can be readily separated from each other for purposes of inspection, cleaning and replacement.

While the foregoing statements are indicative of the nature of the invention other more specific objects and advantages will be apparent to those skilled in the art upon a full understanding of the construction, arrangement and operation of the new pump.

A preferred embodiment of the invention is presented herein for the purpose of exemplification, but it will of course be appreciated that the invention is susceptible of incorporation in other modified forms coming equally within the scope of the appended claims.

In the accompanying drawings:

Fig. 1 is a side view of the new pump;

Fig. 2 is a top view;

Fig. 3 is an end view;

Fig. 4 is an opposite end view;

Fig. 5 is a vertical longitudinal section through the pump, taken on approximately the line 5—5 of Fig. 2;

Fig. 6 is a vertical transverse section, taken on approximately the line 6—6 of Fig. 5;

Fig. 7 is another vertical transverse section, taken on approximately the line 7—7 of Fig. 5;

Fig. 8 is a horizontal section, taken on approximately the line 8—8 of Fig. 5;

Fig. 9 is a partially sectioned side elevation of the cam mechanism for reciprocating the pumping plunger, taken on approximately the line 9—9 of Fig. 7;

Fig. 10 is an end view of the distributor head, which forms one end of the casing of the pump;

Fig. 11 is a vertical section through the distributor head, taken on approximately the line 11—11 of Fig. 10;

Fig. 12 is a section similar to Fig. 11, taken through the distributor head at a slightly different angle;

Fig. 13 is a vertical section, taken on the line 13—13 of Fig. 12;

Fig. 14 is a face view of the primary pump cover plate;

Fig. 15 is an opposite face view of the same plate;

Fig. 16 is a vertical section through the plate, taken on the line 16—16 of Fig. 14;

Fig. 17 is a face view of the drive housing;

Fig. 18 is an opposite face view of the same housing;

Fig. 19 is a section through the drive housing, taken at an inclination on the line 19—19 of Fig. 17;

Fig. 20 is a horizontal section through the drive housing, taken on the line 20—20 of Fig. 18;

Fig. 21 is an enlarged longitudinal section through the pumping plunger and associated parts;

Fig. 22 is a vertical transverse section through the connection between the pumping plunger and the push rod, taken on the line 22—22 of Fig. 21;

Fig. 23 is a vertical transverse section through the connection between the push rod and the adjusting screw, taken on the line 23—23 of Fig. 21;

Fig. 24 is a side view of the valve in the pumping plunger;

Fig. 25 is an end view of that valve; and

Fig. 26 is a section through one corner of the casing of the pump, at the location of one of the tie bolts, taken on the line 26—26 of Fig. 3.

The pump shown in the drawings includes a casing 10 of generally rectangular form, which casing is composed of five separate sections 11, 12, 13, 14 and 15. These sections have complementary meeting faces for fluid-tight engagement with each other and are clamped together adjacent their corners by four tie bolts 16.

The section 11, which forms one end of the casing 10, constitutes what may be termed the delivery head of the pump. This delivery head 11 is provided on its outer face with a circumferentially arranged series of delivery unions 17 through which the fuel is pumped into suitable conduits (not shown) leading to the cylinders of the engine with which the pump is connected. Eight unions are shown, for connection with each of the cylinders in an eight-cylinder engine, but it will of course be appreciated that any lesser or greater number of unions may instead be employed, depending on the number of cylinders in the engine.

The distributor head 11 is provided with a portion 18, preferably of generally conical form, which projects into a fuel chamber 19 which is formed in the adjoining section 12 of the casing. The conical portion 18 contains an axially extending cylindrical bore 20 in which a pumping plunger 21 is mounted for both rotation and reciprocation. The portion 18 is provided about the bore 20 with a plurality of fuel delivery passages 22, which passages lead radially from the bore 20 at circumferentially spaced points to the delivery unions 17. The portion 18 is also provided with a spill port 23 which leads from the bore 20 back into the fuel chamber 19.

The plunger 21 pumps equal quantities of fuel under high pressure from the bore 20 into each one of the delivery passages 22 in turn. The fuel which is pumped enters the bore 20 through an inlet passage 24, which passage receives the fuel from the lower portion of the chamber 19 and conducts it upwardly into the bore 20 past an inlet valve 25 in the bore. The end of the bore 20 at the location of the inlet passage 24 is enlarged and is closed beyond the inlet passage by an exteriorly accessible screw plug 26.

The second section 12 of the casing 10 constitutes what may be termed the control housing. This control housing 12 contains the fuel chamber 19. The housing 12 is provided with a port 27 through which the fuel from the chamber 19 enters the inlet passage 24 leading to the bore 20. The port 27 is connected with the passage 24 by a sleeve 28 which bridges the joint between the meeting surfaces of the casing sections 11 and 12, which joint is sealed by a compressible head gasket 29 backed by a thin metal shim 30.

Before entering the port 27 the fuel from the chamber 19 enters the lower end of an open ended bore 31 in which a plunger-type metering valve 32 is reciprocally mounted. The upper end of the valve 32 is exposed to the pressure of the fuel in the chamber 19 through the upper open end of the bore 31. The lower end of the valve 32 is adapted to vary the effective size of the port 27 and is of frusto-conical formation. A second bore 33 of the same cross-sectional area is located below the bore 31 in spaced concentric relation to the latter. The bore 33 extends downwardly through the bottom of the control housing 12. The bore 33 contains a piston 34 which is integrally connected with the valve 32 by a stem 35 of reduced cross section. The lower end of the piston 34 is exposed to what amounts to atmospheric pressure, resulting in a pressure differential on the valve 32 which normally tends to close off the port 27 under the pressure of the fuel in the chamber 19. The valve 32 is controlled through a governor leaf spring 36. The spring 36 is bifurcated at its front end and projects toward the piston 34 through a recess 37 in the lower wall of the control housing 12. The recess 37 intercepts the lower bore 33 in which the piston 34 is mounted, and the piston is provided intermediate its ends with an annular groove 38 in which the bifurcated end of the spring 36 is hooked.

A centrifugal governor 39 is provided within the fuel chamber 19 for controlling the pressure of the fuel which is pumped into that chamber by the hereinafter described primary pump. The governor 39 includes a rotatable cage 40 which is mounted on the front end of a hollow shaft 41. The shaft 41 extends rearwardly through an opening in the third section 13 of the casing, with a certain amount of clearance at 42, the shaft being supported for rotation as hereinafter described. The cage 40 contains a cross bore 43 which is open at both ends, and the shaft 41 contains a sleeve 44 which communicates at its front end with the cross bore 43. The sleeve 44 is stationary and the shaft 41 is rotatably mounted on the sleeve. A piston-like weight 45 is reciprocably mounted in one end of the cross bore 43 and is connected by a rod 46 with a disk valve 47 of slightly smaller cross-sectional area than the weight 45. The valve 47 coacts with a reduced seat 48 in the cross bore 43 near the other end of the latter. The valve 47 is urged toward its seat by a coil spring 49, which spring is compressed between the underside of the seat 48 and the opposite face of the weight 45.

The centrifugal governor 39 controls the pressure of the fuel in the chamber by permitting a portion of the fuel in that chamber to pass back through the bore 40 and sleeve 44 into a fuel intake chamber 50 from which the hereinafter described primary pump is fed. The pressure of the fuel in the chamber 19 tends to increase the opening of the valve 47, due to the pressure differential resulting from the difference in the cross-sectional areas of the valve 47 and weight 45, while the rotation of the cage 40 which contains the valve and the weight tends to decrease the opening of the valve 47, due to the pull of the weight 45 under centrifugal action. These opposing forces are such as to result in the pressure in the fuel chamber 19 being at every speed of the pump an index of that particular speed.

The fuel chamber 19 is intended in operation to be completely filled with fuel. To prevent an accumulation of air or fuel vapor therein a vent 51 controlled by a small non-return valve 52 is located in the upper wall of the housing 12 and is connected by a suitable conduit (not shown) with the fuel supply. The vent 51 serves as a bleeder for the fuel chamber 19, without appreciably affecting the pressure on the fuel therein.

The third section 13 of the casing 10 constitutes what may be termed the primary pump cover plate. This plate 13 is reversible in position between the casing sections 12 and 14, whereby to adapt the pump to either right-hand or left-hand operation.

The plate 13 contains a large circular opening 53 through which the tip of the conical portion 18 of the distributor head 11 projects. It also contains a smaller circular opening 54 through which the shaft 41 of the governor 39 extends. These openings are located on the vertical center line of the plate and assume the same position in the pump irrespective of whether the plate is reversed.

Both faces of the plate 13 are provided with shallow

cavities 55 and 56. These cavities are located adjacent the ends of the openings 53 and 54 but are spaced from such openings and from each other by intervening flush portions 57 of the faces of the plate. The cavities 55 are arranged back-to-back and are relatively large, extending tortuously about both of the openings 53 and 54. The cavities 56 are likewise arranged back-to-back but are relatively small, being localized below the opening 53 to one side of the opening 54.

The cavity 55 in that face of the plate 13 which is positioned against the fourth section 14 of the casing complements a cavity 58 in the confronting face of the fourth section to form a chamber 59, which chamber constitutes the low pressure chamber of the hereinafter described primary pump, while the cavity 56 in the same face of the plate 13 complements a confronting cavity 60 in the fourth section 14 to form a chamber 61, which chamber constitutes the high pressure chamber of the primary pump.

The fuel which is pumped by the primary pump into the chamber 61 passes from that chamber into the fuel chamber 19 through an opening 62 in the plate 13 (see Figs. 14 and 15), which opening is located between the backs of the cavities 56 in the plate 13.

Beneath the circular opening 54 the plate 13 is provided with two laterally offset openings 63, one of which openings will register, irrespective of the position of the plate 13, with the recess 37 in the lower wall of the control housing 12 to provide a passageway for the governor spring 36.

The stationary sleeve 44, which rotatably supports the hollow drive shaft 41 for the governor 39, projects beyond the plate 13 and rotatably supports a gear 64, which gear constitutes the driven gear of the primary pump and also the driving means for the governor 39. The gear 64 is positioned flush against the adjacent face of the plate 13 and is keyed to the shaft 41 by projections 65 on the end of the latter, which projections extend into slots in the gear.

The fourth section 14 of the casing 10 is the largest section and constitutes what may be termed the drive housing. The face of the drive housing 14 which is positioned against the plate 13 contains, in addition to the low and high pressure cavities 58 and 60, two circular recesses 66 and 67 (see Fig. 17).

The recess 66 is of the same size as the gear 64 and affords a closely conforming pocket in which that gear is positioned for rotation. One part of the recess 66 is in communication with the low pressure chamber 59 of the primary pump while another part of the same recess is in communication with the high pressure chamber 61 of the primary pump.

Centrally of the recess 66 the housing 14 is provided with a relatively small circular opening 68 in which the sleeve 44 which extends through and rotatably supports both the shaft 41 and gear 64 is fixedly secured. This opening 68 communicates downwardly beyond the sleeve 41 with a chamber 69 in the bottom of the housing 14. The chamber 69 constitutes the fluid inlet chamber of the primary pump. The chamber 69 registers with the recesses 63 in the plate 13 and provides an extension of the passageway for the governor spring 36. The spring 36 is coiled a part turn and attached, within the chamber 69, to a rock shaft 70, which shaft is journaled at its ends in the sides of the housing 14. One end 71 of the shaft extends outside the housing and is connected with an operating lever 72.

The recess 67 in the housing 14 contains and closely conforms to a relatively large gear 73, the teeth of which mesh with the teeth of the gear 64 to form the primary pump, hereinafter designated generally as 74. Like the recess 66, one part of the recess 67 is in communication with the low pressure chamber 59 while another part of the recess 67 is in communication with the high pressure chamber 61.

The primary pump 74 pumps the fuel from the fuel inlet chamber 69, where the fuel may be under substantially atmospheric pressure, into the high pressure fuel chamber 19, from which latter chamber it flows through the passage 24 into the bore 20, and is there pumped by the plunger 21 through the passages 22 to the delivery unions 17. The teeth of the gear 64 of the primary pump pick up the fuel in the fuel inlet chamber 69, in which chamber the teeth are exposed, and carry it upwardly through one side of the recess

66 into the high pressure chamber 61, where it is forced under pressure into the chamber 19. The fuel inlet chamber 69 is provided with an inlet port 75 which is adapted to be connected with a conduit (not shown) leading from the fuel supply.

The gear 73 is fixedly secured to a rotatable member 76 which forms the primary pump drive part of a two-part drive shaft 77. The other part of the two-part drive shaft is a rotatable member 78 which is fixedly connected with the member 76 in axially spaced relation to the latter by three longitudinally extending machine screws 79 which extend through three circumferentially spaced tubular projections 80 formed on the member 78.

The member 76 is journaled in a bearing 81 which is mounted in a circular opening 82 in the housing 14, while the member 78 is journaled in an axially spaced bearing 83 which is mounted in a circular opening 84 in the fifth and last section 15 of the casing 10, which section forms the end of the casing and may be termed the end plate. The member 78 of the two-part drive shaft extends beyond the end plate 15, where it is provided about its circumference with a plurality of tapped openings 85 to facilitate its connection with a rotating part (not shown) of the associated engine.

The member 76 is provided with an axially extending bore 86, which bore is disposed in alignment with the bore 20 in the distributor head 11. A push rod 87 is reciprocably mounted in the bore 86 of the member 76 and is non-rotatably connected with the pumping plunger 21 in the bore 20. The non-rotatable connection between the push rod 87 and the plunger 21 is a laterally separable one which consists of an undercut and flattened head 88 (see Fig. 21) on one end of the push rod which engages within an undercut and correspondingly flattened cross slot 89 in the adjacent end of the plunger.

The push rod 87 is in turn rotatably connected to an adjusting screw 90, which screw is mounted in a threaded bore 91 located in the center of a circular cam member 92. The rotatable connection between the push rod 87 and the screw 90 is a laterally separable one which consists of an undercut round head 93 (see Fig. 21) on one end of the screw which fits within an undercut slot 94 in the adjacent end of the push rod.

The cam member 92 is reciprocably mounted in the space between the two drive shaft members 76 and 78 and is provided with three circumferentially spaced guide bores 95 through which the tubular projections 80 pass. The projections permit the cam member 92 to reciprocate freely while preventing it from turning relative to the members 76 and 78. The push rod 87, and in turn the pumping plunger 21, is prevented from turning relative to the cam member 92 by the provision of a flange 96 on the slotted end of the push rod, which flange is provided with a radial slot 97 which embraces and interlocks circumferentially with a lug 98 on an adjacent portion of the cam member.

The cam member 92 is of disk-like form and is provided with an undulating rim 99. The undulations provide eight uniformly spaced cam lobes 100 on the front face of the rim and an equal number of similarly arranged cam lobes 101 on the rear face of the rim (see Fig. 9), with the lobes on the rear face alternating in position with respect to the lobes on the front face.

The rim 99 of the cam member is enclosed within a normally stationary but circumferentially adjustable roller cage 102, which cage is composed of a centrally apertured cup 103 and a centrally apertured disk 104. The cup 103 is non-rotatably connected with the disk 104 by an eccentrically disposed dowel pin 105. The cup 103 is journaled on the rear end of the drive shaft member 76 at 106, in axial abutment with an annular shoulder 107 on that member, while the disk 104 is journaled on the front end of the drive shaft member 78 at 108, in axial abutment with an annular surface 109 on the end plate 15.

The cup 103 of the roller cage is provided in the bottom of the same with circumferentially spaced pockets 110 in which cylindrical cam rollers 111 are rotatably mounted, while the disk 104 is provided with oppositely disposed pockets 112 in which other cylindrical cam rollers 113 are rotatably mounted. These two sets of rollers 111 and 113 coact with the cam lobes 100 and 101 on the rim of the cam member 92 to cause the latter to reciprocate back and forth eight times during each complete revolution of the drive shaft. Each set is shown as consisting of four rollers, but this number may be increased or decreased as desired without affecting the number of reciprocating movements imparted to the cam member.

The disk 104 of the roller cage is provided with an integral arm 114 which extends downwardly into a recess 115 in the lower portion of the housing 14. By oscillating the arm 114 in either direction the attitude of the roller cage 102 will be correspondingly varied, causing the time of injection to be either advanced or retarded. The lower end of the arm 114 is connected by a pin 116 to a piston 117 which forms part of a hydraulically operated timing device 118.

The piston 117 is reciprocally mounted in a horizontal transversely extending cylinder 119 which is formed in the lower portion of the housing 14. The cylinder 119 is of somewhat greater length than the piston 117 and is closed at its ends by exteriorly accessible screw plugs 120 and 121 (see Figs. 7 and 8).

The piston 117, which is movable axially within the cylinder 119 between the closure plugs 120 and 121, contains an axial bore 122 which is closed at one end by a sealing plug 123. A plunger type valve 124 is reciprocally mounted in the bore 122 adjacent the plug 123 and is urged in the direction of that plug by a coil spring 125. The spring 125 surrounds a link 126 and is compressed between washers 127 and 128 on the ends of the link. The washers 127 and 128 are mounted on the ends of the link 126 by snap rings seated in suitable grooves. One end of the link 126 abuts the valve 124, with the washer 127 positioned in a recess in the surrounding end of the valve, while the other end of the link 126 projects freely into a recess 129 in a stationary stud 130, with the washer 128 abutting a shoulder 131 within that recess (see Fig. 7). The stud 130, which provides a compression seat for the spring 125 and allows the contained end of the link 126 to reciprocate within the same, is secured to the closure plug 120 and projects therefrom into the adjacent end of the bore 122 in the piston 117, in freely slidable engagement with the bore.

The timing device 118 is hydraulically operated by the pressure of the fuel on the high pressure side of the primary pump 74, which pressure, as previously stated, is caused by the action of the centrifugal governor 39 to be an index of the speed of the pump at all speeds within the normal operating range of the latter.

The cylinder 119 in which the piston 117 is mounted is provided intermediate its ends with two axially spaced ports 132 and 133 (see Fig. 8), which ports are arranged equidistant from the vertical center line of the pump. One of these ports is connected with the high pressure side of the primary pump, in the manner hereinafter described, while the other is connected with the low pressure or inlet side of the primary pump, in the manner hereinafter described. Which of the two ports 132 and 133 is connected with the high pressure side and which of the same is connected with the low pressure side depends on the position in which the reversible plate 13 is placed in the casing assembly. In the particular placement shown, the port 132 is connected with the high pressure side while the port 133 is connected with the low pressure side.

The piston 117 is provided opposite the ports 132 and 133 with relief portions 134 and 135 which connect respectively with radial passages 136 and 137. The passages 136 and 137 in turn connect with the bore 122 in the piston at longitudinally spaced points. The passage 136, which is the high pressure passage, connects with the bore 122 at a point intermediate the ends of the valve 124, while the passage 137, which is the low pressure passage, connects with the bore 122 at a point in the bore in advance of the valve 124.

The space 138 which is present in the cylinder 119 between the closure plug 121 and the rear end of the piston 117 serves as a high pressure chamber and is connected with the bore 122 in the piston by a passage 139 which opens into the bore at two oppositely spaced ports 140 and 141. The opposite space 142 in the cylinder 119 serves as a low pressure chamber and is connected with the bore 122 in the piston by a passage 143 which opens into the bore at a port 144, which port is located intermediate the ports 140 and 141.

The valve 124 is provided on its outer surface with three annular lands 145, 146 and 147, which lands are separated by two annular grooves 148 and 149. The valve

124 is provided interiorly of the same with two longitudinally extending passages 150 and 151. The passage 150 opens into the bore 122 at the rear face of the valve and is connected by a duct 152 with the annular groove 148 in the outer surface of the valve, while the passage 151 opens into the bore 122 in advance of the valve and is connected by a duct 153 with the other annular groove 149 in the outer surface of the valve.

Fuel under the pressure of the high pressure side of the primary pump 74 enters the timing device 118 through the port 132 and flows through the passage 136 in the piston 124 into the annular groove 148 in the valve 124. From the groove 148 this fuel flows through the duct 152 and the passage 150 into the bore 122 in the piston 117 at the rear of the valve 124. The pressure on this fuel acts against the valve 124 to advance the latter against the resistance offered by the spring 125, thereby causing the land 145 on the piston to expose the passage 139 at 140 and allow the fuel to flow into the space 138 behind the piston. The valve 124 will come to rest as soon as the resistance offered by the spring 125 increases to a point where it counterbalances the pressure exerted on the valve by the fuel. The pressure on the fuel in the space 138 will at the same time advance the piston 117 in the cylinder until the piston reaches a position wherein the land 145 on the then stationary valve 124 closes off further flow of fuel into the space 138, at which time the piston 117 will come to rest. This advancement of the piston acts through the arm 114 to correspondingly advance the timing of the fuel injection. As the fuel enters the space 138 in the cylinder 119 at the rear end of the piston 117 a corresponding amount of fuel will be evacuated from the space 142, through the passage 143, groove 149, duct 153, passage 151, bore 122, passage 137 and port 133, back to the low pressure or inlet side of the primary pump 74. The resistance offered by the spring 125 to the movement of the valve 124 determines the extent to which the valve will permit the piston 117 to advance, which can be varied as desired by substituting springs having different resistances to compression.

The two pressure differential ports 132 and 133 are connected respectively with tortuous passages 154 and 155, which passages may be formed by casting long small diameter tubes 156 and 157 (see Fig. 19) within the interior walls 158 of the housing 14 during the formation of the latter. The tubes 156 and 157 extend upwardly, outwardly and forwardly from the location of the ports 132 and 133, and may be formed initially from a single generally V-shaped length of tube by pinching or otherwise closing off the apex of the tube at 159 intermediate the two ports 132 and 133. The passages 154 and 155 terminate in open ends 160 and 161 which are disposed flush with the flat front face of the housing 14.

The plate 13 which fits against the front face of the housing 14 is provided with a hole 162 (see Figs. 14 and 15) which registers with the open end 160 of the high pressure passage 154 in the housing 14. This hole, which is located adjacent one side of the plate 13, extends from one face of the plate to the other and places the passage 154 in direct communication with the fuel chamber 19 in the housing 12, in which chamber the fuel is maintained at the pressure of the high pressure side of the primary pump 74. The plate 13 has no corresponding hole opposite the open end 161 of the other passage 155 in the housing 14 but is instead provided at that location with a branch 163 of the cavity 55 which forms a part of the low pressure chamber 59, with the result that the passage 155 is placed in direct communication with the low pressure or inlet side of the primary pump 74. It will be understood that the position in which the plate 13 is placed determines which of the cylinder ports 132 and 133 is the high pressure port and which the low pressure port, and that such ports can be reversed by merely reversing the plate 13. This reversal is only made when the direction of rotation of the pump is changed, at which time the timing device 118 will also be reversed end for end in the cylinder 119.

The bearings 81 and 83 for the two axially spaced drive shaft members 76 and 78 are supplied with lubricating oil from an inlet 164 in the top of the housing 14, which inlet is adapted to be connected with a conduit (not shown) leading to a supply of lubricating oil. The oil may be conveyed through a tube 165 (see Fig. 20) cast within the upper wall of the housing 14. The tube 165 may be of general U-shaped form and may be connected at its center with the inlet 164. The two branches of the tube 165 may terminate flush with the rear face of the housing 14 and there register with ducts 166 in the opposed face of the end plate 15, which ducts lead downwardly to the bearing 83 in that plate. Other ducts 167 lead downwardly in the housing 14 from the tube 165 to the front bearing 81. The oil which lubricates the bearings 81 and 83 works into the space between the drive shaft members 76 and 78 in which the cam rollers 111 and 113 operate and there lubricates those rollers.

The housing 14 is provided in the upper wall thereof with an opening 168 which is normally covered by a removable plate 169. The opening 168, which is centered between the two branches of the lubricating tube 165, permits observation of the timing device 118 in checking the position of the latter, the cam roller cage 102 being preferably provided in the top of the same with a position-indicating slot 170 for this purpose.

The end plate 15, which completes the casing assembly, may be provided about the bearing opening 84 with a series of circumferentially arranged vents 171, which vents permit the lubricating oil in the housing 14 to flow into the crank case of the associated engine. The end plate 15 forms the mounting base of the casing 10 and is adapted to be bolted solidly against a stationary part 172 (see Fig. 26) of the associated engine by the previously described tie bolts 16.

Reverting to the injection pump in the distributor head 11—the pumping plunger 21, which both rotates and reciprocates, is provided with a longitudinally extending bore 173 (see Fig. 21) in which a plunger-type valve 174 is slidably mounted. A coil spring 175 is positioned under compression in the bore 173 between the rear end of the valve and the rear end of the bore. The valve 174 is retained in the bore 173 by a bushing 176 which is threaded into the open front end of the bore and is provided with a rim 177 against which the front end of the valve abuts.

The valve 174 is provided at its rear end with an annular land 178, and is provided in front of the land with an annular groove 179. The remainder of the valve 174 is of cruciform cross section and is provided with a plurality of longitudinally extending guide ribs 180, which ribs permit fuel entering the bore 173 through the bushing 176 to flow rearwardly about the valve 174 as far as the land 178.

The plunger 21 is provided with a single delivery port 181 which extends radially through the wall of the same from the bore 173 and is adapted during each revolution of the plunger to register separately with each of the delivery passages 22 in turn. The plunger 21 is also provided in its outer surface with an annular groove 182, which groove is connected with the bore 173 by a plurality of radial ducts 183. The groove 182 and ducts 183 are located near the front end of the plunger 21 in axially spaced relation to the delivery port 181 in the plunger. The groove 182 is adapted to register with the spill port 23 during each advance stroke of the plunger 21, which advance stroke occurs eight times during each revolution of the plunger.

The injection of the fuel through the particular delivery passage 22 with which the delivery port 181 is in register is abruptly discontinued during each advance stroke of the plunger as soon as the front edge of the groove 182 in the plunger reaches the rear edge of the spill port 23, since the fuel can then by-pass back into the fuel chamber 19. The quantity of fuel which is injected on each advance stroke of the plunger has previously entered the bore 20 past the valve 25 during the preceding return stroke of the plunger and has become trapped in the bore 20 in front of the advancing piston by the closing action of the valve 25.

The valve 25 includes an annular seat 184 on the rear edge of the rim of the screw plug 26 and a disk 185 which coacts with the seat 184 and is pressed against the same by a coil spring 186. The spring 186 is housed within a centrally apertured cage 187 which is positioned against a shoulder 188 in the front portion of the bore 20.

During each advance stroke of the plunger 21 the valve 174 in the plunger will be moved rearwardly by the trapped fuel to a point where the land 178 will uncover the delivery port 181 in the wall of the plunger, and will be retained in open position until the pressure on the fuel being discharged has been relieved by the action of the groove 182 in uncovering the rear edge of the spill

port 23, whereupon the valve 174 will move forwardly in the still advancing plunger and tend to draw back into the bore 173 a certain amount of fuel from the particular delivery passage 22 with which the delivery port 181 is then still in register, thus further increasing the abruptness of the injection cut-off by a momentary suction-like action.

In manufacturing the casing 10 of the pump the front end section—namely, the distributor head 11—may advantageously be made of steel, and the remaining sections—namely, the control housing 12, plate 13, drive housing 14 and end plate 15—made of die-cast aluminum or other inexpensive non-ferrous material, thereby materially reducing the manufacturing cost of the pump over what that cost would be if all of such sections were made of steel.

Because of the marked difference in the coefficient of expansion between steel and, say, aluminum, this presents a problem, as the slightest expansion or contraction in the over-all length of the casing under varying heat conditions would adversely affect the accuracy and timing of the pumping action of the plunger 21 through alteration of the longitudinal position of the plunger relative to the delivery passages 22 and spill port 23 in the bore 20 at any particular point in the injection cycle.

To overcome this difficulty and yet make the casing 10 for the most part of aluminum or other non-ferrous material the component casing sections are all bolted together in a pack by the tie bolts 16, which bolts, like the front end section 11, are made of steel and therefore expand and contract longitudinally at the same rate and to the same extent as the injection pumping parts—namely, the drive shaft members 76 and 78, cam member 92, cage 102, rollers 111 and 113, adjusting screw 90, push rod 87 and plunger 21—all of which parts are likewise made of steel.

The gap that would otherwise develop in the casing between the steel and non-ferrous sections is occupied by the gasket 29, which gasket, being of thick resiliently compressible material will maintain all the sections in fluid-tight engagement with each other at all times irrespective of their different rates of expansion.

The tie bolts 16 are positioned within steel tubes 189 (see Fig. 26), which tubes act as spacing struts and bear at their ends against the inner faces 190 and 191 respectively of the front and rear end sections 11 and 15 of the casing. The tubes 189 cooperate with the tie bolts 16 to hold the steel end section 11 in proper position relative to the rotating and reciprocating plunger 21, irrespective of any expansion or contraction, and also serve as positioning guides for the gasket 29, the shim 30 and the other sections of the casing. The tubes 189 extend through bores 192 in the casing sections 12, 13 and 14, while the bolts 16 extend beyond the ends of the tubes 189 into smaller bores 193 in the casing sections 11 and 15. The inner face 191 of the end plate 15, against which the tubes 189 abut, constitutes the base plane from which expansion and contraction is reflected, by reason of the axial abutment thereagainst of the cam roller cage 102 at 109.

The casing sections 10 to 15, inclusive, in addition to being held together by the bolts 16 and tubes 189, are provided with a plurality of positioning dowels 194 which extend between indexing apertures 195 in the adjoining sections. Certain of the sections are also provided with fastening screws 196 which engage within apertures 197 in adjoining sections.

Reviewing briefly the operation of the pump—the fuel enters through the inlet port 75 and is picked up in the inlet chamber 69 by the teeth of the lower gear 64 of the primary pump (see Fig. 6) and carried by the teeth through one of the sides of the recess 66 into the chamber 61. From the chamber 61 the fuel is forced under pressure by the meshing of the teeth of the lower gear 64 with the teeth of the upper gear 73 through the opening 62 (see Fig. 15) in the plate 13 into the main fuel chamber 19. From the chamber 19 the fuel flows upwardly through the passage 24 past the check valve 25 into the bore 20, where it is caused by the action of the piston 21 to be discharged in unit quantities through each of the delivery passages 22 in turn.

The rate of flow of the fuel through the passage 24 determines the amount of fuel discharged in each injection, and this rate of flow is adjustably controlled by the setting of the metering valve 32 with respect to the port 27, which valve is moved toward its open or closed position by means of the exteriorly accessible lever 72 and governor spring 36 to vary as desired the amount of fuel discharged upon each injection.

The pressure of the fuel in the chamber 19 tends to move the valve 32 toward its closed position, but this tendency is offset by pressure exerted on the valve by the governor spring 36, which spring in addition to affording such pressure acts as the lever arm between the valve 32 and the shaft 70 on which the control lever 72 is mounted. The governor spring 36 is set in any desired position by the control lever 72.

In the idling speed position of the valve 32 the lever 72 is so positioned by a set screw 198 and a stop 199 as to flex the governor spring 36 slightly and thus place a certain amount of upward pressure on the free end of the spring, which pressure is transmitted to the valve 32 and balances the latter against the pressure of the fuel in the chamber 19. This counterbalancing relationship at idling speed tends to eliminate undesirable speed variations or hunting in the associated engine at that speed, since any momentary faltering of the engine which might tend to stop the same is immediately reflected in the pump by a reduction of the pressure on the fuel in the chamber 19, which reduction permits the pressure in the flexed governor spring 36 to open automatically the valve 32 far enough to bring about a smooth resumption of the desired idling speed.

The pressure of the fuel in the chamber 19 at every speed of the pump within the normal operating range is automatically maintained as a dependable index of that particular speed by the centrifugal governor 39, which governor by permitting variable quantities of the fuel in the chamber 19 to by-pass back into the inlet chamber 69 maintains a predetermined pressure in the chamber 19 for every operating speed.

The timing of the fuel injection is in turn automatically varied in accordance with the speed of the pump by the pressure on the fuel in the chamber 19 through the timing device 118. As the pressure on the fuel in that chamber increases, such pressure is transmitted to the valve 124 (see Fig. 7), against the rear end of the latter, and the valve 124 is advanced in the piston 117 until the pressure on the fuel is counterbalanced by the resistance of the spring 125. This movement of the valve 124 permits the fuel to continue under the same pressure into the space 138 behind the piston 117, where it acts upon the rear end of the piston and advances the latter. After the piston 117 has advanced with respect to the then stationary valve 124 a certain distance further flow of the fuel into the space 138 will be blocked off by the lands on the outer surface of the valve 124 and the piston 117 will come to rest in its advanced position until such time as the pressure on the fuel in the chamber 19 either increases or decreases, whereupon the valve 124 will be correspondingly moved and the piston 117 will assume a new position. As the piston 117 advances or recedes, the timer arm 114 with which it is connected will correspondingly alter the circumferential attitude of the roller cage 102, causing the timing of each injection to be advanced or retarded in accordance with the speed of the pump.

We claim:

1. In a multiple fuel injection pump for internal combustion engines, a distributor head having a centrally arranged bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a spill port leading therefrom in axially spaced relation to the delivery passages, a plunger mounted in the bore, means for rotating the plunger and simultaneously reciprocating the same with a constant stroke, said means being adapted to be operatively connected with a moving part of an associated engine for causing the plunger to rotate and reciprocate in accordance with the speed of the engine, said plunger being provided with a discharge port for directing fuel from the bore into each of said delivery passages in turn, a fuel inlet passage leading into the bore, a fuel pressure chamber surrounding the head in communication with the spill port for supplying fuel to the passage, means operatively connected with said plunger-operating means for supplying fuel under pressure to the chamber, a metering valve for controlling the entry of the fuel into the inlet passage, and means in the chamber governed by

the speed of the pump for causing said pressure to vary in accordance with such speed.

2. In a multiple fuel injection pump for internal combustion engines, a distributor head having a centrally arranged bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a spill port leading therefrom in axially spaced relation to the delivery passages, a plunger mounted in the bore, means for rotating the plunger and simultaneously reciprocating the same with a constant stroke, said means being adapted to be operatively connected with a moving part of an associated engine for causing the plunger to rotate and reciprocate in accordance with the speed of the engine, said plunger being provided with a discharge port for directing fuel from the bore into each of said delivery passages in turn, a fuel inlet passage leading into the bore, a fuel pressure chamber for supplying fuel to the passage, means operatively connected with said plunger-operating means for supplying fuel under pressure to the chamber, a check valve adjacent the discharge end of the inlet passage for permitting the fuel to flow from the inlet passage into the bore while preventing reverse flow of the fuel back through the inlet passage, a metering valve in the chamber for controlling the rate of flow of the fuel from the chamber into the inlet passage, means governed by the speed of the pump for causing said pressure to vary in accordance with such speed, means actuated by an increase or decrease of the pressure in said chamber for advancing or retarding the timing of the reciprocation of the plunger, and exteriorly accessible manually controllable means for regulating the metering valve to vary the rate of flow of the fuel from the chamber into the inlet passage.

3. In a multiple fuel injection pump for internal combustion engines, a distributor head having a bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a spill port leading therefrom in axially spaced relation to the delivery passages, a spring-pressed check valve in the bore, means for supplying fuel to the bore past the check valve, a plunger mounted in the bore in axially spaced relation to the check valve, said plunger containing a bore which receives the fuel from the bore in the head, a radial discharge port in the plunger for directing fuel from the bore in the plunger into each of the delivery passages in turn, an annular by-pass groove in the plunger in axially spaced relation to the discharge port, which groove is connected with the bore in the plunger and serves to direct surplus fuel from that bore into the spill port upon registration of the groove with the latter, and means for both rotating and reciprocating the plunger.

4. In a multiple fuel injection pump for internal combustion engines, a distributor head having a bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a spill port leading therefrom in axially spaced relation to the delivery passages, a spring-pressed check valve in the bore, means for supplying fuel to the bore past the check valve, a plunger mounted in the bore, said plunger containing a bore which receives the fuel from the bore in the head, a radial discharge port in the plunger for directing fuel from the bore in the plunger into each of the delivery passages in turn, an annular by-pass groove in the plunger in axially spaced relation to the discharge port, which groove is connected with the bore in the plunger and serves to direct surplus fuel from that bore into the spill port upon registration of the groove with the latter, a spring-pressed plunger-type valve in the bore in the plunger between the discharge port and the by-pass groove, which valve serves to draw back into the bore in the plunger a certain amount of fuel from the particular delivery passage with which the discharge port is in register upon the by-pass groove registering with the spill port, and means for both rotating and reciprocating the plunger.

5. In a multiple fuel injection pump for internal combustion engines, a fuel pressure chamber, a fuel inlet chamber, a primary pump for receiving fuel from the inlet chamber and discharging it under pressure into the pressure chamber, an injection pump having a plurality of fuel delivery passages, said injection pump receiving fuel from the pressure chamber and discharging it under pressure into each of the fuel delivery passages in turn, and means controlled by the speed of rotation of the pump for maintaining a predetermined pressure on the fuel in the pressure chamber at any particular speed, said means including a fuel by-pass between the pressure chamber and the inlet chamber, a valve controlling the by-pass, a spring tending to close the valve, a centrifugal weight also tending to close the valve, means for rotating the weight in accordance with the speed of rotation of the pump, and means actuated by the pressure on the fuel in the pressure chamber for variably opening the valve against the resistance offered by the spring and weight.

6. In a multiple fuel injection pump for internal combustion engines, a fuel pressure chamber, a fuel inlet chamber, a primary pump for receiving fuel from the inlet chamber and discharging it under pressure into the pressure chamber, an injection pump having a plurality of fuel delivery passages, said injection pump receiving fuel from the pressure chamber and discharging it under pressure into each of the fuel delivery passages in turn, and means controlled by the speed of rotation of the pump for maintaining a predetermined pressure on the fuel in the pressure chamber at any particular speed, said means including a rotatable cage in the pressure chamber containing an open-ended cross bore and a fuel by-pass between the bore and the inlet chamber, means for rotating the cage in accordance with the speed of rotation of the pump, a valve seat in the bore, a disk valve cooperating with the seat to control the flow of fuel through the by-pass, a spring tending to close the valve, and a plunger of larger cross-sectional area than the valve connected with the valve and reciprocably mounted in the bore, the centrifugal action of the plunger in the rotating cage tending to close the valve, and the pressure on the fuel in the pressure chamber acting differentially on the valve and plunger to maintain the valve open in varying degree against the resistance offered by the spring and said centrifugal action.

7. In a multiple fuel injection pump for internal combustion engines, a fuel pressure chamber, a fuel inlet chamber, a primary pump for receiving fuel from the inlet chamber and discharging it under pressure into the pressure chamber, means connected within the pressure chamber to said injection pump for operating the latter, an injection pump having a plurality of fuel delivery passages, said injection pump being located within the pressure chamber and receiving fuel from the pressure chamber and discharging it under pressure into each of the fuel delivery passages in turn, and means controlled by the speed of rotation of the pump for maintaining a predetermined pressure on the fuel in the pressure chamber at any particular speed.

8. In a multiple fuel injection pump for internal combustion engines, of the type characterized by a distributor head having a bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a plunger in the bore; means for rotating and reciprocating the plunger, said means including a drive shaft member arranged in concentric relation to the plunger, means associated with the plunger for directing fuel from the bore into each of the delivery passages in turn, a fuel pressure chamber connected with the bore, a fuel inlet chamber, a primary pump for receiving fuel from the inlet chamber and discharging it under pressure into the pressure chamber, said pump including two intermeshed pumping gears one of which is fixedly secured to said drive shaft member, and a rotatable centrifugally-controlled fuel by-passing member in the fuel pressure chamber in communication with the inlet chamber for returning fuel to the inlet chamber to maintain a predetermined pressure in the pressure chamber at any particular speed, said centrifugally controlled fuel by-passing member being connected with and rotated by the second gear of the primary pump.

9. In a multiple fuel injection pump for internal combustion engines, of the type characterized by a casing, a drive shaft in the casing, and a rotatable injection pump mechanism in the casing operatively connected with the drive shaft, said mechanism having a plurality of delivery passages; a high pressure fuel chamber in the casing from which fuel flows into said mechanism for delivery through said passages, exteriorly accessible means for controlling the rate of said flow, a rotatable primary pump mechanism in the casing operatively connected with the drive shaft, said primary pump mechanism having two later-

ally offset chambers, either of which is capable of receiving fuel from an inlet and the other of which is capable of discharging such fuel under high pressure into said high pressure chamber, depending upon the direction of rotation of the drive shaft, and a plate which forms a removable section of the casing and is provided with a laterally offset opening, said plate being reversible in position whereby the offset opening therein may be so positioned in the casing as to place said high pressure fuel chamber in communication with that one of the said two offset chambers which is serving as the receiving chamber for the primary pump mechanism, irrespective of the direction of rotation of the drive shaft.

10. In a rotary fuel injection pump for internal combustion engines, a fuel pressure chamber in which fuel is adapted to be maintained under pressure, and means controlled by the speed of rotation of the pump for maintaining a predetermined pressure on the fuel in the pressure chamber at any particular speed, said means including a fuel by-pass leading from the pressure chamber, a valve controlling the by-pass, a spring tending to close the valve, a centrifugal weight also tending to close the valve, means for rotating the weight in accordance with the speed of rotation of the pump, and means actuated by the pressure on the fuel in the pressure chamber for variably opening the valve against the resistance offered by the spring and weight.

11. In a fuel injection pump for internal combustion engines, of the type characterized by an injection pumping mechanism, a primary pumping mechanism, and a fuel pressure chamber in which fuel is maintained under pressure by the primary pumping mechanism; a passage leading from the chamber to the injection pumping mechanism, an open-ended cylinder in the pressure chamber in communication at one side with the passage, a plunger type valve reciprocably mounted in the cylinder for controlling the rate of flow of the fuel in the passage, a second cylinder in the pressure chamber beyond the first, with the confronting end only of the second cylinder exposed to the pressure in the pressure chamber, a plunger reciprocably mounted in the second cylinder, a connection between the valve and the plunger, whereby the pressure on the fuel acting on the valve and said one end of the plunger will tend to close the valve upon an increase in pressure and will tend to open the valve upon a decrease in pressure, and a spring member for adjustably opening

the valve, which member is adapted to be biased against the action of the plunger to hold the valve in a predetermined position but will yield to permit further opening of the valve by the plunger upon a decrease in the pressure on the fuel in the chamber.

12. In a fuel injection pump for internal combustion engines, an automatic fuel pressure governor comprising a rotary cage having a cross bore and a valve seat in the bore adjacent one end of the latter, a disk valve for coaction with the seat, a plunger in the bore adjacent the other end of the latter, a connection between the valve and the plunger, said plunger being acted upon by centrifugal force to urge the valve toward its closed position and being acted upon by hydraulic pressure to urge the valve toward its open position, and a spring for urging the valve toward its closed position.

13. In a single plunger type multiple fuel injection pump for internal combustion engines, a distributor head having a bore, which bore is provided with a plurality of fuel delivery passages leading therefrom at circumferentially spaced points, and a spill port leading therefrom in axially spaced relation to the delivery passages, a plunger mounted in the bore, said plunger being provided with a supply port for directing fuel from the bore into each of the delivery passages in turn, a pressure fuel chamber surrounding the head at the location of the spill port and the point of entry of the plunger into the bore, means for supplying fuel under pressure to the chamber, a fuel passage connected at its inlet end with the pressure chamber and at its discharge end with the bore for conducting fuel from the chamber to the bore, and means for simultaneously rotating and reciprocating the plunger.

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