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### (54) Capacity control valve for variable displacement compressor

(57) A compact, low-cost capacity control valve 30 for a variable displacement compressor 1 is an integrated structure of first and at least second control valves 30A, 30B. The first control valve 30A, placed on a discharge passageway, functions as a variable orifice whose cross-sectional area can be set as desired by varying the power of a solenoid unit 30C. Part of refrigerant discharged at pressure  $P_dH$  is supplied to the crank chamber 15, in which the pressure is  $P_c$ . The second control valve 30B controls the flow to the crank chamber 15 such that the differential pressure between upstream and downstream pressures  $P_dH$ ,  $P_dL$  will be regulated at a specified level. This allows to reduce the size of the solenoid unit 30C because the first control valve 30A does not need a large force to yield a small differential pressure as required for the control operation.

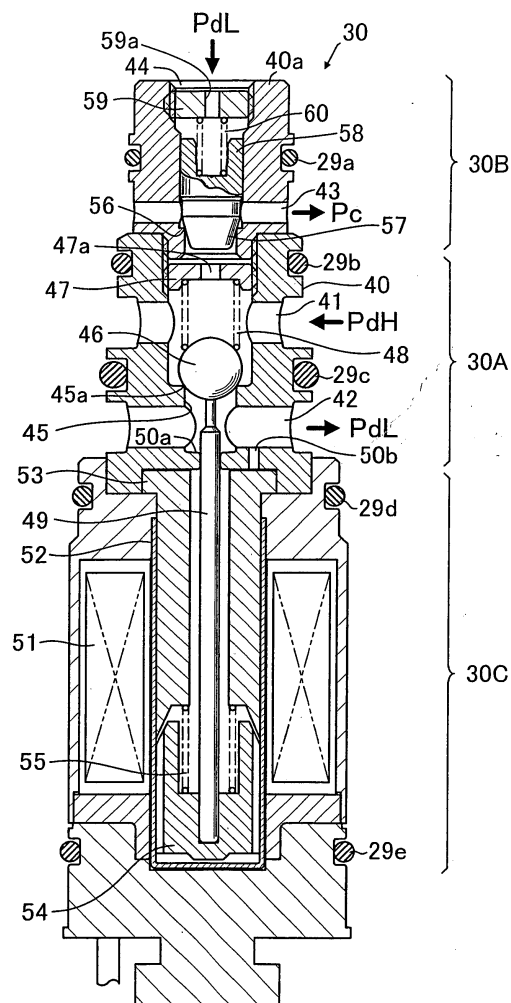


FIG. 2

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## Description

**[0001]** The present invention relates to a capacity control valve according to the preamble part of claim 1.

**[0002]** Automotive air conditioning systems employ a compressor. To obtain a required cooling capacity without restriction by the speed of the driving engine a variable displacement compressor is used allowing to change the capacity independent from the engine speed. The rotation of an inclined wobble plate produces inclination depending piston strokes. The compressor capacity (i.e., the amount of refrigerant being discharged from the compressor) can be varied by changing the wobble plate inclination angle by a part of the pressurized refrigerant introduced into the gastight crank chamber of the compressor. A change of the crank chamber pressure creates a state of balance between opposing pressures exerted on the both ends of each piston linked to the wobble plate and allows to vary the wobble plate inclination angle steplessly. For changing the crank chamber pressure, a capacity control valve is installed either between the refrigerant outlet and the crank chamber or between the refrigerant inlet and the crank chamber. Known capacity control valves open or close such that a certain level of differential pressure between their inlet and outlet will be maintained. More specifically, one can set a desired differential pressure by supplying a capacity control valve with an appropriate control current from an external power source. When the engine speed rises, the capacity control valve raises the pressure of refrigerant supplied to the crank chamber so as to reduce the compressor capacity. When in turn the engine slows down, the capacity control valve decreases the crank chamber pressure so as to increase the compressor capacity.

**[0003]** JP-A-2001-107854 (Fig. 3) describes a capacity control valve that regulates the flow of refrigerant being discharged from a variable displacement compressor. The known capacity control valve needs sensors to detect differential pressure, as well as a controller to control the capacity control valve accordingly. Those extra components push up the cost of variable displacement compressor.

**[0004]** JP-A-2001-107854, describes that the flow of refrigerant that is taken into a suction chamber is determined indirectly by detecting differential pressure between two pressure monitoring points with sensors, and that the intake flow rate from the discharge chamber to the crank chamber is maintained at a constant level to regulate the flow discharged from the compressor.

**[0005]** Another noteworthy aspect of automobile air conditioning systems is what kind of refrigerant to choose. While HFC-134a, a chlorofluorocarbon alternative, was widely used, the recent development of supercritical refrigeration cycles using, for example, carbon dioxide poses another challenge to the compressor design, because the refrigerant then has to function in a region exceeding its critical temperature. In a refrigeration

cycle using carbon dioxide and a capacity control valve to control crank chamber pressure according to the compressor's discharge pressure, the differential pressure between the refrigerant outlet and crank chamber could become extremely high because the refrigerant has to be pressurized up to its supercritical region. Then, a high-power solenoid actuator will be needed to produce a sufficiently large force to deal with the high differential pressure, which leads to increased size and cost of the capacity control valve.

**[0006]** It is an object of the invention to provide a compact capacity control valve for use with a flow-controlled variable displacement compressor, which can be applied not only to ordinary refrigeration cycles using HFC-134a, but also to those using supercritical high-pressure refrigerant, without the needs for high-power solenoids or extra pressure sensors.

**[0007]** To solve the above-described problems, the present invention provides a capacity control valve containing the features of claim 1.

**[0008]** The first control valve functions as a variable orifice for refrigerant, the cross-sectional area of which is set by the solenoid unit according to variations in a given external condition. Responsive to differential pressure developed across the variable orifice, the second control valve controls crank chamber pressure such that the differential pressure will be a specified level. By maintaining the level of differential pressure across the orifice with a certain cross-sectional area, the capacity control valve regulates the suction flow or discharge flow of the variable displacement compressor. Also, since its first and second control valves are formed in an integrated way, the capacity control valve detects differential pressure to control refrigerant flows without the need for extra pressure sensors. This feature enables to produce variable displacement compressors at lower costs. Another advantage is that a smaller solenoid can be used to set a required cross-sectional area of a refrigerant passageway because a small differential pressure will suffice. This feature eliminates the need for increased solenoid power even in a supercritical refrigeration cycle using high-pressure refrigerant, thus contributing to miniaturization of capacity control valves for variable displacement compressors.

**[0009]** The proposed structure employs first and second control valves formed in an integrated way. Here, the first control valve controls the cross-section area at a midway point between low-pressure refrigerant passage and suction chamber, or between discharge chamber and high-pressure refrigerant passage, according to a given external condition. The second control valve, on the other hand, detects differential pressure between upstream end and downstream end of the first control valve and controls the crank chamber pressure in such a way that the differential pressure will be maintained at a specified level. This feature of the present invention enables us to construct a smaller variable displacement compressor at lower cost.

**[0010]** Because the first control valve has only to produce a small amount of differential pressure, the solenoid unit can drive the valve with a small power, and thus it is not necessary to increase its size to achieve the purpose. The present invention can easily be applied to refrigeration cycles using HFC-134a in a system that should operate with small differential pressure between discharge chamber and crank chamber, or crank chamber and suction chamber. In addition, the present invention can also be applied to those using high-pressure refrigerant in its supercritical region.

**[0011]** Embodiments of the present invention will be described with reference to the drawings. The description illustrates capacity control valves for a flow-controlled variable displacement compressor that is supposed to discharge refrigerant at a regulated flow rate. In the drawings is:

Fig. 1 a section of a variable displacement compressor, equipped with a first embodiment of a capacity control valve,

Fig. 2 a detailed section of the capacity control valve of Fig. 1,

Fig. 3 a section of a second embodiment of a capacity control valve,

Fig. 4 a section of a third embodiment of a capacity control valve,

Fig. 5 a section of a fourth embodiment of a capacity control valve,

Fig. 6 a section of a fifth embodiment of a capacity control valve,

Fig. 7 a section of a sixth embodiment of a capacity control valve,

Fig. 8 a section of a seventh embodiment of a capacity control valve,

Fig. 9 a section of an eight embodiment of a capacity control valve,

Fig. 10 a section of a ninth embodiment of a capacity control valve,

Fig. 11 a section of a tenth embodiment of a capacity control valve,

Fig. 12 a section of an eleventh embodiment of a capacity control valve,

Fig. 13 a section of a twelfth embodiment of a capacity control valve,

Fig. 14 a section of a thirteenth embodiment of a capacity control valve,

Fig. 15 a section of a fourteenth embodiment of a capacity control valve,

Fig. 16 a section of a fifteenth embodiment of a capacity control valve,

Fig. 17 a section of a sixteenth embodiment of a capacity control valve,

Fig. 18 a section of a seventeenth embodiment of a capacity control valve,

Fig. 19 a section of an eighteenth embodiment of a capacity control valve,

Fig. 20 a section of a nineteenth embodiment of a capacity control valve.

**[0012]** A variable displacement compressor 1 as shown in Fig. 1 is composed of the following three sections: a driving section 100 that receives drive power from a vehicle engine (not shown); a refrigerant compressing section 200 including a gastight crank chamber 15; and a capacity controlling section 300 that controls discharge capacity. An outlet port 1a is connected to a condenser (or gas cooler) 3 through a high-pressure refrigerant line 2. The refrigerant is then routed from the condenser 3 to an expansion valve 4, an evaporator 5, and a low-pressure refrigerant line 6 in that order, and finally returns to an inlet port 1b (closed circuit for refrigeration cycle).

**[0013]** A drive pulley 13 transmits rotational power to a bracket 14 and a shaft 12 protruding from a front housing 11. The crank chamber 15 is a closed space surrounded by the front housing 11 and by a cylinder block 16. The shaft 12 is rotatably installed in the crank chamber 15, across the length of the front housing 11 and the cylinder block 16.

**[0014]** The drive pulley 13 is supported by an angular bearing 17 at the front housing 11. The rotation is directly transmitted to the variable displacement compressor 1, without an intervening clutch mechanisms (e.g., electromagnetic clutch).

**[0015]** A lip seal 18 is placed between the front housing 11 and the front portion of the rotating shaft 12. A support member 19 is fixed to the shaft 12 in the crank chamber 15. A swash or wobble plate 20 is supported in such a way that it can be inclined at an oblique angle relative to the axis of the shaft 12. The swash plate 20 has a guide pin 22, whose spherical top portion is engaged with a support arm 21 protrudingly mounted on the rotational support member 19. The swash plate 20 rotates together with the rotating shaft 12.

**[0016]** Interposed between the rotational support member 19 and the swash plate 20 an inclination-reduc-

ing spring 23 urges the swash plate 20 in a direction to reduce the inclination angle. The maximum inclination is restricted by a protrusion 20a of the swash plate 20. The shaft 12 is rotatably supported at its rear end by a radial bearing 24 mounted into the cylinder block 16.

**[0017]** The cylinder block 16 has a plurality of cylinder bores 16a housing single-headed pistons 25. The swash plate 20 engages with the head of each piston 25 via a shoe 26. Placed between the rotational support member 19 and the front housing 11 is a thrust bearing 28 which receives reaction forces caused by the compression and acting on the rotational support member 19 via the pistons 25 and the swash plate 20.

**[0018]** The capacity controlling section 300 is separated from the refrigerant compressing section 200 by a valve plate 27. The capacity controlling section 300 has a rear housing 31 located next the valve plate 27 and a capacity control valve 30 in the rear housing 31. The rear housing 31 provides the following separate cavities adjacent to the valve plate 27: suction chambers 32, discharge chambers 33, and a communication passage 34. The suction chambers 32 are cavities containing a suction pressure  $P_s$ . The discharge chambers 33 contain a discharge pressure  $P_dH$  and receive the refrigerant compressed by the refrigerant compressing section 200. The communication passage 34 communicates with the crank chamber 15, and contains a crank chamber pressure  $P_c$ . The rear housing 31 also provides the outlet and the inlet ports 1a, 1b as well as a cavity 35 for the capacity control valve 30. Further, the rear housing 31 has several communication holes 36 to 39 formed in its body. The first communication hole 36 connects the inlet port 1b with the suction chambers 32. The second communication hole 37 connects the housing cavity 35 with the communication passage 34, which further leads to the crank chamber 15. The housing cavity 35 can also communicate with the discharge chambers 33 through the third communication hole 38. The fourth communication hole 39 permits the housing cavity 35 to communicate with the outlet port 1a.

**[0019]** A suction relief valve 32v is placed at each cylinder port connecting to the suction chamber 32, on the side of the valve plate 27 adjacent to the cylinder bores 16a. A discharge relief valve 33v is placed similarly at each cylinder port connecting to its corresponding discharge chambers 33, but on the opposite side of the valve plate 27, remote from the cylinder bores 16a. The suction chambers 32, one for each cylinder bore 16a, communicate with each other in the rear housing 31, as well as with the first communication hole 36. Likewise, the discharge chambers 33 communicate with each other in the rear housing 31, as well as with the third communication hole 38. As the pistons 25 reciprocate, the refrigerant gas in the suction chamber 32 is sucked into each cylinder bore 16a through its corresponding suction relief valve 32v and then discharged from those cylinder bores 16a to their corresponding discharge chambers 33 through respective discharge relief valves 33v.

**[0020]** While not shown in Fig. 1, there is an orifice of fixed cross-section between the crank chamber 15 and the suction chambers 32 to release refrigerant from the crank chamber 15 to the suction chambers 32.

**[0021]** The first embodiment of the capacity control valve 30 (Figs 1 and 2) is made up of a first control valve 30A, a second control valve 30B, and a solenoid unit 30C.

**[0022]** The first control valve 30A has two ports 41 and 42 formed in a body 40. Port 41 receives the discharge pressure  $P_dH$  from the discharge chambers 33 through the third communication hole 38 of the rear housing 31 shown in Fig. 1. Port 42 outputs refrigerant at discharge pressure  $P_dL$  that has been reduced at the first control valve 30A, for delivery through the fourth communication hole 39 and a high-pressure refrigerant line 2. Between ports 41 and 42 is provided a bored valve hole 45 for communication of refrigerant, the upstream edge of which is intended to function as a first valve seat 45a. In an upstream space adjacent to the first valve seat 45a, an e.g. ball-shaped first valve element (ball valve element) 46 is placed. The valve hole space communicating with the port 41 accommodates a coil spring 48 that urges the ball valve element 46 in the direction that it closes the passage, and the amount of that spring load can be adjusted by an adjustment screw 47 screwed into the body 40.

**[0023]** The downstream side of the ball valve element 46 is in contact with one end of a shaft 49 that extends in the axial direction of the solenoid unit 30C through the valve hole of the first valve seat 45a. This shaft 49 is supported by a bearing 50a formed in the body 40, and the bearing 50a has a communication hole 50b to equalize the inside pressure of the solenoid unit 30C with the discharge pressure  $P_dL$ .

**[0024]** The solenoid unit 30C contains a solenoid coil 51 that has a cylindrical cavity, in which a sleeve 52 is fitted. A fixed core 53 is pressed into the sleeve 52. The sleeve 52 contains a plunger 54 that can slide in axial direction while being urged by a coil spring 55 in the downward direction in Fig. 2. The plunger 54 is fixed to the lower end of the shaft 49 running coaxially through the core 53. This arrangement permits the capacity control valve 30 to operate as follows.

**[0025]** When the solenoid coil 51 is in de-energized state, the plunger 54 is set away from the core 53 by coil spring 55, causing the shaft 49 extending from the plunger 54 to loose contact with the ball valve element 46. The first control valve 30A is fully closed because the freed ball valve element 46 is seated on the first valve seat 45a, being urged by another coil spring 48. When the solenoid coil 51 is energized, the plunger 54 is attracted by the magnetized core 53 to push the ball valve element 46 via the shaft 49 in the valve-opening direction. The amount of the thus caused movement, or the valve lift (or opening-degree) of the ball valve element 46 is proportional to the value of the supplied electrical current. This means that the control current deter-

mines the cross-sectional area of the refrigerant passageway of the first control valve 30A which functions as a variable orifice, and changes its cross-sectional size as specified by the control current to allow the discharged refrigerant to pass through.

**[0026]** The solenoid unit 30C described above is intended, not for directly controlling a high-pressure refrigerant flow, but for controlling the first control valve 30A so that a small differential pressure will be produced depending on the discharge flow rate  $Q_d$  of the refrigerant passing therethrough. Since only a small power is needed to achieve the purpose, it is possible to reduce the size of the solenoid unit 30C.

**[0027]** The second control valve 30B has a body 40a screwed into the body 40 of the first control valve 30A so that the two control valves 30A and 30B are stacked in series. The body 40a has two ports 43 and 44. One port 43 is used to apply control pressure  $P_c$  to the crank chamber, and the other port 44 is used to introduce discharge pressure  $P_dL$  that has been reduced at the first control valve 30A. The body 40a has an opening at its bottom end, which communicates with the port 41 to receive discharge pressure  $P_dH$  of the discharge chambers 33 through a communication hole 47a formed on an adjustment screw 47. Between the opening and the port 43, a second valve seat 56 is formed as an integral part of the body 40a. Placed opposite to this second valve seat 56 in the port 43 is a second valve element 57. The second valve element 57 is a taper-shaped member that is integrally formed with a cylindrical piston 58 which is axially movable within a cylinder that is bored on the axis of the body 40a. A coil spring 60 installed at an upper end portion of the piston 58 urges the second valve element 57 in the valve-closing direction. The respective spring load depends on how deep an adjustment screw 59 is screwed into the body 40a. The adjustment screw 59 has a central through hole 59a as a passage introducing the reduced discharge pressure  $P_dL$  from the port 44 to the space above the piston 58. The second valve element 57 and the piston 58 thus receive different axial pressures at their both endfaces. That is, the second valve element 57 receives discharge pressure  $P_dH$  from the port 41, while the piston 58 receives discharge pressure  $P_dL$  from the port 44. The then created differential pressure  $\Delta P$  determines the lift of the second valve element 57. More specifically, the differential pressure  $\Delta P$  is generated when refrigerant flows through a passage with a certain cross-sectional area as determined by the first control valve 30A. Then the second control valve 30B functions as a constant differential pressure valve that controls the amount of refrigerant flowing into the crank chamber 15 in such a way that the above differential pressure  $\Delta P$  will be maintained at a constant level.

**[0028]** Several O-rings are provided around the periphery of the capacity control valve 30, namely an O-ring 29a to seal the gap between the ports 44 and 43, an O-ring 29b between the ports 43 and 41, yet another

O-ring 29c between the ports 41 and 42, another O-ring 29d between the port 42 and the solenoid unit 30C, and yet another O-ring 29e to seal the solenoid unit 30C against the surrounding atmosphere.

**[0029]** When the shaft 12 is driven, the swash plate 20 will wobble to produce reciprocating motion of the pistons 25 linked to the outer regions of the swash plate 20. Refrigerant is sucked from the suction chambers 32 into the cylinder block 16, is compressed and is discharged into the discharge chambers 33.

**[0030]** When the solenoid unit 30C is de-energized the first control valve 30A is fully closed. Refrigerant discharged to the discharge chambers 33 is entered to the crank chamber 15 in its entirety via the second control valve 30B. The variable displacement compressor 1 runs in the minimum capacity mode.

**[0031]** When a predetermined amount of control current is supplied to the solenoid unit 30C, the first control valve 30A shows a predetermined opening degree (valve lift). The first control valve 30A acts as an orifice with a certain cross-sectional size, allowing a flow of refrigerant through the high-pressure refrigerant line 2 leading to the condenser 3. This develops a certain amount of differential pressure  $\Delta P (=P_dH-P_dL)$  across the orifice, depending on the actual discharge flow rate  $Q_d$  of the refrigerant passing through it.

**[0032]** In the second control valve 30B, the second valve element 57 and the piston 58 respond to the differential pressure  $\Delta P$  across the first control valve 30A. The second control valve 30B controls the flow from the discharge chambers 33 to the crank chamber 15 such that the differential pressure  $\Delta P$  will be maintained at a constant level. This control action may vary the capacity of the variable displacement compressor 1 as needed to regulate the discharge flow.

**[0033]** The discharge flow rate is determined depending on how much refrigeration capacity is required in the present refrigeration cycle. Actually, the refrigeration capacity is calculated from various parameters, which include: engine rotation speed, vehicle speed, accelerator pedal position, indoor and outdoor temperatures, set temperatures, and monitoring signals supplied from various temperature and pressure sensors. The amount of the electrical current for the solenoid coil 51 is determined on the basis of this calculation result.

**[0034]** When the engine speed rises, the discharge flow rate is increased accordingly. This develops an increased differential pressure  $\Delta P$  across the first control valve 30A. In response to  $\Delta P$ , the second control valve 30B further lifts its valve element, so that more refrigerant will be supplied from the discharge chambers 33 to the crank chamber 15. The pressure  $P_c$  rises, and the variable displacement compressor 1 is controlled in an output-reducing condition. The compressor 1 now operates with a smaller discharge capacity, suppressing the discharge flow rate of refrigerant, and thus reducing the differential pressure  $\Delta P$ . In this way, the discharge flow rate  $Q_d$  of refrigerant is regulated by controlling the sec-

ond control valve 30B so that the differential pressure across the orifice (i.e., the first control valve 30A being configured as a proportional solenoid valve) will be maintained at a constant level.

**[0035]** When the engine rotation speed drops, the discharge flow rate will decrease and reduce the differential pressure across the first control valve 30A accordingly. The discharge pressure PdH falls, and the second control valve 30B operates such as to reduce the refrigerant flow from the discharge chambers 33 to the crank chamber 15. Pressure Pc in the crank chamber 15 falls and causes the compressor 1 to operate in a capacity-increasing condition, thus recovering the discharge. In this way, the discharge flow rate Qd of refrigerant is maintained at the constant level.

**[0036]** The present invention combines the first control valve 30A that functions as a variable orifice controlled by the solenoid unit 30C and the second control valve 30B that controls the pressure in the crank chamber 15 in an integrated way, thus providing a compact, space-saving design for the capacity control functions (i.e., regulating the flow rate Qd of refrigerant discharged from the variable displacement compressor 1).

**[0037]** The capacity control valve 30 of the second embodiment (Fig. 3) differs from the first embodiment of Fig. 2 in that the ball valve element 46 of its first control valve 30A is arranged such that it will allow more refrigerant to pass through when it is displaced following the stream of the refrigerant. In other words, the ball valve element 46, or the first valve element, is placed on the downstream side of the first valve seat 45a. The plunger 54 and the core 53 have swapped their positions in the solenoid unit 30C.

**[0038]** The first control valve 30A stays fully closed by the coil spring 55 when the solenoid unit 30C is not energized. Refrigerant entering the port 41 at discharge pressure PdH is led to the crank chamber 15 in its entirety through the second control valve 30B, meaning that the variable displacement compressor 1 now operates in the minimum capacity condition.

**[0039]** When a predetermined amount of control current is supplied to the solenoid coil 51, the plunger 54 is attracted by the core 53 and stops at the point where the attraction force comes into balance with the urging force of the coil spring 55. In this state, the ball valve element 46 is lifted, keeping contact with the shaft 49 due to the force of the coil spring 48, and the consequent gap at the valve seat serves as an orifice with a designated size.

**[0040]** Variations of the engine speed affect the compressor discharge flow. Then, the capacity control valve 30 operates in the same way as described earlier for Fig. 2.

**[0041]** The capacity control valve 30 of the third embodiment in Fig. 4 differs from Figs 2, 3 in that a taper-shaped valve element 61 is placed on the upstream side of the first valve seat 45a to receive a force in the valve-opening direction. Instead of port 44, the capacity con-

trol valve 30 in Fig. 4 has a communication hole 62 in the body 40 to serve the same purpose. For that reason, the port 41 for discharge pressure PdH and port 42 for discharge pressure PdL have swapped their positions in Fig. 4 compared to Fig. 3. Another difference is that the crank chamber 15 receives the discharge pressure PdL behind an orifice.

**[0042]** The first control valve 30A in Fig. 4 has a port 41 formed in the body 40 to receive the discharge pressure PdH from the discharge chambers 33. Port 42 serves to supply the high-pressure refrigerant line 2 with discharge pressure PdL that is reduced by the first control valve 30A. The valve hole 45 communicates between the ports 41 and 42, and its upstream-side edge function as the first valve seat 45a. A taper-shaped first valve element 61 is placed in an upstream-side space, opposite to the first valve seat 45a. The first valve element 61 has an integrated circumferential flange 61 a.

**[0043]** The flange 61a retains one end of a coil spring 48 that is placed around the first valve element 61 and abuts around the first valve seat 45a. The coil spring 48 urges the first valve element 61 in opening direction. The valve element 61 is coupled to an end of the axial shaft 49 of the solenoid unit 30C. With the solenoid unit 30C de-energized, the coil spring 55 seats the first valve element 61 on the first valve seat 45a. The shaft 49 is supported by a bearing 50a at its middle portion adjacent to the first control valve 30A, as well as by another bearing 50c at its bottom end. The bottom-end bearing 50c is press-fitted into the central bore of the fixed core 53.

**[0044]** The second control valve 30B is coupled in series with the first control valve 30A, the space above the piston 58 being closed by a lid 59b. The body 40 has a communication hole 62 to communicate that space with the port 41, through which discharge pressure PdH acts on the back face of the piston 58. This arrangement reduces the needed number of ports of the body 40, thus making it easier to manufacture the capacity controlling section 300, and allows to eliminate some of the otherwise needed O-rings.

**[0045]** The first control valve 30A sets a certain cross-sectional area for the refrigerant passageway in accordance with how much the solenoid unit 30C is energized.

The second control valve 30B is responsive to differential pressure developed across the first control valve 30A to control the flow rate of refrigerant supplied from the discharge chambers 33 to the crank chamber 15. The capacity control valve 30 regulates the flow rate Qd that the variable displacement compressor 1 discharges.

**[0046]** Compared with the first embodiment of Fig. 2, the capacity control valve 30 of Fig. 5 is distinct in the following points. The first control valve 30A employs a spool-shaped valve element 63 as the first valve element. The second control valve 30B uses a taper-shaped valve element 64 as a second valve element. As the counterpart of the spool-shaped valve element

63 a first valve seat 63a is provided as an integral part of the second valve element in the second control valve 30B. This first valve seat 63a is designed to set a required cross-section area for refrigerant passage while moving together with the second valve element.

**[0047]** The second control valve 30B in Fig. 5 has a second valve seat 56 and its corresponding second valve element 64 with a tapered shape. The second valve seat 56 is formed as an integral part of the body 40, in the middle of a refrigerant passageway between the two ports 41 and 43, the former receiving refrigerant from the discharge chambers 33 and the latter delivering refrigerant to the crank chamber 15. Opposite the second valve seat 56, the second valve element 64 is located in an upstream-side space (discharge pressure PdH). The second valve element 64 is urged by a coil spring 66 in the valve-opening direction. Integrally formed with the second valve element 64 is a pressure responsive member 64a, whose base portion detects differential pressure between two different discharge pressures PdH and PdL. The pressure responsive member 64a is installed inside the body 40 in a manner that it can come in contact with or move away from the second valve seat 56 according to the differential pressure acting thereon. The pressure responsive member 64a has a central cavity around its axis, the bottom end of which is open. The pressure responsive member 64a has also a hole 64b in its upper portion, which allows the discharge pressure PdH in the port 41 to reach the central cavity.

**[0048]** The first control valve 30A has a first valve seat 63a at the rim of the bottom opening end of the pressure responsive member 64a. The valve seat 63a cooperates with the spool-shaped valve element 63 located below the bottom opening. The first valve seat 63a and first valve element 63 set an appropriate cross-sectional area for a passageway that delivers refrigerant from one port 41 to another port 42 via the hole 64b of the pressure responsive member 64a.

**[0049]** The valve element 63 is integrally formed with a pressure responsive piston 63p having the same cross-sectional area as the valve hole of the first valve seat 63a. A flange 63b is formed around this valve element 63, on a downstream-side portion remote from the first valve seat 63a. This flange 63b receives the force of the coil spring 48 urging the valve element 63 in the valve-opening direction. Another coil spring 60 is disposed between the pressure responsive piston 63p and the pressure responsive member 64a. The pressure responsive piston 63p is slidably supported in a plug 40b, which seals the bottom of the body 40. The pressure responsive piston 63p may also be pressed upward by the shaft 49 between the solenoid unit 30C and the bottom endface of the piston 63p. A pressure balancing hole 65 extends through the pressure responsive piston 63p to introduce the back pressure from the upstream-side cavity adjacent to the first valve seat 63a. This structure permits the discharge pressure PdH from the

port 41 to act equally on both the bottom end of the pressure responsive piston 63p and the top end of the valve element 63. Those two opposing pressure forces cancel each other and the discharge pressure PdH never interferes with the operation of the solenoid unit 30C controlling the position of the valve element 63.

**[0050]** When the solenoid unit 30C in Fig. 5 is de-energized, the coil spring 55 urges the plunger 54 and shaft 49 upward and puts the valve element 63 into the central opening of the pressure responsive member 64a. The first control valve 30A is fully closed, while the second control valve 30B fully opens in an attempt to obtain a predetermined differential pressure between discharge pressures PdH and PdL acting on the pressure responsive member 64a.

**[0051]** When the solenoid unit 30C in Fig. 5 is energized, the shaft 49 moves downward and allows the valve element 63 to leave the first valve seat 63a and to form a gap of a certain width around the first valve seat 63a. The refrigerant in the port 41 at discharge pressure PdH flows out of the port 42 through the first control valve 30A. Then, in the second control valve 30B, the pressure responsive member 64a receives the differential pressure between discharge pressures PdH and PdL, which moves the second valve element 64 so that the differential pressure will become a predetermined level. With this movement of the second valve element 64, the second control valve 30B controls the refrigerant being delivered from the port 43 to the crank chamber 15.

**[0052]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across valve 30A, causing the second valve element 64 to move in the valve-opening direction, so that the second control valve 30B supplies more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover the original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flow into the crank chamber 15. The compressor 1 operates with a larger displacement to regulate the discharge flow rate Qd.

**[0053]** The fifth embodiment of the capacity control valve 30 of Fig. 6 differs from the fourth embodiment (Fig. 5) in that the port 41 for discharge pressure PdH and the port 42 for discharge pressure PdL have swapped their positions. Instead of the taper-shaped valve element 64 as the second valve element, the fifth embodiment in Fig. 6 employs a ball valve element 67 for that purpose. This ball valve element 67 is located downstream of the second valve seat 56 and is loaded in valve-opening direction by a stem 68 extending through the valve hole and the first control valve 30A.

**[0054]** The second control valve 30B has the second valve seat 56 formed as an integral part of the body 40. A ball valve element 67 is located in a downstream-side space adjacent to the second valve seat 56 and is urged

by the coil spring 60 in valve-closing direction. The spring load can be adjusted by an adjustment screw 59 which has a central through hole 59a constituting the port 43 for delivery of refrigerant to the crank chamber 15 (control pressure  $P_c$ ).

**[0055]** The first control valve 30A has the first valve seat 63a at the bottom opening end of the pressure responsive member 64a which is integral with the shaft or stem 68. The upper end of the shaft or stem 68 contacts the ball valve element 67.

**[0056]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction to put the valve element 63 into the central opening of the pressure responsive member 64a. The first control valve 30A is fully closed, while the second control valve 30B is fully opened because of the differential pressure that acts on the pressure responsive member 64a.

**[0057]** When the solenoid unit 30C is energized, the shaft 49 moves downward and allows the valve element 63 to leave the first valve seat 63a and to form a gap of a certain width around the first valve seat 63a. The refrigerant in the port 41 at discharge pressure  $P_dH$  flows out of the port 42 through the first control valve 30A. In the second control valve 30B, the pressure responsive member 64a receives differential pressure between discharge pressures  $P_dH$  and  $P_dL$ , which moves the ball valve element 67, relative to the second valve seat 56 so that the differential pressure will have a predetermined level. By the movement of the element 67, the second control valve 30B controls the flow rate of the refrigerant as delivered from the port 43 to the crank chamber 15.

**[0058]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across that valve 30A. With the increased differential pressure, the ball valve element 67 will open further so that the second control valve 30B supplies more refrigerant into the crank chamber 15. The compressor 1 operates with smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction, thus reducing the refrigerant flow into the crank chamber 15. The compressor 1 operates with a larger displacement so as to regulate the discharge flow rate  $Q_d$ .

**[0059]** The capacity control valve 30 of the sixth embodiment in Fig. 7 differs from the fifth embodiment (Fig. 6) in that its first control valve 30A employs a taper-shaped valve element 61 as a first valve element located in an upstream-side space adjacent to the first valve seat 63a, being urged in the valve-opening direction.

**[0060]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and shaft 49 upward to seat the taper-shaped valve element 61 on the first valve seat 63a. The first control valve 30A is fully closed, while the second control valve 30B is fully

opened because of the differential pressure that acts on the pressure responsive member 64a.

**[0061]** When the solenoid unit 30C is energized, the shaft 49 moves downward and allows the valve element 61 to leave the first valve seat 63a to form a gap of a certain width at the first valve seat 63a. The refrigerant from the port 41 at discharge pressure  $P_dH$  flows out of the port 42 through the first control valve 30A. The pressure responsive member 64a receives the differential pressure between discharge pressures  $P_dH$  and  $P_dL$ , which moves the ball valve element 67 so that the differential pressure will have a predetermined level. By movements of the element 67, the second control valve 30B controls the flow rate of the refrigerant being delivered from the port 43 to the crank chamber 15.

**[0062]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be generated across the valve 30A and valve element 67 opens further to supply more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in valve-closing direction reducing the refrigerant flow into the crank chamber 15. The compressor 1 operates with a larger displacement so as to regulate the discharge flow rate  $Q_d$ .

**[0063]** The capacity control valve 30 of the seventh embodiment in Fig. 8 differs from the sixth embodiment (Fig. 7) in that its first valve element 61 is designed to cancel the back pressure in order to prevent the discharge pressure  $P_dH$  from affecting operation of the first control valve 30A. This concept is described for the fifth embodiment (Fig. 6) already.

**[0064]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and shaft 49 in the upward direction to seat the taper-shaped valve element 61 on the first valve seat 63a. The first control valve 30A is fully closed, while the second control valve 30B is fully opened.

**[0065]** When the solenoid unit 30C is energized, the shaft 49 moves downward and allows the valve element 61 to leave the first valve seat 63a and form a gap of a certain width at the first valve seat 63a. The refrigerant at port 41 at discharge pressure  $P_dH$  flows out of the port 42 through the first control valve 30A. The pressure responsive member 64a receives differential pressure between discharge pressures  $P_dH$  and  $P_dL$  and moves the valve element 67 so that the differential pressure will have a predetermined level. By the movements of the ball valve element 67, the second control valve 30B controls the flow rate at port 43 to the crank chamber 15.

**[0066]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across the valve 30A such that the ball valve element 67 opens further until the second control valve 30B supplies more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller



displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in a valve-closing direction to reduce the refrigerant flow into the crank chamber 15. The variable displacement compressor 1 operates with a larger displacement so as to regulate the discharge flow rate Qd.

**[0067]** The capacity control valve 30 of the eighth embodiment in Fig. 9 differs from the fourth embodiment (Fig. 5) in that its first valve seat 45a is not designed to move, but is an integral part of the body 40 of the first control valve 30A. Another difference is that a plurality of ball valve elements 46 are employed to serve with several valve seats the function of the first valve element.

**[0068]** The body 40 in Fig. 9 has a plurality of valve holes 45 bored along a pitch circle concentric with a cross section of the body 40 itself. The bottom-end edge of each valve hole 45 serves as the first valve seat 45a. Each ball valve element 46 is placed at a downstream-side space adjacent to each first valve seat 45a. All ball valve elements 46 rest on a downstream-side surface of a support member 70, which is urged by a coil spring 60 in the downward direction. The support member 70 also receives an upward force of the coil spring 55 in the solenoid unit 30C, via the plunger 54 and the shaft 49.

**[0069]** The second control valve 30B has the pressure responsive member 64a urged upward by the coil spring 66. Since the pressure responsive member 64a is integral with the second valve element 64, the urging force of the coil spring 66 also acts on the second valve element 64 in the valve-closing direction. The pressure responsive member 64a, in combination with the second valve element 64, is supposed to be responsive to differential pressure  $\Delta P$  between two different discharge pressures PdH and PdL, on the upstream and downstream ends of the first control valve 30A, respectively.

**[0070]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 in the upward direction and seats the ball valve elements 46 at the first valve seats 45a. The first control valve 30A is fully closed. With the discharge pressure PdH in the port 41, the maximum differential pressure acts on the pressure responsive member 64a, and fully opens the second control valve 30B. The compressor 1 operates in the minimum capacity condition.

**[0071]** When the solenoid unit 30C is energized, the shaft 49 moves downward. In conjunction with the force of the coil spring 60 the support member 70 follows in the same direction while keeping contact with the shaft 49. Each ball valve element 46 leaves the corresponding first valve seat 45a and maintains a gap of a certain width at the valve seat 45a. Refrigerant in the port 41 at discharge pressure PdH flows out of the port 42 through the first control valve 30A. Then, the pressure responsive member 64a adjacent to the second valve element 64 receives differential pressure between discharge pressures PdH and PdL and moves the second valve

element 64 so that the differential pressure will have a predetermined level controlling the flow rate at discharge pressure PdH through the second control valve 30B.

**[0072]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across the valve 30A acting on the pressure responsive member 64a. The second valve element 64 is moved to further open until the second control valve 30B supplies more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B reduces the refrigerant flow into the crank chamber 15 because the pressure responsive member 64a impels the second valve element 64 in the valve-closing direction. The compressor 1 operates with a larger displacement so as to regulate the discharge flow rate Qd.

**[0073]** The capacity control valve 30 of the ninth embodiment in Fig. 10 differs from the eighth embodiment (Fig. 9) in the structure of the first valve element and the first valve seat.

**[0074]** The body 40 has a doughnut-shaped valve hole 45 hollowed along a circle that is concentric with a cross section of the body 40, and the bottom-end edge of the valve hole serves as the first valve seat 45a. The doughnut-shaped valve hole 45 does not go through the body 40 over the entire circumference, but some middle part of a floor of the body 40 remains solid at a few places. This measure connects the central portion of the floor to the circumferential portion of the body 40 in order to house the pressure responsive member 64a. As a counterpart of the first valve seat 45a, a flat annular valve element 71 is disposed on the downstream side, together with a plug 40b that supports the flat valve element 71 axially movable.

**[0075]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 in the upward direction to abut the flat valve element 71 on the first valve seat 45a. The first control valve 30A is fully closed. With the discharge pressure PdH in the port 41, the maximum differential pressure acts on the pressure responsive member 64a fully opening the second control valve 30B. The compressor 1 operates in the minimum capacity condition.

**[0076]** When the solenoid unit 30C is energized, the shaft 49 moves downward. This movement, in conjunction with the force of the coil spring 60, allows the flat valve element 71 to follow in the same direction while keeping contact with the shaft 49. The valve element 71 leaves the first valve seat 45a and forms a gap of a certain width at the valve seat 45a. Refrigerant in the port 41 at discharge pressure PdH flows out of the port 42 through the first control valve 30A. The pressure responsive member 64a adjacent to the second valve element 64 receives differential pressure between discharge pressures PdH and PdL, which moves the second valve

element 64 so that the differential pressure will reach a predetermined level controlling the flow rate at discharge pressure PdH through the second control valve 30B.

**[0077]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across the valve 30A, acting on the pressure responsive member 64a. The second valve element 64 is moved to further open and to supply more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B reduces the refrigerant flow into the crank chamber 15 because its pressure responsive member 64a impels the second valve element 64 in the valve-closing direction. The compressor 1 operates with a larger displacement so as to regulate the discharge flow rate Qd. The capacity control valve 30 of the tenth embodiment in Fig. 11 differs from the first embodiment (Fig. 2). The most prominent difference is that the tenth embodiment uses, in its first control valve 30A, a diaphragm 72 to detect differential pressure between upstream and downstream sides.

**[0078]** In a central region of the body 40, an integral cylinder 40c is formed serving as the valve hole 45 interconnecting the ports 41 and 42. The bottom end of the cylinder 40c functions as the first valve seat 45a for the first control valve 30A. In the downstream-side space communicating with the port 42, a taper-shaped valve element 61 is placed opposite the first valve seat 45a. This valve element 61 is integral with the plunger 54 of the solenoid unit 30C and has a circumferential groove 61b in the round periphery. The groove 61b receives a piston ring 74 slidably supporting the plunger 54 on the inner wall of the sleeve 52, and also centering the valve element 61 on the axis of the sleeve 52.

**[0079]** In the second control valve 30B a valve hole is bored between ports 41, 43. The bottom end of the valve hole is the second valve seat 56. In the upstream-side space adjacent to the second valve seat 56, the taper-shaped second valve element 64 is placed. Integrally formed on top of the second valve element 64 are a shaft 64c and a piston 64d. This piston 64d has the same outer diameter as the valve hole of the second valve seat 56. The endface of the piston 64d remote from the second valve element 64 receives, through a communication hole 62, discharge pressure PdH in the port 41, so that the second valve element 64 is actuated solely by the differential pressure between discharge pressures PdH and PdL, without being affected by the absolute value of discharge pressure PdH. The second valve element 64 is integral with a base member 64e, which is larger in diameter than the second valve element 64. The base member 64e has a hole 64b to introduce discharge pressure PdH from the port 41 into the cylinder 40c.

**[0080]** A sliding member 73 surrounds the outer sur-

face of the cylinder 40c in body 40 in a way that it can move in the vertical direction. This sliding member 73 is connected with the inner surface of the cavities of the bodies 40 and 40a via a diaphragm 72, which is a doughnut-shaped sheet with a center hole. The outer circumference of the diaphragm 72 is clamped between the bodies 40 and 40a. The body 40a is pressed into the body 40. The inner circumference of the diaphragm 72 is clamped between the sliding member 73 and a ring 73a. The base member 64e of the second valve element 64 may rest on the sliding member 73. Two coil springs 60 and 66 urge the members 64e and 73 such that they will keep contact. The diaphragm 72 receives differential pressure between the discharge pressure PdH at the port 41 and the discharge pressure PdL at the port 42. This differential pressure displaces the sliding member 73 in axial direction, causing the second valve element 64 to move toward or away from the second valve seat 56.

**[0081]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 upward to seat the valve element 61 sit on the first valve seat 45a. The first control valve 30A is fully closed. With the discharge pressure PdH present in the port 41, the maximum differential pressure acts on the diaphragm 72 to fully open the second control valve 30B. The compressor 1 operates in the minimum capacity condition.

**[0082]** When the solenoid unit 30C is energized, the plunger 54 moves downward allowing the valve element 61 to leave the first valve seat 45a and to form a gap of a certain width at the first valve seat 45a. Refrigerant in the port 41 at the discharge pressure PdH flows out of the port 42 through the hole 64b of the second valve element 64, the central cavity of the cylinder 40c, and the first control valve 30A. The diaphragm 72 receives the differential pressure between the two different discharge pressures PdH and PdL, and moves the sliding member 73 upward so that the differential pressure will reach a predetermined level. The second valve element 64 follows this movement of the sliding member 73, thus controlling the refrigerant at the discharge pressure PdH that flows through the second control valve 30B.

**[0083]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across the valve 30A and acts on the diaphragm 72. The second valve element 64 in the second control valve 30B is impelled in the direction to further open and to supply more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the diaphragm 72 receives a reduced differential pressure and reduces the refrigerant flowing into the crank chamber 15 because its sliding member 73 impels the second valve element 64 in the valve-closing direction. The compressor 1 operates with a larger displacement so as to regulate the discharge flow rate Qd. The second control valve 30B is controlled to re-

spond only to the differential pressure between the two different discharge pressures  $P_{dH}$  and  $P_{dL}$ , because the second valve element 64 is unaffected by variations of the discharge pressure  $P_{dH}$ .

**[0084]** The capacity control valve 30 of the eleventh embodiment of Fig. 12 differs from the tenth embodiment (Fig. 11) in that the taper-shaped valve element (first valve element) 61 in the first control valve 30A is disposed in an upstream-side space of the first valve seat 45b formed at the top end of the cylinder 40c. For this reason, in the solenoid unit 30C, the plunger 54 and core 53 have swapped their positions on the axis (relative to Fig. 11). The shaft 49 connects the first valve element 61 with the plunger 54 in the solenoid unit 30C. The first valve element 61 is urged by the coil spring 55 in the valve-closing direction.

**[0085]** The eleventh embodiment operates basically in the same way as the tenth embodiment. However, the widened base member 64e of the second valve element 64 has at least one round hole 64f in a cylindrical skirt portion in addition to the hole 64b to deliver the discharge pressure  $P_{dH}$  from the port 41 to the upstream side of the first valve element 61.

**[0086]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 to seat the first valve element 61 on the first valve seat 45b. The first control valve 30A is fully closed. With the discharge pressure  $P_{dH}$  in the port 41, the maximum differential pressure acts on the diaphragm 72 which fully opens the second control valve 30B. The variable displacement compressor 1 operates in the minimum capacity condition.

**[0087]** When the solenoid unit 30C is energized, the plunger 54 moves upward and allows the taper-shaped valve element 61 to leave the first valve seat 45b and form a gap of a certain width at the first valve seat 45b. Refrigerant in the port 41 at discharge pressure  $P_{dH}$  flows out of the port 42 through the round hole 64f and the hole 64b of the second valve element 64, the first control valve 30A, and the central cavity of the cylinder 40c. The diaphragm 72 receives differential pressure between the two different discharge pressures  $P_{dH}$  and  $P_{dL}$  and moves the sliding member 73 upward so that the differential pressure will reach a predetermined level. The second valve element 64 follows this upward movement of the sliding member 73, thus controlling the flow rate of the refrigerant at discharge pressure  $P_{dH}$  that flows through the second control valve 30B.

**[0088]** If the refrigerant flow through the first control valve 30A increases, a larger differential pressure will be produced across the valve 30A acting on the diaphragm 72. The second valve element 64 is moved to further open and to supply more refrigerant into the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the diaphragm 72 detects and responds to a reduced differential pressure. The

sliding member 73 impels the second valve element 64 in the valve-closing direction. The second control valve 30B reduces the refrigerant supplied to the crank chamber 15, and the compressor 1 operates with a larger displacement so as to regulate the discharge flow rate  $Q_d$ .

**[0089]** In the embodiment of Fig. 4 the first control valve 30A is located between the discharge chambers 33 and crank chamber 15, and the pressure in the crank chamber 15 is controlled by varying the flow rate at discharge pressure  $P_{dL}$  from the discharge chambers 33 into the crank chamber 15. Unlike the third embodiment of Fig. 4, the twelfth embodiment of Fig. 13 controls the flow rate returning from the crank chamber 15 back into the suction chambers 32. In this alternative arrangement, the variable displacement compressor 1 has an orifice of fixed cross-section in the middle of a passageway that delivers refrigerant from the discharge chambers 33 to the crank chamber 15.

**[0090]** The first control valve 30A and solenoid unit 30C of the capacity control valve 30 of Fig. 13 have almost the same structure shown in the third embodiment of Fig. 4. An exception is that the first control valve 30A is designed to route the discharged refrigerant in the direction that the stream pushes the taper-shaped valve element 61 in opening direction away from the first valve seat 45a.

**[0091]** In the second control valve 30B, two pistons 58 and 58a are integrally formed with the second valve element 57. The pistons 58 and 58a have the same outer diameter as the valve hole of the second valve seat 56. Discharge pressure  $P_{dH}$  acts on the piston 58a and discharge pressure  $P_{dL}$  propagates through a communication hole 62 and acts on one endface of the piston 58. Pressure  $P_c$  of the crank chamber 15 is led from the port 43 to an upstream-side cavity adjacent to the second valve element 57. The downstream-side room communicates with the suction chambers 32 at suction pressure  $P_s$  via the port 75. With such an arrangement of the second control valve 30B, the second valve element 57 and piston 58 are responsive to the differential pressure  $\Delta P$  developed across the first control valve 30A, which is functioning here as an orifice. The second control valve 30B thus controls the flow rate from the crank chamber 15 to the suction chambers 32 such that the differential pressure  $\Delta P$  will be maintained at a constant level. This control action varies the capacity of the variable displacement compressor 1 so as to regulate the discharge flow rate.

**[0092]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 upward to seat the first valve element 61 on the first valve seat 45a. The first control valve 30A is fully closed.

**[0093]** When the solenoid unit 30C is energized, the plunger 54 moves downward and allows the first valve element 61 to leave the first valve seat 45a and to form a gap of a certain width at the first valve seat 45a. Refrigerant in the port 41 at discharge pressure  $P_{dH}$  flows out of the port 42 through the first control valve 30A. In

the second control valve 30B, the second valve element 57 and the piston 58 (a single integrated member) receive differential pressure between the two different discharge pressures  $P_{dH}$  and  $P_{dL}$ , in addition to the force of the coil spring 60. The second valve element 57 moves to a point at which all those forces and pressures are in balance. The refrigerant in the crank chamber 15 at pressure  $P_c$  is allowed to flow back to the suction chambers 32. The second control valve 30B now may control the discharge capacity of the variable displacement compressor 1 by varying the crank chamber pressure  $P_c$ .

**[0094]** The refrigerant flow through the first control valve 30A may increase due to, for example, sudden acceleration of the driving engine. Then, a larger differential pressure will occur across the valve 30A and will actuate the second control valve 30B to further close and to reduce the flow rate out of the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-opening direction, thus increasing the flow rate out of the crank chamber 15. The compressor 1 operates with a larger displacement, thus regulating the discharge flow rate  $Q_d$ .

**[0095]** The third embodiment (Fig. 4) has a capacity control valve 30 controlling the flow rate into the crank chamber 15 (inflow control). The twelfth embodiment (Fig. 13) manipulates the flow rate out of the crank chamber 15 (outflow control). The thirteenth embodiment of Fig. 14 relates to a capacity control valve 30 that employs both in-flow and out-flow control mechanisms. The capacity control valve 30 of Fig. 14 has a first control valve 30A placed in a passageway leading from the discharge chambers 33 to the solenoid unit 30C that governs the cross-sectional area of the passageway. In addition, the capacity control valve 30 of Fig. 14 has the second 30B and a third control valves 30D that both detect the differential pressure developed across the first control valve 30A and which control the pressure in the crank chamber 15 such that the differential pressure will reach a specified level.

**[0096]** The second and third control valves 30B and 30D accommodate the following components in a common valve hole: a piston 58, a second valve element 57, and a third valve element 76, all integrally formed as a single member. One control edge formed in the valve hole serves as a third valve seat 77. The piston 58 has the same outer diameter as the valve seat 77. The second valve element 57 receives the discharge pressure  $P_{dH}$  on its bottom endface, while the piston 58 receives the discharge pressure  $P_{dL}$  through a communication hole 62 of the body 40. The upstream-side room adjacent to the second valve element 57 is at discharge pressure  $P_{dH}$  introduced from the port 41. The downstream side communicates with the crank chamber 15 through another port 43a, the pressure at which is  $P_{c1}$ .

The upstream-side space adjacent to the third valve element 76 receives pressure  $P_{c2}$  from the crank chamber 15 via yet another port 43b. The downstream-side space adjacent to the third valve element 76 communicates with the suction chambers 32 at suction pressure  $P_s$  via still another port 75.

**[0097]** The piston 58 and the second valve element 57 move together in response to the differential pressure  $\Delta P$  across the first control valve 30A, which is functioning here as an orifice. The second and third control valves 30B and 30D act as a three-way valve that controls the inflow of refrigerant from the discharge chambers 33 into the crank chamber 15, simultaneously with the outflow from the crank chamber 15 to the suction chambers 32, so that the differential pressure  $\Delta P$  will be maintained at a constant level.

**[0098]** The capacity control valve 30 with the above construction operates as follows. When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 upward to seat the first valve element 61 on the first valve seat 45a. The first control valve 30A is fully closed.

**[0099]** When the solenoid unit 30C is energized, the plunger 54 moves downward causing the first valve element 61 to leave the first valve seat 45a and to form a gap of a certain width. Refrigerant in the port 41 at discharge pressure  $P_{dH}$  flows out of the port 42 through the first control valve 30A. In the second control valve 30B, the unified valve member (i.e., second valve element 57, third valve element 76, and the piston 58) receives the differential pressure between the two different discharge pressures  $P_{dH}$  and  $P_{dL}$  while being pushed downward by the coil spring 60, and moves to a point at which all those forces and pressures are in balance. The second control valve 30B supplies the crank chamber 15 with refrigerant at discharge pressure  $P_{dH}$ , and the third control valve 30D allows the refrigerant at pressure  $P_c$  to flow back into the suction chambers 32. The capacity control valve 30 varies the crank chamber pressure  $P_c$  in this way, thus being able to control the discharge capacity of the variable displacement compressor 1.

**[0100]** The refrigerant flowing through the first control valve 30A may increase due to, for example, sudden acceleration of the driving engine. Then, a larger differential pressure will occur across the valve 30A. The increased differential pressure opens the second control valve 30B wider, while it actuates the third control valve 30D in valve-closing direction. This results in an increased inflow to the crank chamber 15, along with a decreased outflow from the crank chamber 15. The variable displacement compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in valve-closing direction, thus producing a decreased inflow to the crank chamber 15. At the same time, the third control valve 30D is impelled in the valve-

opening direction, increasing the outflow from the crank chamber 15. The compressor 1 now operates with a larger displacement, resulting in a regulated discharge flow rate  $Q_d$ .

**[0101]** Compared to the twelfth embodiment (Fig. 13), the capacity control valve 30 of Fig. 15 is designed to control how much of the discharged refrigerant is supplied to the crank chamber 15. Another difference is that the second valve element 57 of the second control valve 30B is provided as a discrete component, and is not integrated with a pressure sensing member that responds to differential pressure across the first control valve 30A.

**[0102]** A piston 58 is located inside the body 40. The communication hole 62 in the body 40 applies the discharge pressure  $P_dL$  to the piston 58. A refrigerant passageway laterally branches halfway from the communication hole 62, to the port 43 for the crank chamber 15. In the middle of this passageway, the second valve seat 56 is formed as an integral part of the body 40. Located downstream of the second valve seat 56 the second valve element 57 is integrally formed with the piston 58. The piston 58 receives the discharge pressure  $P_dL$  on its distal end. A piston 78, a coil spring 79, and a spring seat 80 are installed coaxially with the second valve element 57 and the piston 58 in a space formed between the port 41 and the communication hole 62. The discharge pressure  $P_dH$  is available in this space. A shaft integral with the second valve element 57 extends toward the piston 78 in a space that communicates with the communication hole 62. The piston 78 is forced against the shaft by the coil spring 79. Discharge pressure  $P_dL$  does not affect the movement of the second valve element 57 and the piston 58 because their pressure-receiving areas are substantially equal. In the second control valve 30B, the piston 78 responds to the differential pressure  $\Delta P$  developed across the first control valve 30A, which functions as an orifice. The second control valve 30B controls the flow rate from the discharge chambers 33 to the crank chamber 15 such that the differential pressure  $\Delta P$  will be maintained at a constant level. This mechanism varies the capacity of the variable displacement compressor 1 so as to regulate the discharge flow rate.

**[0103]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 to seat the first valve element 61 on the first valve seat 45a. The first control valve 30A is fully closed.

**[0104]** When the solenoid unit 30C is energized, the plunger 54 moves downward and allows the first valve element 61 to leave the first valve seat 45a and to form a gap of a certain width at the first valve seat 45a. Refrigerant in the port 41 at discharge pressure  $P_dH$  flows out of the port 42 through the first control valve 30A. The piston 78 receives the differential pressure between the two different discharge pressures  $P_dH$  and  $P_dL$  while being pushed by two coil springs 60 and 79, and thus moves to a point at which all those forces and pressures are in balance. Refrigerant at discharge pressure  $P_dH$

flows into the crank chamber 15. The second control valve 30B now controls the discharge capacity of the variable displacement compressor 1 by varying the crank chamber pressure  $P_c$ .

**[0105]** The refrigerant flow through the first control valve 30A may increase due to, for example, sudden acceleration of the engine. Then, a larger differential pressure will occur across the valve 30A and will further open the second control valve 30B. This produces an increased inflow to the crank chamber 15, and as a result, the variable displacement compressor 1 operates with a smaller displacement, thus recovering its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in valve-closing direction to decrease the inflow to the crank chamber 15. The variable displacement compressor 1 operates with a larger displacement, resulting in a regulated discharge flow rate  $Q_d$ .

**[0106]** The capacity control valve 30 of the fifteenth embodiment in Fig. 16 is similar to the twelfth embodiment (Fig. 13) and controls the outflow from the crank chamber 15 to the suction chambers 32, however, differs in that the second valve element 57 in its second control valve 30B is provided as a discrete component, not integrated with a pressure sensing member that responds to differential pressure across the first control valve 30A.

**[0107]** To detect differential pressure across the first control valve 30A in Fig. 16, the second control valve 30B employs the piston 78, the coil spring 79, and the spring seat 80. The ports 43 and 75 are disposed to communicate with the crank chamber 15 and suction chambers 32, respectively. Between the ports 43, 75, a second valve seat 56 is formed as an integral part of the body 40. The second valve element 57 is installed in an upstream-side space near the port 43. Integrally formed with this second valve element 57 is the piston 58 with the same diameter as the valve hole of the second valve seat 56. Discharge pressure  $P_dL$  propagates through the communication hole 62 of the body and acts on one endface of the piston 58. The second valve element 57 is integral with another piston 58a having nearly the same diameter as the valve hole of the second valve seat 56. This piston 58a is held in the body 40 in an airtight manner, movably in its axial direction, receiving discharge pressure  $P_dL$ . The lower end of the piston 58a abuts on the piston 78. Discharge pressure  $P_dL$  does not affect movement of the pistons 58a and 58 because their diameters are substantially the same. The piston 78 responds to the differential pressure  $\Delta P$  across the first control valve 30A, functioning as an orifice. The second control valve 30B controls the outflow from the crank chamber 15 to the suction chambers 32 such that the differential pressure  $\Delta P$  will be maintained at a constant level. This control action varies the capacity of the variable displacement compressor 1 to regulate the discharge flow rate.

**[0108]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 to seat the first valve element 61 on the first valve seat 45a. The first control valve 30A is fully closed.

**[0109]** When the solenoid unit 30C is energized, the plunger 54 moves downward to allow the first valve element 61 to leave the first valve seat 45a and to form a gap of a certain width at the first valve seat 45a. Refrigerant in the port 41 at discharge pressure PdH flows out of the port 42 through the first control valve 30A. The piston 78 receives the differential pressure between discharge pressures PdH and PdL, while being pushed by the coil springs 60 and 79, and moves to a point at which all those forces and pressures are in balance. Refrigerant in the crank chamber 15 at pressure Pc is allowed to flow back into the suction chambers 32. The second control valve 30B now controls the discharge capacity of the variable displacement compressor 1 by varying the crank chamber pressure Pc.

**[0110]** The refrigerant flow through the first control valve 30A may increase due to, for example, sudden acceleration of the driving engine. Then, a larger differential pressure will occur across the valve 30A and will actuate the second control valve 30B to further close and to reduce the flow rate out of the crank chamber 15 and to raise the crank chamber pressure Pc. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-opening direction, increasing the outflow from the crank chamber 15. Since the crank chamber pressure Pc drops, the variable displacement compressor 1 now operates with a larger displacement, resulting in a regulated discharge flow rate Qd.

**[0111]** The capacity control valve 30 of the sixteenth embodiment in Fig. 17 is similar to the thirteenth embodiment (Fig. 14), however, differs in that the second valve element 57 in the second control valve 30B is a discrete component, not integrated with a member that senses differential pressure across the first control valve 30A. As the pressure responsive member, the sixteenth embodiment in Fig. 17 has a similar structure to the fifteenth embodiment (Fig. 16).

**[0112]** Inside the second and third control valves 30B and 30D in Fig. 17, the piston 58, the second valve element 57, and the third valve element 76 are disposed in an integrated manner. The piston 58 has the same outer diameter as the valve holes of second and third valve seats 56 and 77 so as to avoid the effect of discharge pressure PdL acting thereon. The piston 58 and second valve element 57 move together in response to the differential pressure  $\Delta P$  across the first control valve 30A, functioning as an orifice. The second and third control valves 30B and 30D serve as a three-way valve controlling the inflow from the discharge chambers 33 into the crank chamber 15, and simultaneously the outflow from the crank chamber 15 to the suction chambers 32,

such that the differential pressure  $\Delta P$  will be maintained at a constant level.

**[0113]** When the solenoid unit 30C is de-energized, the coil spring 55 urges the plunger 54 and the shaft 49 upward to seat the first valve element 61 on the first valve seat 45a. The first control valve 30A is fully closed.

**[0114]** When the solenoid unit 30C is energized, the plunger 54 moves downward to allow the first valve element 61 to leave the first valve seat 45a and to form a gap of a certain width. Refrigerant in the port 41 at discharge pressure PdH flows out of the port 42 through the first control valve 30A. In the second control valve 30B, the unified valve member (i.e., second valve element 57, third valve element 76, and piston 58) receives the differential pressure between discharge pressures PdH and PdL while being pushed by the coil springs 60 and 79, and thus moves to a point at which all those forces and pressures are balanced. The second control valve 30B supplies refrigerant at pressure Pc1 to the crank chamber 15 by controlling the refrigerant at discharge pressure PdL, and at the same time, the third control valve 30D allows the refrigerant at pressure Pc2 in the crank chamber 15 to flow back into the suction chambers 32. The capacity control valve 30 varies the crank chamber pressure Pc in this way to control the discharge capacity of the variable displacement compressor 1.

**[0115]** The refrigerant flow through the first control valve 30A may increase due to, for example, sudden acceleration of the driving engine. Then, a larger differential pressure will occur across that valve 30A which opens the second control valve 30B wider, while actuating the third control valve 30D in valve-closing direction. This control action produces an increased inflow to the crank chamber 15, together with a decreased outflow from the crank chamber 15. The compressor 1 operates with a smaller displacement so as to recover its original discharge flow rate. If the refrigerant flow through the first control valve 30A decreases, the second control valve 30B is actuated in the valve-closing direction, and the third control valve 30D in the valve-opening direction, thus producing a decreased inflow to the crank chamber 15 and an increased outflow from the crank chamber 15. The compressor 1 operates with a larger displacement, resulting in a regulated discharge flow rate Qd.

**[0116]** As in the fourteenth embodiment (Fig. 15), the capacity control valve 30 of the seventeenth embodiment in Fig. 18 is designed to control how much of the discharged refrigerant to supply to the crank chamber 15. Another similarity is that the second valve element 57 of the second control valve 30B is provided as a discrete component, not integrated with a member that responds to differential pressure across the first control valve 30A. The seventeenth embodiment is, however, different in that it has no back-pressure cancellation mechanism for the second valve element 57.

**[0117]** The second control valve 30B in Fig. 18 is con-

structed as follows. The second valve element 57 is urged by the coil spring 60 in valve-closing direction. The discharge pressure PdL is routed through the communication hole 62 in the body 40 and acts only on the piston 78 and the second valve element 57. The upper end of the coil spring 60 is supported by a lid 59c having a vent. An O-ring 29b is used for sealing the capacity control valve 30 when installed in the variable displacement compressor 1. The upper space above the level of this O-ring 29b will be at pressure Pc, i.e., the pressure in the port 43, meaning that the same pressure Pc will be available in the cavity of the coil spring 60.

**[0118]** The capacity control valve 30 of Fig. 18 bears close resemblance to the fourteenth embodiment (Fig. 15) in terms of the structure, except for the fact that the second valve element 57 is not free from back pressures. When the solenoid unit 30C is de-energized, the capacity control valve 30 operates as the fourteenth embodiment. This similarity of the control operations also applies when the solenoid unit 30C is energized, or when the engine rotation speed varies.

**[0119]** As in the fifteenth embodiment (Fig. 16), the capacity control valve 30 of the eighteenth embodiment in Fig. 19 is designed to control the outflow of the crank chamber 15 to the suction chambers 32. The second valve element 57 of the second control valve 30B is a discrete component, not integrated with a member that senses differential pressure across the first control valve 30A. The eighteenth embodiment is, however, different in that it has no back-pressure cancellation mechanism for the second valve element 57.

**[0120]** The second control valve 30B in Fig. 19 is constructed as follows. The second valve element 57 is urged against the piston 78 by the coil spring 60 in the valve-opening direction. The discharge pressure PdL is routed through the communication hole 62 such that it acts only on the piston 78 and on another piston that extends from the second valve element 57. The piston 58 is integrally formed with the second valve element 57, and the coil spring 60 is accommodated in a space between this piston 58 and the lid 59c having a vent. The coil spring space is pressurized at Ps through the vent in the lid 59c.

**[0121]** The capacity control valve 30 of Fig. 19 bears close resemblance to the fifteenth embodiment (Fig. 16) in terms of the structure, except for the fact that the second valve element 57 is not free from back pressures. When the solenoid unit 30C is de-energized, the capacity control valve 30 operates in the same way as described in the fifteenth embodiment. This similarity in its control operations also applies when the solenoid unit 30C is energized and when the engine rotation speed varies.

**[0122]** The capacity control valve 30 of the nineteenth embodiment in Fig. 20 controls both inflow and outflow of refrigerant to/from the crank chamber 15, as the seventeenth embodiment (Fig. 18).

**[0123]** The second and third control valves 30B and

30D are constructed as follows. The second valve element 57 and the third valve element 76, which constitute a three-way valve, are urged by the coil spring 60 in valve-closing direction and in the valve-opening direction, respectively, where discharge pressure PdL is routed through the communication hole 62 such that it acts only on the second valve element 57 and the piston 78. The piston 58 is integrally formed with the second and third valve elements 57 and 76. The coil spring 60 is accommodated in a space between the piston 58 and the lid 59c having a vent. The coil spring space is pressurized at Ps through the vent in the lid 59c.

**[0124]** The capacity control valve 30 of Fig. 20 bears close resemblance to the sixteenth embodiment (Fig. 17) in terms of the structure, except for the fact that the second valve element 57 and third valve element 76 are not free from back pressures. When the solenoid unit 30C is de-energized, the capacity control valve 30 of Fig. 20 operates as the sixteenth embodiment. This similarity also applies when the solenoid unit 30C is energized and as when the engine rotation speed varies.

**[0125]** All embodiments of the capacity control valves 30 share a common concept. The first control valve 30A controls the cross-sectional area of a passageway for discharged refrigerant. The second control valve 30B (and the third control valve 30D if provided in several cases) controls the pressure Pc in the crank chamber 15 such that the differential pressure across the controlled passageway will be maintained at a specified level. The capacity control valves as described should not be limited to a structure that uses the differential pressure on the discharge side of the valves. Rather, there could be provided a structure using the differential pressure on the suction side. That is, the first control valve 30A may control the cross-section area of a passageway into the compressor, and the second control valve 30B (and the third control valve 30D if provided) controls the pressure Pc in the crank chamber 15 such that the differential pressure across the controlled passageway will be maintained at a specified level.

## Claims

1. A capacity control valve (30) for a variable displacement compressor (1) that regulates a flow rate of refrigerant discharged by the compressor, **characterised by**
  - a first control valve (30A) that sets a specific cross-sectional area of a passageway that leads to a suction chamber (32) or a discharge chamber (33) of the compressor (1);
  - at least a second control valve (30B) that senses a differential pressure developed across the first control valve (30A) and controls a flow rate supplied to or coming out of a crank chamber (15) of the compressor such that the differential pressure will be maintained at a specified level;

a solenoid unit (30C) for actuation of the first control valve (30A) to set the cross-sectional area of the refrigerant passageway according to variations in a given external condition, and by the first control valve (30A) the, second control valve (30B), and the solenoid unit (30C) being formed integrally with each other.

2. A capacity control valve according to claim 1, **characterised by**

a third control valve (30D) that together with the second control valve (30B) senses the differential pressure developed across the first control valve (30A) and respectively control flow rates supplied to and coming out of the crank chamber (15) such that the differential pressure will be maintained at a specified level; and

by the first, second, and third control valves (30A, 30B, 30D) and the solenoid unit (30C) being formed integrally with each other.

3. The capacity control valve according to claim 1, **characterised in that**

the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway leading from the discharge chamber (33), and

a first valve element (46) located opposite the first valve seat (45a) to set the cross-sectional area of the refrigerant passageway, the first valve element (46) being actuated by an upstream force produced and controlled by the solenoid unit (30C) while being urged by a downstream force in a valve-closing direction; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

that a second valve element (57) is located opposite the second valve seat (56), allowed to move upstream toward and downstream away from the second valve seat while being urged by upstream-end pressure of the first control valve (30A), and

that a piston (58) is provided for receiving downstream-end pressure of the first control valve (30A) and for impelling the second valve element (57) in a valve-closing direction with the received downstream-end pressure.

4. The capacity control valve according to claim 1, **characterised in that**

the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway leading from the discharge chamber (33), and

a first valve element (46) located opposite the first valve seat (45a) to set the cross-sectional area of the refrigerant passageway, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being urged in a valve-closing direction when the solenoid unit (30C) is de-energized; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (57) located opposite the second valve seat (56), allowed to move upstream toward and downstream away from the second valve seat while being urged by upstream-end pressure of the first control valve (30A), and that

a piston (58) is provided for receiving downstream-end pressure of the first control valve (30A) and impelling the second valve element (57) in a valve-closing direction with the received downstream-end pressure.

5. The capacity control valve according to claim 1, **characterised in that**

the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway leading from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat (45a) to set the cross-sectional area of the refrigerant passageway, allowed to move downstream toward and upstream away from the first valve seat while being forced in a valve-opening direction, actuated by a force that is produced and controlled by the solenoid unit (30C);

that the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the down-



stream end of the first control valve (30A) to the crank chamber (15),

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat while receiving downstream-end pressure of the first control valve (30A), and

a piston (58) is provided for receiving, on one endface thereof, the upstream-end pressure of the first control valve (30A) and thereby impelling the second valve element (57) in a valve-closing direction; and

that the capacity control valve (30) further comprises a communication hole (62) between the first and second control valves (30A, 30B) to connect a space adjacent to the pressure receiving endface of the piston (58) with an upstream-end space of the first control valve (30A).

6. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a spool valve disposed in the refrigerant passageway coming from the discharge chamber (33), comprising a spool-shaped first valve element (63), and

a pressure responsive piston (63p) that is integrally formed with, and has the same diameter as, the first valve element (63) of the spool valve, having a pressure balancing hole (65) therethrough to cause an endface thereof remote from the first valve element (63) to receive valve hole pressure of the spool valve; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (64) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat, and that

a pressure responsive member (64a) is integrally formed with the second valve element (64), one end thereof serving as a first valve seat (63a) for receiving the first valve element (63) of the spool valve, and for im-

PELLING the second valve element (64) in response to a differential

pressure developed across the spool valve.

7. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a spool valve disposed in the refrigerant passageway coming from the discharge chamber (33), comprising a spool-shaped first valve element (63), and

a pressure responsive piston (63p) that is integrally formed with, and has the same diameter as, the first valve element (63) of the spool valve, having a pressure balancing hole (65) therethrough to cause valve hole pressure of the spool valve to act on an endface of the pressure responsive piston (63) remote from the first valve element (63); that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (67) located opposite the second valve seat (56), allowed to move upstream toward and downstream away from the second valve seat while being forced in a valve-closing direction, and that

a pressure responsive member (64a) is provided for impelling the second valve element (67) through a valve hole thereof in response to a differential pressure developed across the spool valve, one end of the pressure responsive member (64a) serving as a first valve seat (63) for receiving the first valve element (63) of the spool valve.

8. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a first valve element (61) with a taper-shaped end, disposed in the refrigerant passageway coming from the discharge chamber (33), being urged by a downstream force in a valve-closing direction that is produced by the solenoid unit (30C) in de-energized state; that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15), 5

a second valve element (67) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat while being forced in a valve-closing direction, and that 10

a pressure responsive member (64a) is provided for impelling the second valve element (67) through a valve hole thereof in response to a differential pressure developed across the first control valve (30A), one end of the pressure responsive member (64a) serving as a first valve seat (63a) for receiving the first valve element (61) of the first control valve (30A). 15 20

9. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises: 25

a taper valve with a first valve element (61) disposed in the refrigerant passageway coming from the discharge chamber (33), being urged by a downstream force that is produced by the solenoid unit (30C) in de-energized state and acts on the first valve element (61) in a valve-closing direction, 30

a pressure responsive piston (61p) integrally formed with the first valve element (61), with the same diameter as a valve hole of the taper valve, having a pressure balancing hole (65) therethrough to cause valve hole pressure of the taper valve to act on an endface of the pressure responsive piston (61p) remote from the first valve element; that 35 40

the second control valve (30B) comprises: 45

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15), 50

a second valve element (67) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat while being forced in a valve-closing direction, and that 55

a pressure responsive member (4a) is pro-

vided for impelling the second valve element (67) through a valve hole thereof in response to a differential pressure developed across the taper valve, one end of the pressure responsive member (64a) serving as a first valve seat (63a) for receiving the first valve element (61) of the taper valve.

10. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a plurality of first valve seats (45a) formed as downstream-side edges of a plurality of valve holes (45), the valve holes (45) being arranged along a pitch circle so as to constitute a part of the refrigerant passageway coming from the discharge chamber (33), and

a plurality of ball-shaped first valve elements (46) disposed in a downstream-side space adjacent to the respective first valve seats (45a), being urged by an upstream force in a valve-closing direction that is produced by the solenoid unit (30C) in de-energized state; that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (64) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat, and that

a pressure responsive member (64a) integrally formed with the second valve element is provided for impelling the second valve element (64) in response to a differential pressure developed across the first control valve (30A).

11. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a first valve seat (54a) formed as a downstream-side edge of a doughnut-shaped valve hole (45), the valve hole (45) being hollowed so as to constitute a part of the refrigerant passageway coming from the discharge chamber (33),

a first valve element (71) located opposite the

first valve seat (45a), being urged by an upstream force in a valve-closing direction that is produced by the solenoid unit (30C) in de-energized state; that

5

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

10

a second valve element (64) located opposite the second valve seat (56), allowed to move downstream toward and upstream away from the second valve seat, and that

15

a pressure responsive member (64a) integrally formed with the second valve element is provided for impelling the second valve element (64) in response to a differential pressure developed across the first control valve (30A).

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12. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

25

a cylinder (40c) constituting a part of the refrigerant passageway coming from the discharge chamber (33), the downstream end the cylinder (40c) serving as a first valve seat (45), and

30

a first valve element (61) located opposite the first valve seat (45a), integrally formed with a plunger (54) of the solenoid unit (30C), being urged by a force in a valve-closing direction that is produced by the solenoid unit (30C) in de-energized state; that

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the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

45

a second valve element (64) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat (56),

50

a pressure responsive piston (64d) integrally formed with the second valve element (64), with the same diameter as a valve hole of the second valve seat (52),

55

a communication hole (62) that propagates

upstream-end pressure of the first control valve (30A) to an endface of the pressure responsive piston (64d) remote from the second valve element (64),

a sliding member (73) slidably fitted on an outer surface of the cylinder, (40c) and

a diaphragm (72) disposed between the sliding member (73) and a body (40, 40a), impelling the second valve element (64) in response to a differential pressure developed across the first control valve (30A).

13. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a cylinder (40c) constituting a part of the refrigerant passageway coming from the discharge chamber (33), the upstream end the cylinder (40c) serving as a first valve seat (45b), and

a first valve element (61) located opposite the first valve seat, being urged by a force in a valve-closing direction that is produced by the solenoid unit (30C) in de-energized state; that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (64) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat (56),

a sliding member (7) slidably fitted on an outer surface of the cylinder (40c), and

a diaphragm (72) disposed between the sliding member and a body (40, 40a), impelling the second valve element (64) in response to a differential pressure developed across the first control valve (30A).

14. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway leading from the discharge chamber (33), and

a first valve element (61) located opposite the

first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the crank chamber (15) to the suction chamber (32),

a second valve element (57) located opposite the second valve seat (56), allowed to move downstream toward and upstream away from the second valve seat,

a first piston (58a) integrally formed with the second valve element (57), receiving upstream-end pressure of the first control valve (30A) and impelling the second valve element (57) in valve-closing direction with the received upstream-end pressure, and

a second piston (58) integrally formed with the second valve element (57), receiving downstream-end pressure of the first control valve (30A) and impelling the second valve element (57) in valve-opening direction with the received downstream-end pressure.

**15. The capacity control valve according to claim 1, characterised in that**

the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat (45a), actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat (56),

a piston (58) integrally formed with the second valve element (57), having substantially the same diameter as a valve hole of the second valve seat (56), receiving downstream-end pressure of the first control valve (30A), and

a pressure responsive piston (76) installed coaxially with the second valve element, causing the second valve element (57) to move in a valve-opening direction in response to upstream-end pressure of the first control valve (30A), also causing the second valve element (57) to move in a valve-closing direction in response to the downstream-end pressure of the first control valve (30A).

**16. The capacity control valve according to claim 1, characterised in that**

the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat (45a), actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the crank chamber (15) to the suction chamber (32),

a second valve element (57) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat, and

first and second pistons (58, 58a) formed integrally and coaxially with the second valve element (57) at both ends thereof, the distal endfaces of the first and second pistons having substantially equal areas to receive downstream-end pressure of the first control valve (30A), and

a pressure responsive piston (78) installed coaxially with the second valve element (57), causing the second valve element to move in a valve-closing direction in response to upstream-end pressure of the first control valve (30A), also causing the second valve element (57) to move in a valve-opening direction in response to downstream-end pressure of the first control valve (30A).

17. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15),

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat, and

a pressure responsive piston (78) installed coaxially with the second valve element (57), causing the second valve element to move in a valve-opening direction in response to upstream-end pressure of the first control valve (30A), also causing the second valve element (57) to move in a valve-closing direction in response to downstream-end pressure of the first control valve (30A).

18. The capacity control valve according to claim 1, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45A) formed as part of the refrigerant passageway coming from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; and that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the crank chamber (15) to the suction chamber (32),

a second valve element (57) located opposite the second valve seat, allowed to move downstream toward and upstream away from the second valve seat, and

a pressure responsive piston (78) installed coaxially with the second valve element (57), causing the second valve element to move in a valve-closing direction in response to upstream-end pressure of the first control valve (30A), also causing the second valve element (57) to move in a valve-opening direction in response to downstream-end pressure of the first control valve (30A).

19. The capacity control valve according to claim 2, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33), and

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction;

that the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the upstream end of the first control valve (30A) to the crank chamber (15), and

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away

from the second valve seat; and that

the third control valve (30D) comprises:

a third valve seat (77) formed as part of a passageway that leads from the crank chamber (15) to the suction chamber (32), 5

a third valve element (76) integrally formed with the second valve element (57), located opposite the third valve seat, allowed to move downstream toward and upstream away from the third valve seat (77), and 10 15

a piston (58) integrally formed with the third valve element (46), receiving downstream-end pressure of the first control valve (30A) and impelling the second valve element (57) in a valve-closing direction and the third valve element (76) in a valve-opening direction with the received downstream-end pressure. 20 25

20. The capacity control valve according to claim 2, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33), and 30

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; that 35 40

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A) to the crank chamber (15), and 45

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat; and that 50

the third control valve (30D) comprises:

a third valve seat (77) formed as part

of a passageway that leads from the crank chamber (15) to the suction chamber (32),

a third valve element (76) integrally formed with the second valve element (57), located opposite the third valve seat, allowed to move downstream toward and upstream away from the third valve seat (77),

a piston (58) integrally formed with the third valve element (76), receiving downstream-end pressure of the first control valve (30A) and impelling the second valve element (57) in a valve-closing direction and the third valve element (76) in a valve-opening direction with the received downstream-end pressure, and

a pressure responsive piston (78) installed coaxially with the second and third valve elements (57, 76), actuating the second valve element (57) in a valve-opening direction and the third valve element (76) in a valve-closing direction in response to upstream-end pressure of the first control valve (30A), also actuating the second valve element (57) in the valve-closing direction and the third valve element (76) in the valve-opening direction in response to downstream-end pressure of the first control valve (30A).

21. The capacity control valve according to claim 2, **characterised in that** the first control valve (30A) comprises:

a first valve seat (45a) formed as part of the refrigerant passageway coming from the discharge chamber (33),

a first valve element (61) located opposite the first valve seat to set the cross-sectional area of the refrigerant passageway, allowed to move upstream toward and downstream away from the first valve seat, actuated by a downstream force that is produced and controlled by the solenoid unit (30C) while being forced in a valve-closing direction; that

the second control valve (30B) comprises:

a second valve seat (56) formed as part of a passageway that leads from the downstream end of the first control valve (30A)

to the crank chamber (15), and

a second valve element (57) located opposite the second valve seat, allowed to move upstream toward and downstream away from the second valve seat (56); and that

the third control valve (30D) comprises:

a third valve seat (77) formed as part of a passageway that leads from the crank chamber (15) to the suction chamber,

a third valve element (76) integrally formed with the second valve element (57), located opposite the third valve seat, allowed to move downstream toward and upstream away from the third valve seat (77), and

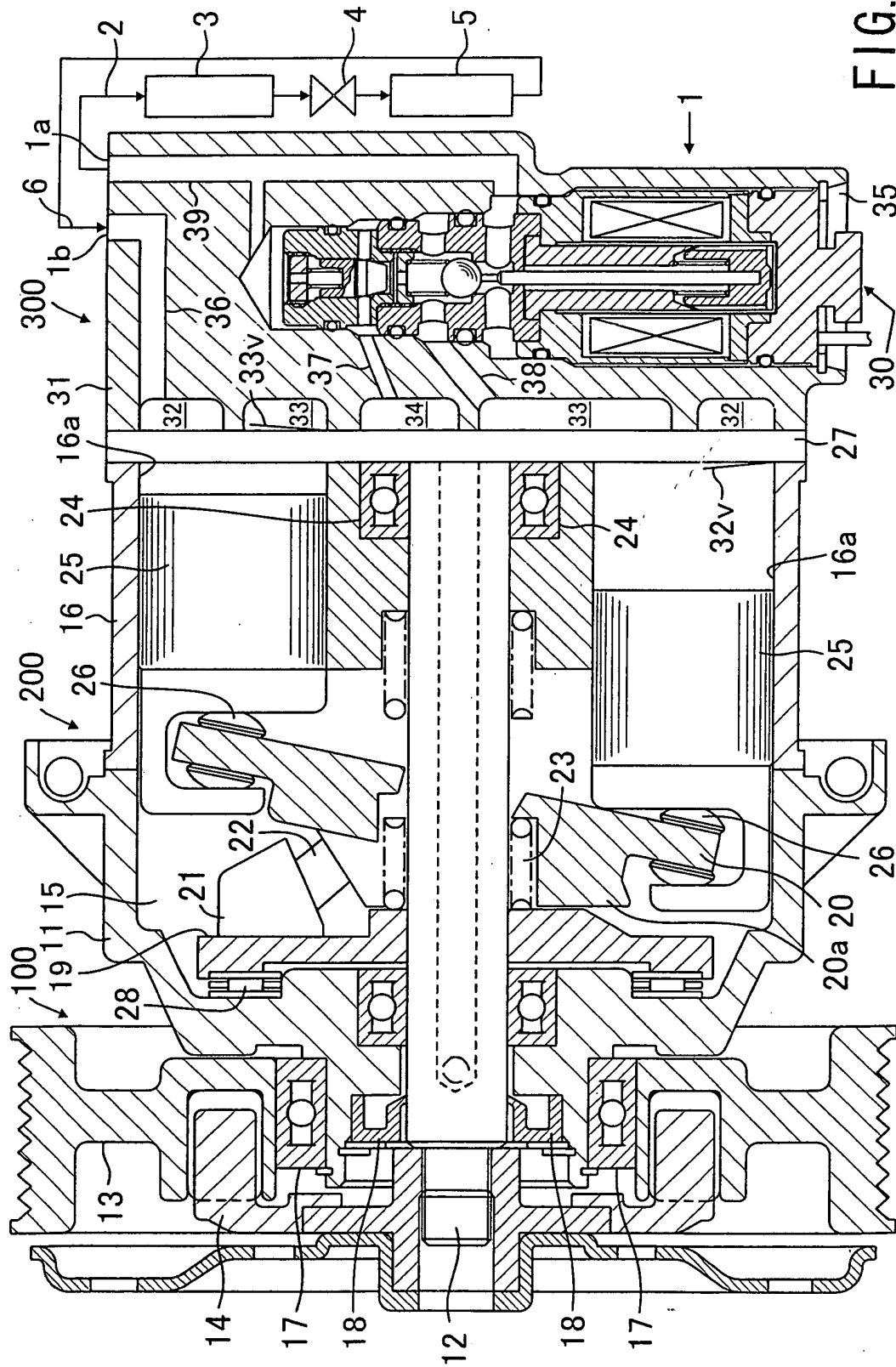
a pressure responsive piston (78) installed coaxially with the second and third valve elements (57, 76), causing the second valve element (57) to move in a valve-opening direction and the third valve element (76) in a valve-closing direction in response to upstream-end pressure of the first control valve (30A), also causing the second valve element (57) to move in the valve-closing direction and the third valve element (76) in the valve-opening direction in response to downstream-end pressure of the first control valve (30A).

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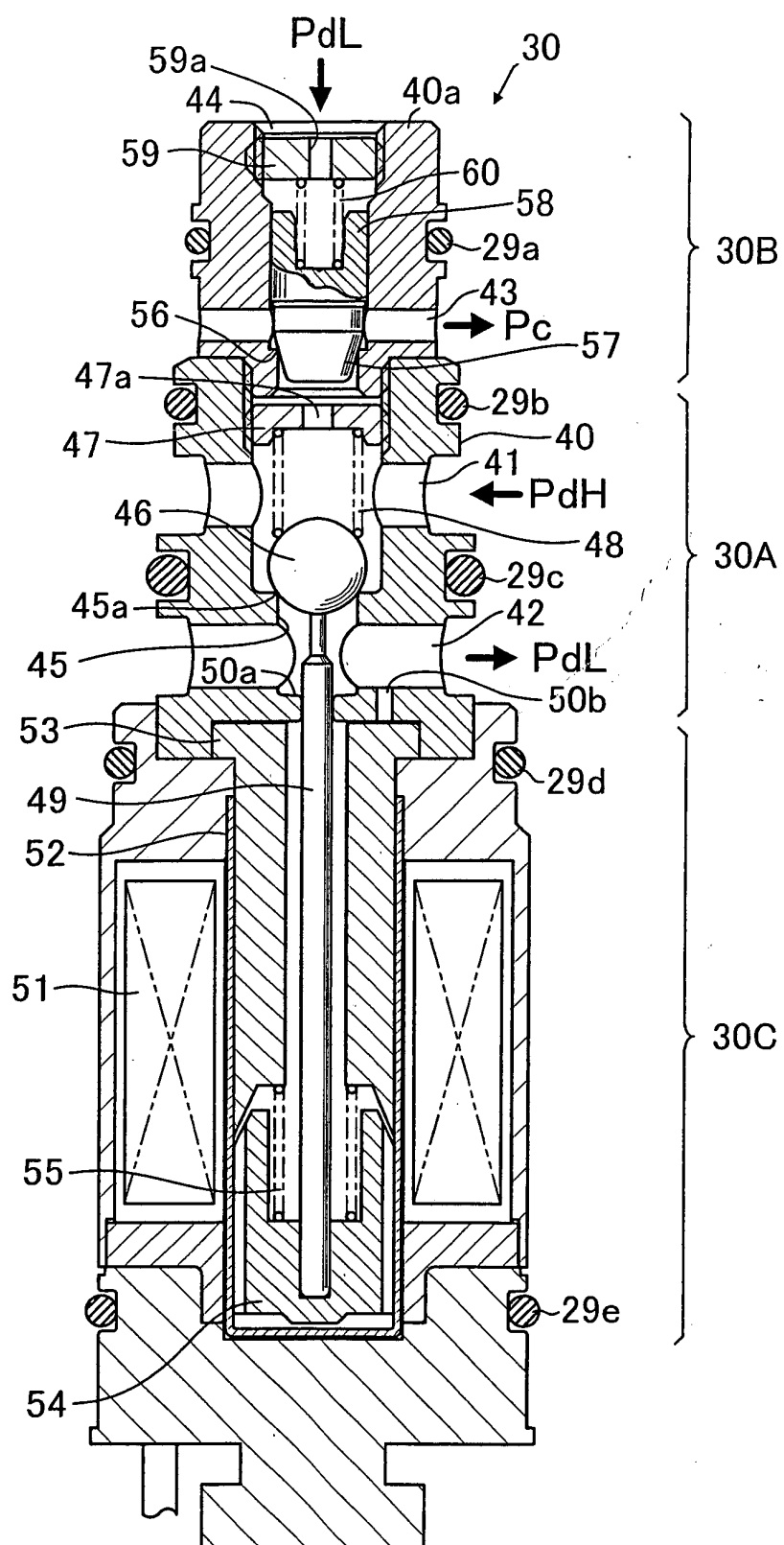


FIG. 2

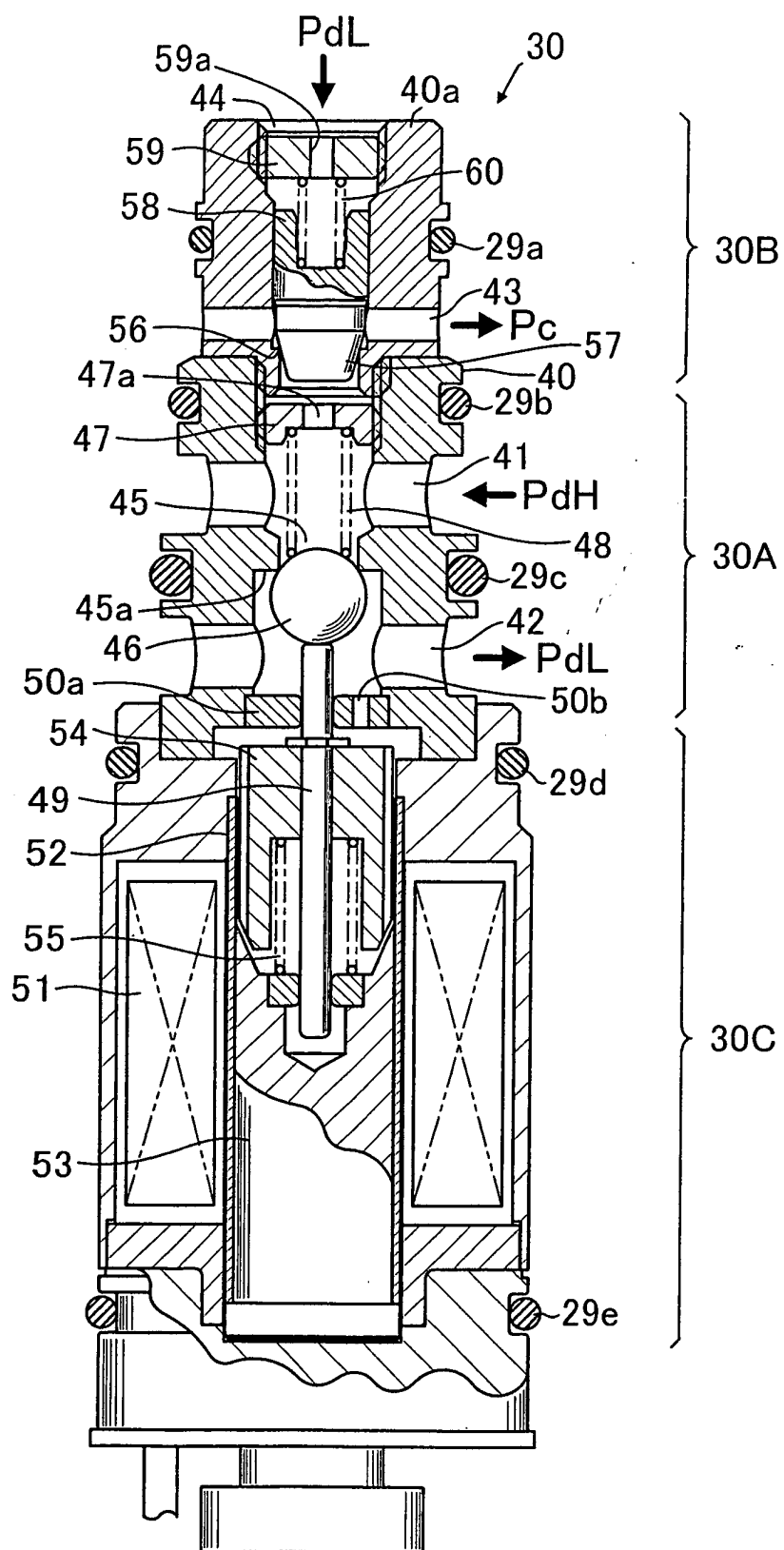


FIG. 3

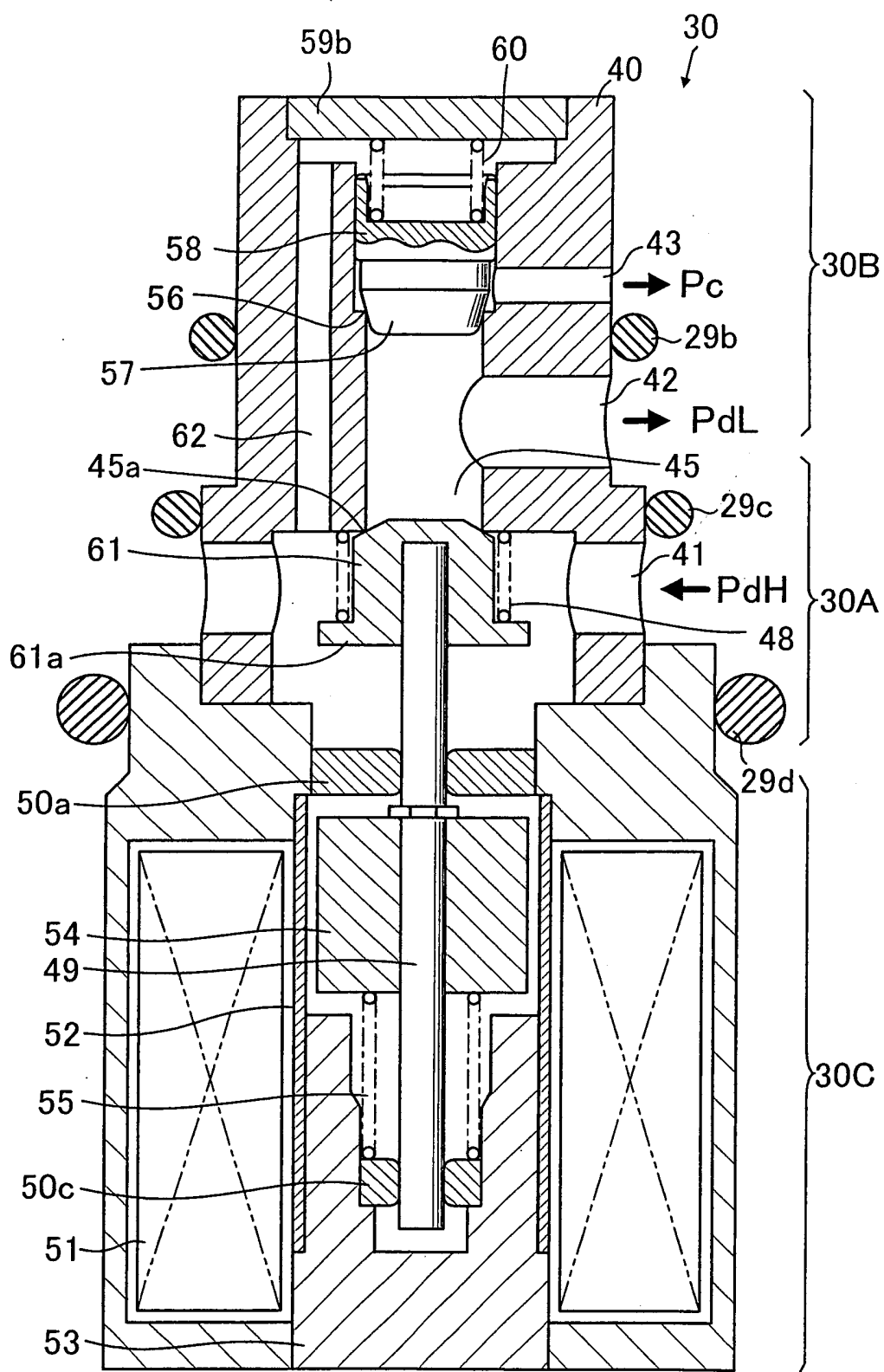


FIG. 4

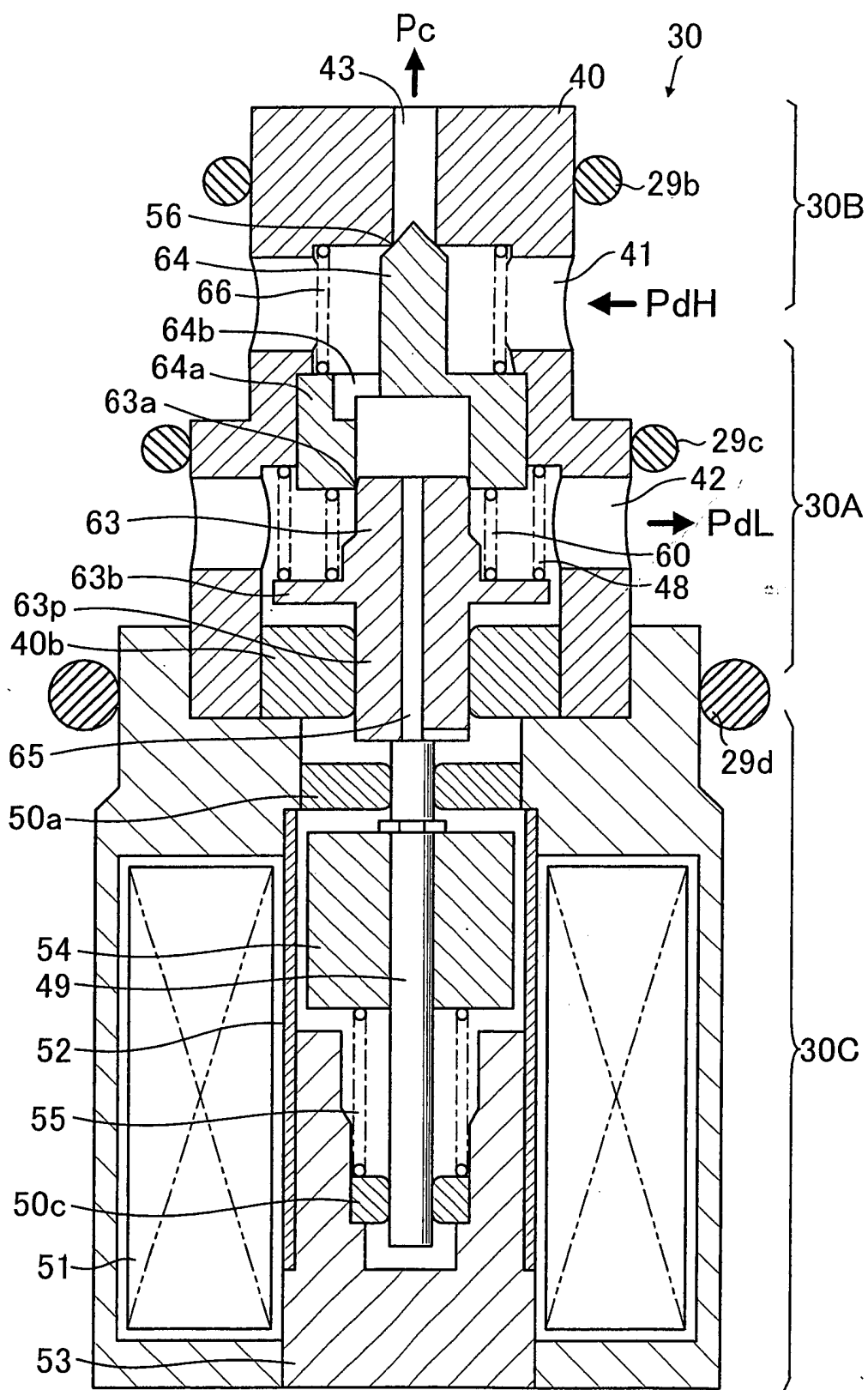


FIG. 5

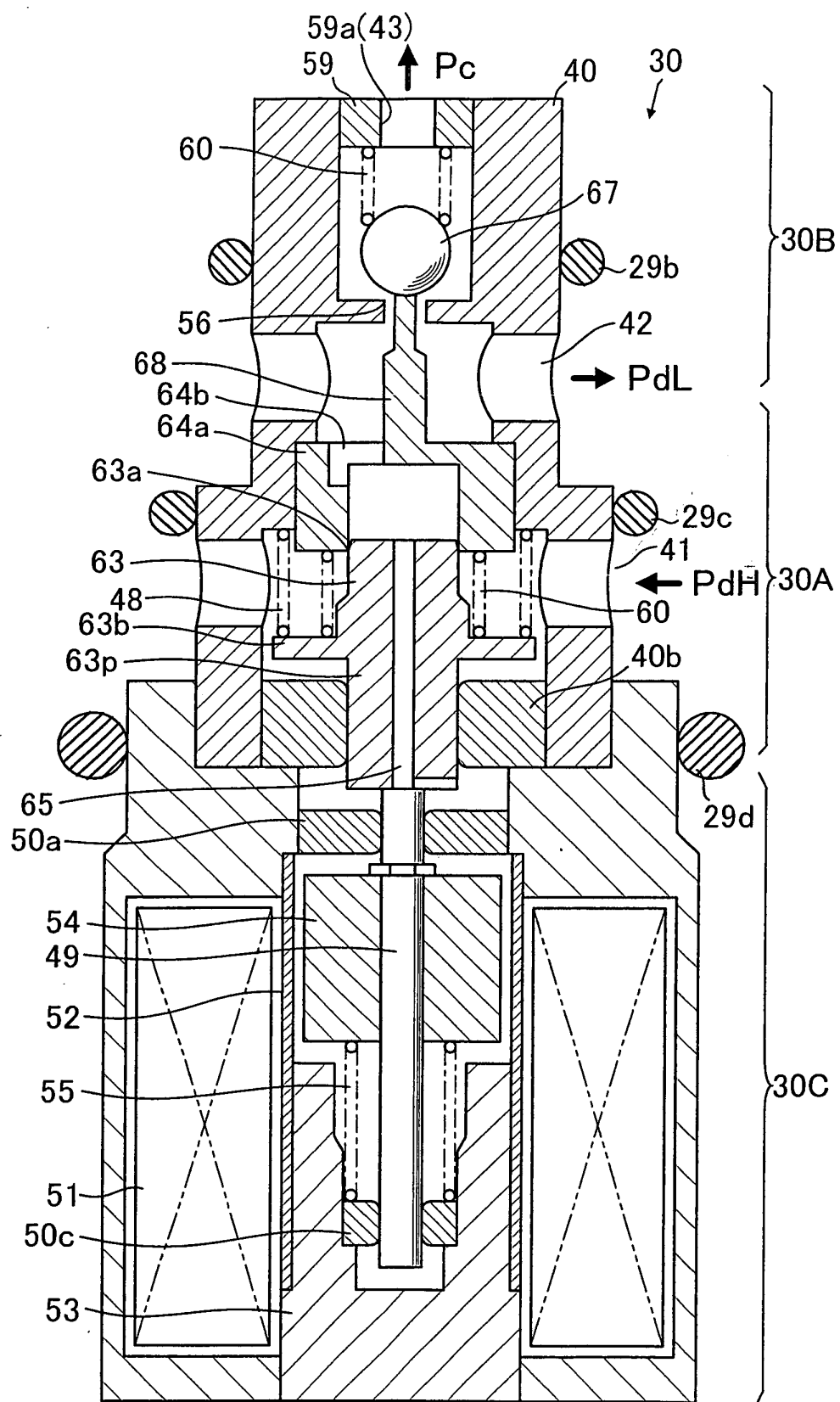


FIG. 6

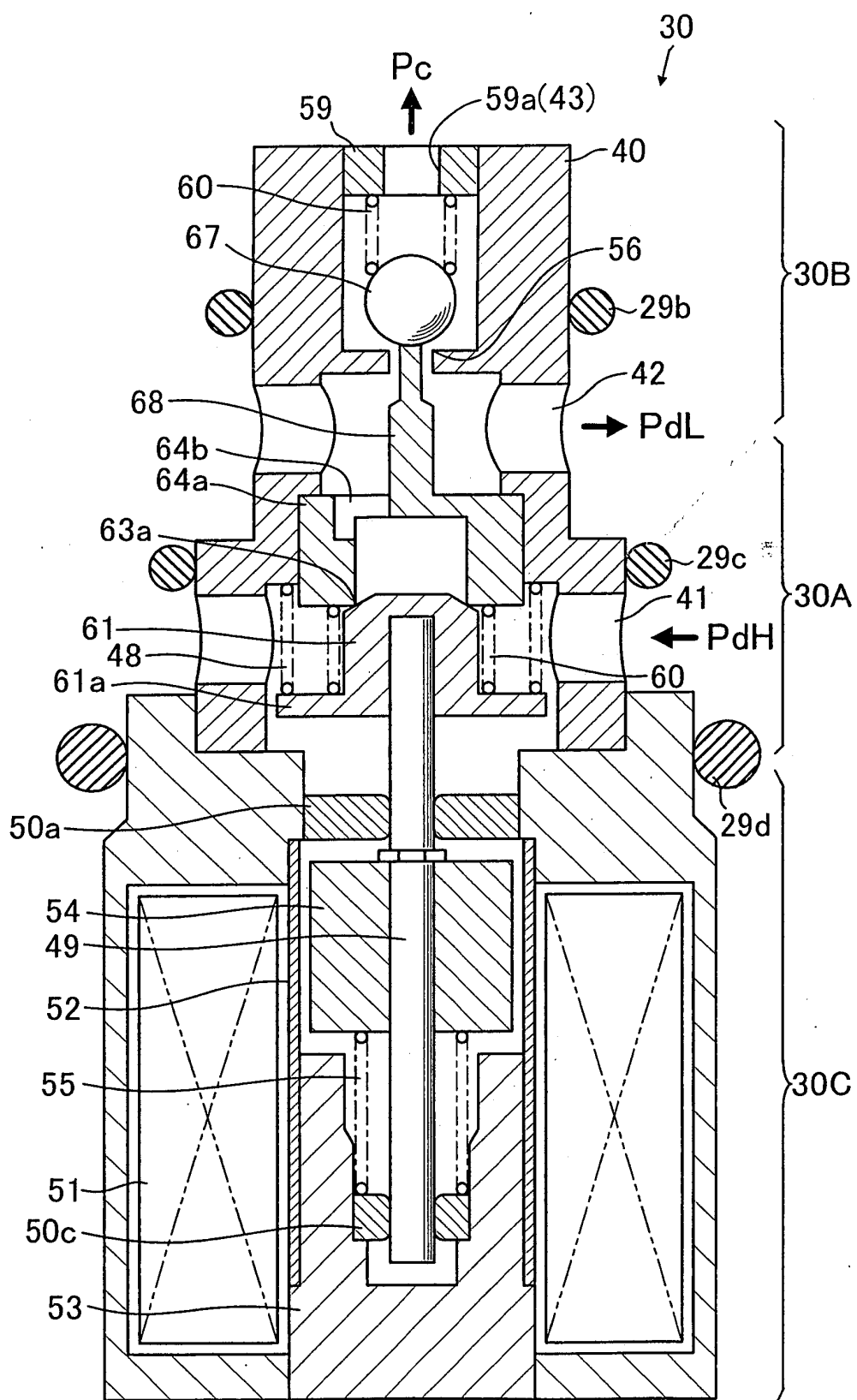


FIG. 7

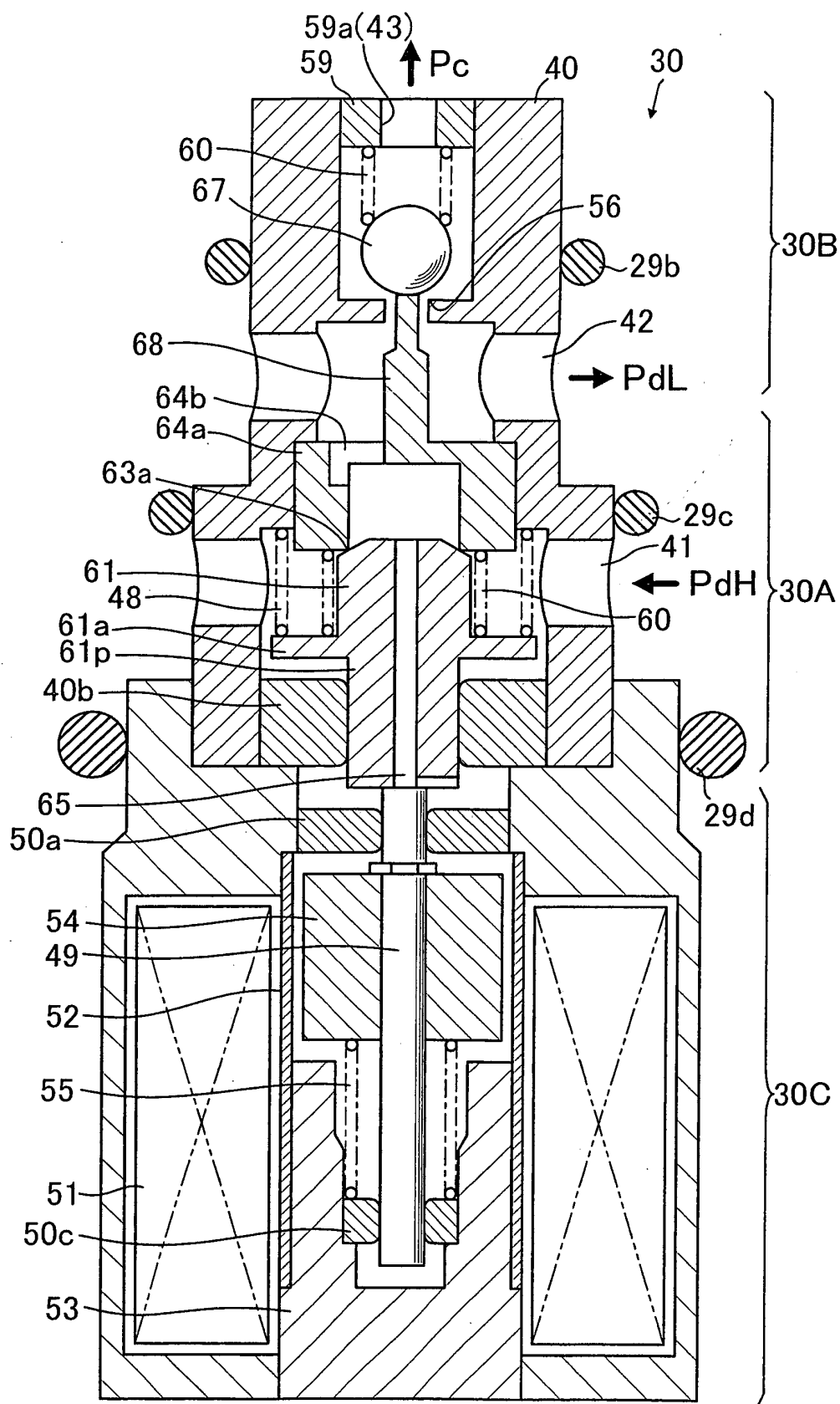


FIG. 8

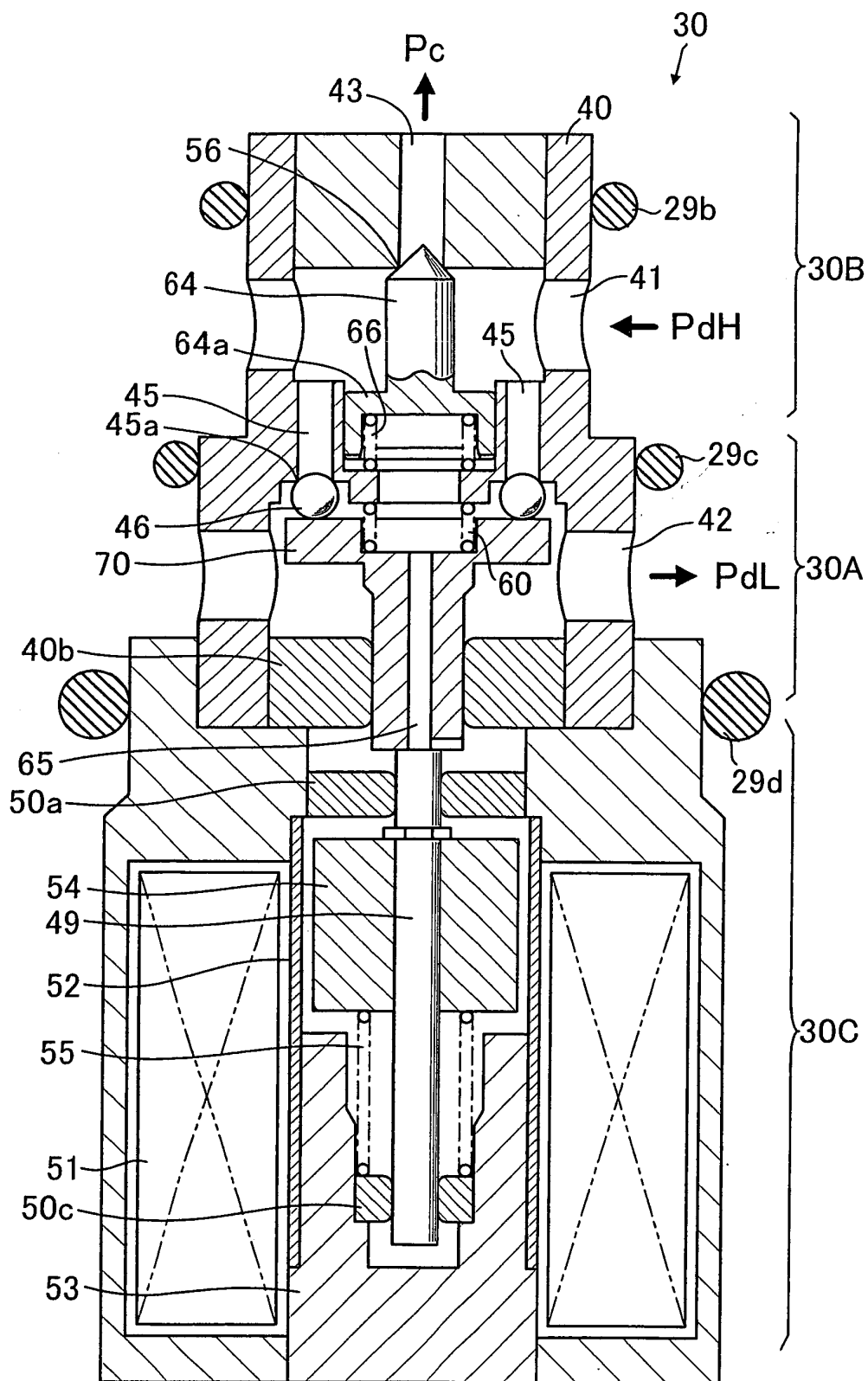


FIG. 9



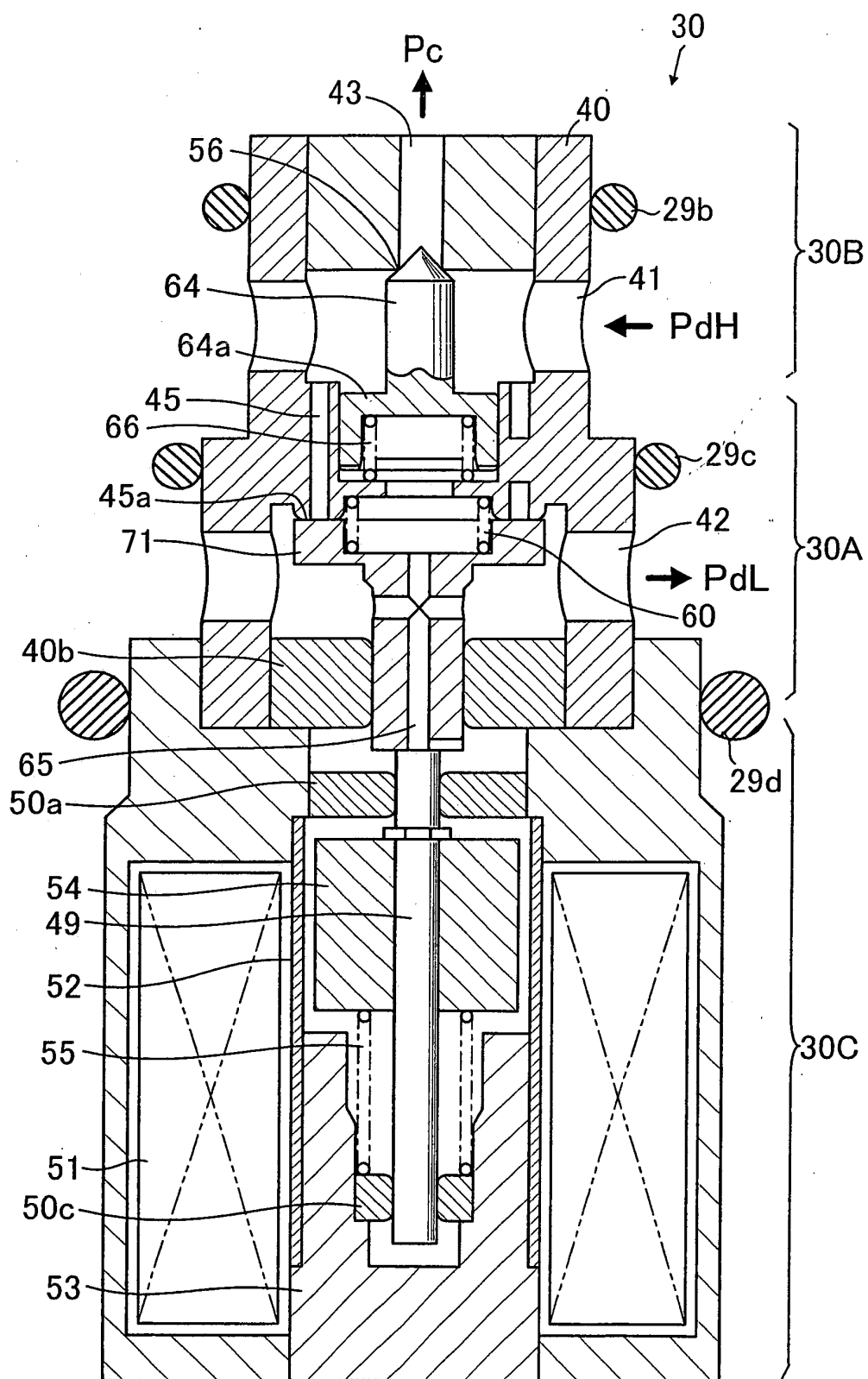


FIG. 10

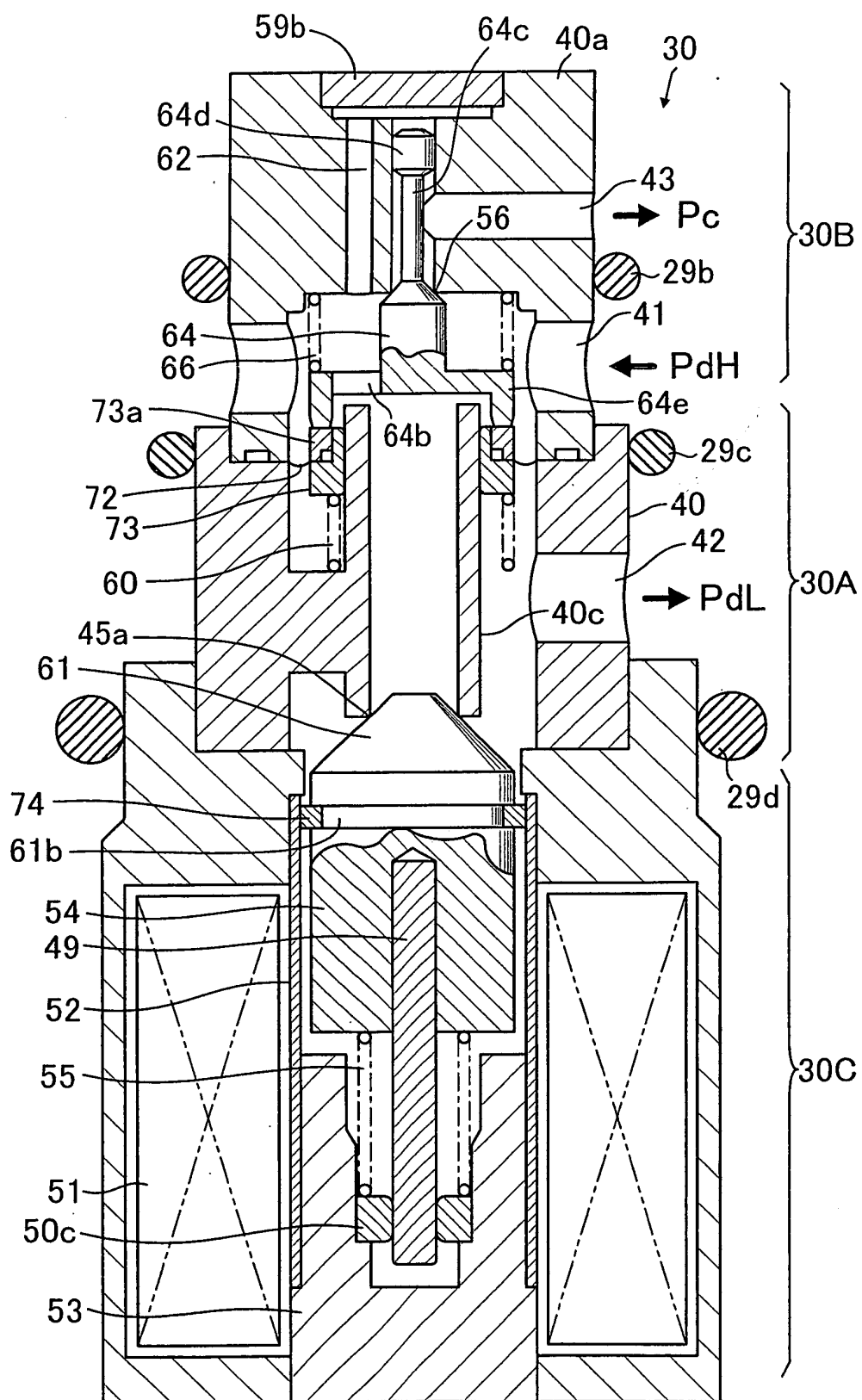


FIG. 11

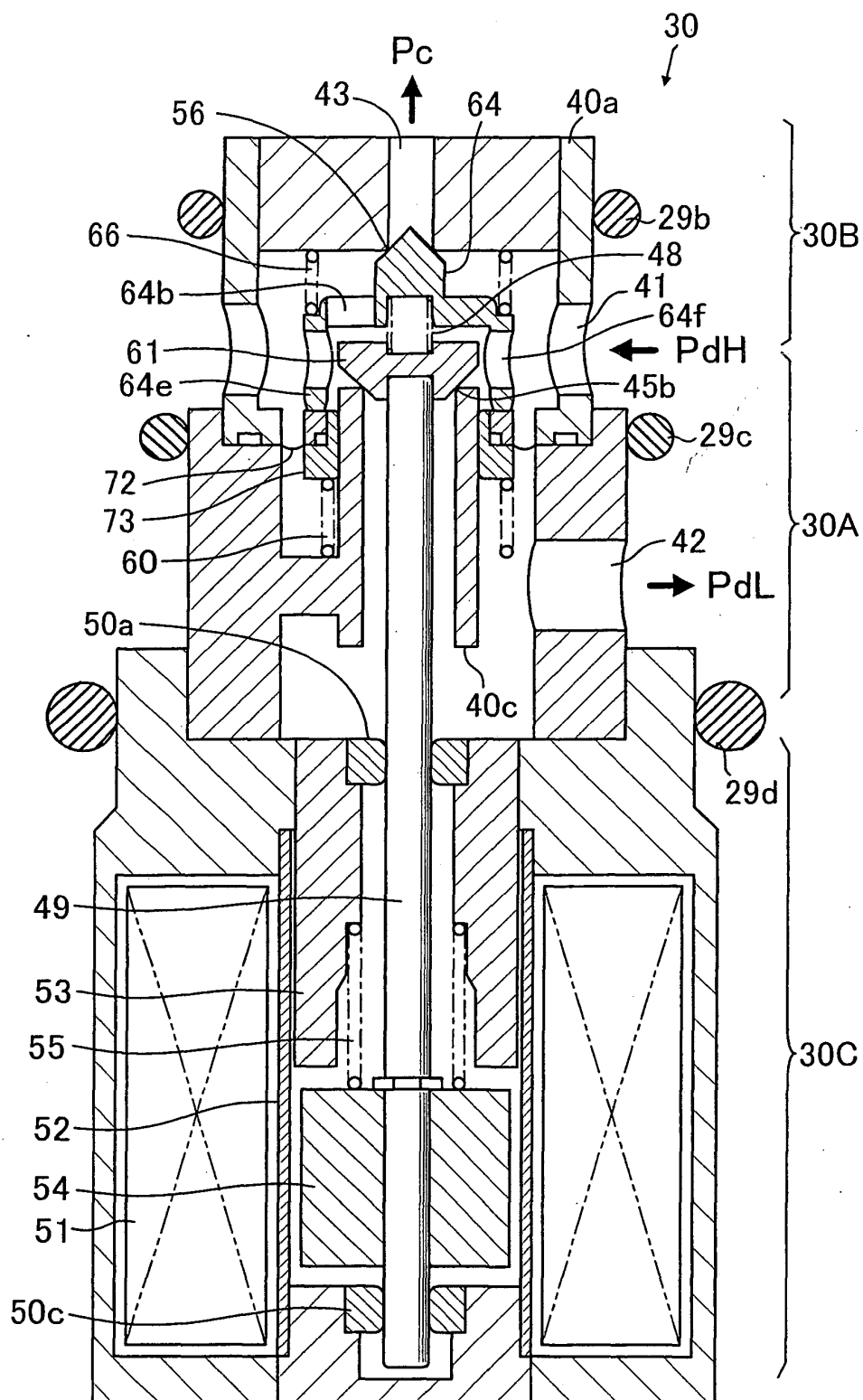


FIG. 12

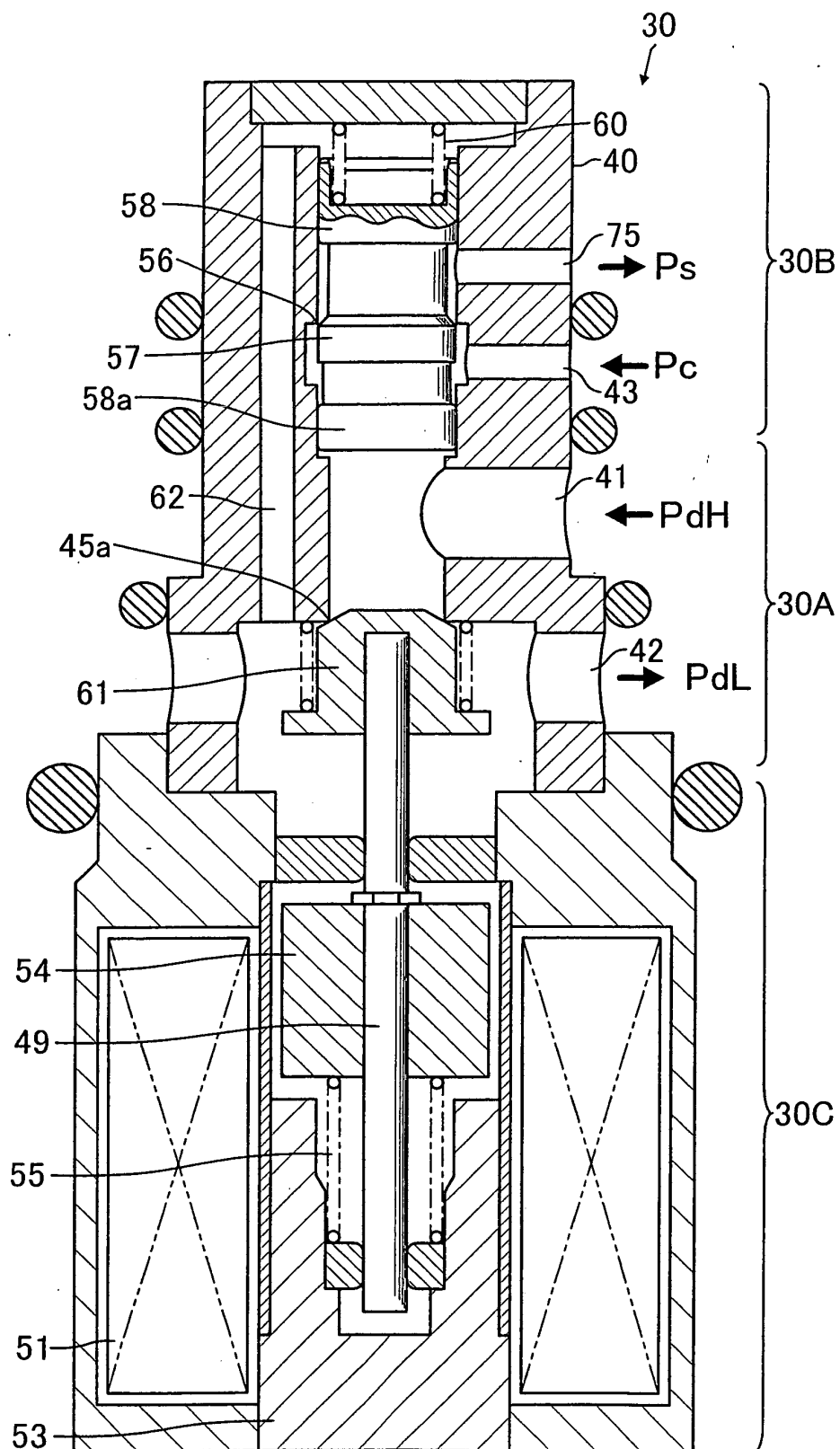


FIG. 13

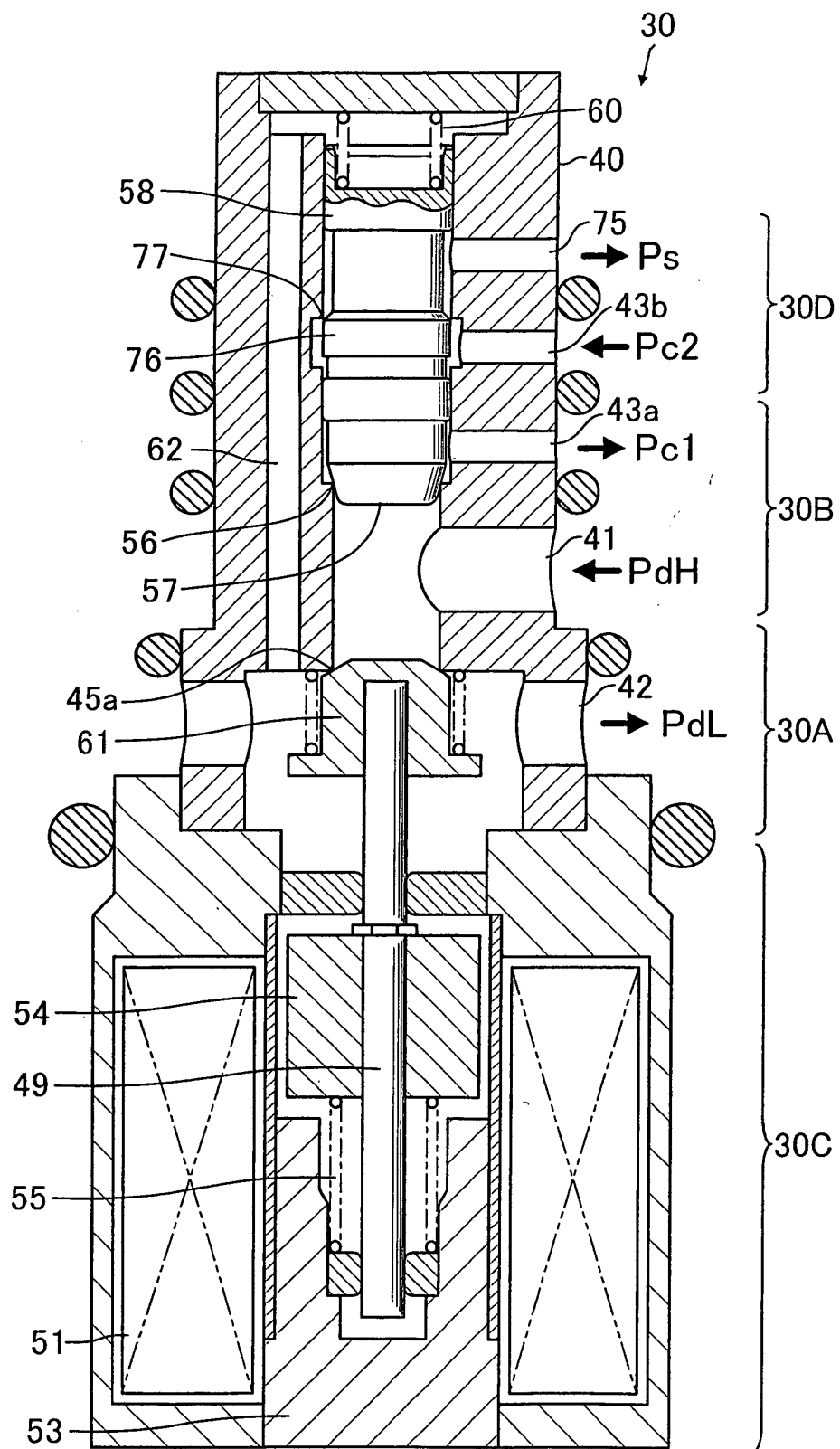


FIG. 14

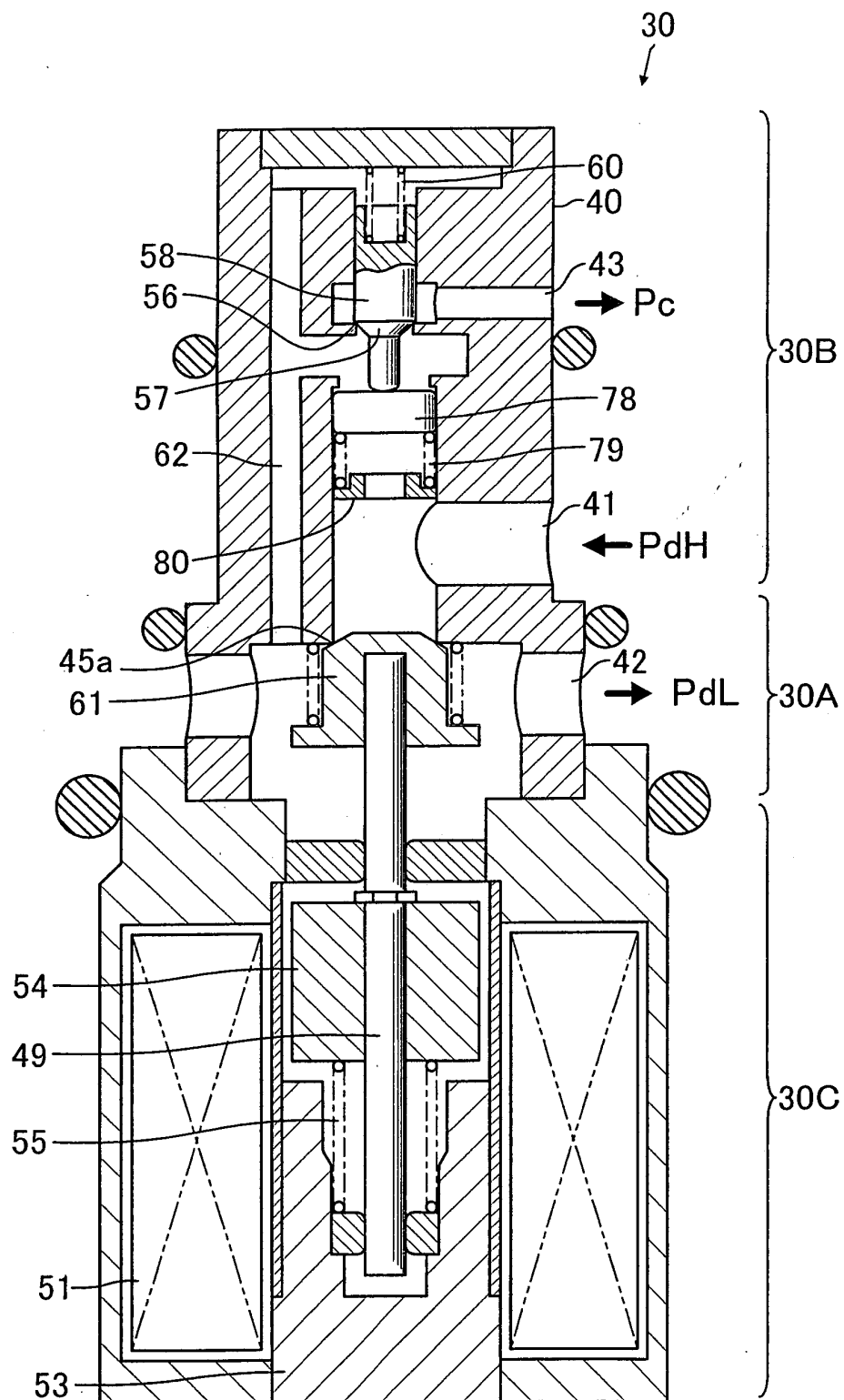


FIG. 15

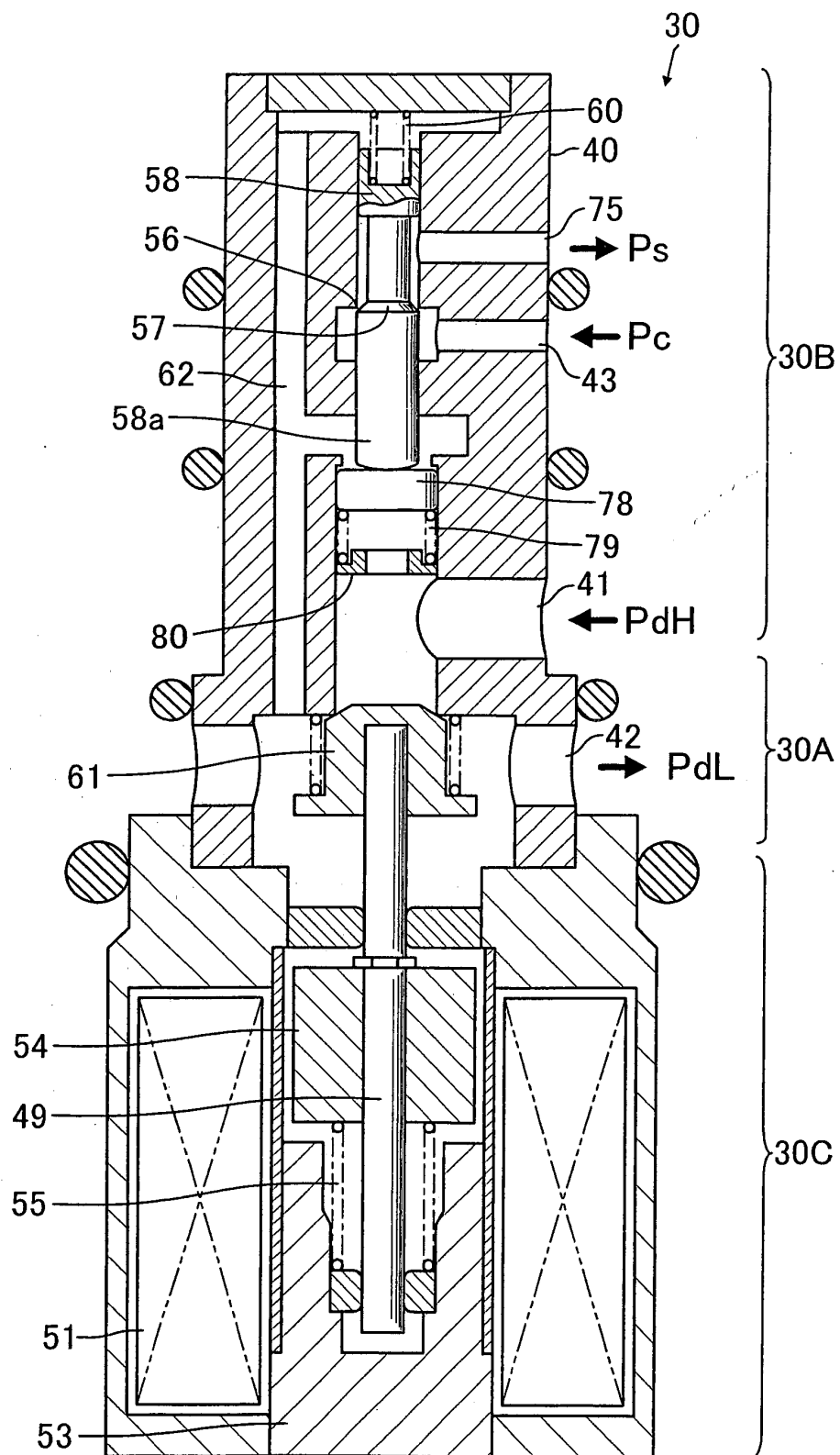


FIG. 16

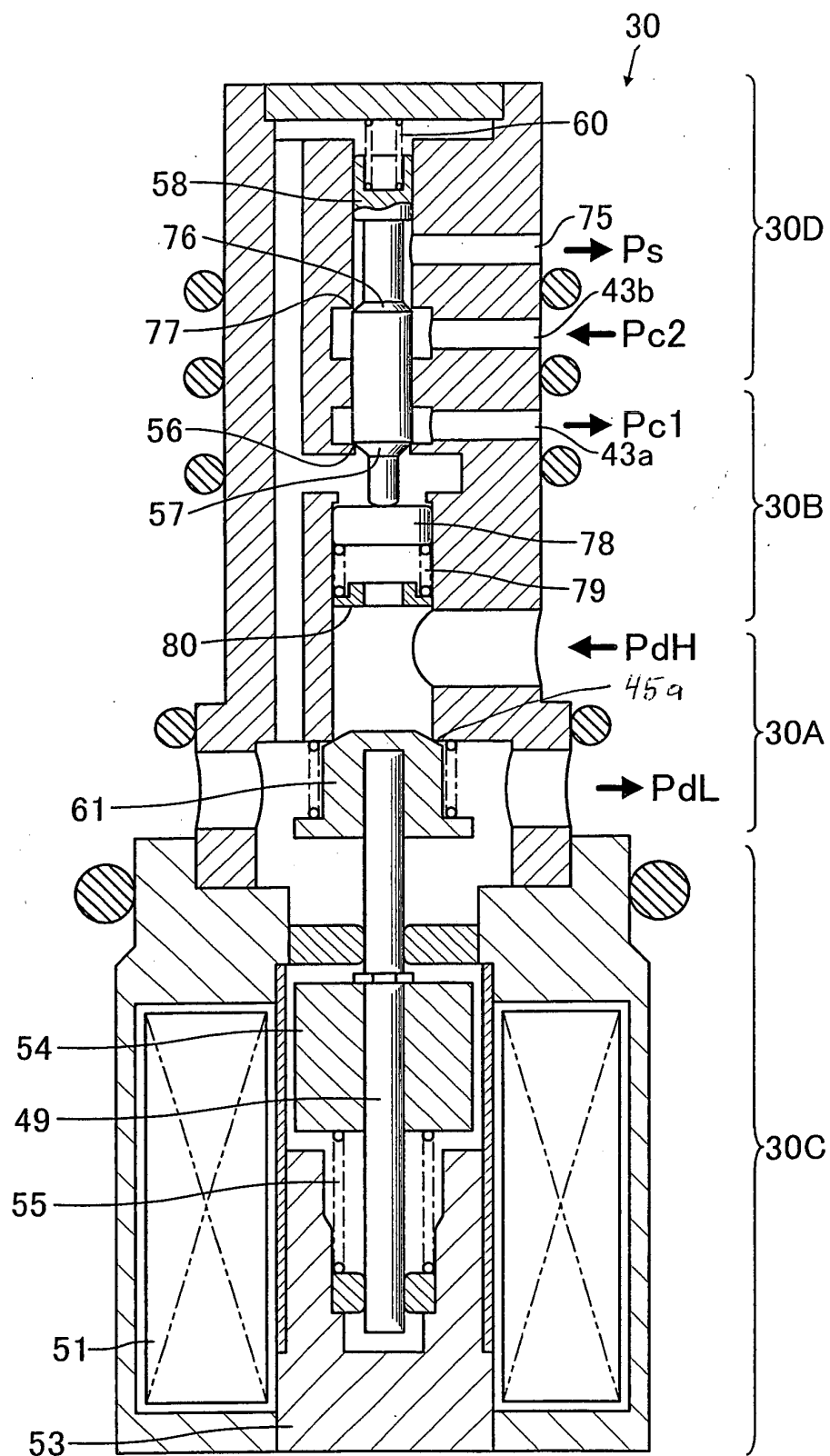


FIG. 17



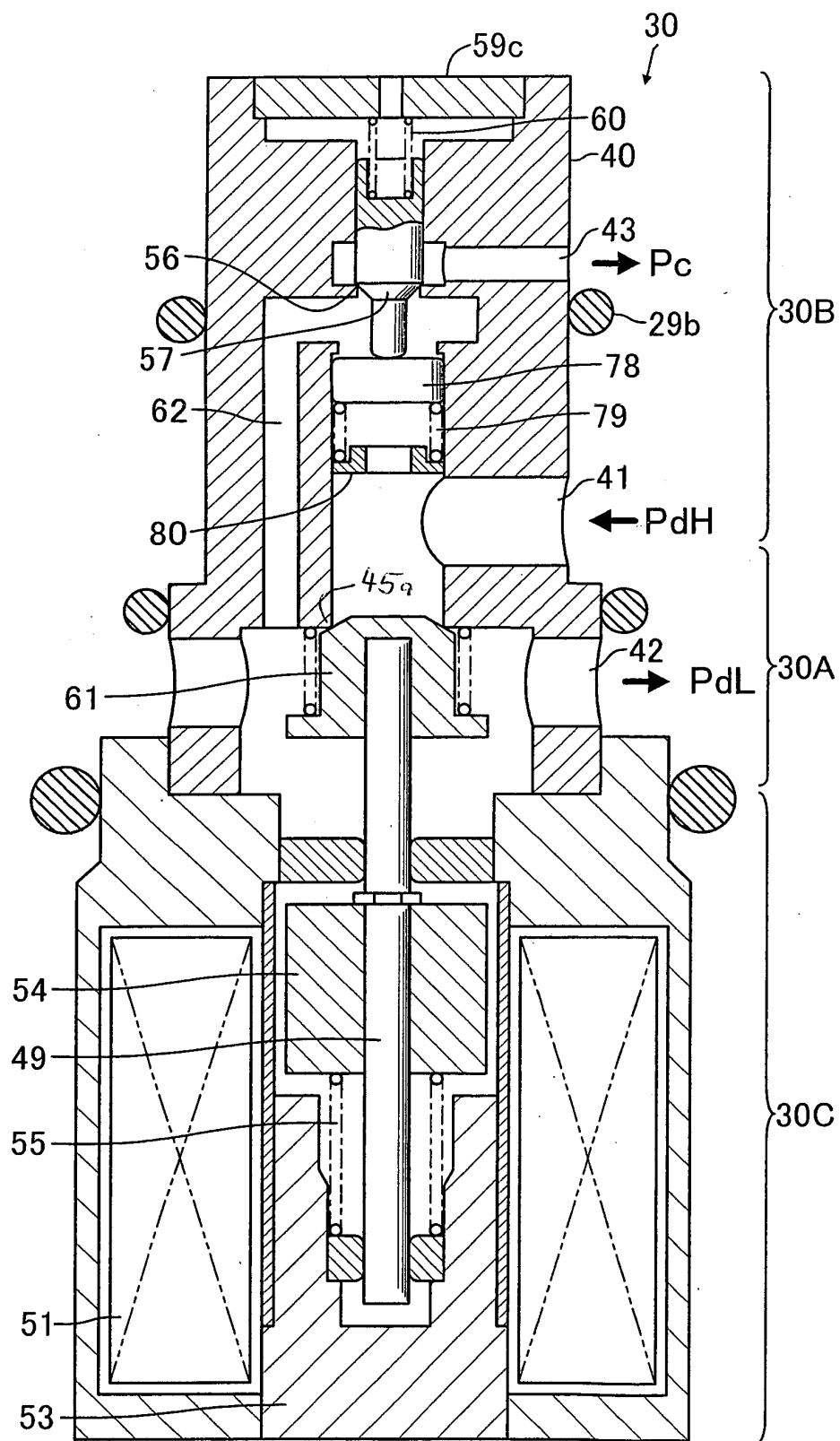


FIG. 18

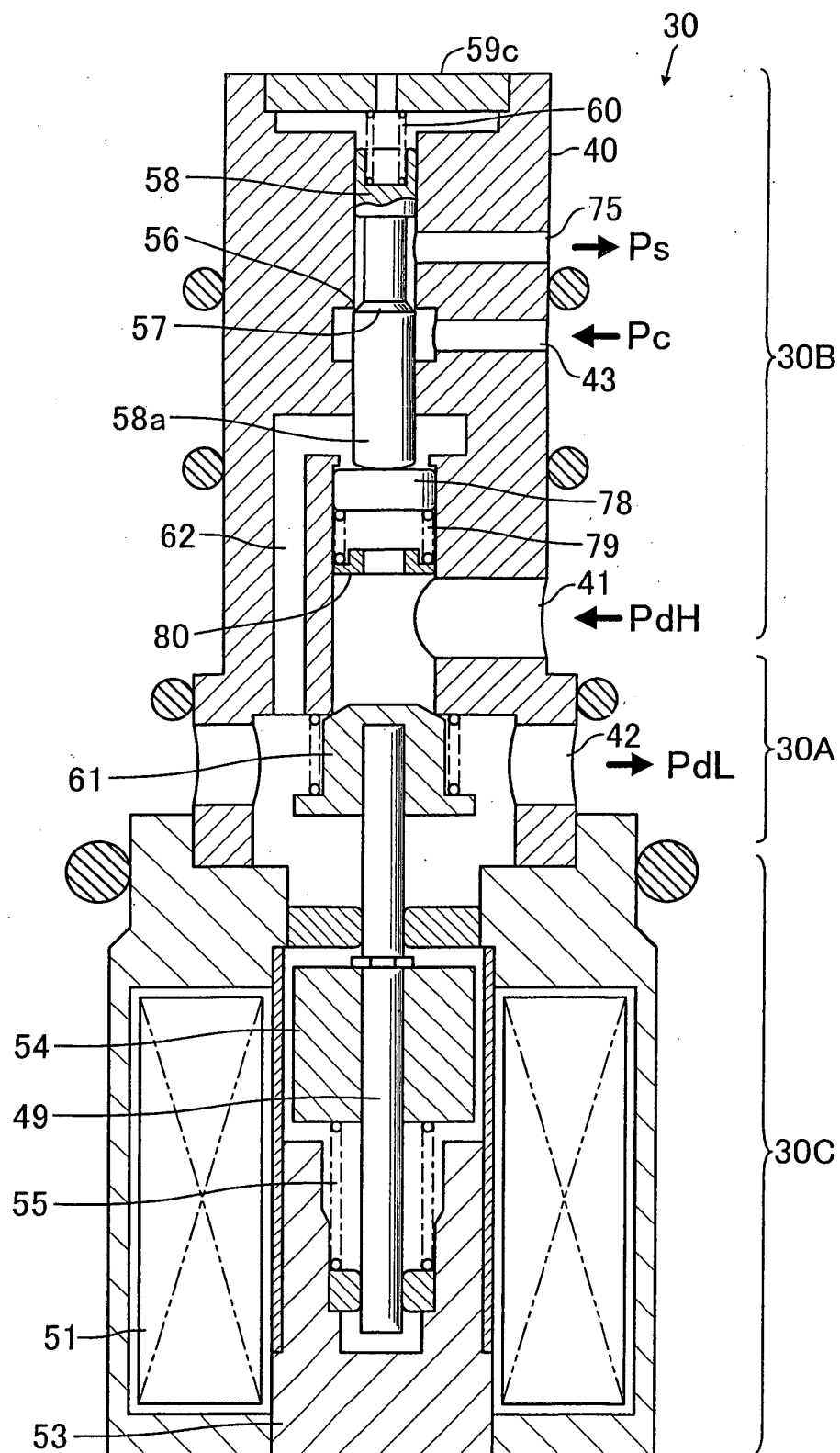


FIG. 19

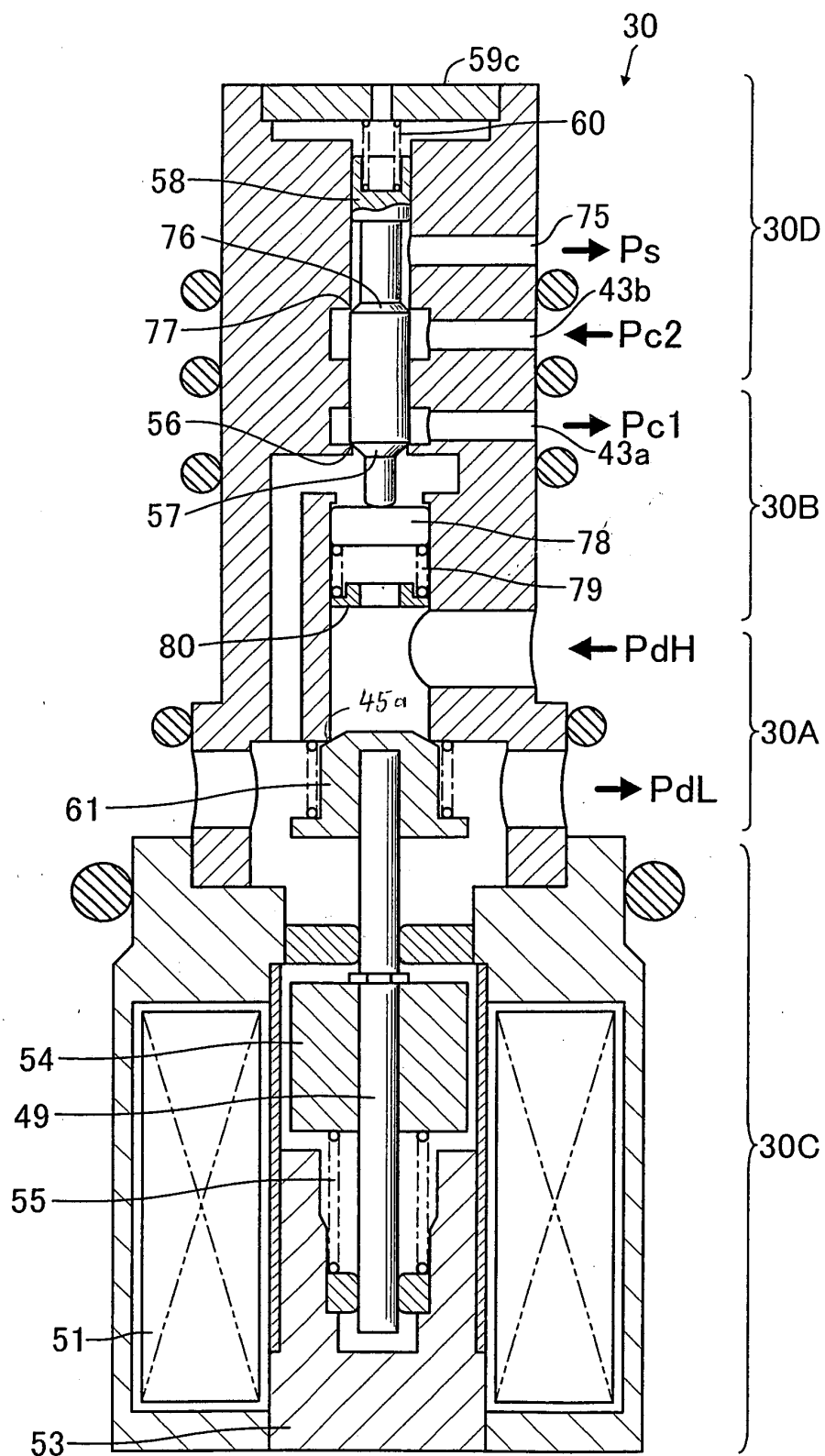


FIG. 20