

Description

[0001] The invention relates to a control valve according to the preamble part of claim 1. More particularly, such a control valve equipped with a variable displacement compressor may be used in a refrigeration cycle of an automotive air conditioner.

[0002] A wobble plate type variable displacement compressor in the refrigeration cycle of an automotive air conditioner is driven by an engine and is adapted to independently vary the refrigerant compression capacity so as to obtain an adequate cooling capacity without being constrained by the rotational speed of the engine. Pistons are connected to a wobble plate on a shaft driven by the engine. By varying the inclination angle of the wobble plate, the stroke of the pistons is varied to vary the refrigerant discharge amount. The inclination angle is continuously changed by introducing part of compressed refrigerant into a hermetically closed crankcase, thereby changing the balance of pressures acting on the opposite sides of each piston.

[0003] A known variable displacement compressor control valve known from (JP-2001 132650 A, Fig. 1, corresponding to EP1 098 091 A) is disposed either between the discharge chamber and the crankcase, or between the crankcase and a suction chamber. The control valve adjusts the controlled pressure in the crankcase by changing the flow either between the discharge chamber and the crankcase, or between the crankcase and the suction chamber. A valve section is disposed in a refrigerant passage between the discharge chamber and the crankcase. A refrigerant path having an orifice is provided between the suction chamber and the crankcase. A valve element receives discharge pressure P_d in valve-opening direction. A piston rod integrally formed on a rear side of the valve element has approximately the same diameter as the valve seat. An end face of the piston rod receives in valve-closing direction the suction pressure P_s and the load of a solenoid for setting the discharge capacity of the compressor by an external signal. The discharge pressure P_d and the suction pressure P_s load the opposite ends of the valve element and the piston rod on the same effective pressure-receiving areas. The differential pressure ($P_d - P_s$) causes the valve element to perform a valve opening/closing operation to control the flow rate between the discharge chamber and the crankcase.

[0004] When the rotational compressor speed increases the discharge capacity increases, and the discharge pressure P_d increases while the suction pressure P_s decreases to increase the value of the differential pressure ($P_d - P_s$). The valve lift of the valve element increases. The valve section operates depending on the value of the differential pressure ($P_d - P_s$), so that the control valve increases the flow rate of refrigerant to the crankcase to increase the controlled pressure P_c in the crankcase. The discharge capacity decreases, thereby decreasing the value of the differential pressure ($P_d - P_s$). The flow

rate into the crankcase is controlled such that the differential pressure ($P_d - P_s$) is held at a predetermined value, which can be set by the value of electric current supplied to the solenoid.

[0005] An engine speed depending change of the rotational compressor changes the discharge capacity. The pressure P_c first is adjusted after the value of the differential pressure ($P_d - P_s$) is changed due to the change in the discharge capacity. Therefore, in a transient period in which the rotational engine speed has rapidly changed, the compressor discharge capacity can be temporarily largely changed due to a delayed compressor responsiveness. This drawback could be somewhat mitigated by improving the control sensitivity by increasing the pressure receiving area. However, in the control valve the solenoid load is applied to the piston rod counter to the high load of the discharge pressure P_d on the valve element. Hence, increasing the sensitivity by increasing the pressure-receiving area is impractical since then the required solenoid load has to be extreme, which needs a very large and powerful solenoid.

[0006] It is an object of the invention to provide a control valve capable of promptly restoring the compressor to a predetermined discharge capacity even when the rotational engine speed has rapidly changed.

[0007] This object is achieved by the features of claim 1.

[0008] When only a gradual pressure change is caused by a gentle change of the rotational speed of the engine and the compressor, the pressure-sensing section remains insensitive to the occurring gradual pressure change. The same operation is performed as in the conventional compressor. However, whenever a significant pressure change is caused by a rapid change of the rotational speed of the engine and the compressor, the pressure-sensing section will respond to the pressure change by accelerating the valve motion in valve-opening and/or closing direction, which motion is initiated by the pressure change. The variable displacement compressor promptly starts changing the discharge capacity to thereby promptly restore a predetermined discharge capacity. The pressure sensing section then operates as a pressure change compensating the valve section servo-drive.

[0009] Whenever the compressor rotational speed changes rapidly the pressure-sensing section responds to the pressure change caused by the rapid speed change and accelerates the motion of the valve section in the valve-opening and/or closing direction. The motion per se is performed in response to the pressure change. The response sensitivity of the control valve hence is enhanced only temporarily when the compressor rotational speed changes rapidly.

[0010] Embodiments of the invention will be described with reference to the drawings.

Fig. 1 is a longitudinal section of a first embodiment of a control valve for a variable displacement

- compressor,
- Fig. 2 is a diagram explaining the operation of the control valve,
- Fig. 3 is a longitudinal section of a second embodiment of the control valve,
- Fig. 4 is a longitudinal section of a third embodiment of the control valve,
- Fig. 5 is a longitudinal section of a fourth embodiment of the control valve,
- Fig. 6 is a longitudinal section of a fifth embodiment of the control valve,
- Fig. 7 is a longitudinal section of a sixth embodiment of the control valve,
- Fig. 8 is an enlarged fragmentary longitudinal section of essential parts of a seventh embodiment of the control valve,
- Fig. 9 is an enlarged fragmentary longitudinal section of essential parts of an eighth embodiment of the control valve,
- Fig. 10 is an enlarged fragmentary longitudinal section of the eighth embodiment in an operative state after the compressor discharge pressure has rapidly decreased,
- Fig. 11 is an enlarged fragmentary longitudinal section of a ninth embodiment of the control valve in states in which the discharge pressure has rapidly increased and rapidly decreased, and
- Fig. 12 is an enlarged fragmentary longitudinal section of a tenth embodiment of the control valve.

[0011] The control valve 11 in Fig. 1 comprises a pressure-sensing section 12 for particularly responding to a rapid change in discharge pressure P_d , a valve section 13 for responding to the value of the differential pressure ($P_d - P_s$) between the discharge pressure P_d and the suction pressure P_s to control the flow rate between the discharge chamber and the crankcase of a variable displacement compressor (not shown), and a solenoid 14 for setting a predetermined value of the differential pressure ($P_d - P_s$) which value is to be controlled by the control valve 11. These sections are arranged on the same axis.

[0012] The pressure-sensing section 12 and the valve section 13 are contained in a first body 15 and in a second body 16 into which the first body 15 is press-fitted. The first body 15 defines a cylinder 17. The open upper cylinder end defines a high-pressure port 18 communicating with the discharge chamber. A pressure-sensing piston

19 is axially movably disposed within the cylinder 17 and is urged in valve opening direction by a spring 21 backed up by a stopper 20. A hollow cylindrical valve seat-forming member 22 is press-fitted into a central hole in the cylinder bottom. An upper part of the valve seat-forming member 22 is inserted into a cylinder formed in the centre of the lower end of the pressure-sensing piston 19 such that an annular pressure-adjusting chamber 23 is defined by the first body 15 and the pressure-sensing piston 19. A through hole of the pressure-sensing piston 19 connects the cylinder in the pressure-sensing piston 19 and the high-pressure port 18. The high-pressure port 18 communicates via the through hole of the pressure-sensing piston 19 with an axial passage or valve hole extending through the valve seat-forming member 22. One end of a shaft 24 is fixed to the pressure-sensing piston 19. The shaft 24 extends further through the valve hole in the valve seat-forming member 22.

[0013] A lower end of the valve seat-forming member 22 forms a valve seat 22a. A valve element 25 is axially movably disposed opposed to the valve seat 22a. The valve element 25 is integral with a piston rod 26. The piston rod 26 is axially movably guided by the second body 16 and has an outer diameter which is equal to the inner diameter of the valve hole. The valve element 25 abuts at the other end of the shaft. The piston rod 26 is urged by a spring 27 in valve opening direction. A space containing the valve element 25 communicates with a medium-pressure port 28 supplying controlled pressure P_c to the crankcase. A space containing the spring 27 communicates with a low-pressure port 29 receiving the suction pressure P_s .

[0014] A hole is formed in the centre of a lower part of the second body 16. The rim of an opening of a bottomed sleeve 30 is tightly connected to the hole. The bottomed sleeve 30 contains a fixed core 31 and an axially movable plunger 32 of the solenoid 14. The core 31 is fixed to the hole of the second body 16 and the bottomed sleeve 30 by press-fitting. The plunger 32 is fixed to one end of a shaft 33 axially extending through the core 31. The plunger 32 is urged toward the core 31 by a spring 34 such that the other end of the shaft 33 abuts at a lower end face of the piston rod 26. A coil 35 surrounds the bottomed sleeve 30. A harness 36 for supplying electric current to the coil 35 leads to the outside of the solenoid 14.

[0015] The force of the spring 27 is set larger than the force of the spring 34. When the solenoid 14 is de-energized, the valve element 25 is kept away from the valve seat 22a. The valve section 13 is in the fully open state. Discharge pressure P_d at the high-pressure port 18 passes through the valve section 13. Refrigerant flows from the medium-pressure port 28 into the crankcase. The pressure P_c is brought close to the discharge pressure P_d . The compressor operates with minimum discharge capacity.

[0016] When the automotive air conditioner is started or when the cooling load is a maximum, the value of electric current supplied to the solenoid 14 is a maximum as

well. The plunger 32 is attracted with maximum attractive force by the core 31. The piston rod 26 is pushed by the shaft 33 in valve-closing direction against the force of the spring 27. The valve element 25 is seated on the valve seat 22a to place the valve section 13 in the fully closed state. The discharge pressure P_d is blocked at the high-pressure port 18 by the closed valve section 13. The pressure P_c is brought close to the suction pressure P_s . The compressor operates with maximum discharge capacity.

[0017] When the value of the electric current is set to a predetermined value, the valve element 25 is stopped at a valve lift position where the loads of the springs 21 and 27 in valve-opening direction, the solenoid load in valve-closing direction, the force generated by the discharge pressure P_d on the valve element 25 in valve-opening direction, and the force of the suction pressure P_s in valve-closing direction are balanced.

[0018] In the above balanced state, when the compressor rotational speed is increased e.g. by an increase in the engine rotational speed, to increase the discharge capacity the discharge pressure P_d increases and the suction pressure P_s decreases so that the value of the differential pressure ($P_d - P_s$) increases. This generates a force in valve-opening direction on the valve element 25 and the piston rod 26. The valve element 25 is lifted. Refrigerant flows at an increased flow rate from the discharge chamber into the crankcase. The pressure P_c is increased. The compressor is switched to operate in a discharge capacity reducing direction. The differential pressure ($P_d - P_s$) is controlled to the predetermined value set by the solenoid 14. When then the engine rotational speed has decreased, the control valve 11 will operate inversely, whereby the compressor is controlled such that the differential pressure ($P_d - P_s$) again becomes equal to the predetermined value set by the solenoid 14.

[0019] When the compressor rotational speed only is gently or gradually changed, e.g. when the automotive vehicle is cruising at approximately constant speed, the pressure-sensing section 12 will remain insensitive and does not respond independently of the response of the valve section. To the contrary, the control valve 11 will operate differently when the compressor rotational speed is rapidly changed by a rapid change in the engine rotational speed, e.g. when the automotive vehicle has been rapidly accelerated or decelerated.

[0020] The diagram in Fig. 2 explains such operations of the control valve when the compressor rotational speed is rapidly increased.

[0021] The compressor first steadily runs at e.g. 800 rpm, before the rotational speed has been rapidly increased to 2000 rpm. The valve lift increases due to a rise of the discharge pressure P_d and a drop of the suction pressure P_s . The control valve 11 tends to increase the pressure P_c in the crankcase, as indicated by broken lines in Fig. 2. At this time, the pressure-sensing section 12 receives the discharge pressure P_d , which has rapidly

increased, at the pressure-sensing piston 19 having a larger pressure-receiving area than that of the valve element 25. On the other hand, in the pressure-adjusting chamber 23, pressure $P_d(av)$, which is an average pressure derived from the discharge pressure P_d before it has rapidly increased, is maintained. The differential pressure ($P_d - P_d(av)$) generates a force on the pressure-sensing piston 19 in a direction toward the valve section 13. This force is applied by the shaft 24 to the valve element 25, and hence in addition to the rapidly increased discharge pressure P_d , the differential pressure ($P_d - P_d(av)$) of the pressure-sensing section 12 is additionally applied to the valve element 25. As a result, as indicated by solid lines in Fig. 2, the valve lift is increased more promptly, so that the control valve 11 causes the pressure P_c in the crankcase to increase more promptly. After that, in the pressure-sensing section 12, the rapidly increased discharge pressure P_d is promptly introduced into the pressure-adjusting chamber 23 via the clearance between the cylinder 17 and the pressure-sensing piston 19 and the clearance between the pressure-sensing piston 19 and the valve seat-forming member 22, until the differential pressure ($P_d - P_d(av)$) becomes zero. At this time, the function of the pressure-sensing section 12 is nullified. The pressure-sensing section 12 only has the function of a derivative element for sensing a rapid increase in the discharge pressure P_d , and for temporarily accelerating the motion of the valve section 13 e.g. in valve-opening direction only. This enables the control valve 11 to promptly restore the compressor to the predetermined discharge capacity.

[0022] The control valve 11 operates similarly but inversely when the compressor rotational speed is rapidly decreased. Then, the differential pressure ($P_d(av) - P_d$) moves the pressure-sensing piston 19 away from the valve section 13. The urging force of the spring 21 in valve-opening direction via the pressure-sensing piston 19 and the shaft 24 is weakened, which causes the valve element 25 to move in valve-closing direction.

[0023] In other words, any pressure rises or pressure drops are transmitted into or out of the pressure-adjusting chamber 23 in retarded fashion in order to generate a boosting effect on the valve section, but only when the pressure drops or pressure rises originate from rapid compressor speed changes. This effect is caused by flow restricting means in the flow path into or out of the pressure-adjusting chamber 23.

[0024] When the compressor rotational speed is rapidly changed, the shaft 24 sometimes even may separate from the valve element 25, depending on the setting of the spring 21. In such a case, the speed of the motion of the valve element 25 in valve-closing direction is slower, so that the valve-opening characteristic of the control valve becomes asymmetric such that the valve opens differently between when the compressor rotational speed is rapidly increased or rapidly decreased. In this case, when the compressor rotational speed is rapidly increased, if the discharge capacity of the compressor is

not promptly decreased, the compressor load applied to the engine becomes more significant for the engine, whereas in the opposite case, even if the compressor discharge capacity is not promptly increased, the compressor load applied to the engine only decreases the vehicle speed, and hence there is no problem even if the valve opening characteristic is asymmetric as mentioned.

[0025] In the control valve 11a of Fig. 3 (second embodiment) the shaft 24 fixed to the pressure sensing piston 19, the valve element 25, and the piston rod 26 are formed integrally, and the spring 21 is eliminated.

[0026] The pressure-sensing section 12 of the control valve 11a directly exerts influence on the valve section 13 in both cases (compressor rotational speed has rapidly increased or has rapidly decreased). When the compressor rotational speed is rapidly increased, the control valve 11a operates like the control valve 11 of Fig. 1, but when the compressor rotational speed is rapidly decreased, the pressure-sensing piston 19 directly actuates the shaft 24, the valve element 25, and the piston rod 26 in valve closing direction. This control valve 11a is suitable when a symmetric valve-opening characteristic is desired between when the rotational speed of the compressor is rapidly increased and when it is rapidly decreased.

[0027] The control valve 11b in Fig. 4 (third embodiment) comprises flow rate-adjusting means at a location between the cylinder 17 of the pressure-sensing section 12 and the pressure-sensing piston 19 and between the pressure-sensing piston 19 and the valve seat-forming member 22 for adjusting an amount of refrigerant leakage flowing into or out of the pressure-adjusting chamber 23. Grooves in the peripheries of the pressure-sensing piston 19 and of the valve seat-forming member 22 contain sealing members 37 and 38, such as piston rings. Each of the sealing members 37 and 38 has the shape of a C-shaped ring which is circumferentially partially cut out, and is made of a material having low sliding resistance, such as polytetrafluoroethylene.

[0028] The circumferential length of each cut-off portion of the members 37 and 38 is adjusted to determine the refrigerant flow rate in both directions between the high-pressure port 18 and the pressure-adjusting chamber 23. This setting adjusts the rise and fall characteristics of the valve lift.

[0029] The control valve 11c in Fig. 5 (fourth embodiment) response to a rapid change of the controlled pressure Pc for controlling the valve lift of the valve section 13. In the control valve 11c the pressure-sensing section 12 is disposed between the valve section 13 and the solenoid 14. The pressure-sensing piston 19 that receives the pressure Pc is fixed to the piston rod 26 integrally formed with the valve element 25. The annular pressure-adjusting chamber 23 is defined by the first body cylinder 17 and the pressure-sensing piston 19. A spring 39 urges the piston rod 26 via the pressure-sensing piston 19 in valve-opening direction and against the dis-

charge pressure Pd.

[0030] When the discharge pressure Pd rapidly increases and the suction pressure Ps rapidly decreases, the value of the differential pressure ($P_d - P_s$) between the opposite ends of the valve element 25 and the piston rod 26 increases, whereby the valve lift is increased. The pressure Pc on the downstream side of the valve section 13 as well rapidly increases. At this time, since the pressure-sensing piston 19 has a sufficiently larger pressure-receiving area than the valve element 25, a force is generated which causes the pressure-sensing piston 19 to further move away from the valve section 13, and the force causes the piston rod 26 fixed to the pressure-sensing piston 19 to act in valve-opening direction. The force of the pressure-sensing piston 19 acting in valve-opening direction is additionally applied to the valve element 25, and thereby causes the valve lift to promptly increase, and hence the discharge pressure Pd and the pressure Pc in the crankcase to sharply increase. In a short time, when the pressure in the pressure-adjusting chamber 23 becomes equal to the pressure Pc, the discharge pressure Pd, the pressure Pc in the crankcase, the suction pressure Ps, and the valve lift promptly returns to their original states. Of course, also when the compressor rotational speed is rapidly decreased, the control valve 11c operates promptly, similarly to the above, to thereby make it possible to promptly restore the compressor to the predetermined discharge capacity.

[0031] The control valve 11c of Fig. 6 (fifth embodiment) responds to a rapid change of the suction pressure Ps for controlling the valve lift of the valve section 13. The cylinder 17 of the first body 15 in this case is formed in an end facing toward the solenoid 14. The first body 15 guides the piston rod 26. The pressure-sensing piston 19 which is fixed to the piston rod 26 integrally formed with the valve element 25 is disposed in the cylinder 17. A spring 27 urging the piston rod 26 in valve-opening direction via the pressure-sensing piston 19 is arranged in the annular pressure-adjusting chamber.

[0032] When the discharge pressure Pd rapidly increases, and the suction pressure Ps rapidly decreases, the differential pressure ($P_d - P_s$) between the opposite ends of the valve element 25 and the piston rod 26 increases, whereby the valve lift is increased. At this time, since the pressure-sensing piston 19 has a sufficiently larger pressure-receiving area than the valve element 25, a force is generated which causes the pressure-sensing piston 19 to further move away from the valve section 13, and the force causes the piston rod 26 fixed to the pressure-sensing piston 19 to act in valve-opening direction. Therefore, the force in valve-opening direction is additionally applied to the valve element 25, and causes the valve lift to promptly increase, and hence the pressure Pc in the crankcase to sharply increase, to thereby promptly cause the discharge capacity to change in decreasing direction. In a short time, namely when the pressure in the pressure-adjusting chamber 23 becomes equal to the suction pressure Ps, the discharge pressure

P_d , the pressure P_c in the crankcase, the suction pressure P_s , and the valve lift promptly return to their original states. When the compressor rotational speed is rapidly decreased, the control valve 11c operates promptly, similarly to the above, to promptly restore the compressor to the predetermined discharge capacity.

[0033] In the control valve 11e of Fig. 7 (sixth embodiment) the pressure-sensing section 12 sensitively responds to a rapidly increasing change of the discharge pressure P_d but insensitively responds to a rapidly decreasing change of the discharge pressure P_d . A main passage for high-pressure refrigerant in this case does not extend through the pressure-sensing section 12.

[0034] The pressure-sensing piston 19 is equipped with a check valve mechanism for switching sensitivity between when a rapidly increasing change of the discharge pressure P_d occurs and when a rapidly decreasing change occurs. The check valve mechanism comprises a passage with a stepped portion in the pressure-sensing piston 19 for communication between the high-pressure port 18 and the pressure-adjusting chamber 23. A ball-shaped valve element 40 is arranged in a large-diameter passage extending toward the high-pressure port 18. The pressure-sensing piston 19 is urged by a leaf spring 42 engaged with the open end of a cylinder-forming member 41 which is integral with the valve seat-forming member 22. The cylinder-forming member 41 accommodates the pressure-sensing piston 19, such that the pressure-sensing piston 19 is in contact with the shaft 24. The leaf spring 42 also serves to prevent that the valve element 40 may fall out. The shaft 24 extends axially movably through the cylinder-forming member 41 with a predetermined clearance. The valve hole of the valve seat-forming member 22 directly opens into the high-pressure port 18. The first body 15 carries a strainer 43 covering the high-pressure port 18 and the pressure-sensing section 12.

[0035] When the discharge pressure P_d rapidly increases, the check valve mechanism in the pressure-sensing piston 19 is closed. The pressure-sensing piston 19 having a larger pressure-receiving area than the valve element 25 senses the change of the discharge pressure P_d which has rapidly increased, and causes the valve section 13 to rapidly operate in valve-opening direction, thereby causing the pressure P_c in the crankcase to rise more promptly such that the discharge capacity is promptly controlled in decreasing direction. Inversely, if the discharge pressure P_d has rapidly decreased, the check valve mechanism provided in the pressure-sensing piston 19 is opened by the differential pressure between the rapidly-lowered discharge pressure P_d and the pressure in the pressure-adjusting chamber 23, so that the pressure-sensing piston 19 becomes only little sensitive to the change of the rapidly-lowered discharge pressure P_d . This means that the control valve 11e has asymmetric valve-opening characteristics, i.e. high responsiveness to a rapidly increasing change of the discharge pressure P_d , but low responsiveness to a

rapidly decreasing change of the discharge pressure P_d . Therefore, e.g. even if the compressor performs a transient response to a rapid change in the discharge pressure P_d in the increasing direction to cause the discharge pressure P_d to rapidly change in the decreasing direction, the compressor is prevented from performing a transient response to a rapid change in the discharge pressure P_d in the decreasing direction. This prevents occurrence of a hunting phenomenon.

[0036] In the control valve 11f of Fig. 8 (seventh embodiment) the check valve mechanism of the pressure-sensing section 12 is using a poppet valve instead.

[0037] A valve element 40a in the form of a mushroom is arranged in a large-diameter passage in the pressure-sensing piston 19 extending toward the high-pressure port 18. The passage connects the high-pressure port 18 and the pressure-adjusting chamber 23. A spring 44 which has low spring force urges the valve element 40a in valve-closing direction.

[0038] The check valve mechanism in the control valve 11g of Figs 9 and 10 (eighth embodiment) is designed as a reed valve.

[0039] The check valve mechanism comprises a through hole formed through the pressure-sensing piston 19 such that the through hole communicates between the high-pressure port 18 and the pressure-adjusting chamber 23. A valve element 40b is movably arranged such that it opens and closes the through hole at the end of the pressure-sensing piston 19 facing toward the high-pressure port 18. The valve element 40b comprises a film-like part which is easily bent in response to the differential pressure between the discharge pressure P_d and the pressure in the pressure-adjusting chamber 23, and a base part fixed to the pressure-sensing piston 19. Both parts are integrally formed of rubber or a flexible resin. The base part is fitted in a fixing through hole formed through the pressure-sensing piston 19. A portion of the film-like part close to the base part is retained by the leaf spring 42 securing the valve element 40b to the pressure-sensing piston 19.

[0040] When the compressor rotational speed changes only gently, and also when the compressor rotational speed rapidly increase to increase the discharge pressure P_d , the check valve mechanism of the pressure-sensing section 12 is closed (Fig. 9). When the compressor rotational speed rapidly decreases to rapidly decrease the discharge pressure P_d , the check valve mechanism is opened by the differential pressure between the discharge pressure P_d and the pressure in the pressure-adjusting chamber 23, (Fig. 10).

[0041] The control valve 11h in Fig. 11 (ninth embodiment) has a check valve mechanism with a differently configured a reed valve.

[0042] A valve hole is formed by a gap between the outer periphery of the pressure-sensing piston 19 and the inner wall of the cylinder forming part 41 of the member 22. A valve element 40c can block the valve hole at the side of the high-pressure port 18. A central portion

of the valve element 40c is sandwiched between the leaf spring 42 and the pressure-sensing piston 19. The valve element 40c may be formed by a circular film made of rubber or a flexible resin.

[0043] When the compressor rotational speed changes gently only and also when the compressor rotational speed rapidly increases to increase the discharge pressure P_d , the valve element 40c is brought into intimate contact with upper end faces of the pressure-sensing piston 19 and the cylinder forming part 41 to close the valve hole (left half of Fig. 11). When the compressor rotational speed is rapidly decreased to rapidly decrease the discharge pressure P_d , the valve element 40c is bent upward in an outer edge portion due to the differential pressure between the discharge pressure P_d and the pressure in the pressure-adjusting chamber 23, to clear the valve hole (right half of Fig. 11).

[0044] The control valve 11i in Fig. 12 (tenth embodiment) includes a sensitivity-switching mechanism for switching the sensitivity or the responsiveness between when the discharge pressure P_d rapidly increases and when it rapidly decreases.

[0045] The sensitivity-switching mechanism switches ease of a refrigerant flow into or out of the pressure-adjusting chamber 23. The outer periphery of the pressure-sensing piston 19 is tapered such that the outer diameter progressively decreases from the side of the high-pressure port 18 to the side of the pressure-adjusting chamber 23. A gap between the outer periphery of the pressure-sensing piston 19 and the cylinder wall of the cylinder forming part 41 has the narrowest restriction at an upper end of the gap. From there, the gap progressively increases in passage cross-sectional area into the pressure-adjusting chamber 23. Assuming that the cross-sectional area of the refrigerant passage is suddenly expanded on the high-pressure port 18 side of the restriction, and refrigerant fails to flow from the restriction into the suddenly-expanded portion, a contracted flow is produced there. The pressure-sensing section 12 has a characteristic that insofar as the differential pressure between pressure in the high-pressure port 18 and the pressure in the pressure-adjusting chamber 23 is the same, the flow rate is smaller when refrigerant having entered the high-pressure port 18 passes through a restriction after being suddenly restricted in flow than when refrigerant in the pressure-adjusting chamber 23 passes through the restriction after being progressively restricted in flow. For the same value of the differential pressure the flow rate into the chamber 23 will be smaller than the flow rate out of the chamber 23.

[0046] When the compressor rotational speed rapidly increases to rapidly increase the discharge pressure P_d , refrigerant tends to flow from the side of the high-pressure port 18 through the gap between the outer periphery of the pressure-sensing piston 19 and the part 41 into the pressure-adjusting chamber 23. Inversely, when the rotational speed of the compressor rapidly decreases to rapidly decrease the discharge pressure P_d , refrigerant

tends to flow from the pressure-adjusting chamber 23 through the gap around the outer periphery of the pressure-sensing piston 19 toward the high-pressure port 18. There occurs a difference in the flow rates flowing through the gap between when the discharge pressure P_d has rapidly increased and when it has rapidly decreased. The force of the pressure-sensing piston 19 on the valve element 25 in valve-opening direction when the discharge pressure P_d has rapidly increased can be larger than the force of the pressure-sensing piston 19 on the valve element 25 in valve-closing direction when the discharge pressure P_d has rapidly decreased.

15 Claims

1. A control valve (11, 11a, 11b, 11c, 11d, 11e, 11f, 11g, 11h, 11i) for a variable displacement compressor for sensing and responding to a differential pressure between a discharge pressure (P_d) in a compressor discharge chamber and a suction pressure (P_s) in a compressor suction chamber, and for controlling a refrigerant flow rate between the compressor discharge chamber and a compressor crankcase to vary the refrigerant discharge capacity by varying a controlled pressure (P_c) in the crankcase, **characterised by:**

a pressure-sensing section (12) adapted to sense and to respond to a pressure change caused by a rapid compressor rotational speed change by accelerating a pressure change depending valve section motion in valve-opening and/or closing direction.

2. The control valve according to claim 1, **characterised in that** the pressure-sensing section (12) comprises a pressure-sensing piston (19) having a larger pressure-receiving area than a valve element (25) of the valve section (3), that the pressure-sensing piston (19) is coupled to the valve element (25) at least in one of two opposite force transmitting directions, the two force transmitting directions corresponding to the valve opening direction and the valve closing direction of the valve section (12), that the smaller valve element pressure-receiving area is loaded by the discharge pressure (P_d) in valve opening direction away from a valve seat (22a) of the valve section (13), the valve seat (22a) being situated between a discharge pressure high pressure port (18) and a controlled pressure medium pressure port (28), and that the pressure-sensing piston pressure receiving area is loaded in one of the two force transmitting directions by one of the discharge pressure (P_d), of the suction pressure (P_s) or of the controlled pressure (P_c), and in the other opposite force transmitting direction by a pressure in a pressure adjusting chamber (23) bounded by the pressure sensing

piston (19), the pressure in the pressure adjusting chamber (23) being derived from one of the discharge pressure (Pd), of the suction pressure (Ps) or of the controlled pressure (Pc), respectively.

3. The control valve as in claim 1, **characterised in that** the pressure-sensing piston (19) is coupled to the valve element (25) for a force transmission in both opposite force transmitting directions..
4. The control valve as in claim 2, **characterised in that** the pressure-sensing piston (19) is disposed in a discharge pressure introducing high-pressure port (18), that a shaft (24) is secured to the pressure-sensing piston (19) and extends through the valve seat (22a) to the valve element (25), and that the shaft (24) either is separate from or is integral with the valve element (25).
5. The control valve according to claim 2, **characterised in that** a piston rod (26) is integral with the valve element (25), and that an end face of the piston rod (26) remote from an end face of the valve element (25) loaded by the discharge pressure (Pd) is loaded by the suction pressure (Ps).
6. The control valve according to claim 2, **characterised in that** the pressure-sensing piston (19) is equipped with flow rate-adjusting means for adjusting an amount of refrigerant leakage between the high-pressure port (18) and the pressure-adjusting chamber (23).
7. The control valve according to claim 4, **characterised in that** the flow rate-adjusting means includes a C-shaped ring (37, 38) provided in a sliding surface of the pressure-sensing piston (19), that the C-shaped ring (37, 38) is formed with a circumferential cut-out the length of which corresponds to the desired amount of leakage, and that the C-shaped ring (37, 38) consists of a material which has low sliding resistance.
8. The control valve as in at least one of claims 1 and 7, **characterised in that** the pressure-sensing piston (19) and the valve element (25) are fixedly interconnected by the piston rod (26), and that the pressure-sensing piston (19) either is disposed in a medium-pressure port (28) from which control pressure (Pc) controlled by the valve section (12) is delivered to the crankcase, or in a low pressure port (29) into which the suction pressure (Ps) is introduced.
9. The control valve according to claim 2, **characterised in that** the pressure-sensing section (13) is equipped with a sensitivity-switching means for increasing the force exerted by the pressure-sensing piston (19) on the valve element (25) when the dis-

charge pressure (Pd) has rapidly increased in comparison to a lower force exerted when the discharge pressure (Pd) has rapidly decreased.

10. The control valve according to claim 9, **characterised in that** the sensitivity-switching means includes a check valve in a passage extending through the pressure-sensing piston (19) between the high-pressure port (18) and the pressure-adjusting chamber (23), the check valve blocking flow from the high-pressure port (18) to the pressure-adjusting chamber (23), and allowing flow from the pressure-adjusting chamber (23) to the high-pressure port (18).
11. The control valve according to claim 9, **characterised in that** the sensitivity-switching means includes a check valve comprising a film-like valve element (40b) disposed on the pressure-sensing piston (19) such that the film-like valve element (40b) extends over a gap formed around an outer periphery of the pressure-sensing piston (19) at the side of the high-pressure port (18) for blocking flow via the gap from the high-pressure port (18) to the pressure-adjusting chamber (23) and allowing flow from the pressure-adjusting chamber (23) to the high-pressure port (18).
12. The control valve according to claim 8, **characterised in that** the sensitivity-switching means includes a gap formed around the outer periphery of the pressure-sensing piston (19), that the outer periphery of the pressure-sensing piston (19) is tapered in the gap such that the flow passage cross-sectional area of a gap progressively increases from the side of the high-pressure port (18) to the side of the pressure-adjusting chamber (23).

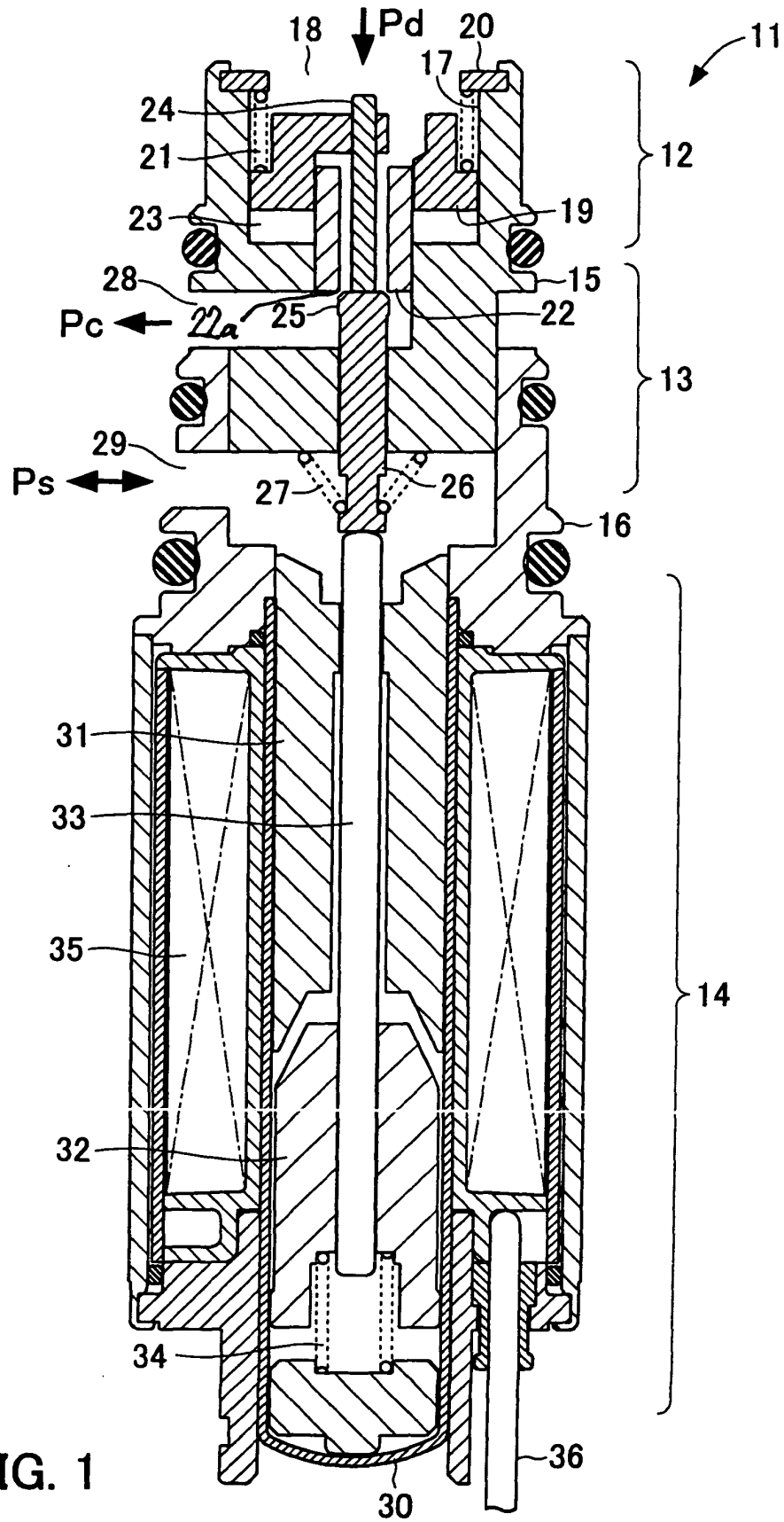


FIG. 1

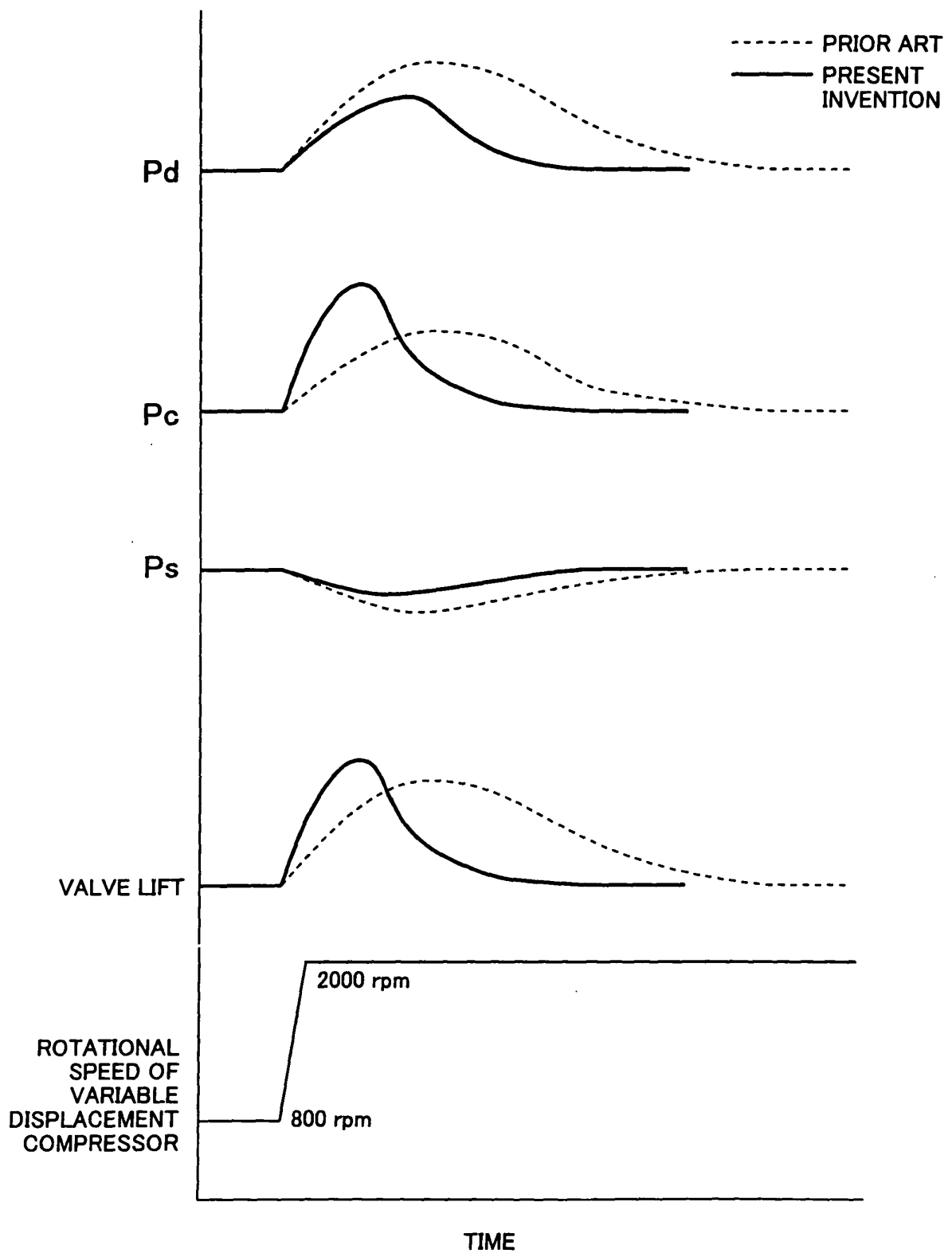
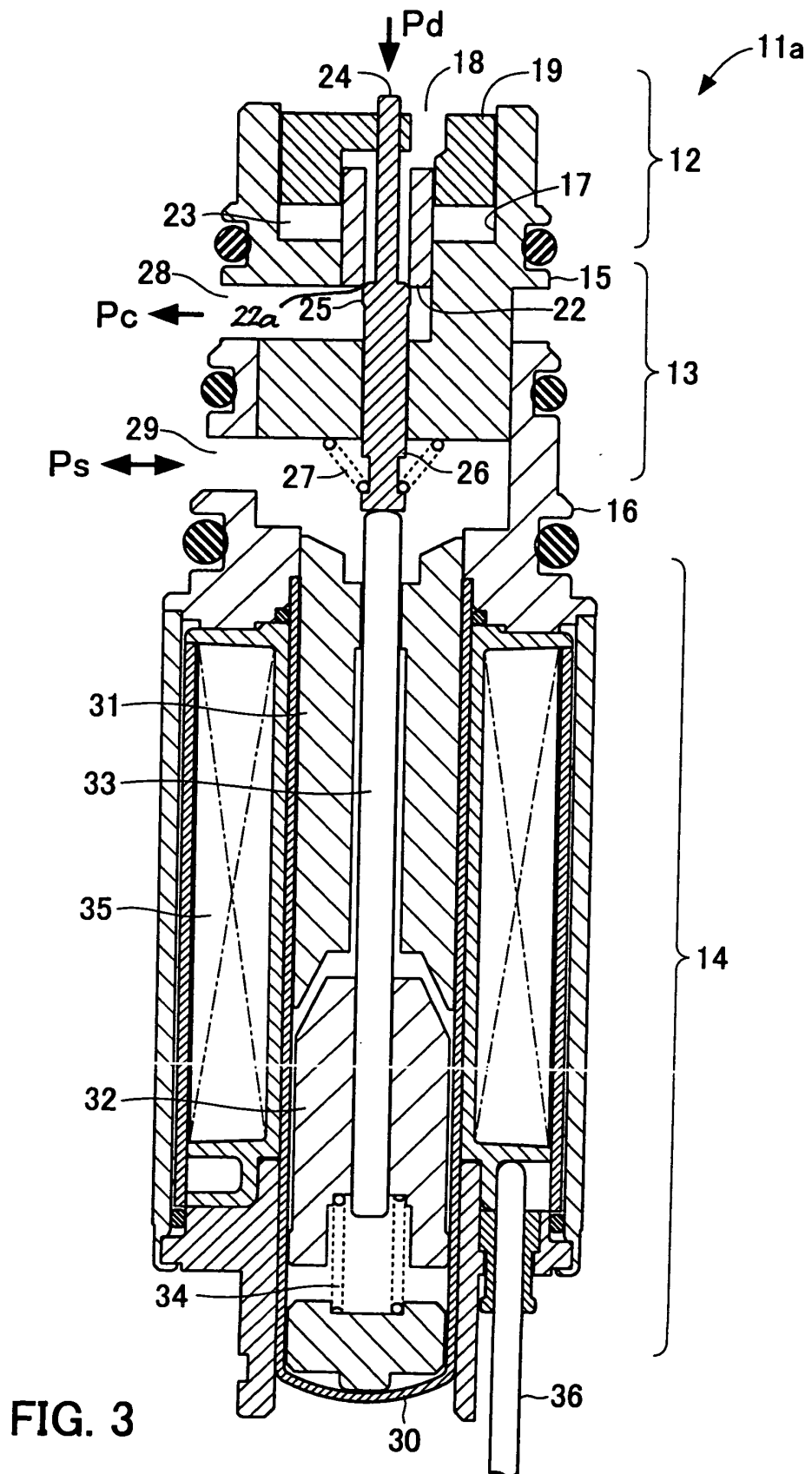
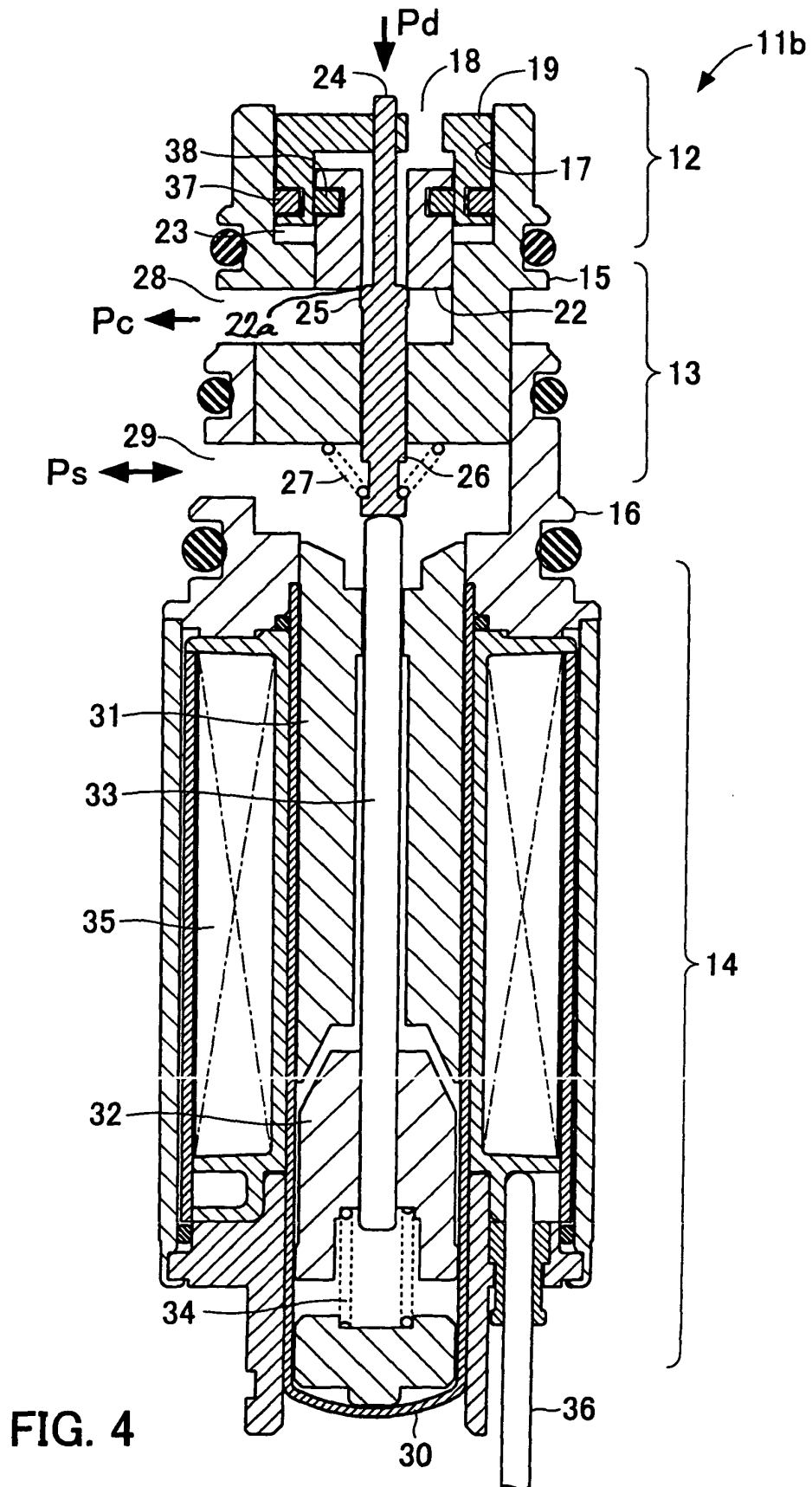


FIG. 2





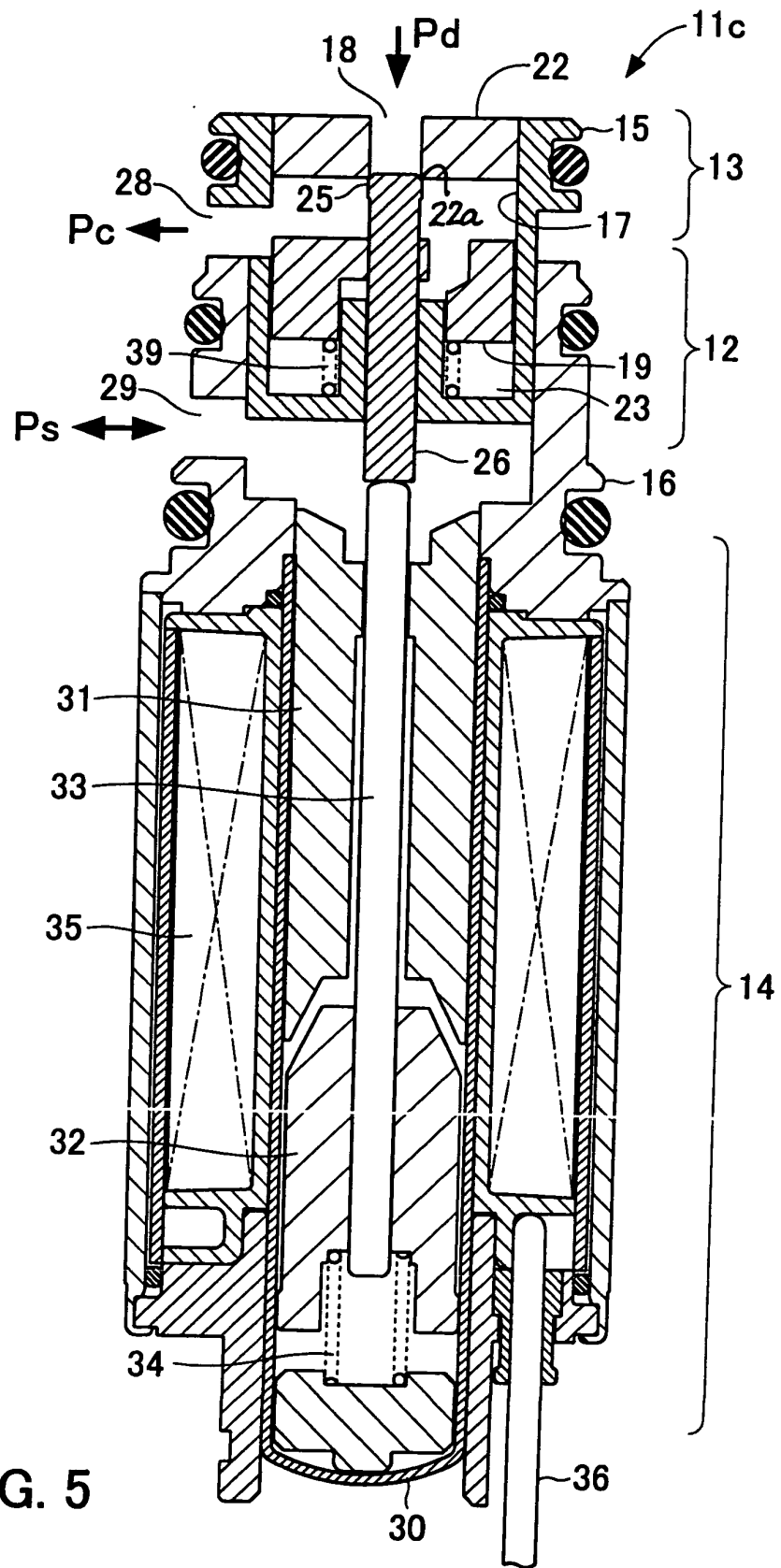


FIG. 5

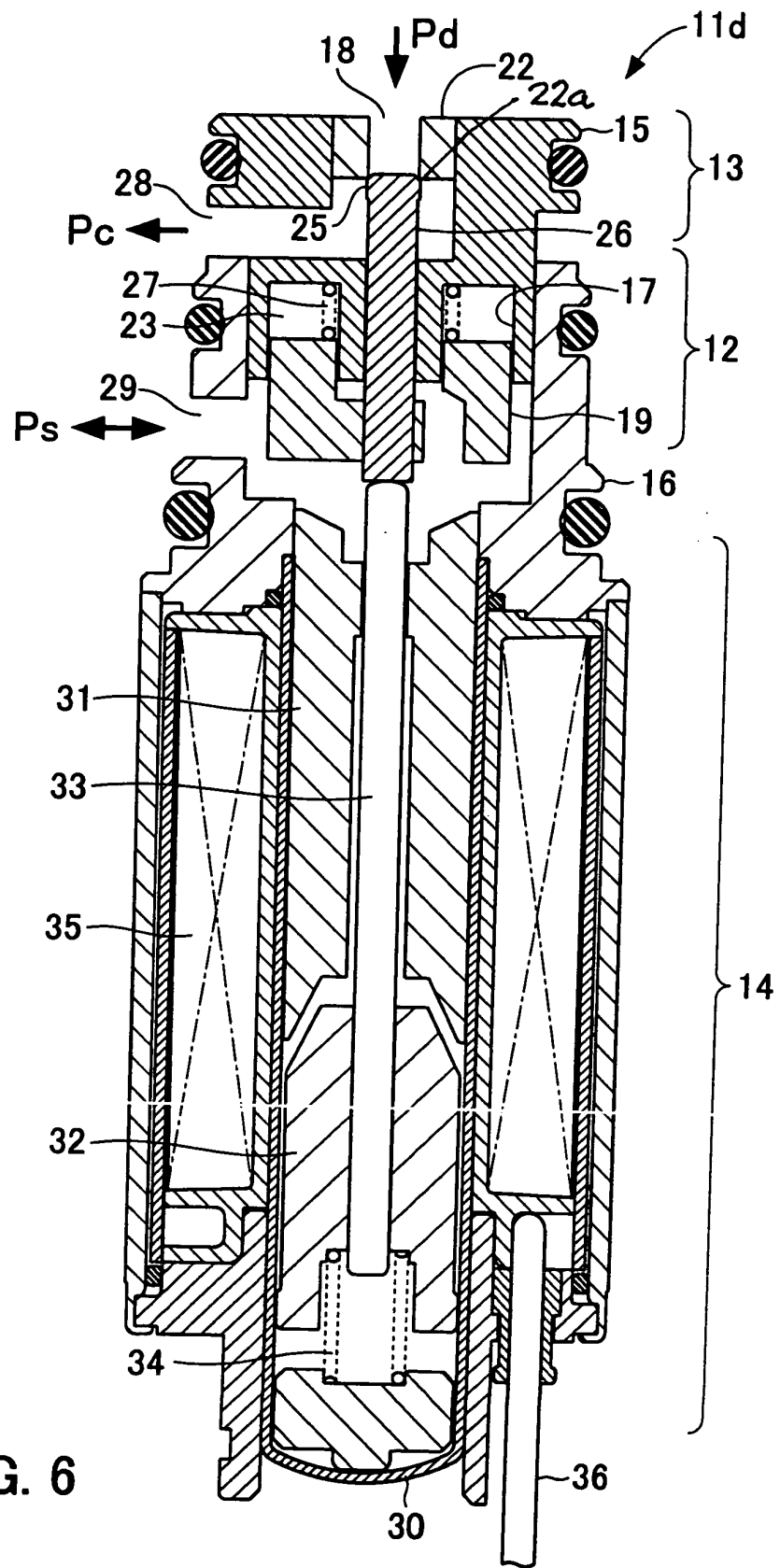


FIG. 6

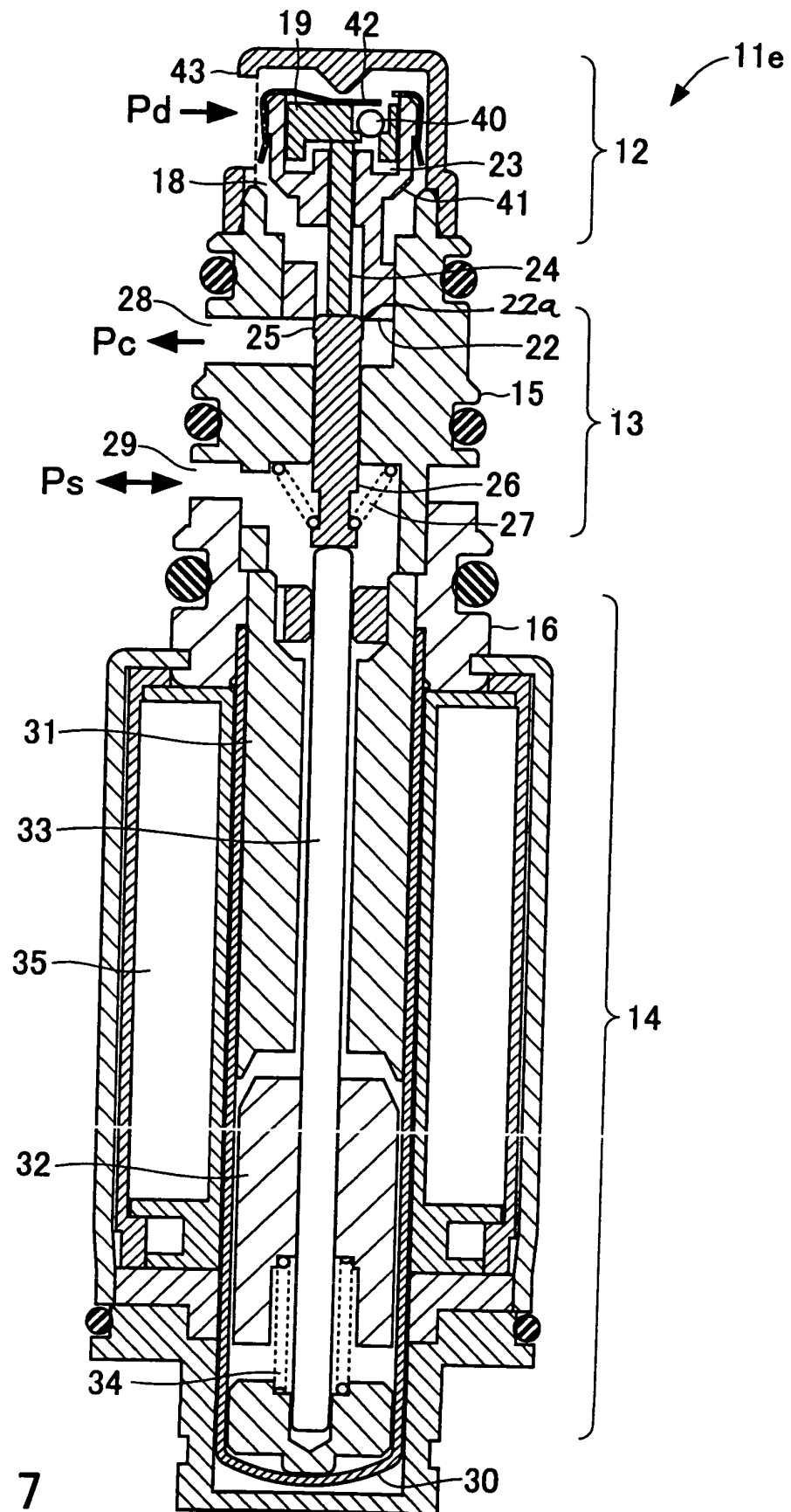


FIG. 7

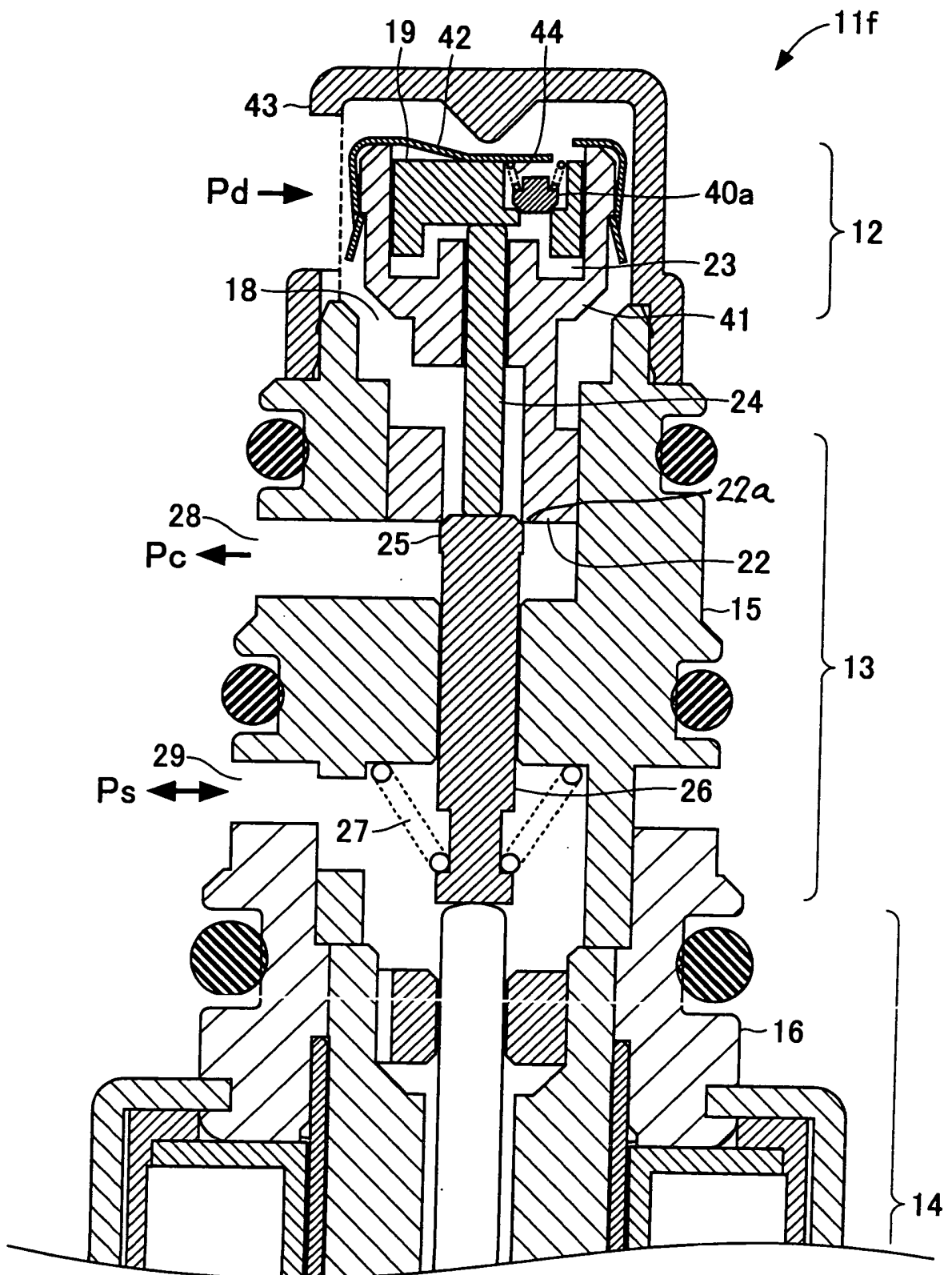


FIG. 8

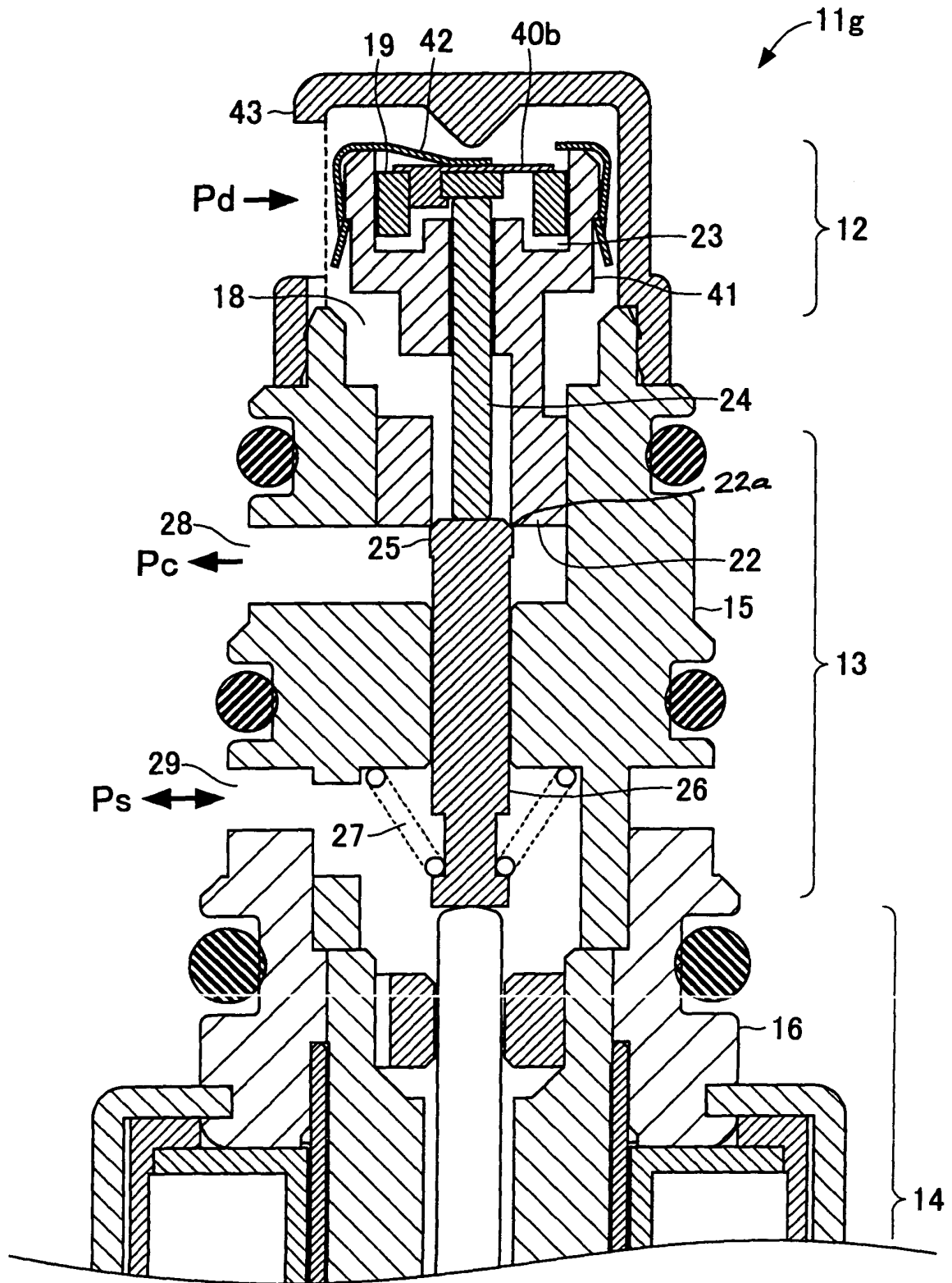


FIG. 9

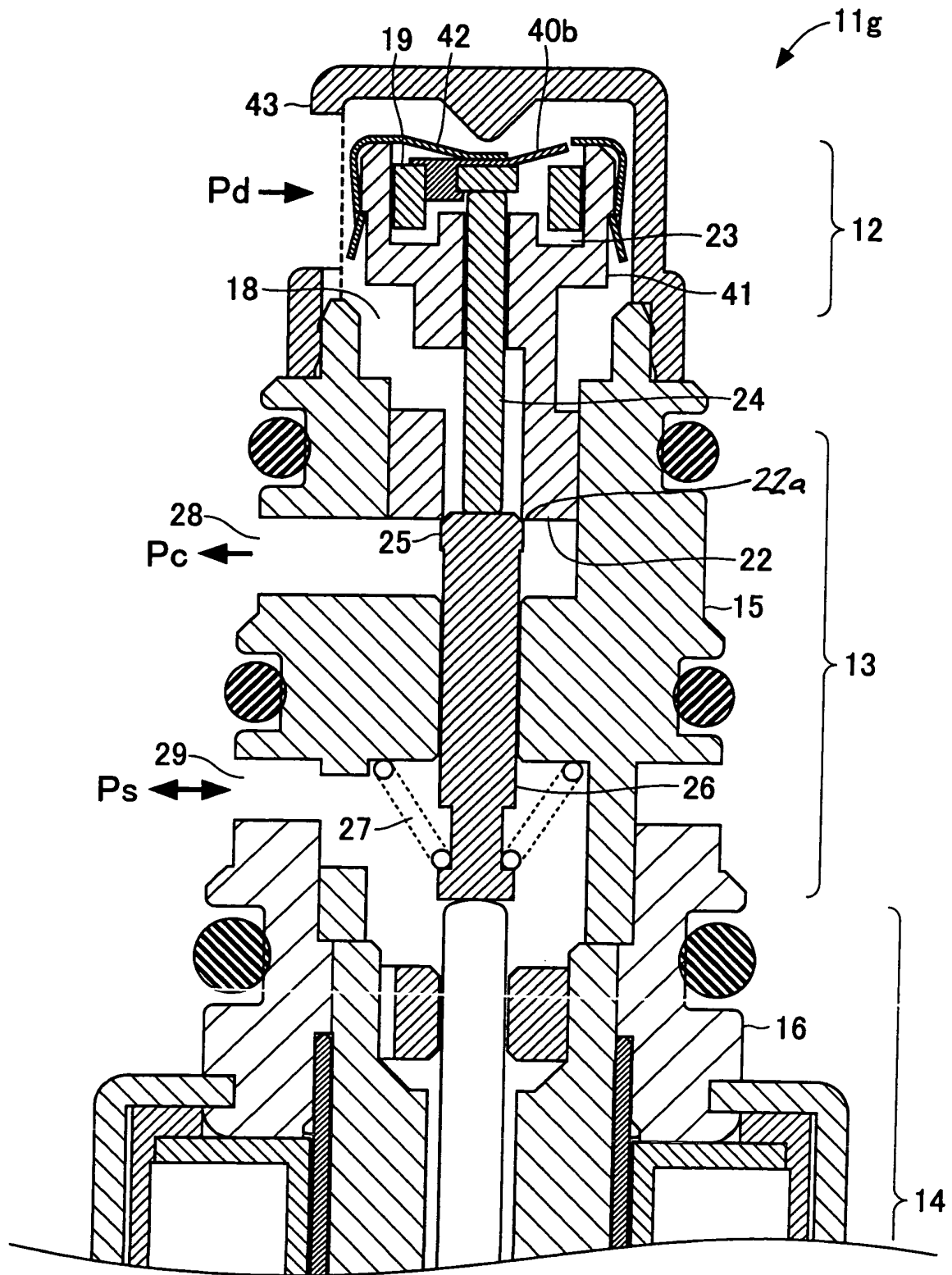


FIG. 10

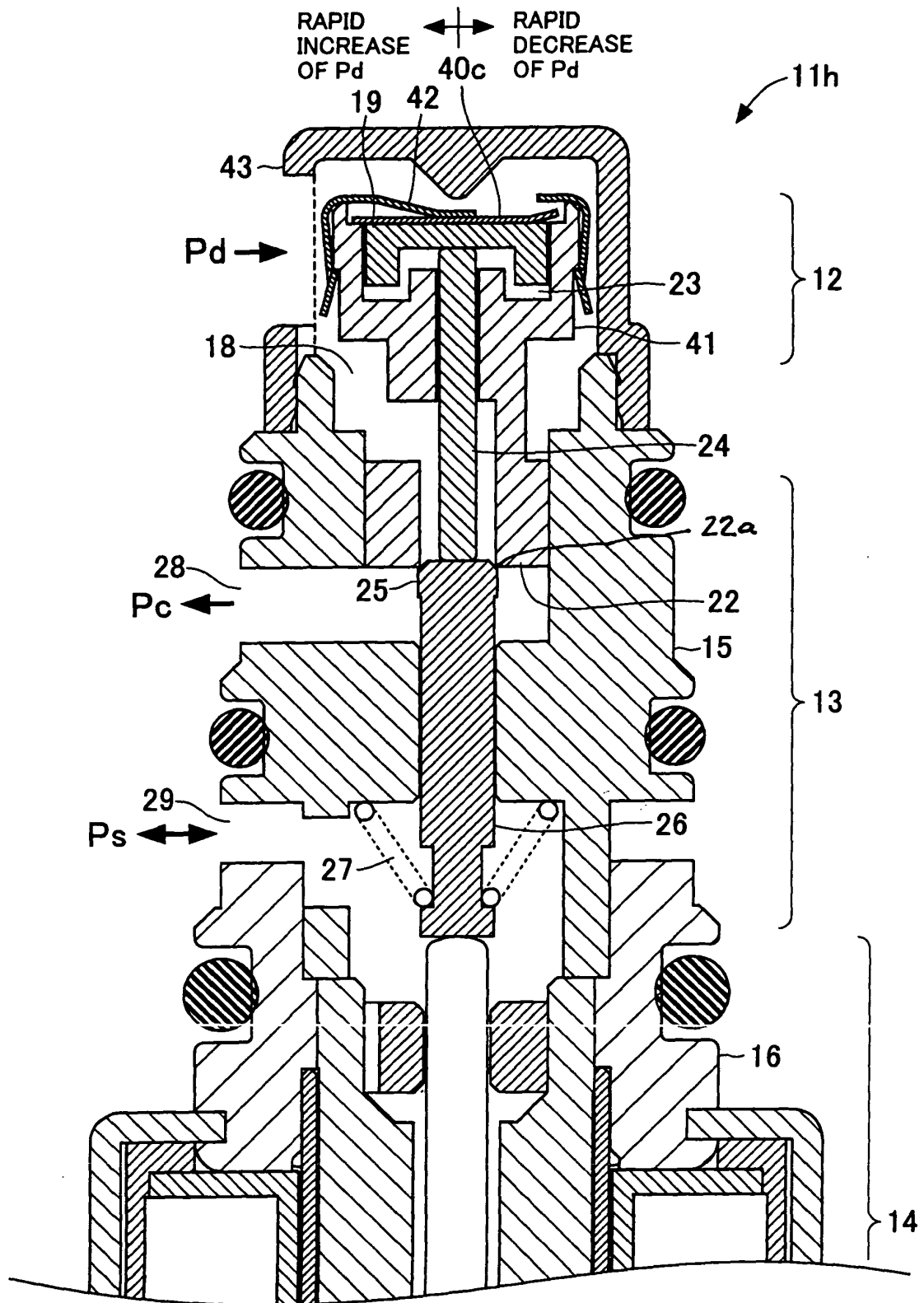


FIG. 11

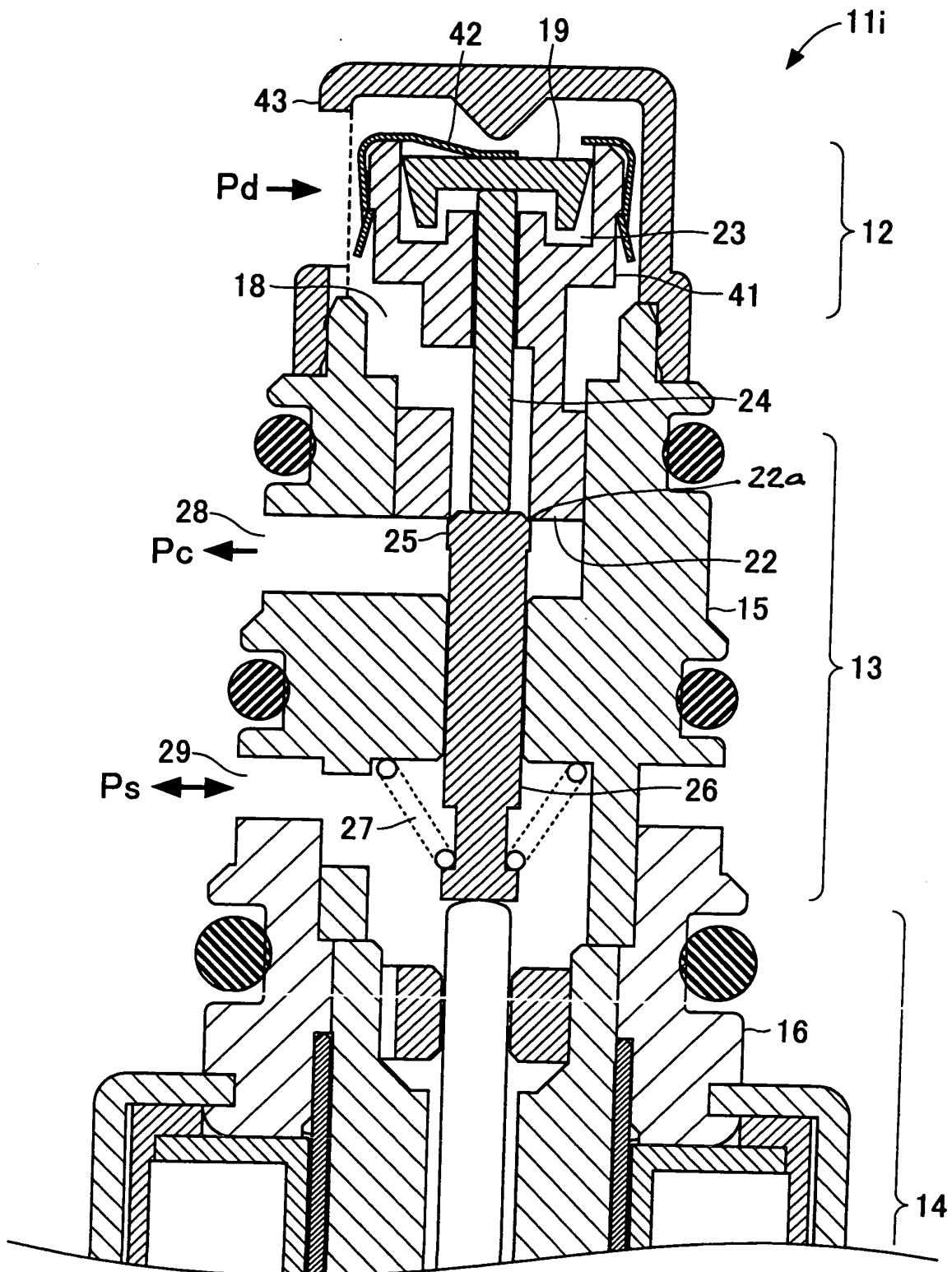


FIG. 12



European Patent
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EUROPEAN SEARCH REPORT

Application Number
EP 05 01 6774

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			F04B
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 13 December 2005	Examiner Pinna, S
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13-12-2005

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