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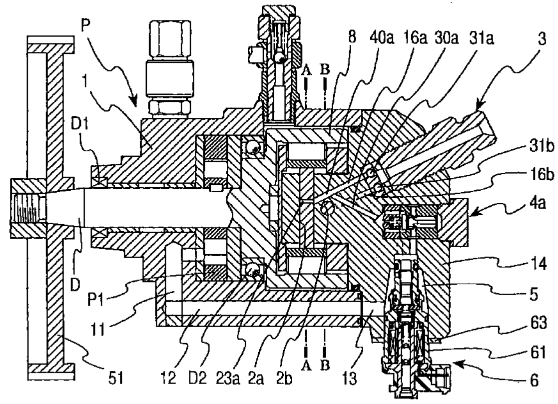
(54) Variable-discharge-rate high-pressure pump

(57) An object of this invention is to reduce the drive torque required by a variable-discharge-rate high-pressure pump. The reduction in the drive torque enhances the durability of a timing belt used in a mechanism for power transmission to the high-pressure pump. The high-pressure pump has a plurality of cylinders (2a, 2b). A pair of plungers (21a, 21c) are disposed in the cylinder (2a). A pair of plungers (21b, 21d) are disposed in the cylinder (2b). A pressure chamber (23a) is defined between opposing end surfaces of the plungers (21a, 21c). A pressure chamber (23b) is defined between opposing end surfaces of the plungers (21b, 21d). Low-pressure fuel which is metered by an electromagnetic valve (6) for flow rate control is sequentially fed to the pressure chambers (23a, 23b). An inner cam (8) is provided which drives the plungers (21a, 21b, 21c, 21d). The inner cam (8) is common to the plungers (21a, 21b, 21c, 21d). Fuel pressurization and fuel pumping are alternately implemented in the pressure chambers (23a, 23b) in accordance with rotation of the inner cam (8). According to the alternate fuel pressurization and fuel pumping, the peak value of the drive torque required by the high-pressure pump can be reduced.

FIG. 1



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Description

[0001] This invention generally relates to a common-rail fuel injection system which includes a common rail (a high pressure accumulation chamber or a pressure accumulation pipe) storing pressurized fuel, and devices supplied with fuel from the common rail and injecting fuel into cylinders of a diesel engine. This invention particularly relates to a variable-discharge-rate high-pressure pump (a variable-displacement high-pressure pump) in such a common-rail fuel injection system which pumps high-pressure fuel to a common rail therein.

[0002] A common-rail fuel injection system is known as a system for injecting fuel into a diesel engine. The common-rail fuel injection system includes a pressure accumulation chamber (a common rail) connected to cylinders of the engine. A variable-discharge-rate high-pressure pump or a variable-displacement high-pressure pump feeds high-pressure fuel to the common rail at a desired flow rate. Generally, the pressure of fuel in the common rail is regulated at a constant level. The high-pressure fuel is fed from the common rail to fuel injectors before being injected thereby into the cylinders of the engine at desired timings. An example of the common-rail fuel injection system is disclosed in Japanese published unexamined patent application 64-73166 (application number 62-231349).

[0003] Fig. 14 shows an example of a prior-art high-pressure pump used in a common-rail fuel injection system. The prior-art pump of Fig. 14 includes a cylinder 91 in which a plunger 92 is movably disposed. The plunger 92 is driven by a cam (not shown), being reciprocated within the cylinder 91. A pressure chamber 93 is defined by the inner wall surfaces of the cylinder 91 and the upper end surface of the plunger 92. An electromagnetic valve (a solenoid valve) 94 is located in a position above the pressure chamber 93. The electromagnetic valve 94 has a low-pressure passage 95, and a valve member 96 for blocking and unblocking communication of the low-pressure passage 95 with the pressure chamber 93. The electromagnetic valve 94 also includes a coil 97 for driving the valve member 96.

[0004] In the prior-art pump of Fig. 14, when the coil 97 is deenergized, the valve member 96 assumes an open position. In the case where the valve member 96 remains in its open position, as the plunger 92 moves downward, fuel is transmitted from a low-pressure feed pump (not shown) to the pressure chamber 93 via the low-pressure passage 95 and the gap or the opening between the valve member 96 and its seat 98 thereabound. When the coil 97 is energized, the valve member 96 is attracted upward into contact with its seat 98. Thus, the valve member 96 assumes a closed position. In the case where the valve member 96 remains in its closed position, as the plunger 92 moves upward, the fuel is pressurized and is pumped from the pressure chamber 93 toward a common rail via a high-pressure passage 99 extending in side walls of the cylinder 91.

[0005] During the upward movement of the plunger 92, the pressure of fuel in the pressure chamber 93 forces the valve member 96 toward its closed position. Accordingly, in this case, after the valve member 96 assumes its closed position, the valve member 96 continues to be held in its closed position even if the coil 97 is deenergized. The flow rate of fuel transmitted to the common rail from the prior-art pump of Fig. 14 is adjusted by controlling a timing at which the valve member 96 is changed from its open position to its closed position. Such control is referred to as prestroke control. Specifically, after the plunger 92 starts to move upward, the valve member 96 remains held in its open position and hence allows fuel to return from the pressure chamber 93 to the low-pressure passage 95 until the amount of fuel in the pressure chamber 93 decreases to a desired value. When the amount of fuel in the pressure chamber 93 reaches the desired value, the valve member 96 is changed from its open position to its closed position so that fuel starts to be pumped from the pressure chamber 96 toward the common rail. Fuel remains pumped so that the desired amount of fuel will be transmitted to the common rail.

[0006] The prior-art pump of Fig. 14 is powered by a related diesel engine. As the rotational speed of the engine rises, the rate of fuel transmission by the prior-art pump of Fig. 14 increases. In the case where the rate of fuel transmission by the prior-art pump of Fig. 14 excessively increases, the valve member 96 tends to be spontaneously locked to its closed position even though the coil 97 is de-energized. Causes of such a problem are as follows. During upward movement of the plunger 92, the lower end surface of the valve member 96 is directly subjected to a dynamic pressure of fuel in the pressure chamber 93, and the valve member 96 is forced upward thereby. Fuel which is flowing toward the low-pressure passage 95 through the opening between the valve member 96 and its seat 98 undergoes the throttling process and hence forces the valve member 96 toward its closed position. In the presence of such a problem, it tends to be difficult to properly control the rate of fuel transmission by the prior-art pump of Fig. 14.

[0007] A first conceivable way of solving the previously-indicated problem is to set the stroke of the valve member 96 to a large value. A second conceivable way of solving the previously-indicated problem is to use a strong return spring which urges the valve member 96. Both the first and second ways cause deteriorations in response characteristics of the prior-art pump of Fig. 14. Thus, to provide acceptable response characteristics of the prior-art pump of Fig. 14, it is necessary to increase the valve-member attracting force developed in the electromagnetic valve 94. The increase in the valve-member attracting force is implemented by raising electric power of energizing the coil 97 or enlarging the electromagnetic valve 94. The raised electric power causes a problem related to power consumption rate.

The enlarged electromagnetic valve 94 causes a problem related to manufacturing cost.

[0008] In the prior-art pump of Fig. 14, the communication of the low-pressure passage 95 with the pressure chamber 93 is blocked and unblocked by the electromagnetic valve 94. There is a certain time lag between the moment of application of a valve-closing signal to the electromagnetic valve 94 and the moment of movement of the valve member 96 to its closed position. Generally, the time lag is calculated in advance, and the calculated time lag is considered in controlling the timing at which the valve member 96 is changed to its closed position. In the case where the rotational speed of the engine considerably rises so that the rate of fuel transmission by the prior-art pump of Fig. 14 significantly increases, the timings at which the valve member 96 assumes its closed position and its open position tend to deviate from desired timings. Thus, in this case, the control of the rate of fuel transmission by the prior-art pump of Fig. 14 tends to be improper.

[0009] Inventors among the inventors of the present application have proposed an advanced high-pressure pump of the variable-discharge-rate type in Japanese published unexamined patent application 10-73064 (application numbers 8-195653 and 9-100939). The advanced high-pressure pump is designed to reliably control the rate of fuel transmission to a common rail (a pressure accumulation chamber) even when the rotational speed of a related diesel engine considerably rises. The advanced high-pressure pump is good in size and power consumption rate. In the advanced high-pressure pump, a valve member for controlling the rate of fuel flow into a pressure chamber from a low-pressure passage is separate from a valve member for blocking and unblocking communication between the pressure chamber and the low-pressure passage.

[0010] Fig. 15 shows the advanced high-pressure pump. The high-pressure pump of Fig. 15 includes a pump housing 100 into which a drive shaft 101 extends. The drive shaft 101 is supported by the pump housing 100. A feed pump 102 rotates together with the drive shaft 101. The feed pump 102 drives fuel into a low-pressure chamber or a fuel reservoir 105 via low-pressure passages 103 and 104.

[0011] In the high-pressure pump of Fig. 15, an inner cam 106 is formed integrally on the right-hand end of the drive shaft 101. The left-hand end of a pump head 107 extends into a recess in the inner cam 106. The left-hand end of the pump head 107 has four radial holes 108 (only two of which are shown in Fig. 15) corresponding to cylinders respectively. Plungers 109 are disposed in the radial holes 108, respectively. The plungers 109 can reciprocate within the radial holes 108. A pressure chamber 110 is defined by the inner end surfaces of the plungers 109 and the inner wall surfaces in the radial holes 108. Reciprocating motion of the plungers 109 draws fuel into the pressure chamber 110, and pressurizes and pumps fuel.

[0012] In the high-pressure pump of Fig. 15, the fuel reservoir 105 and the pressure chamber 110 is connected by a fuel passage in which an electromagnetic valve 111 and a check valve 112 are provided successively along the direction from the upstream side toward the downstream side. While the electromagnetic valve 111 remains open, the check valve 112 continues to be opened by the pressure of incoming fuel. When the electromagnetic valve 111 is closed, the check valve 112 is also closed. Alter a desired amount of fuel has been supplied to the pressure chamber 110 via the electromagnetic valve 111, fuel is pumped out of the pressure chamber 110 by the plungers 109. In this case, the check valve 112 continues to block the fuel passage to the pressure chamber 110 during the time interval from the moment of the start of fuel pressurization to the moment of the end thereof. Accordingly, the electromagnetic valve 111 receives a pressure whose highest level is equal to a fuel feed pressure (about 15 atm). Thus, the electromagnetic valve 111 having a small size suffices, and a cost reduction is available.

[0013] In the high-pressure pump of Fig. 15, since the four plungers 109 are synchronously moved to pressurize fuel, pumping fuel tends to require a great drive torque. Fig. 8(a) indicates the drive torque (the drive torque which occurs at the maximum fuel discharge rate) required by the high-pressure pump of Fig. 15 which is designed such that the total number of the plungers 109 is equal to four, and the inner circumferential surfaces of the inner cam 106 have four cam projections. During every revolution of the inner cam 106, that is, during every revolution of the drive shaft 101, pressurizing and pumping fuel are implemented four times. The time interval of each pumping of fuel out of the pressure chamber 110 corresponds to an angular interval of about 45°. The time interval of each feed of fuel to the pressure chamber 110 corresponds to an angular interval of about 45°. The pumping of fuel out of the pressure chamber 110 is intermittently carried out. As shown in Fig. 8(a), the maximum drive torque required by the high-pressure pump of Fig. 15 is equal to 50 Nm.

[0014] In the high-pressure pump of Fig. 15, a mechanism for rotating the drive shaft 101 and the inner cam 106 has a timing belt (not shown) which is connected to the crankshaft or the camshaft of the engine. As shown in Fig. 15, a timing pulley 113 is mounted on the left-hand end of the drive shaft 101. The timing pulley 113 engages the timing belt. A rotational force is transmitted from the engine to the inner cam 106 via the members including the timing belt. Thus, the inner cam 106 is driven and rotated by the engine. In the case where the maximum drive torque is equal to a large value, for example, 50 Nm, the timing belt tends to be subjected to a great stress. The great stress causes a deterioration in durability of the timing belt.

[0015] Accordingly, it is an object of this invention to provide a variable-discharge-rate high-pressure pump which requires only a relatively-low drive torque, and hence which makes a timing belt adequately durable.

[0016] A first aspect of this invention provides a variable-discharge-rate high-pressure pump comprising a plunger movably disposed in a cylinder, a cam for reciprocating the plunger in the cylinder, a pressure chamber defined by an inner wall surface in the cylinder and an end surface of the plunger for receiving low-pressure fuel from a low-pressure fuel passage and pressurizing low-pressure fuel in accordance with reciprocation of the plunger, and a means for transmitting pressurized fuel from the pressure chamber to a high-pressure fuel passage, characterized in that there are provided a plurality of pressure chambers, and a means for feeding low-pressure fuel to the pressure chambers; and the cam is common to the pressure chambers, and the cam causes fuel pressurizations and fuel pumpings in the pressure chambers to alternate with each other.

[0017] A second aspect of this invention is based on the first aspect thereof, and provides a variable-discharge-rate high-pressure pump characterized in that the feeding means comprises an electromagnetic valve for adjusting rates of fuel feed to the pressure chambers, and check valves provided between the electromagnetic valve and the pressure chambers for allowing fuel flow only in directions from the low-pressure fuel passage to the pressure chambers.

[0018] A third aspect of this invention is based on the first aspect or the second aspect thereof, and provides a variable-discharge-rate high-pressure pump characterized in that there are provided a plurality of cylinders, axes of the cylinders are separate from each other in an axial direction of a drive shaft to which the cam is connected, two plungers are slidably disposed in each of the cylinders, and the pressure chambers are formed between opposing end surfaces of the plungers.

[0019] A fourth aspect of this invention is based on the second aspect thereof, and provides a variable-discharge-rate high-pressure pump characterized in that there are provided a plurality of electromagnetic valves which correspond to the pressure chambers respectively.

[0020] A fifth aspect of this invention is based on the third aspect thereof, and provides a variable-discharge-rate high-pressure pump characterized in that the number of the cylinders is two, and the axes of the cylinders are perpendicular to each other.

[0021] A sixth aspect of this invention is based on the fifth aspect thereof, and provides a variable-discharge-rate high-pressure pump characterized in that the cam has a cylindrical shape, the cam has inner circumferential surfaces formed with projections opposed to each other, and the cylinders are in a recess of the cam.

[0022] A seventh aspect of this invention provides a variable-discharge-rate high-pressure pump comprising first and second cylinders; a first plunger movably disposed in the first cylinder and defining a part of a first pressure chamber; a second plunger movably disposed in the second cylinder and defining a part of a second pressure chamber; means for feeding fuel to the first and second pressure chambers; means for reciprocating the first and second plungers relative to the first and second cylinders to periodically contract the first and second pressure chambers and to periodically pressurize fuel in the first and second pressure chambers; and wherein the reciprocation of the first plunger differs in phase from the reciprocation of the second plunger so that a peak value of a mechanical power required to reciprocate the first and second plungers can be reduced.

[0023] An eighth aspect of this invention provides a variable-discharge-rate high-pressure pump comprising first pumping means including a first cylinder, a first plunger disposed in the first cylinder and being able to reciprocate, and a first pressure chamber defined by an inner wall surface of the first cylinder and an end surface of the first plunger and pressurizing low-pressure fuel in accordance with reciprocation of the first cylinder; second pumping means including a second cylinder, a second plunger disposed in the second cylinder and being able to reciprocate, and a second pressure chamber defined by an inner wall surface of the second cylinder and an end surface of the second plunger and pressurizing low-pressure fuel in accordance with reciprocation of the second cylinder; and a cam for sequentially reciprocating the first plunger and the second plunger; wherein the cam sequentially drives the first plunger and the second plunger so that fuel will be sequentially pumped from the first pressure chamber and the second pressure chamber into a pressure accumulation chamber.

[0024] A ninth aspect of this invention is based on the eighth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the cam has a common cam surface for driving the first plunger and the second plunger.

[0025] A tenth aspect of this invention is based on the ninth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the cam comprises an inner cam having an inner circumferential surface forming the cam surface.

[0026] An eleventh aspect of this invention is based on one of the eighth aspect to the tenth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein low-pressure fuel is introduced into the first pressure chamber and the second pressure chamber from a low-pressure passage, an electromagnetic valve is provided which adjusts a rate of low-pressure fuel flow into the first pressure chamber and the second pressure chamber, a first check valve is provided between the electromagnetic valve and the first pressure chamber to inhibit fuel return from the first pressure chamber toward the low-pressure passage, and a second check valve is provided between the electromagnetic valve and the second pressure chamber to inhibit fuel return from the second pressure chamber toward the low-pressure passage.

[0027] A twelfth aspect of this invention is based on the eleventh aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the electromagnetic valve includes a plurality of sub electromagnetic valves corresponding to the first pumping means and the second pumping means respectively.

[0028] A thirteenth aspect of this invention is based on one of the eighth aspect to the twelfth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein axes of the first cylinder and the second cylinder are separate from each other in an axial direction of a drive shaft to which the cam is connected.

[0029] A fourteenth aspect of this invention is based on the thirteenth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the axes of the first cylinder and the second cylinder are perpendicular to each other.

[0030] A fifteenth aspect of this invention is based on one of the ninth aspect to the fourteenth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the first plunger includes a pair of plungers, the second plunger includes a pair of plungers, and the cam surface of the cam has two projections each corresponding to the pair of the plungers.

[0031] A sixteenth aspect of this invention is based on one of the ninth aspect to the fifteenth aspect thereof, and provides a variable-discharge-rate high-pressure pump wherein the cam surface of the cam has a first region for driving the first plunger and a second region for driving the second plunger, and the first region and the second region partially overlap each other.

Fig. 1 is a longitudinal section view of a variable-discharge-rate high-pressure pump according to a first embodiment of this invention.

Fig. 2 is a diagram of a common-rail fuel injection system including the high-pressure pump of Fig. 1.

Fig. 3 is a plan view of a mechanism for power transmission from a diesel engine to the high-pressure pump of Fig. 1.

Fig. 4 is an enlarged section view of a portion of Fig. 1.

Fig. 5(A) is a sectional view taken along the line A-A in Fig. 4.

Fig. 5(B) is a sectional view taken along the line B-B in Fig. 4.

Fig. 6 is a diagram of a major portion of the high-pressure pump in Fig. 1.

Fig. 7 is a diagram of operating conditions of the high-pressure pump in Fig. 1.

Fig. 8(a) is a diagram of pumping and pressurizing characteristics of a variable-discharge-rate high-pressure pump in Fig. 15.

Fig. 8(b) is a diagram of pumping and pressurizing characteristics of the high-pressure pump in Fig. 1.

Fig. 9 is a diagram of a major portion of a variable-discharge-rate high-pressure pump according to a second embodiment of this invention.

Fig. 10 is a diagram of operating conditions of the high-pressure pump in Fig. 9.

Fig. 11 is a diagram of operating conditions of the high-pressure pump in Fig. 1 which occur at a small fuel discharge rate.

Fig. 12 is a sectional view taken along the line C-C in Fig. 4.

Fig. 13 is a perspective view of a shoe guide in the high-pressure pump of Fig. 1.

Fig. 14 is a sectional view of a prior-art high-pressure pump used in a common-rail fuel injection system.

Fig. 15 is a longitudinal section view of a variable-discharge-rate high-pressure pump.

First Embodiment of the Invention

[0032] With reference to Fig. 2, a common-rail fuel injection system includes a plurality of injectors "I" which extend into combustion chambers (cylinders) of a diesel engine "E" respectively. The injectors "I" are connected to a pressure accumulation chamber "R" via electromagnetic valves (solenoid valves) B1. The pressure accumulation chamber "R" is common to the cylinders of the engine "E". Thus, the pressure accumulation chamber "R" is also referred to as the common rail "R". The common rail "R" is supplied with pressurized fuel.

[0033] The electromagnetic valves B1 are used for fuel injection control. The electromagnetic valves B1 are changed between open positions (ON states) and closed positions (OFF states) in response to electric control signals applied thereto. When the electromagnetic valves B1 are open, fuel is transmitted from the common rail "R" to the injectors "I" and is injected into the cylinders of the engine "E" via the injectors "I". When the electromagnetic valves B1 are closed, the transmission of fuel from the common rail "R" to the injectors "I" is inhibited so that the fuel injection into the engine "E" is interrupted.

[0034] Preferably, the pressure of fuel in the common rail "R" is continuously maintained at a predetermined high level. The common rail "R" is connected to a variable-discharge-rate high-pressure pump "P" via a feed pipe R1 and discharge valves B2.

[0035] The high-pressure pump "P" is connected to a fuel tank "T" via a feed pump P1. The feed pump P1 drives fuel from the fuel tank "T" to the high-pressure pump "P". The pressure of fuel fed to the high-pressure pump "P" is relatively low. Thus, the high-pressure pump "P" is supplied with low-pressure fuel. The high-pressure pump "P" pressurizes low-pressure fuel into high-pressure fuel, and pumps high-pressure fuel into the common rail "R" via the discharge valves B2.

[0036] A pressure sensor S1 located in the common rail "R" detects the pressure of fuel in the common rail "R". The pressure sensor S1 outputs an electric signal representing the detected fuel pressure.

[0037] An engine speed sensor S2 associated with the camshaft or the crankshaft of the engine "E" detects the rotational speed of the engine "E". The engine speed sensor S2 outputs an electric signal representing the detected rotational engine speed.

[0038] An engine load sensor S3 associated with the engine "E" detects a load on the engine "E". The engine load sensor S3 outputs an electric signal representing the detected engine load.

[0039] The high-pressure pump "P" includes an electromagnetic valve (a solenoid valve) P2 which forms at least a portion of a discharge control device. The electromagnetic valve P2 adjusts the rate of fuel discharge from the high-pressure pump "P" in response to an electric control signal applied thereto.

[0040] An electronic control unit ECU is connected to the sensors S1, S2, and S3. The electronic control unit ECU receives the output signals of the sensors S1, S2, and S3. The electronic control unit ECU is also connected to the electromagnetic valves B1 and P2. The electronic control unit ECU generates control signals in response to the output signals of the sensors S1, S2, and S3, and outputs the control signals to the electromagnetic valves B1 and P2. The electromagnetic valves B1 and P2 respond to the control signals.

[0041] The electronic control unit ECU includes a combination of a CPU, a ROM, a RAM, and an input/output port. The electronic control unit ECU operates in accordance with a program stored in the ROM.

[0042] The program has a segment (a subroutine) for controlling the pressure of fuel in the common rail "R". According to this program segment, the electronic control unit ECU operates as follows. The electronic control unit ECU derives the current values of the rotational engine speed and the engine load from the output signals of the engine speed sensor S2 and the engine load sensor S3. The electronic control unit ECU calculates a desired pressure of fuel in the common rail "R" from the current values of the rotational engine speed and the engine load. The electronic control unit ECU derives information of an actual pressure of fuel in the common rail "R" from the output signal of the pressure sensor S1. The electronic control unit ECU calculates the difference (the error) between the actual pressure and the desired pressure of fuel in the common rail "R". The electronic control unit ECU drives the electromagnetic valve P2 in response to the calculated pressure difference (the calculated pressure error). Thus, the electronic control unit ECU adjusts the rate of fuel discharge from the high-pressure pump "P" in response to the calculated pressure difference. The control of the fuel discharge rate is designed to equalize the actual fuel pressure in the common rail "R" to the desired fuel pressure therein.

[0043] The program has a segment (a subroutine) for controlling the timing and the rate of fuel injection into the engine "E". According to this program segment, the electronic control unit ECU operates as follows. The electronic control unit ECU derives the current values of the rotational engine speed and the engine load from the output signals of the engine speed sensor S2 and the engine load sensor S3. The electronic control unit ECU calculates a desired fuel injection timing and a desired fuel injection rate from the current values of the rotational engine speed and the engine load. The electronic control unit ECU drives the electromagnetic valves B1 in response to the desired fuel injection timing and the desired fuel injection rate so that actual fuel injection into the engine "E" will be implemented at a timing and a rate equal to the desired fuel injection timing and the desired fuel injection rate respectively.

[0044] With reference to Fig. 1, the high-pressure pump "P" includes a pump housing 1 into which a drive shaft "D" extends. The drive shaft "D" is supported by the pump housing 1. The drive shaft "D" is rotated by the engine "E". The speed of rotation of the drive shaft "D" is equal to half the speed of rotation of the crankshaft of the engine "E". A timing pulley 51 is mounted on the left-hand end of the drive shaft "D".

[0045] As shown in Fig. 3, the timing pulley 51 engages a timing belt 52. A timing pulley 53 is mounted on the camshaft of the engine "E". A timing pulley 54 is mounted on the crankshaft of the engine "E". The timing pulleys 53 and 54 mesh with the timing belt 52. As the crankshaft of the engine "E" rotates, the timing pulley 54 rotates so that the timing belt 52 moves. The timing pulleys 51 and 53 rotate in accordance with movement of the timing belt 52. The drive shaft "D" rotates together with the timing pulley 51. In this way, the drive shaft "D" is rotated by the engine "E". Idler pulleys 55 and 56 engage the timing belt 52. A spring 57 presses the idler pulley 56 against the timing belt 52, thereby applying a suitable tension to the timing belt 52 to prevent the occurrence of looseness thereof.

[0046] As shown in Fig. 1, the pump housing 1 accommodates the feed pump P1. In other words, the feed pump P1 is built in the high-pressure pump "P". The feed pump P1 is connected to the drive shaft "D". The feed pump P1 is of a vane type. The vanes of the feed pump P1 rotate together with the drive shaft "D". The feed pump P1 drives fuel from the fuel tank "T" (see Fig. 2) to the high-pressure pump "P". The feed pump P1 pressurizes fuel at a low level, and supplies low-pressure fuel to the high-pressure pump "P".

[0047] A pump head 14 is attached to the pump housing 1. The pump head 14 has a low-pressure chamber or a fuel reservoir 5. The walls of the pump housing 1 have low-pressure passages 11 and 12 connected to each other. The low-pressure passage 11 extends from the outlet of the feed pump P1. The pump head 14 has a low-pressure passage 13 connected between the low-pressure passage 12 and the fuel reservoir 5. The feed pump P1 draws fuel from the fuel tank "T" (see Fig. 2), and drives fuel to the fuel reservoir 5 via the low-pressure passages 11, 12, and 13. The inlet and the outlet of the feed pump P1 are connected via a pressure adjustment valve (not shown). The pressure of fuel at the outlet of the feed pump P1 is controlled or regulated by the pressure adjustment valve.

[0048] Bearings D1 and D2 rotatably support the drive shaft "D" on the pump housing 1. An inner cam 8 is integrally formed on the right-hand end of the drive shaft "D". Thus, the inner cam 8 rotates together with the drive shaft "D". The pump head 14 fits into a right-hand end opening in the pump housing 1. The left-hand end of the pump head 14 has a central portion which extends into a recess in the inner cam 8. A check valve 4a is located in a central portion of the right-hand end of the pump head 14. The check valve 4a will be explained later. Another check valve 4b (not shown in Fig. 1) is provided in the pump head 14. The check valve 4b will be explained later.

[0049] An electromagnetic valve 6 forming the electromagnetic valve P2 is attached to the lower end of the pump head 14. The electromagnetic valve 6 extends into the pump head 14. The electromagnetic valve 6 has a casing 61 formed with a flange 63 which is fixed to the walls of the pump head 14 by a bolt or bolts (not shown). The electromagnetic valve 6 and the check valves 4a and 4b compose the previously-indicated discharge control device. The electromagnetic valve 6 adjusts the rate of low-pressure fuel flow into a pressure chamber in response to an electric signal applied thereto.

[0050] It should be noted that the drive shaft "D" and the inner cam 8 may be separate members connected by a joint.

[0051] As shown in Fig. 4, the casing 61 of the electromagnetic valve 6 accommodates a coil 62. The electromagnetic valve 6 has a valve body 68 which securely fits into the upper end of the casing 61. The valve body 68 has a cylinder 69 in which a valve member 73 is slidably disposed. An annular passage 74a extends around an upper end portion of the valve member 73. The annular passage 74a communicates with the fuel reservoir 5 via a radial passage 74b extending in the valve body 68. The annular passage 74a can communicate with an axial passage 74c extending in the valve body 68. The axial passage 74c leads to the check valve 4a via a fuel passage 72 extending in the pump head 14. Also, the axial passage 74c leads to the check valve 4b (not shown in Fig. 4) via a fuel passage 71 extending in the pump head 14 and being parallel with the fuel passage 72.

[0052] An armature 64 is fixed to the lower end of the valve member 73. The armature 64 is opposed to a stator 65. The armature 64 is spaced from the stator 65. The coil 62 extends outward of the stator 65. The stator 65 has a chamber 66 in which a spring 67 is disposed. The spring 67 is connected between the armature 64 and a member connected to the stator 65. The spring 67 urges the armature 64 upward relative to the stator 65.

[0053] The valve body 68 is formed with a valve seat 75 extending at the lower end of the axial passage 74c. The valve seat 75 has an approximately conical shape. The coil 62 receives an electric control signal from the electronic control unit ECU (see Fig. 2). The coil 62 is energized and de-energized by the electric control signal. When the coil 62 is de-energized, the upper end of the valve member 73 contacts the valve seat 75 and hence the electromagnetic valve 6 falls into a closed position. In this case, the valve member 73 blocks the communication between the annular passage 74a and the axial passage 74c. When the coil 62 is energized, the armature 64 is attracted toward the stator 65. In this case, the valve member 73 moves together with the armature 64, and the upper end thereof separates from the valve seat 75 so that the electromagnetic valve 6 changes to an open position in which the communication between the annular passage 74a and the axial passage 74c is unblocked. Since the electromagnetic valve 6 is closed when being de-energized, there is the advantage that fuel supply will be suspended if the coil 62 is damaged.

[0054] As shown in Fig. 4, the central portion of the left-hand end of the pump head 14 has diametrical holes 2a and 2b corresponding to cylinders respectively. As shown in Fig. 5(A), a pair of plungers 21a and 21c are slidably disposed in the diametrical hole 2a. The plungers 21a and 21c are opposed to each other. The plungers 21a and 21c can reciprocate relative to the diametrical hole 2a. The opposing end surfaces (the inner end surfaces) of the plungers 21a and 21c, and the inner surfaces of the pump head 14 which are exposed at the diametrical hole 2a define a pressure chamber 23a. The diametrical hole 2a, the plungers 21a and 21c, and the pressure chamber 23a form a first high-pressure pumping device.

[0055] As shown in Fig. 5(B), a pair of plungers 21b and 21d are slidably disposed in the diametrical hole 2b. The plungers 21b and 21d are opposed to each other. The plungers 21b and 21d can reciprocate relative to the diametrical hole 2b. The opposing end surfaces (the inner end surfaces) of the plungers 21b and 21d, and the inner surfaces of the pump head 14 which are exposed at the diametrical hole 2b define a pressure chamber 23b. The diametrical hole 2b, the plungers 21b and 21d, and the pressure chamber 23b form a second high-pressure pumping device.

[0056] The plunger 21a is shorter than the plunger 21c. Thus, the center of the pressure chamber 23a is offset from the center of the diametrical hole 2a. Similarly, the plunger 21d is shorter than the plunger 21b. Thus, the center of the pressure chamber 23b is offset from the center of the diametrical hole 2b. This arrangement has the advantage that fuel passages 40a, 40b, 30a, and 30b explained later can be easily formed.

[0057] The outer ends of the plungers 21a, 21b, 21c, and 21d are provided with shoes 24a, 24b, 24c, and 24d which retain rotatable rollers 22a, 22b, 22c, and 22d, respectively. The shoes 24a, 24b, 24c, and 24d are slidably supported by a shoe guide 15.

[0058] The diametrical holes 2a and 2b are separate from each other by a predetermined interval extending along the axial direction of the drive shaft "D". The axes of the diametrical holes 2a and 2b are perpendicular to each other. In addition, the axes of the diametrical holes 2a and 2b are perpendicular to the axis of the drive shaft "D".

[0059] The inner cam 8 is common to the diametrical holes 2a and 2b. The plungers 21a, 21b, 21c, and 21d reciprocate relative to and within the diametrical holes 2a and 2b as the inner cam 8 rotates. The inner cam 8 has inner circumferential surfaces forming cam surfaces 81. The cam surfaces 81 have a plurality of projections. The rollers 22a, 22b, 22c, and 22d contact the cam surfaces 81. During rotation of the inner cam 8, the rollers 22a, 22b, 22c, and 22d follow the cam surfaces 81 while being radially reciprocated. The cam surfaces 81 are approximately elliptical in cross section, and have two projections spaced by an angular interval of 180°.

[0060] Regarding the combination of the plungers 21a and 21c and the pressure chamber 23a, every revolution of the inner cam 8 is divided into first, second, third, and fourth successive stages. During the first stage, the plungers 21a and 21c are moved inward so that the pressure chamber 23a contracts. In this case, fuel in the pressure chamber 23a is pressurized, and is pumped out thereof. Accordingly, during the first stage, there occurs a pressurization stroke related to the combination of the plungers 21a and 21c and the pressure chamber 23a. The first stage corresponds to an angular interval of, for example, about 120°. During the second stage which follows the first stage, the plungers 21a and 21c are moved outward so that the pressure chamber 23a expands. In this case, fuel can be drawn into the chamber 23a. Accordingly, during the second stage, there occurs a suction stroke related to the combination of the plungers 21a and 21c and the pressure chamber 23a. The second stage corresponds to an angular interval of, for example, about 60°. During the third stage which follows the second stage, the plungers 21a and 21c are moved inward so that the pressure chamber 23a contracts. In this case, fuel in the pressure chamber 23a is pressurized, and is pumped out thereof. Accordingly, during the third stage, there occurs a pressurization stroke related to the combination of the plungers 21a and 21c and the pressure chamber 23a. The third stage corresponds to an angular interval of, for example, about 120°. During the fourth stage which follows the third stage, the plungers 21a and 21c are moved outward so that the pressure chamber 23a expands. In this case, fuel can be drawn into the chamber 23a. Accordingly, during the fourth stage, there occurs a suction stroke related to the combination of the plungers 21a and 21c and the pressure chamber 23a. The fourth stage corresponds to an angular interval of, for example, about 60°.

[0061] Regarding the combination of the plungers 21b and 21d and the pressure chamber 23b, every revolution of the inner cam 8 is divided into first, second, third, and fourth successive stages. During the first stage, the plungers 21b and 21d are moved inward so that the pressure chamber 23b contracts. In this case, fuel in the pressure chamber 23b is pressurized, and is pumped out thereof. Accordingly, during the first stage, there occurs a pressurization stroke related to the combination of the plungers 21b and 21d and the pressure chamber 23b. The first stage corresponds to an angular interval of, for example, about 120°. During the second stage which follows the first stage, the plungers 21b and 21d are moved outward so that the pressure chamber 23b expands. In this case, fuel can be drawn into the chamber 23b. Accordingly, during the second stage, there occurs a suction stroke related to the combination of the plungers 21b and 21d and the pressure chamber 23b. The second stage corresponds to an angular interval of, for example, about 60°. During the third stage which follows the second stage, the plungers 21b and 21d are moved inward so that the pressure chamber 23b contracts. In this case, fuel in the pressure chamber 23b is pressurized, and is pumped out thereof. Accordingly, during the third stage, there occurs a pressurization stroke related to the combination of the plungers 21b and 21d and the pressure chamber 23b. The third stage corresponds to an angular interval of, for example, about 120°. During the fourth stage which follows the third stage, the plungers 21b and 21d are moved outward so that the pressure chamber 23b expands. In this case, fuel can be drawn into the chamber 23b. Accordingly, during the fourth stage, there occurs a suction stroke related to the combination of the plungers 21b and 21d and the pressure chamber 23b. The fourth stage corresponds to an angular interval of, for example, about 60°.

[0062] The combination of the plungers 21a and 21c and the pressure chamber 23a is offset in operation phase from the combinations of the plungers 21b and 21d and the pressure chamber 23b so that the fuel pressurization by the plungers 21a and 21c will substantially alternate with the fuel pressurisation by the plungers 21b and 21d. Thus, the pair of the plungers 21a and 21c and the pair of the plungers 21b and 21d alternately pressurize and pump fuel. Accordingly, it is possible to effectively reduce the maximum torque required by the high-pressure pump "P".

[0063] As shown in Fig. 4, a plate 7 is provided on the left-hand end of the pump head 14. Bolts (not shown) fix the plate 7 and the shoe guide 15 to the pump head 14. A washer 76 is provided between the plate 7 and the drive shaft "D". The washer 76 is slidable and rotatable relative to the drive shaft "D". Also, the washer 76 is slidable and rotatable relative to the plate 7.

[0064] The check valve 4a has a casing 42, and a valve member 44 disposed in the casing 42. The casing 42 is fixed to the pump head 14 by a screw 47. The casing 42 has a fuel passage 43 which extends along the left-right direction in Fig. 4. In addition, the casing 42 has an annular passage 48, and radial passages 49 communicating with the annular passage 48. The annular passage 48 and the radial passages 49 connect the fuel passage 43 with the fuel passage 72. The valve member 44 selectively blocks and unblocks the fuel passage 43. Inner surfaces of the casing 42 have a step forming a conical valve seat 45 which extends at a mid point in the fuel passage 43. A stopper 41 for a spring 46 is coaxially connected to the casing 42. The spring 46 extends into the stopper 41. The spring 46 engages the valve member 44. The spring 46 urges the valve member 44 toward the valve seat 45 relative to the stopper 41 along the rightward direction in Fig. 4. Normally, the valve member 44 contacts the valve seat 45 so that the fuel passage 43 is blocked. Accordingly, the check valve 4a is of the normally closed type. When the electromagnetic valve 6 is opened and hence low-pressure fuel flows from the fuel reservoir 5 into the fuel passage 43, the valve member 44 is separated from the valve seat 45 by the pressure of fuel in the fuel passage 43. Thus, in this case, the check valve 4a is opened. The fuel passage 43 leads to the pressure chamber 23a via fuel passages 30a, 40a, and 50. The fuel passages 30a and 40a extend in the pump head 14. The fuel passage 50 extends in the spring stopper 41. When the check valve 4a is open, low-pressure fuel flows into the pressure chamber 23a via the fuel passage 72, the annular passage 48, the radial passages 49, and the fuel passages 43, 50, 30a, and 40a.

[0065] The check valve 4b (not shown in Fig. 4) is similar in structure to the check valve 4a. The check valve 4b is connected between the electromagnetic valve 6 and the pressure chamber 23b.

[0066] With reference to Fig. 1, the fuel reservoir 5 is filled with low-pressure fuel. Fuel in the reservoir 5 is supplied from the feed pump P1. The pressure of fuel in the reservoir 5 is equal to about 15 atm. When the electromagnetic valve 6 and the check valve 4a are open, low-pressure fuel is transmitted from the fuel reservoir 5 to the pressure chamber 23a via the fuel passages including the fuel passages 30a and 40a. The pressure chamber 23a is connected to a delivery valve 3 via a discharge port 16a extending in the pump head 14. The delivery valve 3 correspond to the discharge valves B2 in Fig. 2. The delivery valve 3 is connected to the common rail "R" via the feed pipe R1 (see Fig. 2). The plungers 21a and 21c pressurize fuel in the pressure chamber 23a into high-pressure fuel, and pump high-pressure fuel from the pressure chamber 23a to the common rail "R" via the discharge port 16a, the delivery valve 3, and the feed pipe R1. The pressure of fuel fed to the common rail "R" can depend on the operating conditions of the engine "E". For example, the pressure of fuel fed to the common rail "R" can be in the range of 200 to 1,200 atm.

[0067] The delivery valve 3 serves as check valves. The delivery valve 3 include valve balls 31a and 31b. The valve ball 31a selectively blocks and unblocks a fuel passage communicating with the discharge port 16a. The discharge port 16a is connected to the pressure chamber 23a. The valve ball 31b selectively blocks and unblocks a fuel passage communicating with a discharge port 16b (see Fig. 6). The discharge port 16b is connected to the pressure chamber 23b (see Fig. 6).

[0068] As shown in Fig. 6, the pair of the plungers 21a and 21c are disposed in the diametrical hole 2a. The pressure chamber 23a is defined between the plungers 21a and 21c. The diametrical hole 2a, the plungers 21a and 21c, and the pressure chamber 23a form the first high-pressure pumping device. Similarly, the pair of the plungers 21b and 21d are disposed in the diametrical hole 2b. The pressure chamber 23b is defined between the plungers 21b and 21d. The diametrical hole 2b, the plungers 21b and 21d, and the pressure chamber 23b form the second high-pressure pumping device. The pair of the plungers 21a and 21c in the first high-pressure pumping device, and the pair of the plungers 21b and 21d in the second high-pressure pumping device contract the pressure chambers 23a and 23b during substantially non-overlapping time periods, respectively. Thus, the pair of the plungers 21a and 21c in the first high-pressure pumping device, and the pair of the plungers 21b and 21d in the second high-pressure pumping device alternately pressurize fuel. The pressure chamber 23a is connected via the fuel passage 40a and the discharge port 16a to a section of the delivery valve 3 which includes the valve ball 31a. The pressure chamber 23b is connected via the fuel passage 40b and the discharge port 16b to a section of the delivery valve 3 which includes the valve ball 31b. When the pressure of fuel applied to the valve ball 31a exceeds the pressure of fuel in the common rail "R", the valve ball 31a separates from its seat so that high-pressure fuel flows into the common rail "R" via the opening around the valve ball 31a. When the pressure of fuel applied to the valve ball 31a is equal to or lower than the pressure of fuel in the common rail "R", the valve ball 31a contacts its seat and hence inhibits the flow of high-pressure fuel into the common rail "R". When the pressure of fuel applied to the valve ball 31b exceeds the pressure of fuel in the common rail "R", the valve ball 31b separates from its seat so that high-pressure fuel flows into the common rail "R" via the opening around the valve ball 31b. When the pressure of fuel applied to the valve ball 31b is equal to or lower than the pressure of fuel in the common rail "R", the valve ball 31b contacts its seat and hence inhibits the flow of high-pressure fuel into the common rail "R". Since the pressurizing of fuel in the pressure chamber 23a alternates with the pressurizing of fuel in the pressure chamber 23b, the feed of high-pressure fuel to the common rail "R" via the opening around the valve ball 31a alternates with the feed of high-pressure fuel to the common rail "R" via the opening around the valve ball 31b.

[0069] With reference to Fig. 7, points on the cam surfaces 81 which face and contact the plungers 21a and

21c are denoted by characters 81a and 81c respectively. In addition, points on the cam surfaces 81 which face and contact the plungers 21b and 21d are denoted by characters 81b and 81d respectively. The profile of the cam surfaces 81 is designed so that a lift (a, c) at the points 81a and 81c and a lift (b, d) at the points 81b and 81d will vary in accordance with rotation of the inner cam 8 as shown in Fig. 7.

[0070] With reference to Fig. 7, at a moment (a), the lift (a, c) starts to decrease. Thus, at the moment (a), operation of the pair of the plungers 21a and 21c enters a suction stroke. The coil 62 of the electromagnetic valve 6 is energized by the electronic control unit ECU at a timing prior to the moment (a) so that the valve member 73 of the electromagnetic valve 6 will surely move to its open position at the moment (a). When the electromagnetic valve 6 is opened, low-pressure fuel flows from the fuel reservoir 5 into the pressure chamber 23a via the fuel passages including the fuel passages 30a and 40a. At this time, the plungers 21a and 21c are forced toward the cam surfaces 81 by incoming fuel. Low-pressure fuel continues to flow into the pressure chamber 23a until the valve member 73 of the electromagnetic valve 6 falls into its closed position.

[0071] The coil 62 of the electromagnetic valve 6 is de-energized by the electronic control unit ECU, and hence the valve member 73 of the electromagnetic valve 6 falls into its closed position at a moment (b) which follows the moment (a). When the electromagnetic valve 6 is closed, the communication between the fuel reservoir 5 and the pressure chamber 23a is blocked so that the fuel feed to the pressure chamber 23a is interrupted. Then, the plungers 21a and 21c stop radial movement although the lift (a, c) continues to decrease, and the rollers 22a and 22c separate from the cam surfaces 81 of the inner cam 8.

[0072] As shown in Fig. 7, at a moment (c) which follows the moment (b), the lift (a, c) starts to increase. Thus, at the moment (c), operation of the pair of the plungers 21a and 21b changes from the suction stroke to a pressurization stroke. At a moment (d) following the moment (c), the rollers 22a and 22c contact the cam surfaces 81 of the inner cam 8. Accordingly, at the moment (d), the plungers 21a and 21c start to move inward. During the pressurization stroke, the check valve 4a remains closed. As the plungers 21a and 21c move inward, the pressure chamber 23a contracts and hence the pressure of fuel therein rises. When the pressure of fuel in the pressure chamber 23a exceeds a given level, high-pressure fuel is transmitted from the pressure chamber 23a to the common rail "R" via the discharge port 16a, the delivery valve 3, and the feed pipe R1. At a moment (f) following the moment (d), the plungers 21a and 21c reach the innermost positions, and pumping fuel to the common rail "R" terminates.

[0073] As shown in Fig. 7, at a moment (e) between the moments (d) and (f), the valve member 73 of the electromagnetic valve 6 moves to its open position, and the lift (b, d) starts to decrease. Thus, at the moment (e), operation of the pair of the plungers 21b and 21d enters a suction stroke. Then, the plungers 21b and 21d move outward, and low-pressure fuel is drawn into the pressure chamber 23b similarly to the previously-indicated operation of the combination of the plungers 21a and 21c and the pressure chamber 23a.

[0074] As shown in Fig. 7, the suction stroke in the operation of the combination of the plungers 21a and 21c and the pressure chamber 23a is offset by an angle of 90° from the suction stroke in the operation of the combination of the plungers 21b and 21d and the pressure chamber 23b. Similarly, the pressurization stroke in the operation of the combination of the plungers 21a and 21c and the pressure chamber 23a is offset by an angle of 90° from the pressurization stroke in the operation of the combination of the plungers 21b and 21d and the pressure chamber 23b.

[0075] Fig. 8(a) shows the drive torque required by the high-pressure pump of Fig. 15. In the high-pressure pump of Fig. 15, every pressurization stroke corresponds to an angular interval of about 45°. Fig. 8(b) shows the drive torque required by the high-pressure pump "P" of Fig. 1. Specifically, the solid line in Fig. 8(b) denotes a component of the required drive torque which relates to the operation of the plungers 21a and 21c, and the broken line in Fig. 8(b) denotes a component of the required drive torque which relates to the operation of the plungers 21b and 21d. The drive torque required by the high-pressure pump "P" of Fig. 1 is equal to the sum of the torque components denoted by the solid line and the broken line in Fig. 8(b). In the high-pressure pump "P" of Fig. 1, every pressurization stroke corresponds to an angular interval of about 120°. The pressurization stroke in the high-pressure pump "P" of Fig. 1 is longer than that in the high-pressure pump of Fig. 15. The torque components denoted by the solid line and the broken line in Fig. 8(b) are different in phase on a stagger basis. Accordingly, the maximum drive torque (the drive torque which occurs at the maximum fuel discharge rate) required by the high-pressure pump "P" of Fig. 1 is smaller than that required by the high-pressure pump of Fig. 15.

[0076] Fig. 12 shows the diametrical holes 2a and 2b, and adjacent members. Fig. 13 shows the shoe guide 15. As shown in Fig. 13, the shoe guide 15 has a cylindrical shape. The circumferential walls of the shoe guide 15 have a pair of grooves 151b and 151d extending from the right-hand end face of the shoe guide 15. As shown in Fig. 12, the shoes 24b and 24d are slidably supported in the grooves 151b and 151d of the shoe guide 15 respectively. Similarly, the circumferential walls of the shoe guide 15 have a pair of grooves 151a and 151c extending from the left-hand end face of the shoe guide 15. As shown in Fig. 4, the shoes 24a and 24c are slidably supported in the grooves 151a and 151c of the shoe guide 15 respectively. The grooves 151a, 151b, 151c, and 151d of the shoe guide 15 are spaced at angular intervals of 90°. As shown in Fig. 13, the shoe guide 15 has a plurality of axial holes 152. Bolts (not shown) extending through the axial holes 152 fix the shoe guide 15 to the pump head 14. The shoes 24a, 24b, 24c, and 24d are held between the shoe guide 15 and the pump head 14, or between the shoe guide 15 and the plate 7. The shoes 24a, 24b, 24c, and 24d can move radially with respect to the shoe guide 15. Movement of the shoes 24a, 24b, 24c, and 24d in the circumferential direction of the shoe guide 15 is limited by the walls of the shoe guide 15.

[0077] The axial length of the shoe guide 15 is slightly greater than the axial dimension of the grooves 151a, 151b, 151c, and 151d. The axial position range of the grooves 151b and 151d partially overlaps the axial position range of the grooves 151a and 151c. This overlap design enables a relatively short axial length of the shoe guide 15. As shown in Fig. 12, the axial position range RG1 of the rollers 22b and 22d partially overlaps the axial position range RG2 of the rollers 22a and 22c. This overlap design enables a relatively short axial length of the inner cam 8.

[0078] The inner cam 8 is common to the rollers 22a, 22b, 22c, and 22d. In other words, the inner cam 8 is common to the plungers 21a, 21b, 21c, and 21d. The shoe guide 15 is common to the shoes 24a, 24b, 24c, and 24d. In other words, the shoe guide 15 is common to the plungers 21a, 21b, 21c, and 21d. As previously indicated, the axial length of the inner cam 8 is relatively small. In addition, the axial length of the shoe guide 15 is relatively small. Thus, the axial length of the high-pressure pump "P" can be relatively small.

<u>Second Embodiment of the Invention</u>

[0079] A variable-discharge-rate high-pressure pump according to a second embodiment of this invention is similar to the variable-discharge-rate high-pressure pump of the first embodiment thereof except for design changes indicated hereinafter.

[0080] Fig. 9 shows a variable-discharge-rate high-pressure pump according to the second embodiment of this invention. The high-pressure pump of Fig. 9 includes an electromagnetic valve 6a for adjusting the rate of low-pressure fuel feed to the pressure chamber 23a in the first high-pressure pumping device, and an electromagnetic valve 6b for adjusting the rate of low-pressure fuel feed to the pressure chamber 23b in the second high-pressure pumping device. A fuel reservoir 5a extends around an end of the electromagnetic valve 6a. A fuel reservoir 5b extends around an end of the electromagnetic valve 6b. The electromagnetic valves 6a and 6b are similar in structure to the electromagnetic valve 6 in the first embodiment of this invention.

[0081] As shown in Fig. 10, at a moment (a), the lift (a, c) starts to decrease. Thus, at the moment (a), operation of the pair of the plungers 21a and 21c enters a suction stroke. The coil 62 of the electromagnetic valve 6a is energized by the electronic control unit ECU at a timing prior to the moment (a) so that the valve member 73 of the electromagnetic valve 6a will surely move to its open position at the moment (a). When the electromagnetic valve 6a is opened, low-pressure fuel flows from the fuel reservoir 5a into the pressure chamber 23a. At this time, the plungers 21a and 21c are forced toward the cam surfaces 81 by incoming fuel Low-pressure fuel continues to flow into the pressure chamber 23a until the valve member 73 of the electromagnetic valve 6a falls into its closed position.

[0082] The coil 62 of the electromagnetic valve 6a is de-energized by the electronic control unit ECU, and hence the valve member 73 of the electromagnetic valve 6a falls into its closed position at a moment (b) which follows the moment (a). When the electromagnetic valve 6a is closed, the communication between the fuel reservoir 5a and the pressure chamber 23a is blocked so that the fuel feed to the pressure chamber 23a is interrupted. Then, the plungers 21a and 21c stop radial movement although the lift (a, c) continues to decrease, and the rollers 22a and 22c separate from the cam surfaces 81 of the inner cam 8.

[0083] As shown in Fig. 10, at a moment (c) which follows the moment (b), the lift (a, c) starts to increase. Thus, at the moment (c), operation of the pair of the plungers 21a and 21c changes from the suction stroke to a pressurization stroke. At a moment (e) following the moment (c), the rollers 22a and 22c contact the cam surfaces 81 of the inner cam 8. Accordingly, at the moment (e), the plungers 21a and 21c start to move inward. During the pressurization stroke, the check valve 4a remains closed. As the plungers 21a and 21c move inward, the pressure chamber 23a contracts and hence the pressure of fuel therein rises. When the pressure of fuel in the pressure chamber 23a exceeds a given level, high-pressure fuel is transmitted from the pressure chamber 23a to the common rail "R" via the discharge port 16a, the delivery valve 3, and the feed pipe R1. At a moment (f) following the moment (e), the plungers 21a and 21 c reach the innermost positions, and pumping fuel to the common rail "R" terminates.

[0084] As shown in Fig. 10, at a moment (d) between the moments (c) and (d), the valve member 73 of the electromagnetic valve 6b moves to its open position, and the lift (b, d) starts to decrease. Thus, at the moment (d), operation of the pair of the plungers 21b and 21d enters a suction stroke. Then, the plungers 21b and 21d move outward, and low-pressure fuel is drawn into the pressure chamber 23b similarly to the previously-indicated operation of the combination of the plungers 21a and 21c and the pressure chamber 23a.

[0085] The maximum drive torque required by the high-pressure pump of Fig. 9 is smaller than that required by the high-pressure pump of Fig. 15. The high-pressure pump of Fig. 9 is advantageous over the high-pressure pump of Fig. 1 as follows.

[0086] Fig. 7 shows the operating conditions of the high-pressure pump of Fig. 1 which occur at a relatively great rate of fuel discharge. On the other hand, Fig. 11 shows the operating conditions of the high-pressure pump of Fig. 1 which occur at a relatively small rate of fuel discharge. With reference to Fig. 11, during the time interval from a moment (a) to a moment (c), the valve member 73 of the electromagnetic valve 6 remains in its open position, and the fuel feed to the pressure chamber 23b is implemented while the plungers 21b and 21d move outward. In this case, since the fuel reservoir 5 and the pressure chamber 23a are also in communication with each other, the plungers 21a and 21c tend to be moved outward and hence the fuel feed to the pressure chamber 23a tends to be also implemented during the time interval from the moment (a) to a moment (b). The outward movement of the plungers 21a and 21c during the time interval between the moments (a) and (b) does not respond to the profile of the cam surfaces 81. Accordingly, there is a chance of reducing the accuracy of the fuel discharge rate control in the high-pressure pump of Fig. 1. On the other hand, in the high-pressure pump of Fig. 9, during the fuel feed to the pressure chamber 23b, the electromagnetic valve 6a inhibits the fuel feed to the pressure chamber 23a. In addition, during the fuel feed to the pressure chamber 23a, the electromagnetic valve 6b inhibits the fuel feed to the pressure chamber 23b. Therefore, in the high-pressure pump of Fig. 9, the accuracy of the fuel discharge rate control can be maintained at an acceptable level even when the fuel discharge rate is relatively small.

[0087] An object of this invention is to reduce the drive torque required by a variable-discharge-rate high-pressure pump. The reduction in the drive torque enhances the durability of a timing belt used in a mechanism for power transmission to the high-pressure pump. The high-pressure pump has a plurality of cylinders (2a, 2b). A pair of plungers (21a, 21c) are disposed in the cylinder (2a). A pair of plungers (21b, 21d) are disposed in the cylinder (2b). A pressure chamber (23a) is defined between opposing end surfaces of the plungers (21a, 21c). A pressure chamber (23b) is defined between opposing end surfaces of the plungers (21b, 21d). Low-pressure fuel which is metered by an electromagnetic valve (6) for flow rate control is sequentially fed to the pressure chambers (23a, 23b). An inner cam (8) is provided which drives the plungers (21a, 21b, 21c, 21d). The inner cam (8) is common to the plungers (21a, 21b, 21c, 21d). Fuel pressurization and fuel pumping are alternately implemented in the pressure chambers (23a, 23b) in accordance with rotation of the inner cam (8). According to the alternate fuel pressurization and fuel pumping, the peak value of the drive torque required by the high-pressure pump can be reduced.

Claims

1. A variable-discharge-rate high-pressure pump comprising a plunger movably disposed in a cylinder, a cam for reciprocating the plunger in the cylinder, a pressure chamber defined by an inner wall surface in the cylinder and an end surface of the plunger for receiving low-pressure fuel from a low-pressure fuel passage and pressurizing low-pressure fuel in accordance with reciprocation of the plunger, and a means for transmitting pressurized fuel from the pressure chamber to a high-pressure fuel passage, characterized in that:

   there are provided a plurality of pressure chambers, and a means for feeding low-pressure fuel to the pressure chambers; and
   the cam is common to the pressure chambers, and the cam causes fuel pressurizations and fuel pumpings in the pressure chambers to alternate with each other.

2. A variable-discharge-rate high-pressure pump according to claim 1, characterized in that the feeding means comprises an electromagnetic valve for adjusting rates of fuel feed to the pressure chambers, and check valves provided between the electromagnetic valve and the pressure chambers for allowing fuel flow only in directions from the low-pressure fuel passage to the pressure chambers.

3. A variable-discharge-rate high-pressure pump according to claim 1 or 2, characterized in that there are provided a plurality of cylinders, axes of the cylinders are separate from each other in an axial direction of a drive shaft to which the cam is connected, two plungers are slidably disposed in each of the cylinders, and the pressure chambers are formed between opposing end surfaces of the plungers.

4. A variable-discharge-rate high-pressure pump according to claim 2, characterized in that there are provided a plurality of electromagnetic valves which correspond to the pressure chambers respectively.

5. A variable-discharge-rate high-pressure pump according to claim 3, characterized in that the number of the cylinders is two, and the axes of the cylinders are perpendicular to each other.

6. A variable-discharge-rate high-pressure pump according to claim 5, characterized in that the cam has a cylindrical shape, the cam has inner circumferential surfaces formed with projections opposed to each other, and the cylinders are in a recess of the cam.

7. A variable-discharge-rate high-pressure pump comprising:

   first and second cylinders;

a first plunger movably disposed in the first cylinder and defining a part of a first pressure chamber;

a second plunger movably disposed in the second cylinder and defining a part of a second pressure chamber;

means for feeding fuel to the first and second pressure chambers;

means for reciprocating the first and second plungers relative to the first and second cylinders to periodically contract the first and second pressure chambers and to periodically pressurize fuel in the first and second pressure chambers; and

wherein the reciprocation of the first plunger differs in phase from the reciprocation of the second plunger so that a peak value of a mechanical power required to reciprocate the first and second plungers can be reduced.

8. A variable-discharge-rate high-pressure pump comprising:

first pumping means including a first cylinder, a first plunger disposed in the first cylinder and being able to reciprocate, and a first pressure chamber defined by an inner wall surface of the first cylinder and an end surface of the first plunger and pressurizing low-pressure fuel in accordance with reciprocation of the first cylinder;

second pumping means including a second cylinder, a second plunger disposed in the second cylinder and being able to reciprocate, and a second pressure chamber defined by an inner wall surface of the second cylinder and an end surface of the second plunger and pressurizing low-pressure fuel in accordance with reciprocation of the second cylinder; and

a cam for sequentially reciprocating the first plunger and the second plunger;

wherein the cam sequentially drives the first plunger and the second plunger so that fuel will be sequentially pumped from the first pressure chamber and the second pressure chamber into a pressure accumulation chamber.

9. A variable-discharge-rate high-pressure pump according to claim 8, wherein the cam has a common cam surface for driving the first plunger and the second plunger.

10. A variable-discharge-rate high-pressure pump according to claim 9, wherein the cam comprises an inner cam having an inner circumferential surface forming the cam surface.

11. A variable-discharge-rate high-pressure pump according to one of claims 8 to 10, wherein low-pressure fuel is introduced into the first pressure chamber and the second pressure chamber from a low-pressure passage, an electromagnetic valve is provided which adjusts a rate of low-pressure fuel flow into the first pressure chamber and the second pressure chamber, a first check valve is provided between the electromagnetic valve and the first pressure chamber to inhibit fuel return from the first pressure chamber toward the low-pressure passage, and a second check valve is provided between the electromagnetic valve and the second pressure chamber to inhibit fuel return from the second pressure chamber toward the low-pressure passage.

12. A variable-discharge-rate high-pressure pump according to claim 11, wherein the electromagnetic valve includes a plurality of sub electromagnetic valves corresponding to the first pumping means and the second pumping means respectively.

13. A variable-discharge-rate high-pressure pump according to one of claims 8 to 12, wherein axes of the first cylinder and the second cylinder are separate from each other in an axial direction of a drive shaft to which the cam is connected.

14. A variable-discharge-rate high-pressure pump according to claim 13, wherein the axes of the first cylinder and the second cylinder are perpendicular to each other.

15. A variable-discharge-rate high-pressure pump according to one of claims 9 to 14, wherein the first plunger includes a pair of plungers, the second plunger includes a pair of plungers, and the cam surface of the cam has two projections each corresponding to the pair of the plungers.

16. A variable-discharge-rate high-pressure pump according to one of claims 9 to 15, wherein the cam surface of the cam has a first region for driving the first plunger and a second region for driving the second plunger, and the first region and the second region partially overlap each other.

FIG. 1

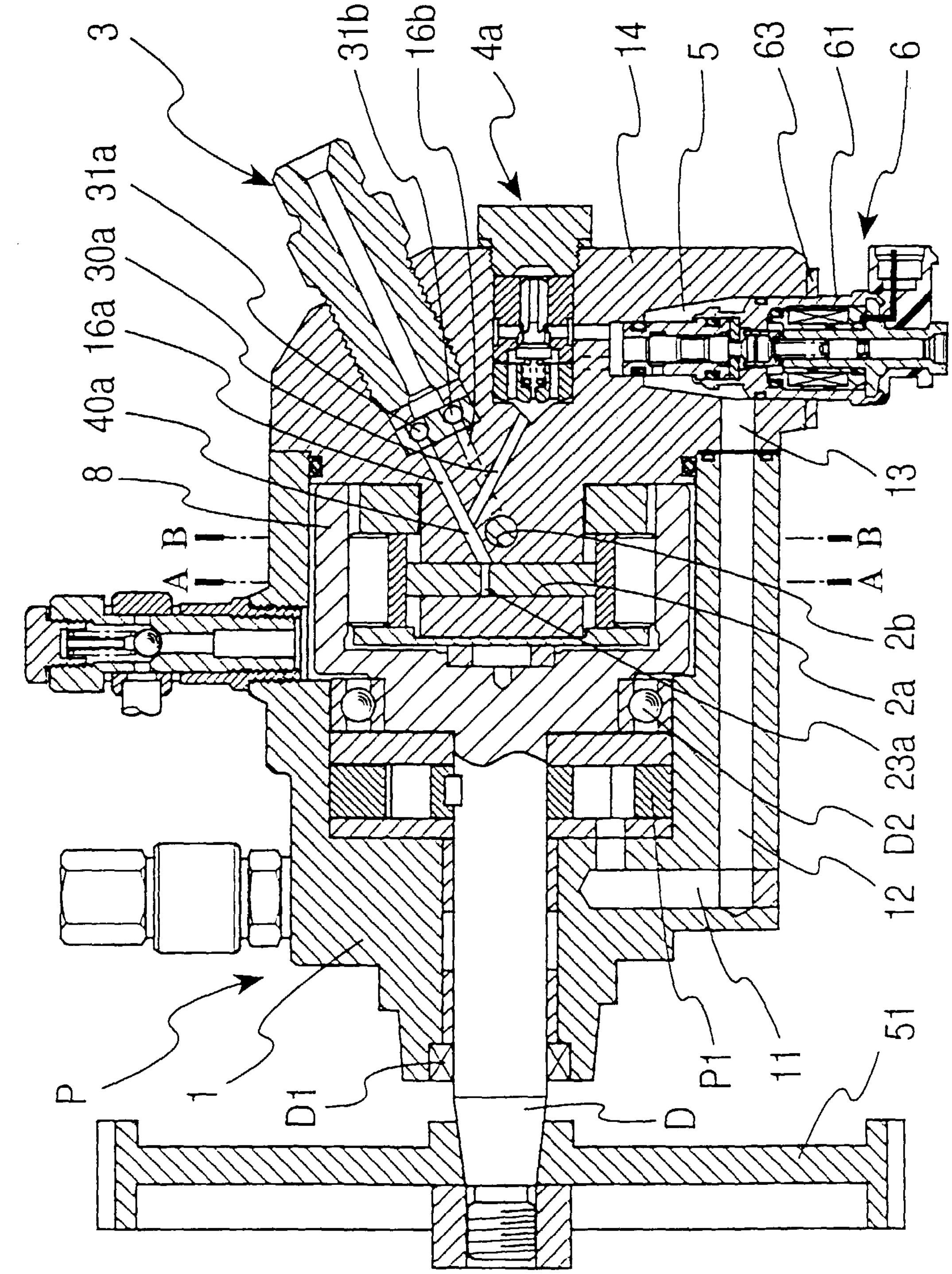

3
31b
16b
4a
14
5
63
61
6
31a
30a
16a
40a
8
A
B
A
B
13
2b
2a
23a
D2
12
P
1
D1
D
P1
11
51

FIG. 2

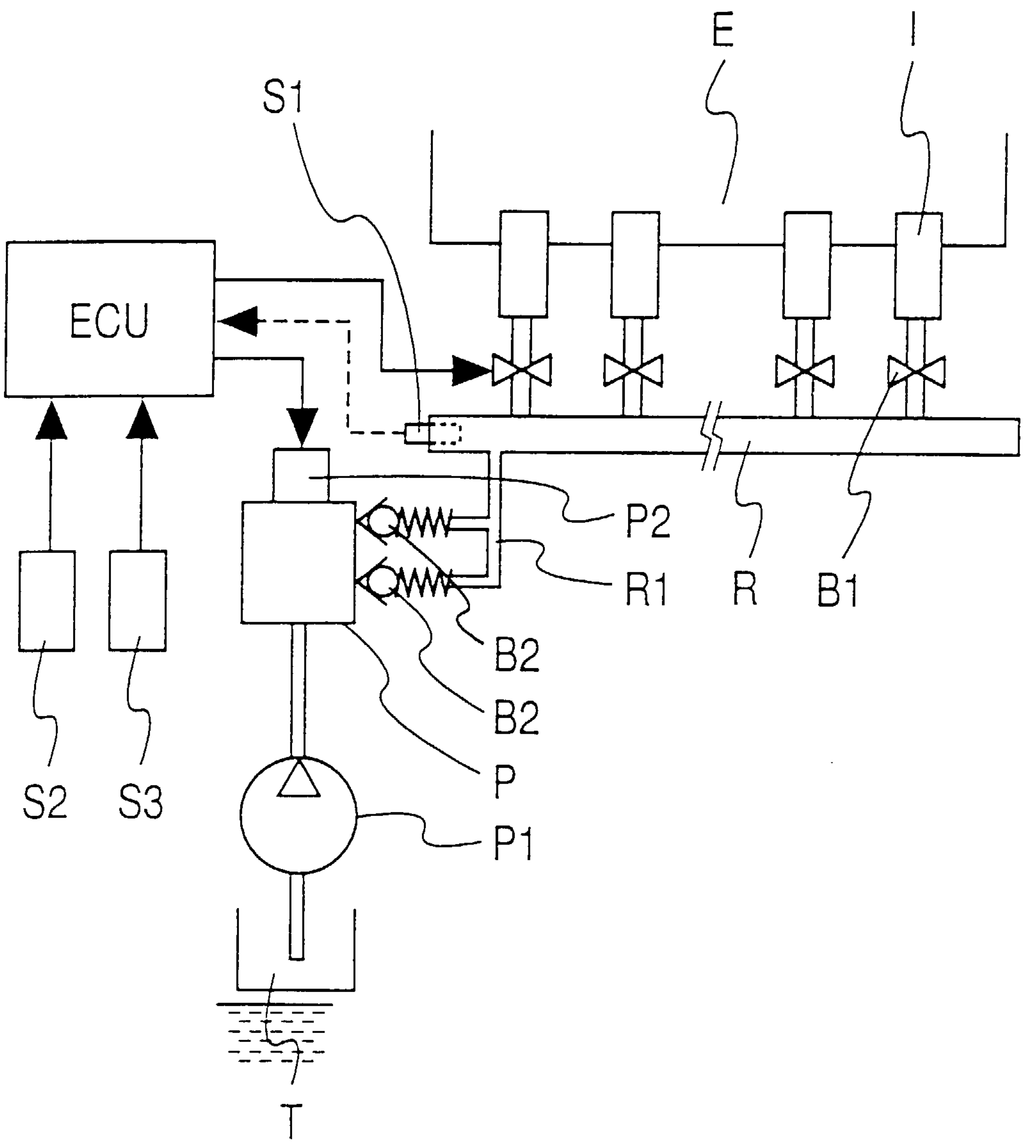


FIG. 3

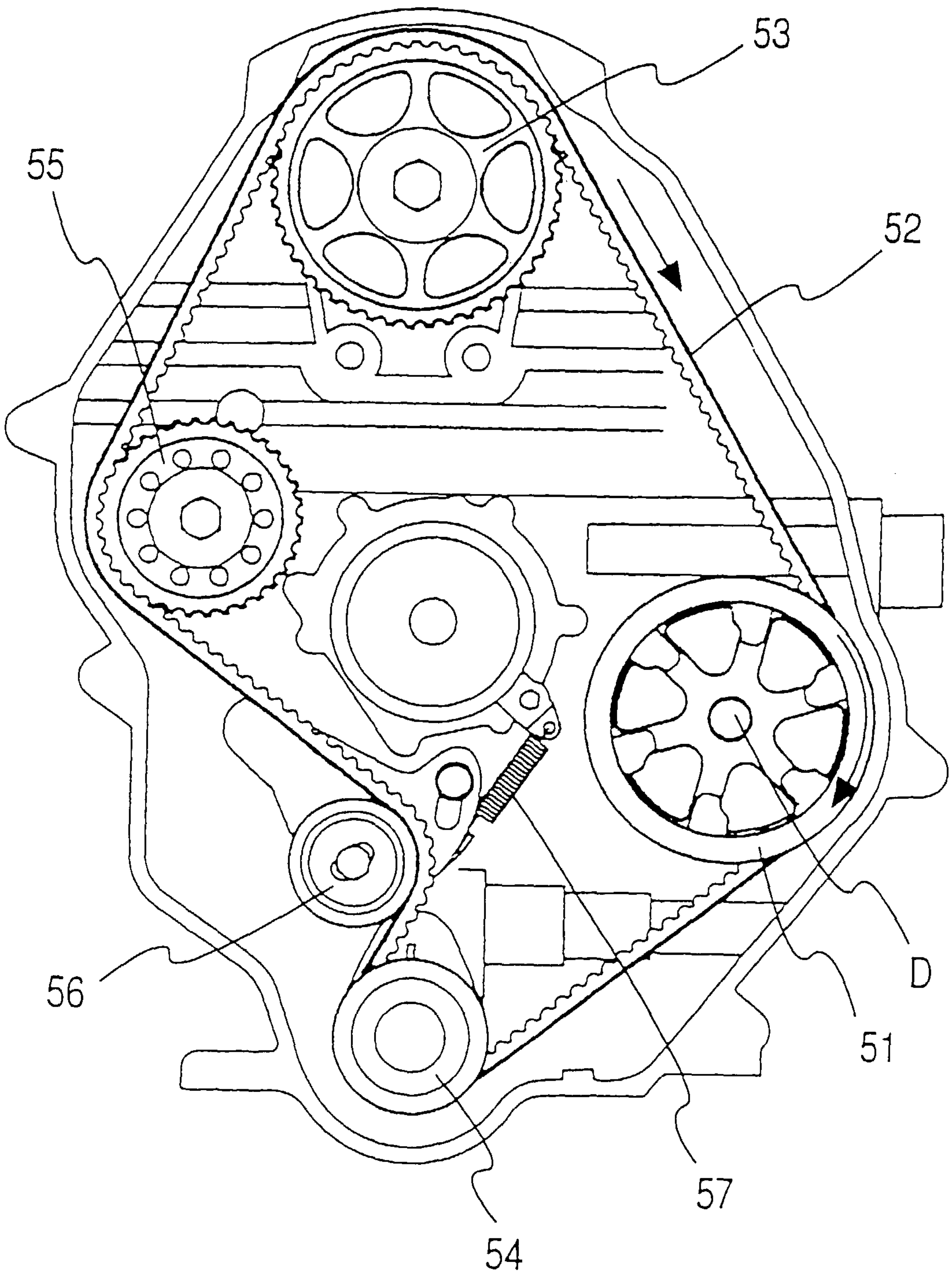
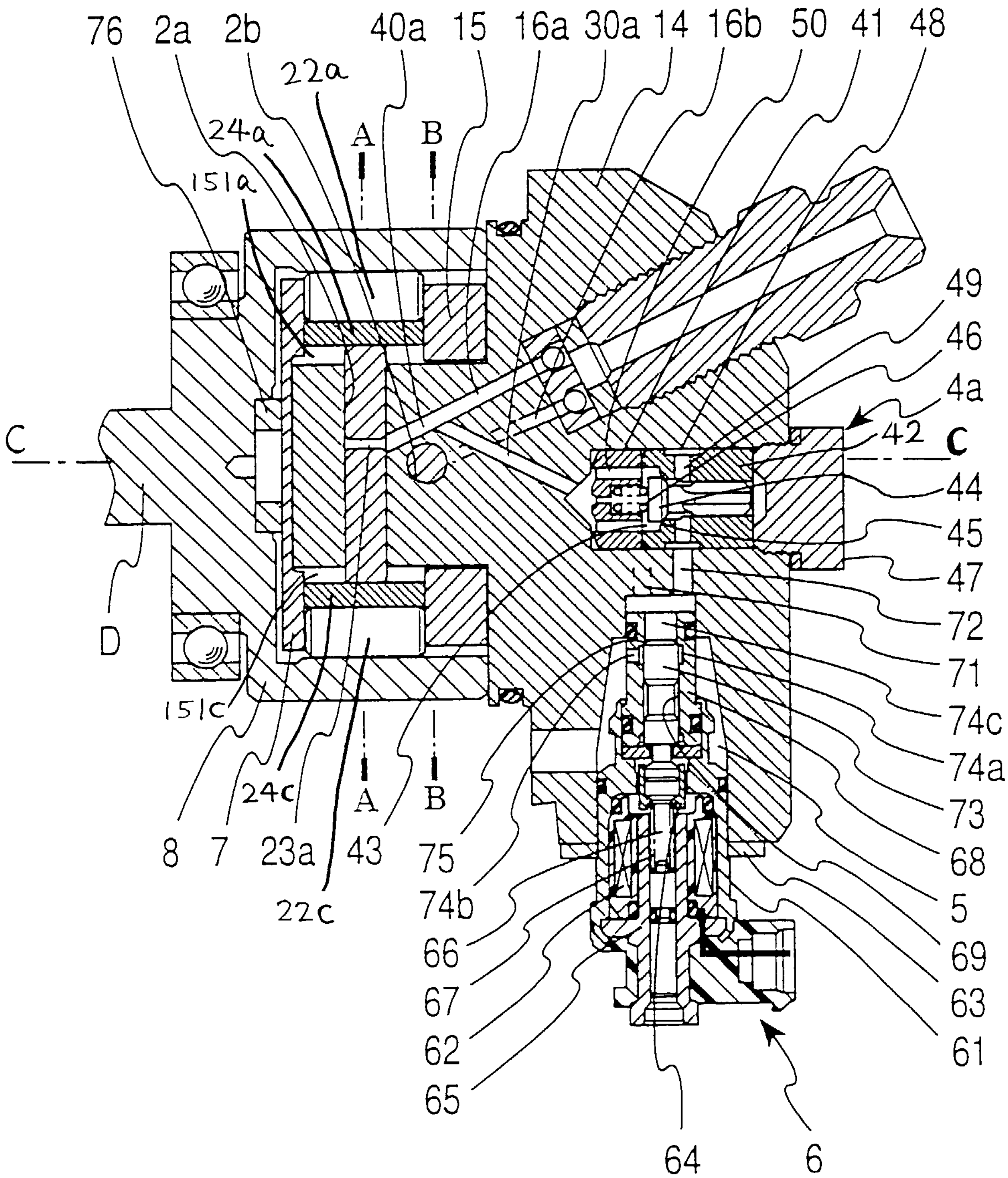


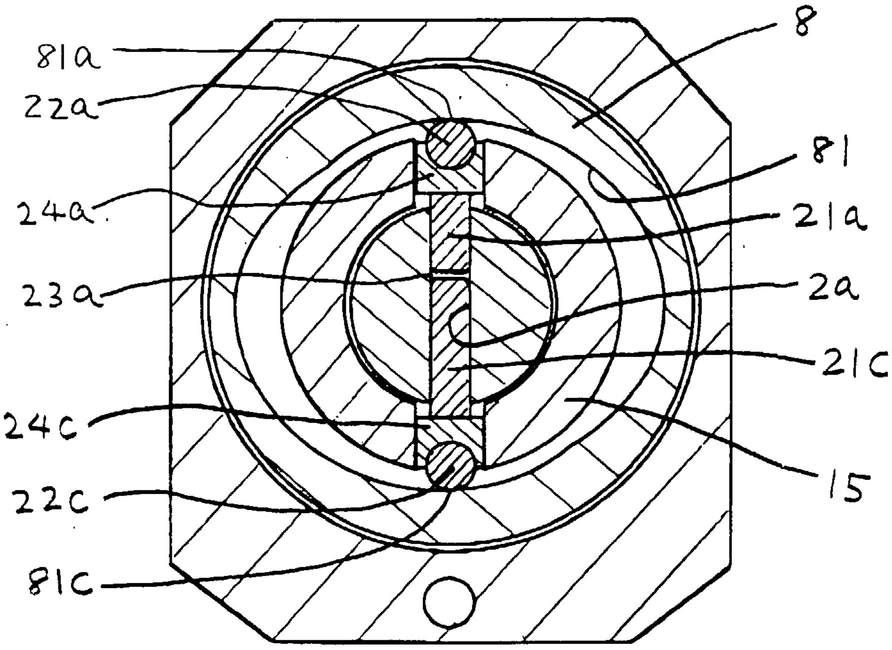
FIG. 4



76
2a
2b
22a
24a
151a
A
B
40a
15
16a
30a
14
16b
50
41
48
49
46
4a
42
C
C
44
45
47
72
71
74c
74a
73
68
5
69
63
61
D
151c
8
7
24c
23a
22c
43
75
74b
66
67
62
65
64
6

FIG. 5

(A)


8
81a
22a
81
24a
21a
23a
2a
21c
24c
15
22c
81c

(B)

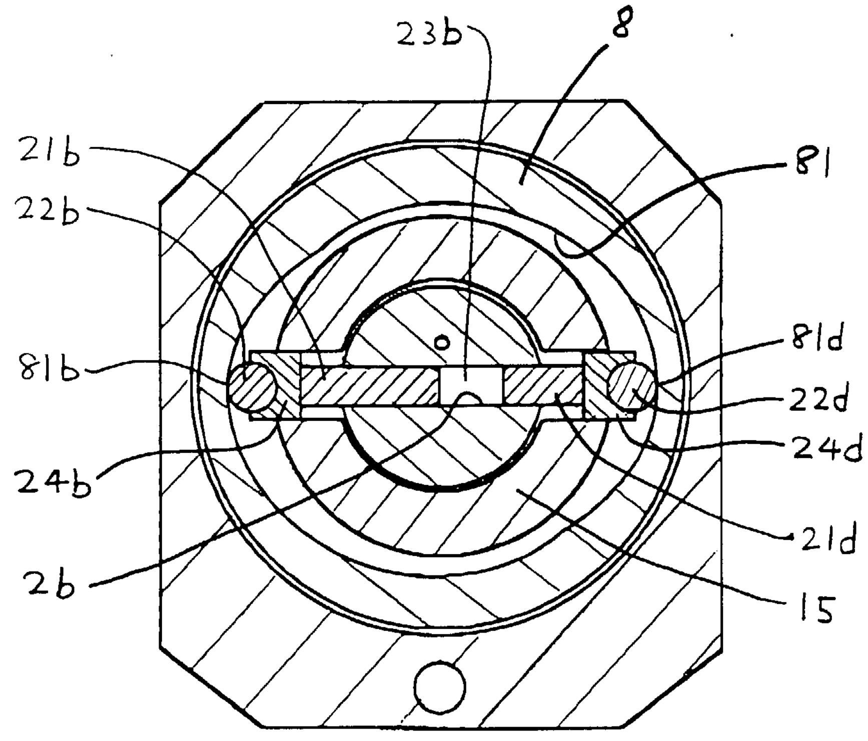

23b
8
21b
81
22b
81d
81b
22d
24d
24b
21d
2b
15

FIG. 6

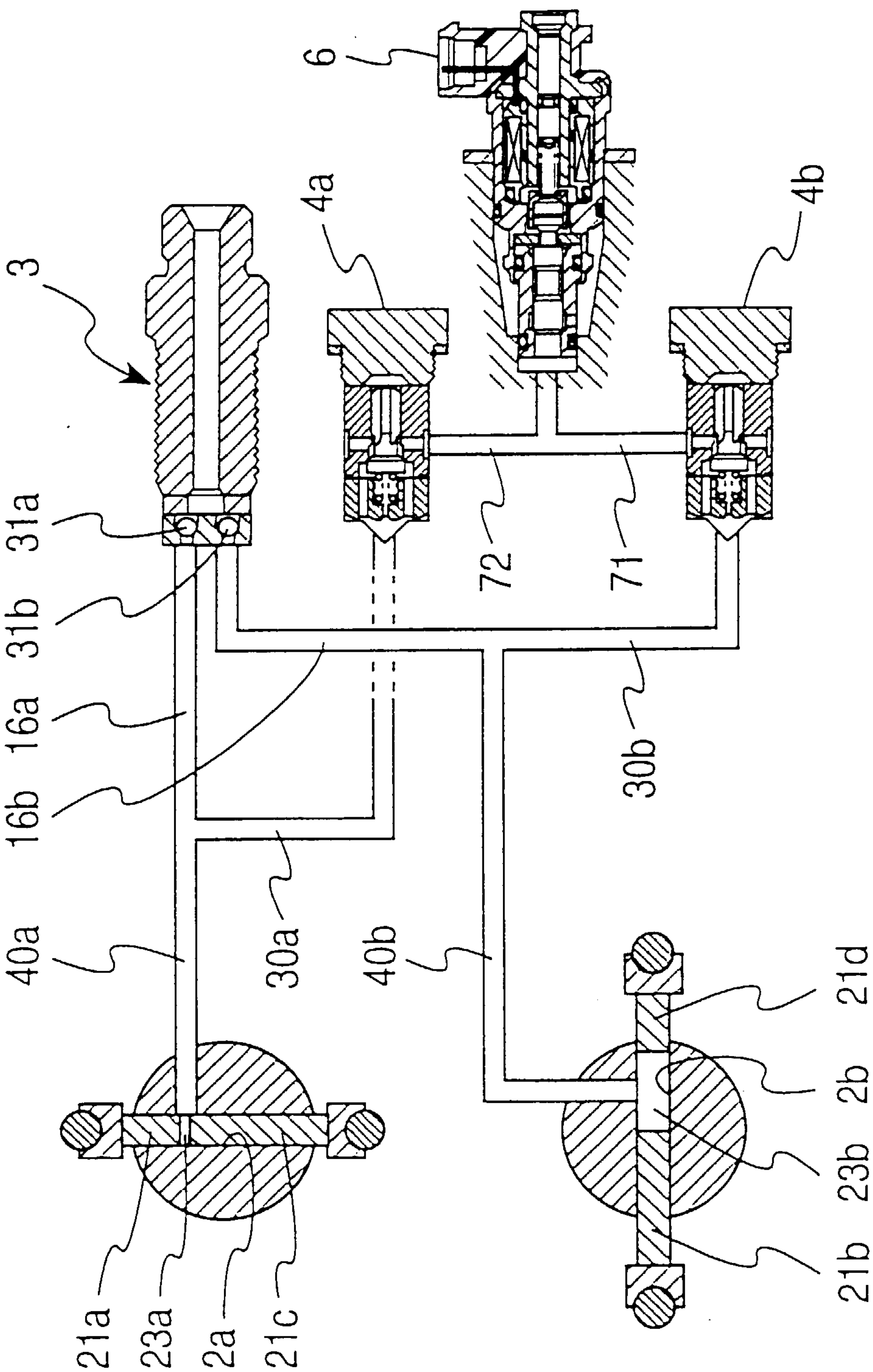
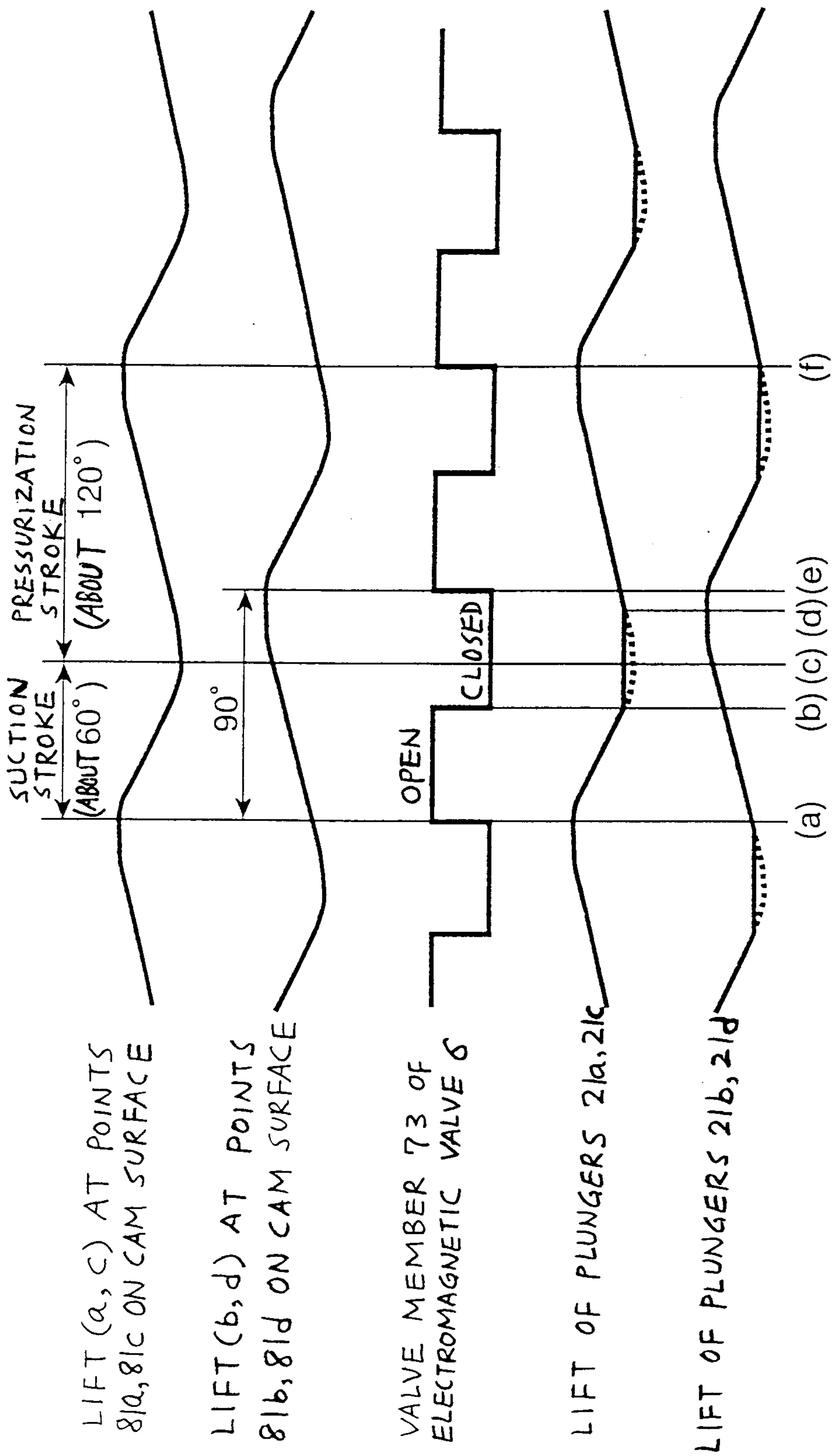

3
31a
31b
16a
16b
40a
30a
40b
21a
23a
2a
21c
6
4a
72
71
30b
4b
21d
2b
23b
21b

FIG. 7


LIFT (a, c) AT POINTS 81a, 81c ON CAM SURFACE
LIFT (b, d) AT POINTS 81b, 81d ON CAM SURFACE
VALVE MEMBER 73 OF ELECTROMAGNETIC VALVE 6
LIFT OF PLUNGERS 21a, 21c
LIFT OF PLUNGERS 21b, 21d
SUCTION STROKE (ABOUT 60°)
PRESSURIZATION STROKE (ABOUT 120°)
90°
OPEN
CLOSED
(a)
(b) (c) (d) (e)
(f)

F I G. 8

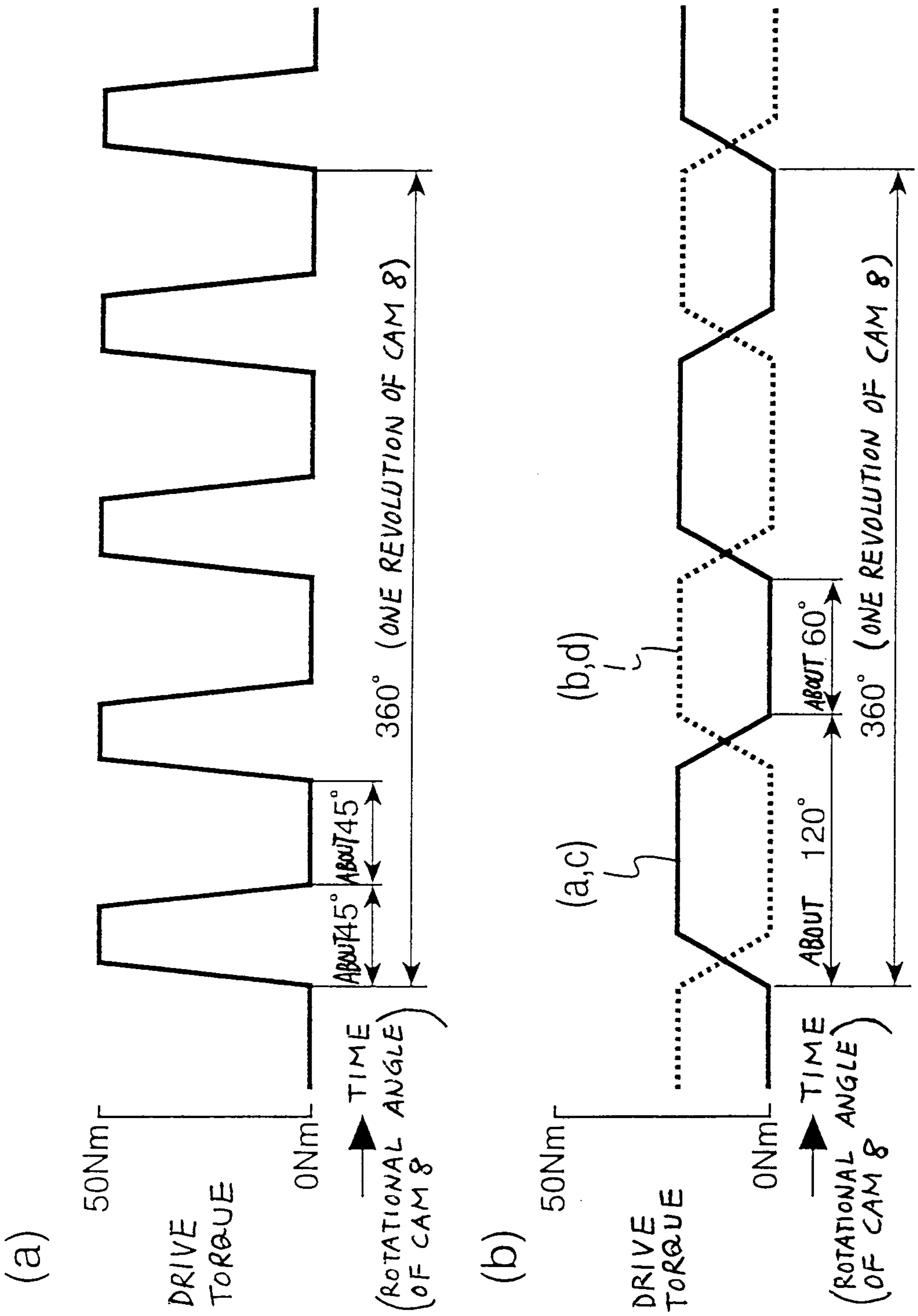

(a)
DRIVE TORQUE
50Nm
0Nm
TIME
(ROTATIONAL ANGLE OF CAM 8)
ABOUT 45°
ABOUT 45°
360° (ONE REVOLUTION OF CAM 8)
(b)
DRIVE TORQUE
50Nm
0Nm
TIME
(ROTATIONAL ANGLE OF CAM 8)
(a,c)
(b,d)
ABOUT 120°
ABOUT 60°
360° (ONE REVOLUTION OF CAM 8)

FIG. 9

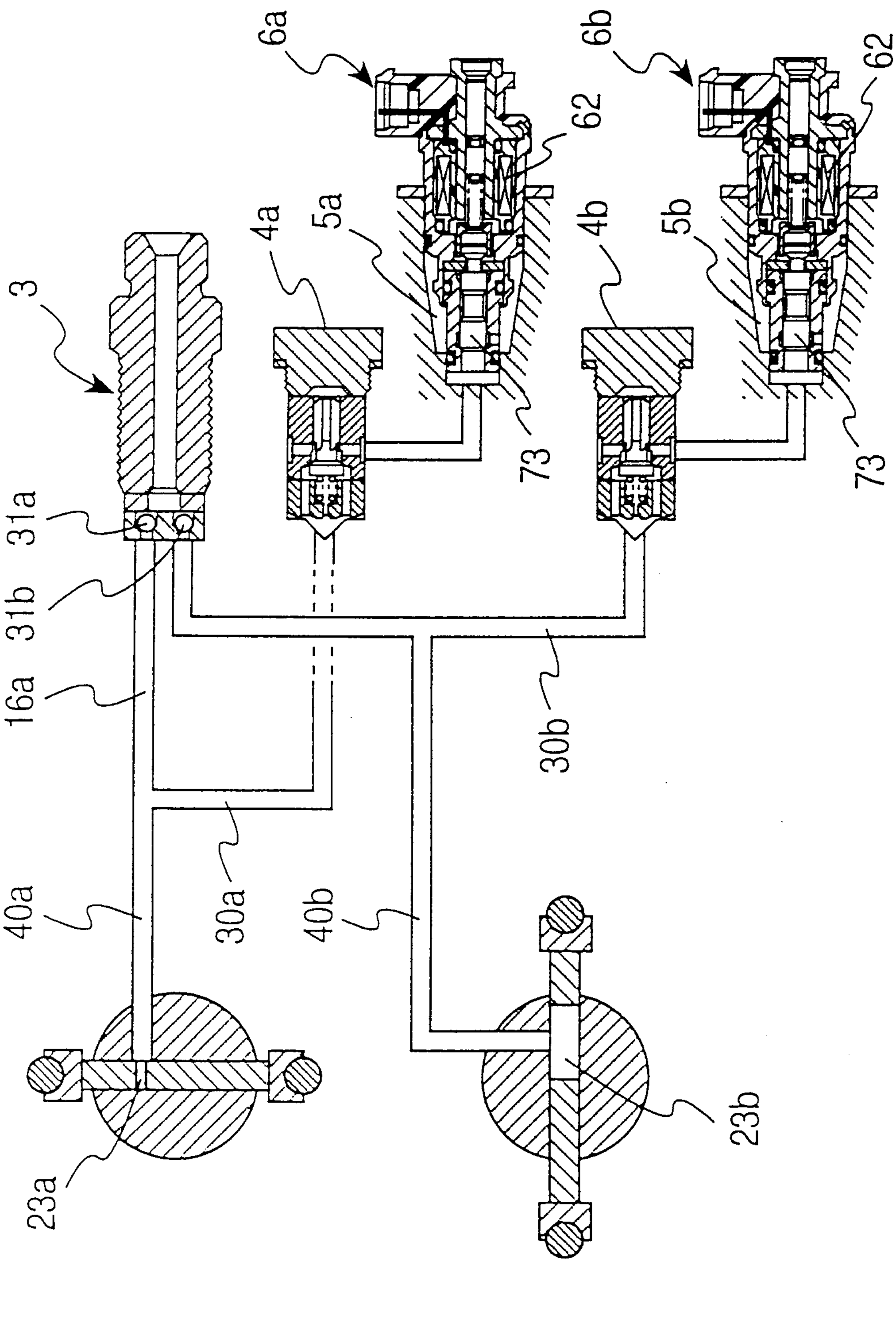


FIG. 10

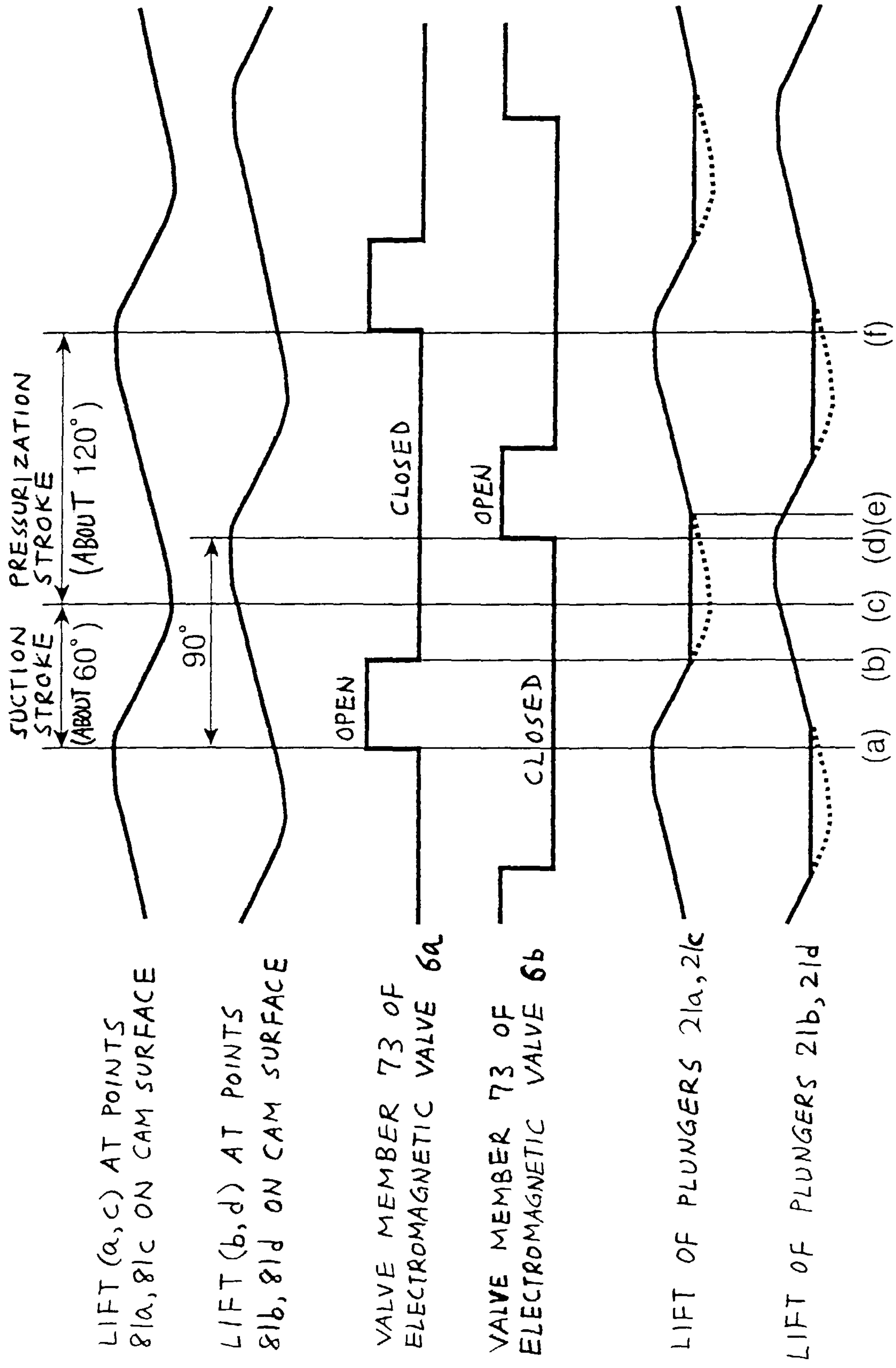

SUCTION STROKE (ABOUT 60°)
PRESSURIZATION STROKE (ABOUT 120°)
90°
OPEN
CLOSED
CLOSED
OPEN
(a)
(b)
(c)
(d)(e)
(f)
LIFT (a, c) AT POINTS 81a, 81c ON CAM SURFACE
LIFT (b, d) AT POINTS 81b, 81d ON CAM SURFACE
VALVE MEMBER 73 OF ELECTROMAGNETIC VALVE 6a
VALVE MEMBER 73 OF ELECTROMAGNETIC VALVE 6b
LIFT OF PLUNGERS 21a, 21c
LIFT OF PLUNGERS 21b, 21d

FIG. 11

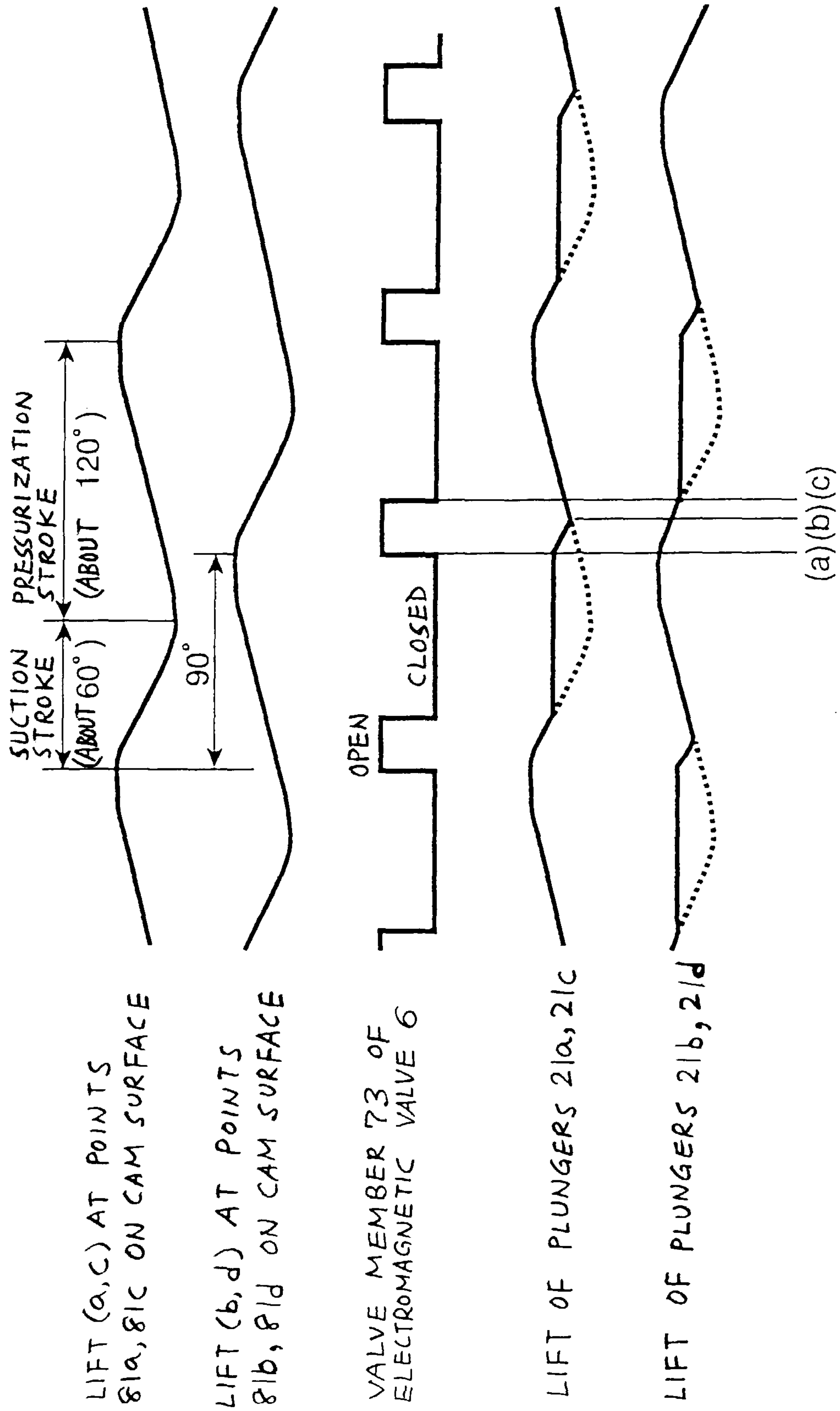

LIFT (a, c) AT POINTS
81a, 81c ON CAM SURFACE
LIFT (b, d) AT POINTS
81b, 81d ON CAM SURFACE
VALVE MEMBER 73 OF
ELECTROMAGNETIC VALVE 6
LIFT OF PLUNGERS 21a, 21c
LIFT OF PLUNGERS 21b, 21d
SUCTION
STROKE
(ABOUT 60°)
PRESSURIZATION
STROKE
(ABOUT 120°)
90°
OPEN
CLOSED
(a)(b)(c)

FIG. 12

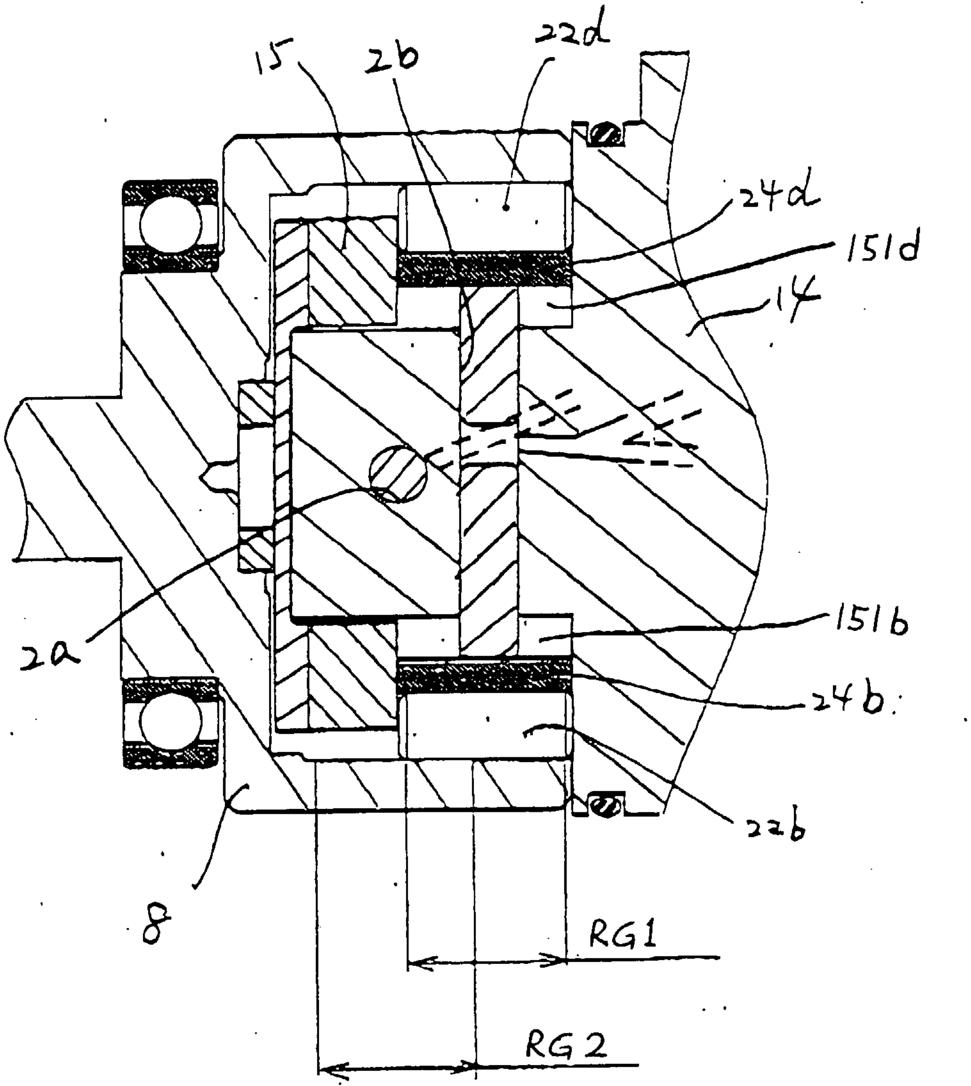


FIG. 13

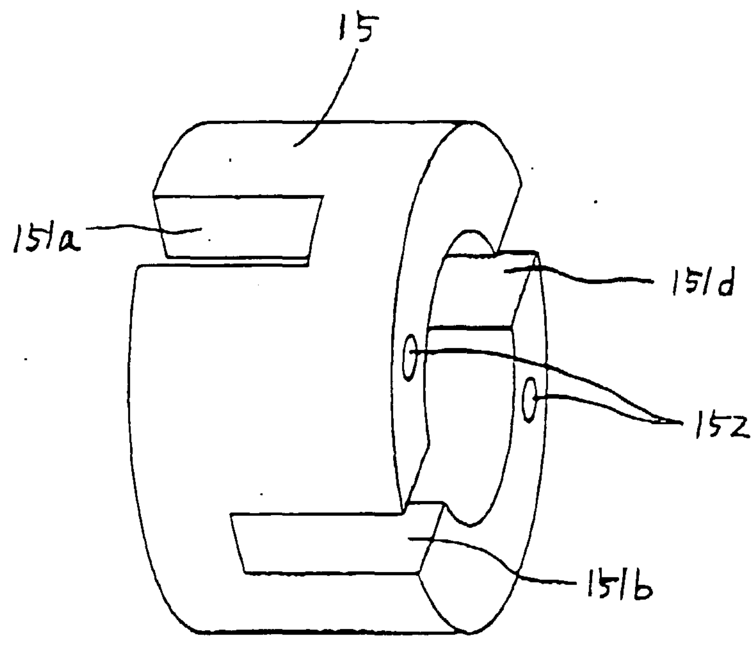
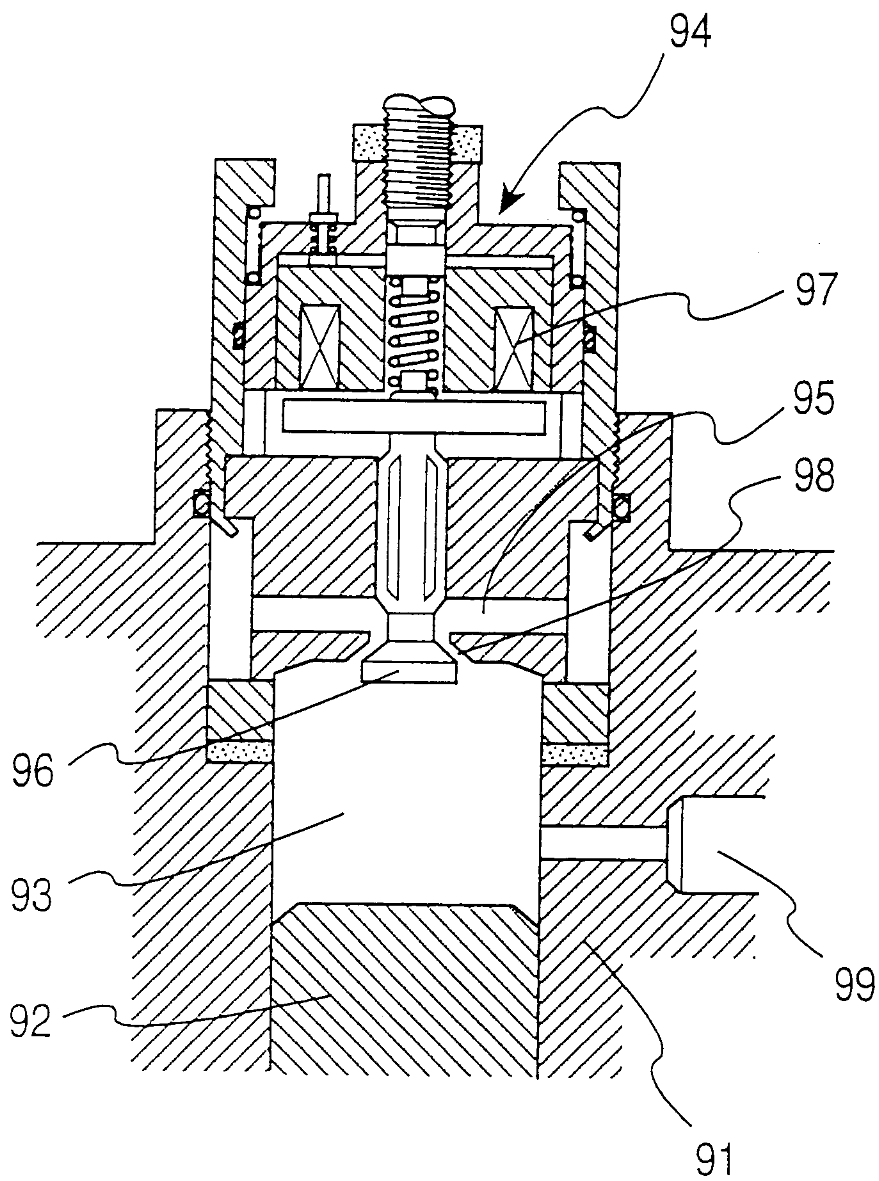


FIG. 14



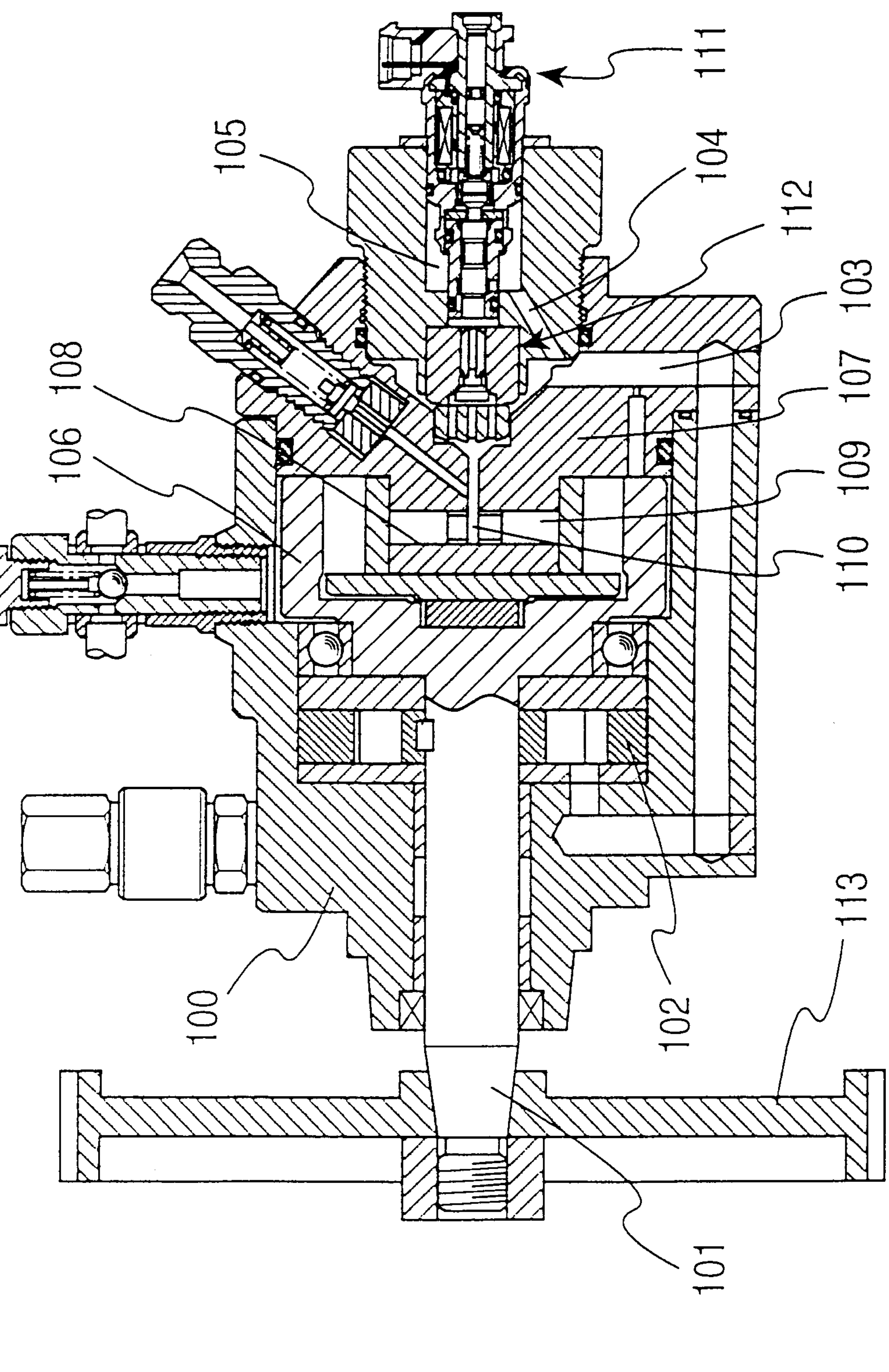


FIG. 15