

(19)



(11)

EP 3 135 899 B1

(12)

EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention
of the grant of the patent:

28.10.2020 Bulletin 2020/44

(51) Int Cl.:

F02M 59/44 (2006.01) **F02M 59/04** (2006.01)
F02M 59/48 (2006.01) **F04B 1/04** (2020.01)
F04B 53/16 (2006.01)

(21) Application number: **15782715.5**

(86) International application number:

PCT/JP2015/061774

(22) Date of filing: **17.04.2015**

(87) International publication number:

WO 2015/163243 (29.10.2015 Gazette 2015/43)

(54) **HIGH-PRESSURE FUEL PUMP**

HOCHDRUCKBRENNSTOFFPUMPE

POMPE À CARBURANT À HAUTE PRESSION

(84) Designated Contracting States:

**AL AT BE BG CH CY CZ DE DK EE ES FI FR GB
GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO
PL PT RO RS SE SI SK SM TR**

(30) Priority: **25.04.2014 JP 2014090821**

(43) Date of publication of application:

01.03.2017 Bulletin 2017/09

(73) Proprietor: **Hitachi Automotive Systems, Ltd.**

Hitachinaka-shi, Ibaraki 312-8503 (JP)

(72) Inventors:

- **SUGANAMI Masayuki**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **YAMADA Hiroyuki**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **USUI Satoshi**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **TOKUO Kenichirou**
Hitachinaka-shi
Ibaraki 312-8503 (JP)

- **SAITO Atsuji**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **YAGAI Masamichi**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **SASO Yuta**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **KOBAYASHI Masayuki**
Hitachinaka-shi
Ibaraki 312-8503 (JP)
- **GUNJI Kenichi**
Hitachinaka-shi
Ibaraki 312-8503 (JP)

(74) Representative: **MERH-IP Matias Erny Reichl**

**Hoffmann
Patentanwälte PartG mbB
Paul-Heyse-Strasse 29
80336 München (DE)**

(56) References cited:

**EP-A2- 2 055 934 WO-A1-2013/080253
JP-A- H06 249 133 JP-A- 2001 295 729
JP-A- 2009 185 613 JP-A- 2009 185 613
JP-A- 2012 211 558 US-A1- 2012 195 780**

Note: Within nine months of the publication of the mention of the grant of the European patent in the European Patent Bulletin, any person may give notice to the European Patent Office of opposition to that patent, in accordance with the Implementing Regulations. Notice of opposition shall not be deemed to have been filed until the opposition fee has been paid. (Art. 99(1) European Patent Convention).

EP 3 135 899 B1

Description

Technical Field

[0001] The present invention relates to the structure of a cylinder of a high-pressure fuel supply pump for an internal-combustion engine of a vehicle.

Background Art

[0002] High-pressure fuel supply pumps that increase the pressure of the fuel are widely used for direct-injection internal-combustion engines in which the fuel is directly injected to the inside of the combustion chamber among internal-combustion engines, for example, of vehicles.

[0003] JP 5178676 B2 describes a high-pressure fuel supply pump having a structure in which the outer periphery of a cylinder is held by a cylindrical fitting part of a cylinder holder, a screw thread put around the outer periphery of the cylinder holder is screwed in a thread put around the pump body so that an end surface of the cylinder is adhered to the pump body and the other end surface of the cylinder is adhered to the pump body, and the cylinder is fixed to the pump body (see PTL 1). PTL 2 relates to a high pressure pump, in which a cylinder is configured into a bottomed tubular form and includes an inner peripheral wall, an inner bottom wall, an outer peripheral wall, an intake hole, and a discharge hole.

[0004] US2012195780 A1 discloses another high-pressure fuel supply pump.

Citation List

Patent Literature

[0005]

PTL 1: JP 517867 B2

PTL 2: US 2012/0195780 A1

Summary of Invention

Technical Problem

[0006] However, in the conventional technique, the cylinder is screwed and fastened to the pump body through the cylinder A holder. Thus, the screw thread needs to have the tightening axial torque that withstands the fuel pressure required for the internal-combustion engine. Recently, in order to deal with environmental regulations, there is the increasing demand for increasing the pressure of the fuel in a direct-injection internal-combustion engine in which the fuel is directly injected to the inside of the combustion chamber among internal-combustion engines of vehicles. In the conventional technique, in order to withstand a higher pressure of the fuel, it is necessary to increase the tightening axial torque of the screw thread and then fix the cylinder to the pump body. This

leads to the increase in size of the screw thread, and thus leads to the increase in size of the pump body. The production cost is increased. The restrictions on attachment of the pump body to the internal-combustion engine are increased. These may decrease the marketability of the pump.

[0007] An objective of the invention is to provide a high-pressure fuel supply pump in which the cylinder can be fixed to the pump body with a simple structure even when the fuel pressure is high. As a result, the size and cost of the pump body can be reduced.

Solution to Problem

[0008] The objective of the present invention can be achieved by a structure in which the cylinder is formed in a cylindrical shape with a bottom, and the cylinder includes a large-diameter part and a small-diameter part so that the surface of the width difference formed between the large-diameter part and the small-diameter part is press-fitted to the pump body in the pressurizing direction in which the plunger is pressurized.

Advantageous Effects of Invention

[0009] According to the invention having the structure described above, the width difference formed between the large-diameter part and small-diameter part of the cylinder is more strongly pressed in a direction in which the surface of the width difference is press-fitted to the pump body in a pressurizing process in which the maximum force acts on the cylinder. In other words, it is unnecessary to consider that the cylinder drops from the pump body due to the pressurizing force. As a result, less fixing force is required in order to fix the cylinder to the pump body. In other words, the cylinder can be fixed to the pump body with a simply structure. As a result, the size and cost of the pump body can be reduced.

Brief Description of Drawings

[0010]

[FIG. 1] FIG. 1 is a vertical cross-sectional view of the whole of a high-pressure fuel supply pump of a first embodiment in which the present invention is implemented.

[FIG. 2] FIG. 2 is a vertical cross-sectional view of the whole of the high-pressure fuel supply pump of the first embodiment in which the present invention is implemented, viewed from another angle, and a cross-sectional view taken along the axis of an intake joint.

[FIG. 3] FIG. 3 is a horizontal cross-sectional view of the whole of the high-pressure fuel supply pump of the first embodiment in which the present invention is implemented, and a cross-sectional view taken along the axis of a fuel discharge outlet.

[FIG. 4] FIG. 4 is a view of the whole configuration of a system.

[FIG. 5] FIG. 5 illustrates the detailed shape of a part including a circular protrusion.

[FIG. 6] FIG. 6 illustrates another embodiment of the circular protrusion.

[FIG. 7] FIG. 7 is a vertical cross-sectional view of the whole of a high-pressure fuel supply pump of a second embodiment in which the present invention is implemented.

[FIGS. 8(a) to 8(c)] FIGS. 8(a) to 8(c) illustrate embodiments in which a cylinder is fixed to a pump body with a ring.

Description of Embodiments

[0011] Hereinafter, an embodiment according to the present invention will be described.

First Embodiment

[0012] The configuration and operation of a system will be described with reference to the view of the whole configuration of the system illustrated in FIG. 4.

[0013] A part surrounded by a dashed line is the body of a high-pressure fuel supply pump (hereinafter, referred to as a high-pressure pump). The mechanism and parts in the dashed line are integrally embedded in a high-pressure pump body 1. The fuel in a fuel tank 20 is pumped up by a feed pump 21, and fed via an intake pipe 28 to an intake joint 10a of the pump body 1.

[0014] After passing through the intake joint 10a, the fuel passes through a pressure pulsation reducing mechanism 9, and an intake path 10b, and reaches an intake port 30a of an electromagnetic inlet valve 30 included in a flow rate control mechanism. The pulsation preventing mechanism 9 will be described below.

[0015] The electromagnetic inlet valve 30 includes an electromagnetic coil 308. When the electromagnetic coil 308 does not conduct electricity, the difference between the biasing force of an anchor spring 303 and the biasing force of a valve spring 304 biases an inlet valve body 301 in a valve-opening direction in which the inlet valve body 301 is opened, and this opens the intake opening 30d. Note that the biasing force of the anchor spring 303 and the biasing force of the valve spring 304 are set so that the biasing force of the anchor spring 303 > the biasing force of the valve spring 304 holds.

[0016] When the electromagnetic coil 308 conducts electricity, a state in which an anchor 305 is moved to the left side of FIG. 4 and the anchor spring 303 is compressed is maintained. An inlet valve body 301 with which the tip of an electromagnetic plunger 305 coaxially has contact seals the intake opening 30d connected to a pressurizing chamber 11 of the high-pressure pump using the biasing force of the valve spring 304.

[0017] The operation of the high-pressure pump will

be described hereinafter.

[0018] When the rotation of a cam described below displaces a plunger 2 downward in FIG. 4 and the plunger 2 is in an intake process, the volume of the pressurizing chamber 11 is increased and the fuel pressure in the pressurizing chamber 11 is decreased. In the intake process, when the fuel pressure in the pressurizing chamber 11 is reduced to a pressure lower than the pressure in the intake path 10b (the intake port 30a), the fuel passes through the opened intake opening 30d and flows into the pressurizing chamber 11. When the plunger 2 completes the intake process and moves to a compression process, the plunger 2 moves to the compression process (a state in which the plunger 2 moves upward in FIG. 1). At that time, a state in which the electromagnetic coil 308 does not conduct electricity is maintained, and thus magnetic biasing force does not act. Thus, the inlet valve body 301 is still opened by the biasing force of the anchor spring 303. The volume of the pressurizing chamber 11 decreases with the compressing motion of the plunger 2. In such a state, the fuel sucked in the pressurizing chamber 11 is returned through the opened inlet valve body 301 to the intake path 10b (the intake port 30a). Thus, the pressure in the pressurizing chamber is not increased. This process is referred to as a return process.

[0019] When a control signal from an engine control unit 27 (hereinafter, referred to as ECU) is applied to the electromagnetic inlet valve 30 in the return process, a current flows through the electromagnetic coil 308 of the electromagnetic inlet valve 30. The magnetic biasing force moves the electromagnetic plunger 305 to the left side of FIG. 4 and a state in which the anchor spring 303 is compressed is maintained. As a result, the biasing force of the anchor spring 303 does not act on the inlet valve body 301. The fluid force due to the biasing force of the valve spring 304 and the flow of the fuel into the intake path 10b (the intake port 30a) acts. This closes the inlet valve 301 and thus closes the intake opening 30d. When the intake opening 30d is closed, the fuel pressure in the pressurizing chamber 11 starts increasing with the upward motion of the plunger 2. When the fuel pressure is larger than or equal to the pressure in the fuel discharge outlet 12, the fuel remaining in the pressurizing chamber 11 is discharged at high pressure through the discharge valve mechanism 8, and fed to the common rail 23. This process is referred to as a discharge process.

[0020] In other words, the compression process of the plunger 2 (a process in which the plunger 2 rises from a lower starting point to an upper starting point) includes the return process and the discharge process. Controlling the timing at which the electromagnetic coil 308 of the electromagnetic inlet valve 30 conducts electricity can control the amount of the high-pressure fuel to be discharged. When the timing at which the electromagnetic coil 308 conducts electricity is hastened, the proportion of the return process is low and the proportion of the discharge process is high to the compression process.

In other words, the amount of fuel to be returned to the intake path 10b (the intake port 30a) is decreased and the amount of fuel to be discharged at high pressure is increased. On the other hand, when the timing at which the electromagnetic coil 308 conducts electricity is delayed, the proportion of the return process is high and the proportion of the discharge process is low to the compression process. In other words, the amount of fuel to be returned to the intake path 10b is increased and the amount of fuel to be discharged at high pressure is decreased. The timing at which the electromagnetic coil 308 conducts electricity is controlled by the instructions from the ECU.

[0021] The configuration described above controls the timing at which the electromagnetic coil 308 conducts electricity. This can control the amount of fuel to be discharged at high pressure in accordance with the amount of fuel that the internal-combustion engine requires.

[0022] The outlet of the pressurizing chamber 11 is provided with a discharge valve mechanism 8. The discharge valve mechanism 8 includes a discharge valve seat 8a, a discharge valve 8b, and a discharge valve spring 8c. When there is no fuel differential pressure between the pressurizing chamber 11 and the fuel discharge outlet 12, the discharge valve 8b is pressed and fixed to the discharge valve seat 8a and closed by the biasing force of the discharge valve spring 8c. When the fuel pressure in the pressurizing chamber 11 exceeds the fuel pressure in the fuel discharge outlet 12, the discharge valve 8b is opened against the discharge valve spring 8c and the fuel in the pressurizing chamber 11 is discharged at high pressure through the fuel discharge outlet 12 to the common rail 23.

[0023] As described above, the fuel guided to the intake joint 10a is pressurized at high pressure by the reciprocation of the plunger 2 in the pressurizing chamber 11 of the pump body 1 as much as necessary, and fed from the fuel discharge outlet 12 to the common rail 23 by the pressure.

[0024] Injectors 24 for direct injection (namely, a direct-injection injectors) and a pressure sensor 26 are attached to the common rail 23. The number of the attached direct-injection injectors 24 corresponds to the number of cylinder engines of the internal-combustion engine. The direct-injection injectors 24 open and close in accordance with the control signal from the engine control unit (ECU) 27 so as to inject the fuel in the cylinder.

[0025] The pump body 1 is further provided with a discharge flow path 110 communicating the downstream part of the discharge valve 8b with the pressurizing chamber 11 and bypassing the discharge valve, separately from the discharge flow path. The discharge flow path 110 is provided with a pressure relief valve 102 that limits the flow of the fuel only to a direction from the discharge flow path to the pressurizing chamber 11. The pressure relief valve 102 is pressed to the pressure relief valve seat 101 by the relief spring 104 that generates pressing force. When the difference between the pressure in the

pressurizing chamber and the pressure in a relief path is larger than or equal to a predetermined pressure, the pressure relief valve 102 moves away from the pressure relief valve seat 101 and opens.

[0026] For example, when a failure of the direct-injection injector 24 causes an excessive high pressure in the common rail 23 and the differential pressure between the discharge flow path 110 and the pressurizing chamber 11 is larger than or equal to the valve-opening pressure at which the pressure relief valve 102 is opened, the pressure relief valve 102 opens and the discharge flow path at the excessive high pressure is returned from the discharge flow path 110 to the pressurizing chamber 11. This protects a high-pressure pipe such as the common rail 23.

[0027] Hereinafter, the configuration and operation of the high-pressure fuel pump will be described in more detail with reference to FIGS. 1 to 4.

[0028] FIG. 1 is a vertical cross-sectional view of the whole of a high-pressure fuel supply pump in which the present invention is implemented, and is a cross-sectional view taken along the axis of a discharge joint. FIG. 2 is a vertical cross-sectional view, viewed from an angle different from FIG. 1, and a cross-sectional view taken along the axis of an intake joint. FIG. 3 is a horizontal cross-sectional view, and a cross-sectional view taken along the axis of a fuel discharge outlet. FIG. 4 illustrates the whole configuration of a fuel supply system.

[0029] A general high-pressure pump is air-tightly sealed and fixed to the flat surface of a cylinder head 41 of the internal-combustion engine with a flange 1e provided to the pump body 1. An O-ring 61 is fitted to the pump body 1 so that the airtightness between the cylinder head and the pump body is retained.

[0030] A cylinder 6 is attached to the pump body 1. The cylinder 6 is formed in a cylinder with a bottom on an end so that the cylinder 6 guides the back-and-forth movement of the plunger 2 and the pressurizing chamber 11 is formed in the cylinder 6. The pressurizing chamber 11 is provided with a plurality of communication holes 11a so that the pressurizing chamber 11 communicates with the electromagnetic inlet valve 30 configured to feed the fuel and the discharge valve mechanism 8 configured to discharge the fuel from the pressurizing chamber 11 to the discharge path.

[0031] The lower end of the plunger 2 is provided with a tappet 3 that converts the rotation movement of a cam 5 attached to a camshaft of the internal-combustion engine into up-and-down movement, and transmits the up-and-down movement to the plunger 2. The plunger 2 is pressed and fixed to the tappet 3 through a retainer 15 with a spring 4. This can move (reciprocate) the plunger 2 up and down with the rotation movement of the cam 5.

[0032] A plunger seal 13 held on the lower end of the inner periphery of the seal holder 7 has slidably contact with the outer periphery of the plunger 2 on the lower end of the cylinder 6 in the drawing. This seals the blow-by gap between the plunger 2 and the cylinder 6 and pre-

vents the fuel from leaking to the outside of the pump. Meanwhile, this prevents the lubricant (including engine oil) that smoothly moves a sliding part of the internal-combustion engine from leaking through the blow-by gap into the pump body 1.

[0033] The fuel sucked by the feed pump 21 is fed through the intake joint 10a coupled with the intake pipe 28 to the pump body 1.

[0034] A damper cover 14 is coupled with the pump body 1 and forms a low-pressure fuel chamber 10. The fuel passing through the inlet joint 10a flows into the low-pressure fuel chamber 10. In order to remove an obstacle such as a metal powder in the fuel, a fuel filter 102 is attached to the upstream part of the low-pressure fuel chamber 10, for example, while being pressed and inserted in the pump body 1.

[0035] A pressure pulsation reducing mechanism 9 is installed in the low-pressure fuel chamber 10 so that the pressure pulsation reducing mechanism 9 reduces the spread of the pressure pulsation generated in the high-pressure pump to a fuel pipe 28. When the fuel sucked in the pressurizing chamber 11 is returned through the opened inlet valve body 301 to the intake path 10b (the intake port 30a) under a state in which the flow rate of the fuel is controlled, the fuel returned to the intake path 10b (the intake port 30a) generates the pressure pulsation in the low-pressure fuel chamber 10. However, the pressure pulsation is absorbed and reduced by the expansion and contraction of a metal damper 9a forming the pressure pulsation reducing mechanism 9 provided to the low-pressure fuel chamber 10. The metal damper 9a is formed of two corrugated metal disks of which outer peripheries are bonded together. Inert gas such as argon is injected in the metal damper 9a. Mounting hardware 9b is configured to fix the metal damper 9a on the inner periphery of the pump body 1.

[0036] The electromagnetic inlet valve 30 is a variable control mechanism that includes the electromagnetic coil 308. The electromagnetic inlet valve 30 is connected to the ECU through the terminal 307 and repeats conduction and non-conduction of electricity so as to open and close the inlet valve and control the flow rate of the fuel.

[0037] When the electromagnetic coil 308 does not conduct electricity, the biasing force of the anchor spring 303 is transmitted to the inlet valve body 301 through the anchor 305 and the anchor rod 302 integrally formed with the anchor 305. The biasing force of the valve spring 304 installed in the inlet valve body is set so that

the biasing force of the anchor spring 303 > the biasing force of the valve spring 304

holds. As a result, the inlet valve body 301 is biased in a valve-opening direction in which the inlet valve body 301 is opened. The intake opening 30d is opened. Meanwhile, the anchor rod 302 has contact with the inlet valve body 301 at a part 302b (in a state illustrated FIG. 1).

[0038] The setting for the magnetic biasing force generated by the electricity conduction through the coil 308 is configured to enable the anchor 305 to overcome the

biasing force of the anchor spring 303 and be sucked into a stator 306. When the coil 308 conducts electricity, the anchor 303 moves toward the stator 306 (the left side of the drawing) and a stopper 302a formed on an end of the anchor rod 302 has contact with an anchor rod bearing 309 and is seized. At that time, the clearance is set so that

the travel distance of the anchor 301 > the travel distance of the inlet valve body 301

holds. The contact part 302b opens between the anchor rod 302 and the inlet valve body 301. As a result, the inlet valve body 301 is biased by the valve spring 304 and the intake opening 30d is closed.

[0039] The electromagnetic inlet valve 30 is fixed to the pump body 1 while an inlet valve seat 310 is hermetically inserted in a tubular boss 1b so that the inlet valve body 301 can seal the intake opening 30d to the pressurizing chamber. When the electromagnetic inlet valve 30 is attached to the pump body 1, the intake port 30a is connected to the intake path 10b.

[0040] The discharge valve mechanism 8 is provided with a plurality of discharge paths radially drilled around the sliding axis of the discharge valve body 8b. The discharge valve mechanism 8 includes a discharge valve seat member 8a and a discharge valve member 8b. The discharge valve seat member 8a is provided with a bearing that can sustain the sliding reciprocation of the discharge valve body 8b at the center of the discharge valve seat member 8a. The discharge valve member 8b has the central axis so as to slide with respect to the bearing of the discharge valve seat member 8a, and has a circular contact surface on the outer periphery. The circular contact surface can retain the airtightness by having contact with the discharge valve seat member 8a. Furthermore, a discharge valve spring 33 is inserted and held in the discharge valve mechanism 8. The discharge valve spring 33 is a coil spring that biases the discharge valve member 8b in a valve-closing direction in which the discharge valve member 8b is closed. The discharge valve seat member, for example, is pressed, inserted and held in the pump body 1. The discharge valve member 8b and the discharge valve spring 33 are further inserted in the pump body 1. A sealing plug 17 seals the pump body 1. This forms the discharge valve mechanism 8. The discharge valve mechanism 8 is formed as described above. The formation causes the discharge valve mechanism 8 to function as a check valve that controls the direction in which the fuel flows.

[0041] The operation of the pressure relief valve mechanism will be described in more detail. A pressure relief valve mechanism 100 includes a pressure relief valve stopper 101, a pressure relief valve 102, a relief seat 103, a relief spring stopper 104, and a relief spring 105 as illustrated. The pressure relief valve seat 103 includes a bearing that enables the pressure relief valve 102 to slide. The pressure relief valve 102 integrally including a sliding shaft is inserted in the pressure relief valve seat 103. After that the position of the relief spring stopper 104 is

determined so that the relief spring 105 has a desired load, and the relief spring stopper 104 is fixed to the pressure relief valve 102, for example, by press-insertion. The valve-opening pressure at which the pressure relief valve 102 is opened is determined depending on the pressing force of the relief spring 104. The pressure relief valve stopper 101 is inserted between the pump body 1 and the pressure relief valve seat 103 so as to function as a stopper that controls how much the pressure relief valve 102 is opened.

[0042] The pressure relief valve mechanism 100 unitized as described above is fixed to the pump body 1 by the press-insertion of the pressure relief valve seat 103 to the inner peripheral wall of a cylindrical pass-through slot 1C provided to the pump body 1. Subsequently, the fuel discharge outlet 12 is fixed so that the fuel discharge outlet 12 blocks the cylindrical pass-through slot 1C of the pump body 1. This prevents the fuel from leaking from the high-pressure pump to the outside and to enable the pressure relief valve mechanism 100 to be connected to a common rail.

[0043] The relief spring 105 is provided to a side of the pressure relief valve 102 facing the fuel discharge outlet 12 as described above. This prevents the volume of the pressurizing chamber 11 from increasing even when the outlet of the pressure relief valve 102 of the pressure relief valve mechanism 100 is opened to the pressurizing chamber 11.

[0044] When the motion of the plunger 2 starts decreasing the volume of the pressurizing chamber 11, the pressure in the pressurizing chamber increases with the decrease in volume. When the pressure in the pressurizing chamber finally exceeds the pressure in the discharge flow path 110, the discharge valve mechanism 8 is opened and the fuel is discharged from the pressurizing chamber 11 to the discharge flow path 110. From the moment the discharge valve mechanism 8 is opened to the time immediately after the opening, the pressure in the pressurizing chamber overshoots and becomes very high. The very high pressure propagates in the discharge flow path and the pressure in the discharge flow path simultaneously overshoots. If the outlet of the pressure relief valve mechanism 100 is connected to the intake flow pass 10b at the overshoot, the overshoot of the pressure in the discharge flow path causes the pressure difference between the inlet and outlet of the pressure relief valve 102 to exceed the valve-opening pressure at which the pressure relief valve mechanism 100 is opened. This causes an error in the pressure relief valve. In light of the foregoing, the outlet of the pressure relief valve mechanism 100 of the embodiment is connected to the pressurizing chamber 11, and thus the pressure in the pressurizing chamber acts on the outlet of the pressure relief valve mechanism 100 and the pressure in the discharge flow path 110 acts on the inlet of the pressure relief valve mechanism 11. The pressure overshoot occurs simultaneously in the pressurizing chamber and the discharge flow path. Thus, the pressures difference between the

inlet and outlet of the pressure relief valve does not exceed the valve-opening pressure at which the pressure relief valve is opened. In other words, an error in the pressure relief valve does not occur.

[0045] When the motion of the plunger 2 starts increasing the volume of the pressurizing chamber 11, the pressure in the pressurizing chamber decreases with the increase in volume. When the pressure in the pressurizing chamber falls below the pressure in the intake path 10b (the intake port 30a), the fuel flows from the intake path 10b (the intake port 30a) into the pressurizing chamber 11. When the motion of the plunger 2 starts decreasing the volume of the pressurizing chamber 11 again, the fuel is pressurized at high pressure and discharged due to the mechanism described above.

[0046] Next, an example in which failure of the direct-injection injector 24 generates an excessive high pressure in the common rail 23 will be described in detail.

[0047] In the event of failure of the direct-injection injector, in other words, when the injection function of the direct-injection injector stops and the direct-injection injector does not feed the fuel fed in the common rail 23 into the combustion chamber of the internal-combustion engine, the fuel accumulates between the discharge valve mechanism 8 and the common rail 23. This causes an excessive high pressure of the fuel. When the fuel pressure moderately increases to the excessive high pressure, the pressure sensor 26 provided to the common rail 23 detects the abnormal pressure. Then, the electromagnetic inlet valve 30 that is a flow rate control mechanism provided in the intake path the intake path 10b (the intake port 30a) is controlled by feedback control. The feedback control operates as a safety function to decrease the amount of the fuel to be discharged. However, the feedback control with the pressure sensor is not effective in dealing with an instantaneous excessive high pressure. When the electromagnetic inlet valve 30 is out of order and keeps the maximum flow rate in an operation state in which the fuel is not required so much, the pressure at which the fuel is discharged excessively increases. In such a case, the excessive high pressure is not dissolved because of the failure of the flow rate control mechanism even when the pressure sensor 26 of the common rail 23 detects the excessive high pressure.

[0048] When the excessive high pressure described above occurs, the pressure relief valve mechanism 100 of the embodiment functions as a safety valve.

[0049] When the motion of the plunger 2 starts increasing the volume of the pressurizing chamber 11, the pressure in the pressurizing chamber decreases with the increase in volume. When the pressure in the inlet of the pressure relief valve mechanism 100, namely, in the discharge flow path is higher than or equal to the pressure in the outlet of the pressure relief valve, namely, in the pressurizing chamber 11 by the valve-opening pressure at which the pressure relief valve mechanism 100 is opened, the pressure relief valve mechanism 100 is

opened and returns the fuel at an excessive high pressure in the common rail to the pressurizing chamber. This return prevents the fuel pressure from being higher than or equal to a predetermined pressure even when an excessive high pressure occurs. This prevention protects the high-pressure pipe system including the common rail 23.

[0050] The structure of a cylinder of the present embodiment will be described in detail.

[0051] A cylinder 6 includes a large-diameter part 6b and a small-diameter part 6c on the outer diameter of the cylinder 6. The small-diameter part is pressed and inserted into the pump body 1 so that the circumferential surface pressure acting on the small-diameter part maintains the pressure in the intake path 10b and the pressurizing chamber 11a. Specifically, the pressure in the intake path 10b is a lower fuel pressure fed to the high-pressure pump by a feed pump, and is about 0.4 MPa. On the other hand, the pressure generated in the pressurizing chamber 11 is a pressure pressurized by the high-pressure pump and the instantaneous pressure reaches about 30 to 50 MPa. The pressurized fuel is fed from the pressurizing chamber 11 through a plurality of communication holes 11a drilled in a side of the cylinder and through the discharge valve mechanism 8 and the fuel discharge outlet 12 to the common rail 23. The setting for the press fit allowance of the small-diameter part is configured to prevent the fuel from leaking to the intake path 10b due to the pressure pressurizing the fuel. On the other hand, if the void between the large-diameter part 6b and the inner diameter of the pump body 1 is zero, the cylinder only needs to be slightly pressed and inserted into the pump body.

[0052] When the plunger 2 in a compression process (the plunger is displaced upward in FIG. 1), the fuel is pressurized in the pressurizing chamber 11 and the pressure pressuring the fuel acts on the bottom surface of the inner diameter of the cylinder 6. As a result, the surface of the width difference 6a between the large-diameter part 6b and the small-diameter part 6c is press-fitted to the pump body 1, and seals the pressurizing chamber 11 so as to prevent the pressurized fuel from leaking to the space formed between the seal holder 7 and the lower end of the cylinder (hereinafter, referred to as an auxiliary pressurizing chamber). The auxiliary pressurizing chamber communicates with the intake path 10b and the pressure in the auxiliary pressurizing chamber is equal to the value of the lower fuel pressure. The fuel pressure generated in the compression process of the plunger 2 acts on the press-fitted surface. The bottom of the cylinder 6 receives the pressure pressuring the fuel and the pressure acts in a direction in which the press-fitted surface is more tightly adhered and the leakage of fuel is prevented.

[0053] The structure in which the cylinder 6 does not drop from the pump body 1 in the compression process in which the maximum pressure acts on the high-pressure pump among the operation processes is important

to ensure the high quality of the high-pressure pump. In the present embodiment as described above, the cylinder 6 receives the pressure to adhere the cylinder 6 to the pump body 1 in the compression process. This is also advantageous for preventing the cylinder 6 from dropping from the pump body 1.

[0054] On the other hand, in the intake process (when the plunger 2 is displaced downward in FIG. 1), the lower fuel pressure in the intake path 10b acts on the cylinder 6 so as to disconnect the cylinder 6 from the pump body 1. As described above, the lower pressure is about 0.4 MPa. Provided that the diameter of the small-diameter part 6c is 13 mm, the disconnecting force acting on the cylinder 6 is about 53 N. The value of the disconnecting force is small enough to hold the cylinder 6 with the press-inserting force between the small-diameter part 6c and the pump body 1.

[0055] Additionally, in order to cause the plunger 2 to follow the rotation of the cam 5 and smoothly slide back and forth, it is necessary to accurately set the coaxiality of the cylinder 6 and the plunger seal 13. The small-diameter part of the cylinder 6 and the plunger seal 13 are pressed and inserted in the pump body 1. This press-insertion allows for high coaxiality in the seal holder 7 into which the small-diameter part of the cylinder 6 and the plunger seal 13 are attached.

[0056] A seal part will be described in detail with reference to FIGS. 5 and 6.

[0057] FIG. 5 is an enlarged view of a part including a circular protrusion. FIG. 6 illustrates another exemplary variation of the part including a circular protrusion.

[0058] In FIG. 5, the width difference 6a between the large-diameter part 6b and small-diameter part 6c of the cylinder 6 is provided with a circular protrusion 6d with a triangular cross-sectional surface.

[0059] When the cylinder 6 is attached into the pump body 1, the circular protrusion 6d comes into contact with the pump body 1 first in the width difference 6a and this contact locally increases the surface pressure. The material of the cylinder 6 has hardness higher than or equal to the hardness of the material of the pump body 1 in order to support the reciprocation of the plunger 2. This causes earlier plastic deformation of the pump body 1 than the cylinder 6. The circular protrusion 6d is engaged in the pump body 1. This engagement can further increase the sealing function of the width difference 6a.

[0060] Alternatively, the circular protrusion 6d can be formed into a shape that does not protrude from the flat surface of the width difference 6a as illustrated in FIG. 6.

[0061] When the cylinder 6 is attached into the pump body 1, the width difference 6a comes into contact with the pump body 1 first. The surface of the pump body having contact with the width difference 6a is slightly plastic deformed. Then, the circular protrusion 6d is engaged in the pump body and this engagement locally increases the surface pressure and enhances the sealing function. In the structure in FIG. 6, the protrusion on the cylinder 6 does not protrude from the surface of the width differ-

ence 6a before the high-pressure pump is assembled. Advantageously, this makes it unnecessary to pay attention to breakage of the protrusion and thus makes it easy to handle the cylinder 6.

[0062] In the present embodiment, the cross-sectional surface of the circular protrusion 6d has a triangular shape. However, for example, a convex shape or a curved surface can have the same effect.

[0063] Alternatively, the circular protrusion can similarly be formed on the pump body 1. This can also achieve the objective.

[0064] A ring 16 will be described in detail with reference to FIGS. 7 and 8(a) to 8(c).

[0065] FIG. 7 is a vertical cross-sectional view of the whole of the high-pressure pump to which the cylinder is fixed with the ring 16.

[0066] In FIG. 7, in order to add a pre-charge pressure to the press-fitted surface 6a of the cylinder 6, an end surface of the large-diameter part 6b of the cylinder is pressed to the pump body with the ring 16. The ring 16 is fixed to the pump body 1 by press-insertion or, for example, with a metal flow part (plastic flow combination) 1d illustrated in FIG. 8(a), or with a swaged part 1f illustrated in FIG. 8(b). In order to apply a desired pressing load onto the cylinder 6 when the cylinder 6 is attached into the pump body 1, the ring 16 is previously pressurized and installed in the pump body 1. Then, the ring 16 is fixed to the pump body 1 by swaging or metal flow.

[0067] As the embodiment illustrated in FIG. 8(c), a spring member 18 can be attached to an end surface of the large-diameter part of the cylinder in order to add a pre-charge pressure to the width difference 6a of the cylinder 6.

[0068] FIGS. 8 (a) to 8 (c) illustrate embodiments in which a ring is used to fix the cylinder to the pump body.

[0069] A void 17 is provided between the large-diameter part 6b of the cylinder 6 and the pump body 1. According to the structure of the cylinder described above, the press-insertion of the small-diameter part 6c in the pump body 1 and press-fitting of the press-fitted surface 6a to the pump body 1 allows the pump body 1 to hold the cylinder 6. Thus, the void provided between the outer diameter part 6b of the cylinder and the pump body 1 does not adversely affect the holding of the cylinder at all.

[0070] The void between the outer diameter of the plunger 2 and the inner diameter of the cylinder 6 greatly affects the pump pressurizing performance. In other words, the larger the void is, the lower the degree of efficiency in pressuring the fuel in the pressurizing chamber 11 during the compression process is. Thus, it is necessary to regulate the size of the void within a range of 5 to 10 μm when the diameter of the plunger is 8 to 10 mm. Accordingly, it is necessary to accurately process the outer diameter of the plunger 2 and the inner diameter of the cylinder 6.

[0071] Thus, when the large-diameter part 6b of the cylinder is pressed and inserted into the pump body 1, the inner diameter of the cylinder is slightly deformed in

a direction in which the cylinder contracts. For example, when the press fit allowance is 10 to 20 μm , the cylinder contracts by about 1 to 2 μm . The amount of deformation is one tenth of the press fit allowance. The void between the outer diameter of the plunger 2 and the inner diameter of the cylinder 6 is 5 to 10 μm . At worst, the contraction may cause the plunger to be burned and seized during the operation of the high-pressure pump. In light of the foregoing, a process for correcting the inner diameter of the cylinder is required after the cylinder 6 is pressed and inserted into the pump body.

[0072] In the present embodiment, the void between the outer diameter of the plunger 2 and the inner diameter of the cylinder 6 exists between the width difference 6a of the large-diameter part of the cylinder and the end surface of the cylinder protruding toward the auxiliary pressurizing chamber. In the present embodiment, the void 17 is provided between the large-diameter part 6b of the cylinder and the pump body 1. Thus, when the cylinder 6 is attached into the pump body 1, a force acting in a direction in which the inner diameter of the cylinder 6 contracts is not generated. Furthermore, the inner diameter of the cylinder is increased between the width difference 6a and the small-diameter part 6c. This increase prevents the decrease in size of the void due to the press-insertion of the small-diameter part and thus prevents the decrease from causing the burning and seizing of the plunger. According to the structure described above, the process for correcting the inner diameter of the cylinder is not required after the cylinder is attached into the pump body. This can reduce the cost for the pump body.

[0073] Note that the present embodiment can be defined as follows. The embodiment is a high-pressure fuel pump that includes: a plunger that reciprocates; a cylinder including a part that guides the reciprocation of the plunger; and a pump body that holds the cylinder. The cylinder is formed in a cylindrical shape with a bottom, and includes a large-diameter part and a small-diameter part. The surface of cylinder is press-fitted to the pump body in a direction in which the plunger reciprocates. Alternatively, the embodiment is a high-pressure fuel pump that includes: a plunger that reciprocates; a cylinder including a part that guides the reciprocation of the plunger; and a pump body that holds the cylinder. The cylinder is formed in a cylindrical shape with a bottom, and includes a large-diameter part and a small-diameter part. The surface of cylinder is press-fitted to the pump body in a direction in which the plunger reciprocates. The surface of the cylinder is a part that does not axially overlap the part that guides the reciprocation of the plunger.

Reference Signs List

[0074]

- | | |
|---|-----------|
| 1 | pump body |
| 2 | plunger |

- 6 cylinder
- 8 discharge valve mechanism
- 9 pressure pulsation reducing mechanism
- 30 electromagnetic inlet valve
- 100 pressure relief valve mechanism

5

Claims

1. A high-pressure fuel supply pump, comprising:
 - a plunger (2) that reciprocates;
 - a cylinder (6) including a part that guides the reciprocation of the plunger (2);
 - a pressurizing chamber (11) formed in the cylinder (6);
 - and
 - a pump body (1) that holds the cylinder (6), wherein the cylinder (6) is formed in a cylindrical shape with a bottom, and includes a large-diameter part (6b) and a small-diameter part (6c), and a surface (6a) of the cylinder (6) is subjected to a surface pressure and pressed to the pump body (1) in a direction in which the plunger (2) reciprocates in a compression process, and the pressed surface (6a) of the cylinder (6) is an annular step surface (6a) formed between the large-diameter part (6b) and an intermediate part of the cylinder (6) having a smaller diameter than the small-diameter part (6c), and the annular step surface (6a) seals the pressurizing chamber (11) so as to prevent the pressurized fuel from leaking.
2. The high-pressure fuel supply pump according to claim 1, wherein the cylinder (6) includes a circular protrusion (6d), a surface of the circular protrusion (6d) is pressed to the pump.
3. The high-pressure fuel supply pump according to claim 1, wherein the large-diameter part (6b) of the cylinder (6) is provided with a different member (16) formed, for example, in a ring shape in order to add a pre-charge pressure to the pressed surface (6a) of the cylinder (6), and the different member (16) is fixed to the pump body (1) by a plastic forming process such as swaging or by press-insertion.
4. The high-pressure fuel supply pump according to claim 1, wherein a void (17) is provided between an outer diameter of the large-diameter part (6b) of the cylinder (6) and an inner diameter of the pump body (1).
5. The high-pressure fuel supply pump according to claim 1, wherein the surface (6a) of the cylinder (6) does not axially overlap the part that guides the reciprocation of the plunger (2).

6. The high -pressure fuel supply pump according to any of the preceding claims, wherein the intermediate part of the cylinder (6) forms a gap between the cylinder (6) and the pump body (1).

Patentansprüche

1. Hochdruckkraftstoffzufuhrpumpe, die Folgendes umfasst:
 - einen Kolben (2), der sich hin- und herbewegt;
 - einen Zylinder (6), der einen Teil enthält, der die Hin- und Herbewegung des Kolbens (2) führt;
 - eine Druckkammer (11), die im Zylinder (6) gebildet ist; und
 - einen Pumpenkörper (1), der den Zylinder (6) hält, wobei
 - der Zylinder (6) in einer zylindrischen Form mit einem Boden gebildet ist und einen Teil (6b) mit großem Durchmesser und einen Teil (6c) mit kleinem Durchmesser enthält, eine Oberfläche (6a) des Zylinders (6) einem Oberflächendruck ausgesetzt ist und in einer Richtung, in der der Kolben (2) sich in einem Kompressionsprozess hin- und herbewegt, zum Pumpenkörper (1) gedrückt wird, und die gedrückte Oberfläche (6a) des Zylinders (6) eine ringförmige Stufenfläche (6a) ist, die zwischen dem Teil (6b) mit großem Durchmesser und einem Zwischenteil des Zylinders (6), der einen kleineren Durchmesser als der Teil (6c) mit kleinem Durchmesser besitzt, gebildet ist, und
 - die ringförmige Stufenfläche (6a) die Druckkammer (11) abdichtet, um zu verhindern, dass der mit Druck beaufschlagte Kraftstoff austritt.
2. Hochdruckkraftstoffzufuhrpumpe nach Anspruch 1, wobei der Zylinder (6) einen kreisförmigen Vorsprung (6d) enthält und eine Oberfläche des kreisförmigen Vorsprungs (6d) zur Pumpe gedrückt wird.
3. Hochdruckkraftstoffzufuhrpumpe nach Anspruch 1, wobei der Teil (6b) mit großem Durchmesser des Zylinders (6) mit einem verschiedenen Element (16) versehen ist, das z. B. in einer Ringform gebildet ist, um einen Vorladedruck auf die gedrückte Oberfläche (6a) des Zylinders (6) zuzugeben, und das verschiedene Element (16) am Pumpenkörper (1) durch einen Kunststoffbildungsprozess wie z. B. Verpressen oder durch Einpressen befestigt ist.
4. Hochdruckkraftstoffzufuhrpumpe nach Anspruch 1, wobei ein Hohlraum (17) zwischen einem Außendurchmesser des Teils (6b) mit großem Durchmesser des Zylinders (6) und einem Innendurchmesser des Pumpenkörpers (1) vorgesehen ist.

5. Hochdruckkraftstoffzufuhrpumpe nach Anspruch 1, wobei die Oberfläche (6a) des Zylinders (6) mit dem Teil, der die Hin- und Herbewegung des Kolbens (2) führt, nicht axial überlappt.
6. Hochdruckkraftstoffzufuhrpumpe nach einem der vorhergehenden Ansprüche, wobei der Zwischenteil des Zylinders (6) eine Lücke zwischen dem Zylinder (6) und dem Pumpenkörper (1) bildet.

Revendications

1. Pompe d'alimentation de carburant à haute pression, comprenant :
 - un piston (2) qui se déplace en va-et-vient ;
 - un cylindre (6) incluant une partie qui guide le déplacement en va-et-vient du piston (2) ;
 - une chambre de pressurisation (11) formée dans le cylindre (6) ; et
 - un corps de pompe (1) qui contient le cylindre (6),
 - dans laquelle le cylindre (6) est formé sous une forme cylindrique avec un fond, et inclut une partie de grand diamètre (6b) et une partie de petit diamètre (6c), et une surface (6a) du cylindre (6) est soumise à une pression surfacique et est pressée vers le corps de pompe (1) dans une direction dans laquelle le piston (2) se déplace en va-et-vient dans un processus de compression, et la surface pressée (6a) du cylindre (6) est une surface en gradin annulaire (6a) formée entre la partie de grand diamètre (6b) et une partie intermédiaire du cylindre (6) ayant un diamètre plus petit que la partie de petit diamètre (6c), et
 - la surface en gradin annulaire (6a) étanche la chambre de pressurisation (11) de manière à empêcher le carburant pressurisé de fuir.
2. Pompe d'alimentation de carburant à haute pression selon la revendication 1, dans laquelle le cylindre (6) inclut une projection circulaire (6d), une surface de la projection circulaire (6d) étant pressée contre la pompe.
3. Pompe d'alimentation de carburant à haute pression selon la revendication 1, dans laquelle la partie de grand diamètre (6b) du cylindre (6) est dotée d'un élément différent (16) formé, par exemple, sous la forme d'un anneau afin d'ajouter une pression de précharge à la surface pressée (6a) du cylindre (6), et l'élément différent (16) est fixé sur le corps de pompe (1) par un processus de déformation plastique comme un sertissage ou par insertion à la presse.

4. Pompe d'alimentation de carburant à haute pression selon la revendication 1, dans laquelle un vide (17) est prévu entre un diamètre extérieur de la partie de grand diamètre (6b) du cylindre (6) et un diamètre intérieur du corps de pompe (1).
5. Pompe d'alimentation de carburant à haute pression selon la revendication 1, dans laquelle la surface (6a) du cylindre (6) ne chevauche pas axialement la partie qui guide le déplacement de va-et-vient du piston (2).
6. Pompe d'alimentation de carburant à haute pression selon l'une quelconque des revendications précédentes, dans laquelle la partie intermédiaire du cylindre (6) forme un intervalle entre le cylindre (6) et le corps de pompe (1).

FIG. 1

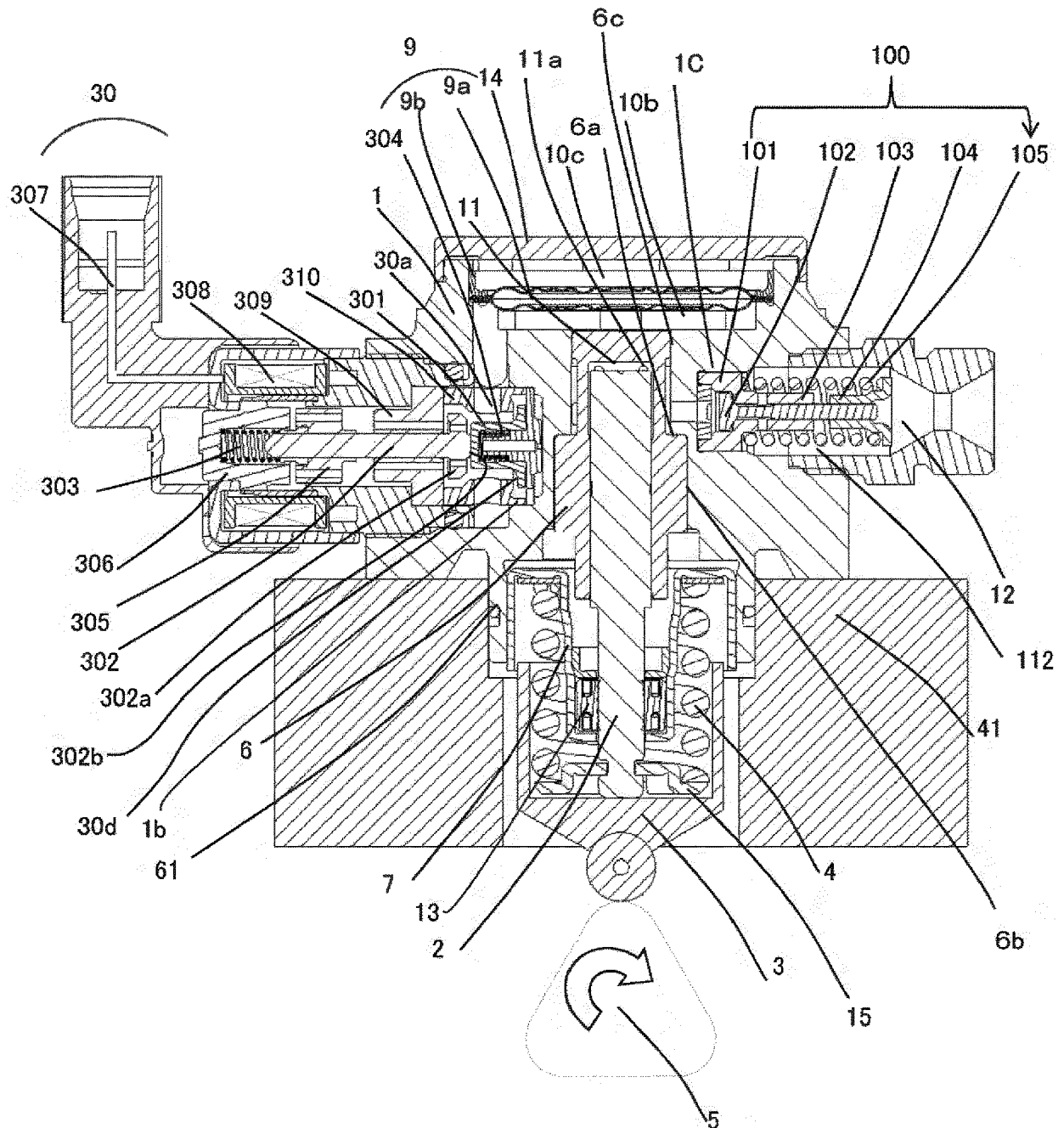


FIG. 2

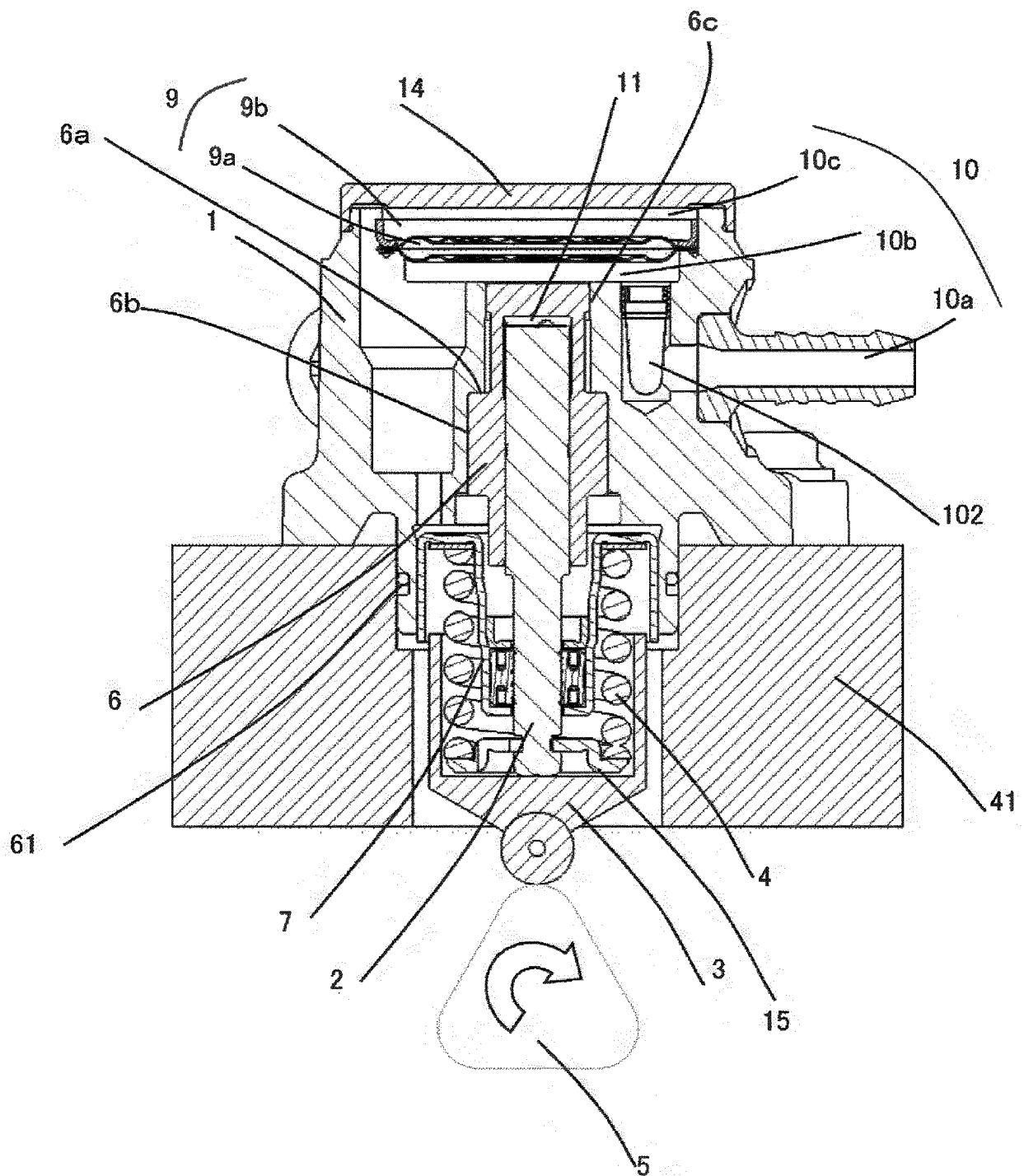


FIG. 3

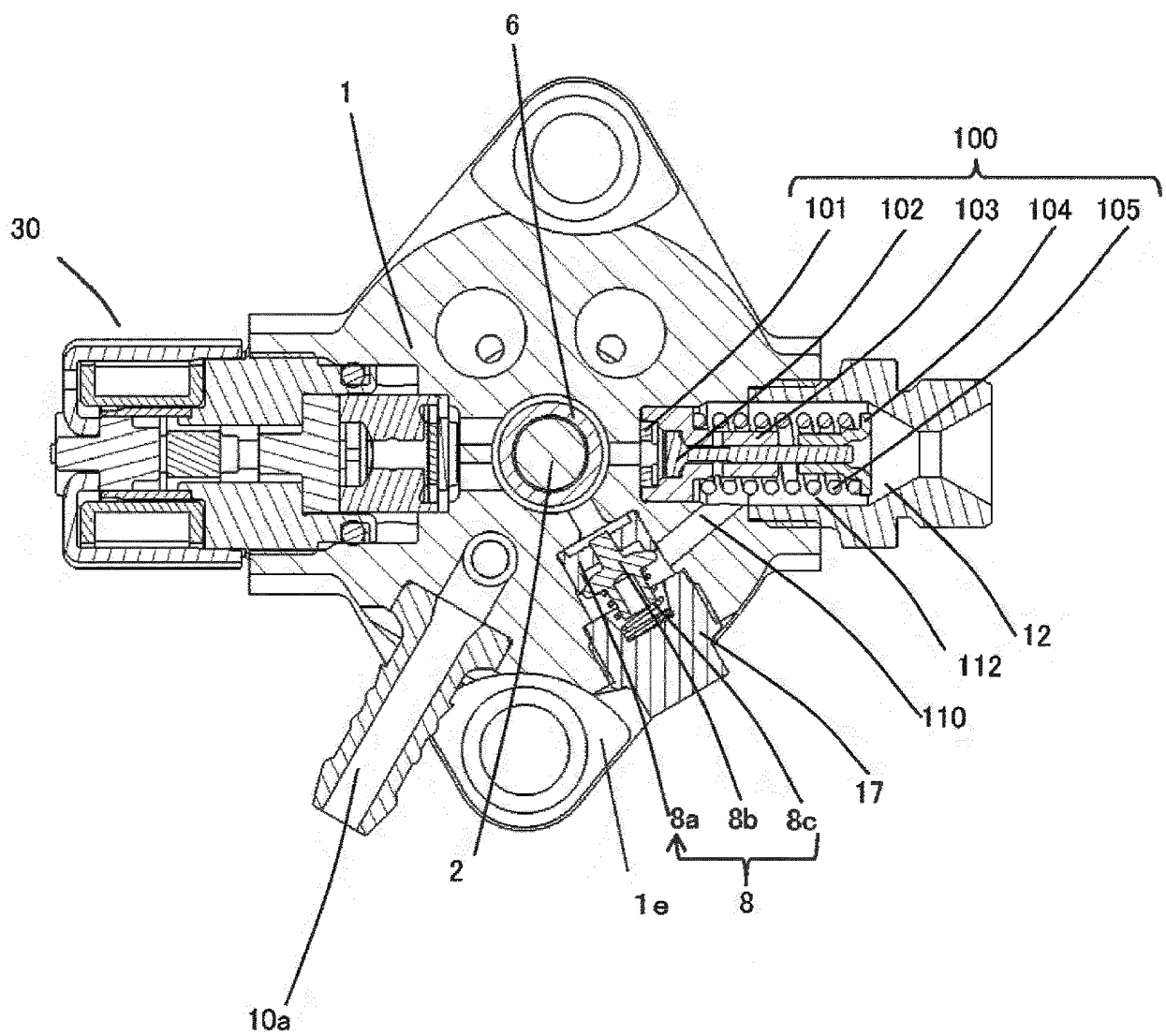


FIG. 4

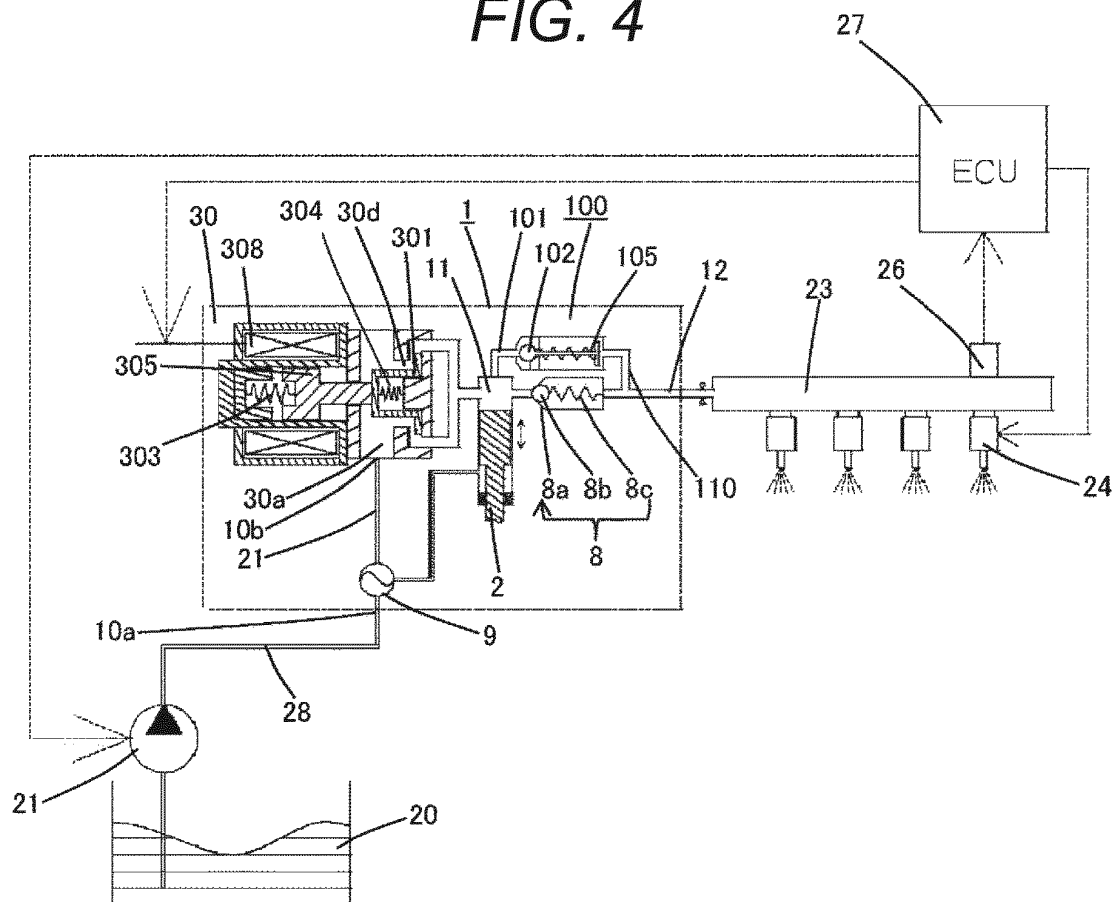


FIG. 5

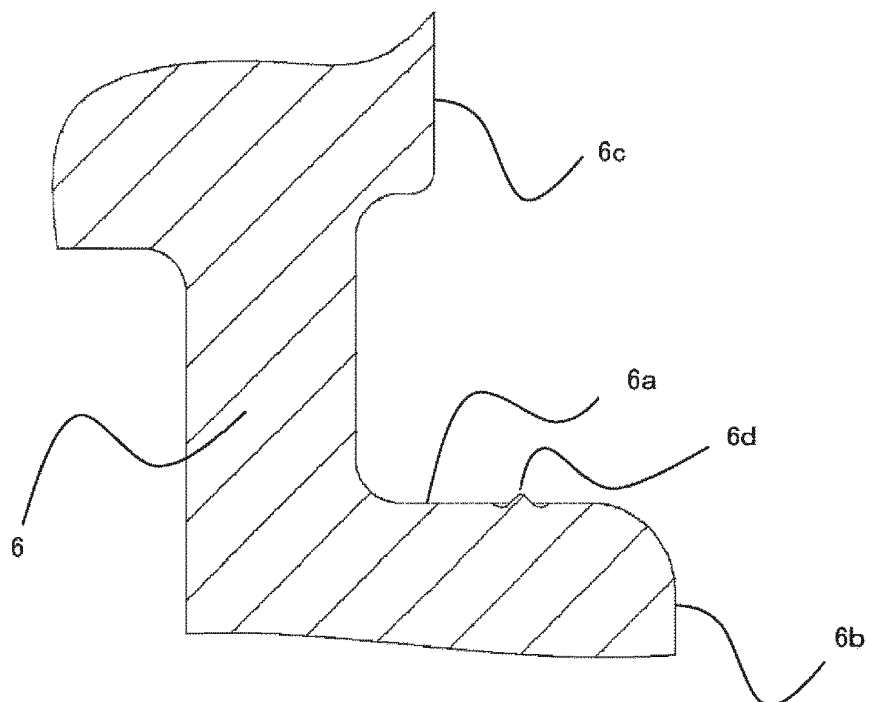


FIG. 6

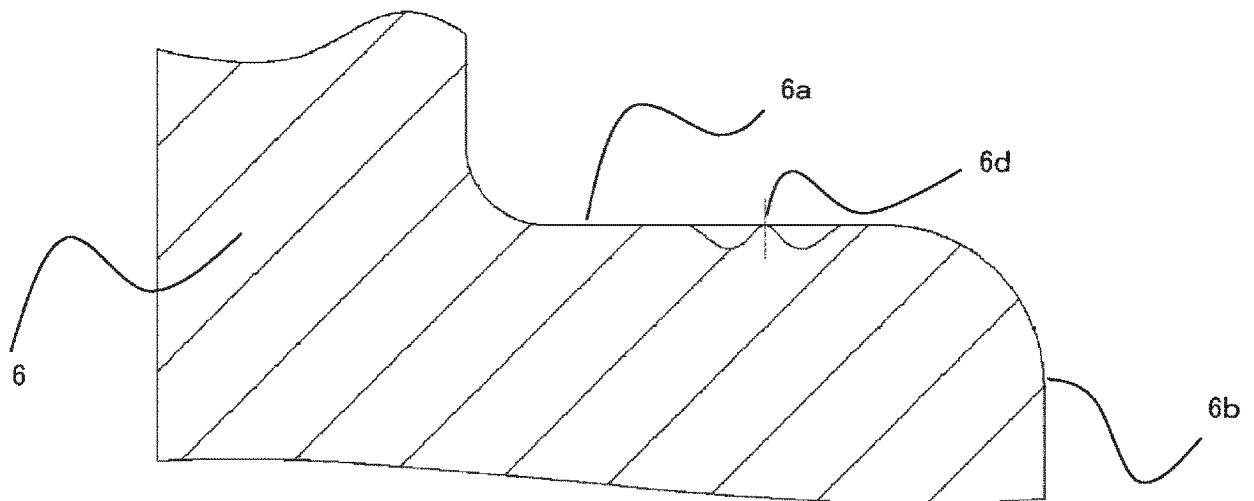


FIG. 7

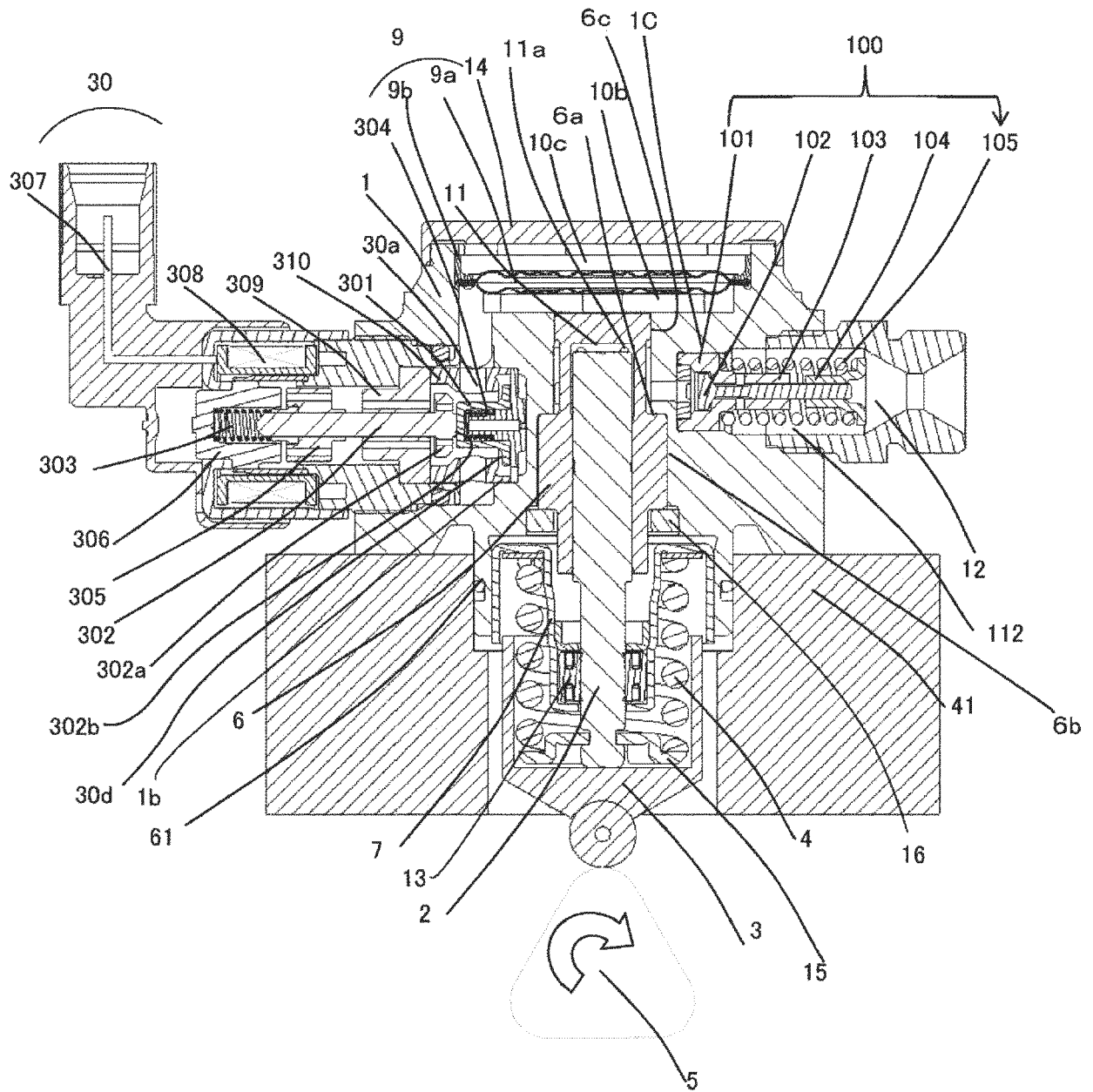
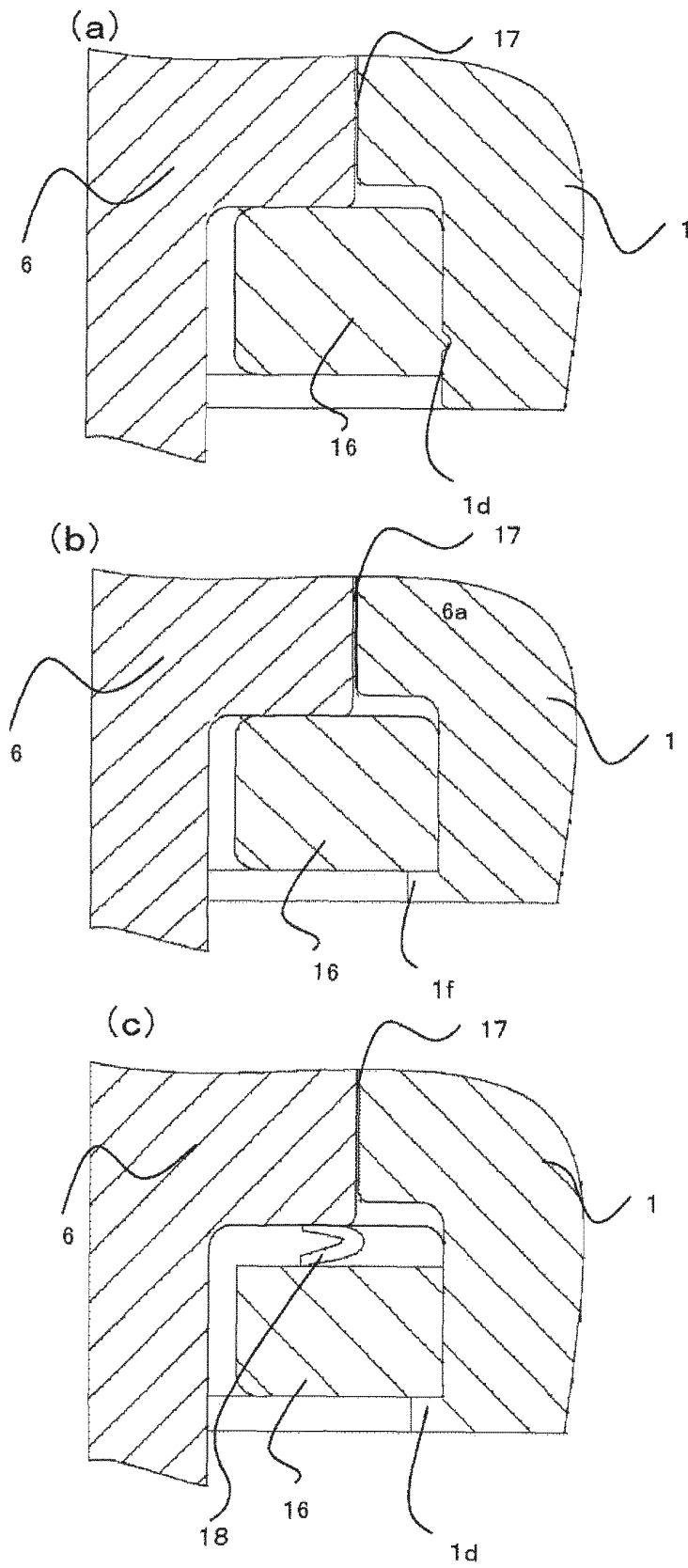


FIG. 8



REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- JP 5178676 B [0003]
- US 2012195780 A1 [0004]
- JP 517867 B [0005]
- US 20120195780 A1 [0005]