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(54) **Control valve for a variable displacement compressor**

(57) A control valve for a variable displacement compressor controls the discharge flow rate to be constant, to avoid a check valve in a compressor outlet port. A first control valve 10A controls the cross-sectional area between a discharge chamber to the outlet port. A second control valve 10B controls the flow rate between the discharge chamber and a crankcase such that a differential pressure (Pdh-Pdl) across the first control valve 10A becomes constant. When a solenoid section 10C is de-energized, a first valve element 18 is engaged with a piston 33 to forcibly fully open the second control valve 10B. The piston outer diameter equals the inner diameter of a second valve seat 17, so that the discharge pressure Pdl does not adversely affect the fully-opening operation of the second control valve 10B, and allows to maintain the first control valve 10A fully-closed.

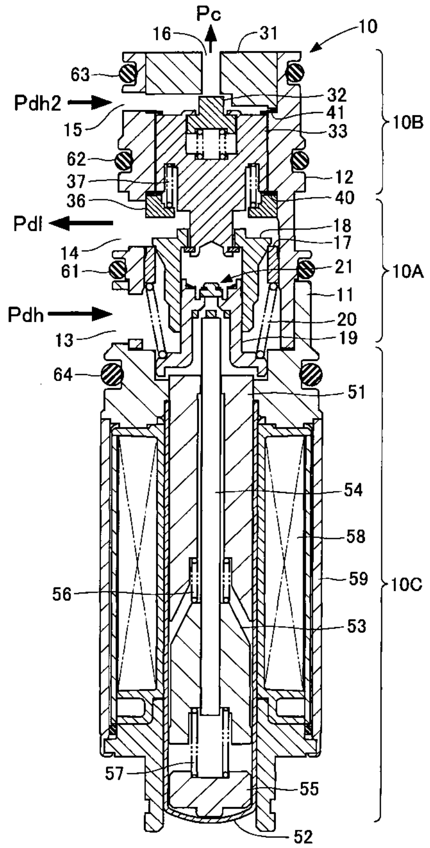

FIG. 1

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Description

[0001] The present invention relates to a control valve according to the preamble of claim 1, particularly for controlling a constant refrigerant discharge flow rate.

[0002] In the refrigeration cycle of an automotive air conditioner, for compressing refrigerant, a variable displacement compressor capable of varying the volume (discharge amount) of refrigerant is employed so as to obtain an adequate cooling capacity without being constrained by the speed of the engine which drives the compressor. Pistons that reciprocate parallel to a shaft driven by the engine are connected to a wobble plate (swash plate) mounted on the shaft. By rotating the wobble plate and varying the plate inclination angle within a crankcase, the stroke of the pistons is varied to control the discharge amount. In order to change the plate inclination angle, the balance of pressures acting on the both sides of each is changed by introducing compressed refrigerant into the crankcase to cause a change in the pressure in the crankcase. In general, pressure in the crankcase is changed by a control valve in a passage communicating between the discharge chamber and the crankcase.. Now, when the control valve is set to a predetermined valve lift, if the speed of the engine increases, pressure introduced from the discharge chamber into the crankcase increases to make the plate inclination angle close to 90°, whereby the compression volume is controlled to be small. Inversely, when the speed of the engine drops, pressure in the crankcase decreases whereby the compression volume is controlled to become larger. The compressor is controlled such that the volume of discharged refrigerant is not varied irrespective of the rotational speed of the engine.

[0003] Among compressor control methods it is generally known, for example, to hold suction pressure Ps in the suction chamber constant, and to hold the differential pressure between the suction pressure Ps and a discharge pressure Pd constant. It is also known to make the discharge flow rate constant (JP-A- 2001-107854).

[0004] According to JP-A- 2001-107854 the differential pressure between two pressure monitoring points is detected by sensors to thereby indirectly grasp the flow rate into the suction chamber. The control valve controls the flow rate between the discharge chamber and the crankcase such that the flow rate into the suction chamber becomes constant, whereby the discharge flow rate is controlled to be constant.

[0005] In contrast, a control valve known from JP-A-2004-116349 comprises a first control valve for the flow rate between the discharge chamber and a refrigerant outlet port, a second control valve that senses the differential pressure across the first control valve using a diaphragm, and which controls the flow rate between the discharge chamber and the crankcase based on the differential pressure, to change the displacement of the compressor, and to control the flow rate through the first control valve to be constant. A solenoid section sets the flow rate through the first control valve. The first control valve forms a variable orifice that has its passage area set by the solenoid section according to changes in external conditions. The second control valve senses the differential pressure across the variable orifice, and controls the crankcase pressure such that the differential pressure equals a predetermined value. The differential pressure across the variable orifice is held at the predetermined value, whereby the discharge flow rate is controlled to be constant. However, when the compressor stops operation, its capability of compressing and discharging refrigerant is suddenly lost. This may invert the relationship in pressure between the discharge chamber that has been at high pressure and the refrigerant outlet port located downstream of the first control valve. This acts not to control the second control valve to the minimum displacement side but to control the same to the maximum displacement side. To solve this problem, the conventional control requires a check valve at the refrigerant outlet port so as to prevent the first control valve from being adversely affected by the pressure at the refrigerant outlet port upon stoppage of the compressor, meaning increased manufacturing costs.

[0006] It is an object of the present invention to provide such a control valve that controls the discharge flow rate, without requiring a check valve at a compressor refrigerant outlet port.

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

[0008] Since the control is insensitive to pressure on the downstream side of the first control valve, pressure at the refrigerant outlet port is prevented from acting on the second control valve in the direction of increasing the displacement of the compressor even when the pressure at the refrigerant outlet port becomes higher than the pressure in the discharge chamber. This makes it possible to dispense with the check valve conventionally provided at the refrigerant outlet port of the compressor, which is advantageous in that it is possible to reduce the costs of the compressor.

[0009] Embodiments of the invention will be explained with the help of the drawings.

Fig. 1	is a central sectional view of a first embodiment of a control valve for a variable displacement compressor,
Fig. 2	is a partial enlarged sectional view of the first embodiment.
Figs. 3 to 9	are sectional views of second to eighth embodimens.

[0010] Figs. 10A to 10C are views explaining characteristics of a diaphragm, among which Fig. 10A shows a pressure-free state, Fig. 10B shows a state in which the diaphragm is displaced by a differential pressure, and Fig. 10C shows a state in which the differential pressure is applied to the displaced diaphragm in a direction opposite to the direction of the displacement.

[0011] Figs. 11A to 11 B are explanatory views of the construction of a differential pressure-sensing section of the eighth embodiment of Fig. 9, among which Fig. 11A shows a case where discharge pressure Pdh_2 is higher than an outlet port pressure Pdl, and Fig. 11 B shows a case where the discharge pressure Pdh_2 is lower than outlet port pressure Pdl.

[0012] The control valve 10 (first embodiment, Figs 1 and 2) for a variable displacement compressor comprises a first control valve 10A that controls a passage cross-sectional area of a refrigerant passage between a discharge chamber of the compressor and a refrigerant outlet port, a second control valve 10B that controls the flow rate between the discharge chamber and a crankcase, and a solenoid section 10C that sets the passage cross-sectional area of the refrigerant passage of the first control valve 10A, all of which are arranged on the same axis.

[0013] The first control valve 10A and the second control valve 10B have a first body 11 and a second body 12 press-fitted into the first body 11. The first body 11 and the second body 12 are provided with ports 13, 14, and 15. When the control valve 10 is mounted in the compressor, the ports 13, 14, and 15 are communicated respectively with the discharge chamber, for introducing discharge pressure Pdh into port 13" with the refrigerant outlet port, for discharging discharge pressure Pdl, and again with the discharge chamber, for introducing discharge pressure Pdh2 into port 15. The second body 12 has another port 16 communicating with the crankcase, for discharging pressure Pc.

[0014] The control valve 10 can be applied to a compressor configured such that the port 13 at the discharge pressure Pdh and the port 15 at the discharge pressure Pdh2 both communicate with the discharge chamber. However, it is preferable that the control valve 10 is applied to a compressor such that the port 13 at discharge pressure Pdh directly communicates with the discharge chamber, and the port 15 at discharge pressure Pdh2 communicates with an outlet port of an oil separator disposed on the downstream side of the discharge chamber. This enables the second control valve 10B to return compressor lubricating oil, contained the in refrigerant in a large amount, while controlling pressure Pc.

[0015] The first control valve 10A has an axial passage through the second body 12 between the ports 13, 14. A first valve seat 17 is press-fitted in the passage, and a first valve element 18 is movably disposed downstream of the first valve seat 17.

[0016] The first valve element 18 has a hollow integral cylindrical portion axially extending through a valve hole. A guide 19 is rigidly press-fitted in the hollow cylindrical portion. The guide 19 is urged by a spring 20 in valve-closing direction of the first control valve 10A. A portion of the guide 19 is in sliding contact with an inner wall of the first body 11 and has an outer diameter equal to the inner diameter of the first valve seat 17, whereby discharge pressure Pdh from port 13 equally acts on the first valve element 18 and the guide 19 in opposite directions to prevent that the discharge pressure Pdh adversely affects control operations of the first control valve 10A.

[0017] The guide 19 forms an axial refrigerant passage, and is equipped with a check valve 21 for opening and closing the refrigerant passage. The check valve 21 has a valve element 22, e.g. made of rubber, disposed on a low-pressure side or port 14 of the refrigerant passage and a leaf spring 23 axially movably holding the valve element 22. In a neutral pressure-free state the valve element 22 is held by the leaf spring at a position where the refrigerant passage is slightly open.

[0018] The second control valve 10B has a second valve seat 31 press-fitted into the foremost end of the second body 12, where the axial port 16 is formed, and a second movable valve element 32 upstream of the second valve seat 31. The second valve element 32 is axially movably guided and captured by a piston 33 forming a differential pressure-sensing section which in turn is axially guided by the second body 12. The piston 33 has a recessed valve element base portion-accommodating portion 34 opposed to the second valve seat 31, containing a spring 35 and a base portion of the second valve element 32. An open end of the accommodating portion 34 is swaged to hold back the second valve element 32. This makes it possible to soften collision impacts between the second valve element 32 and the second valve seat 31 when the second control valve 10B closes quickly.

[0019] The piston 33 is urged in valve-closing direction of the second control valve 10B by a spring 37 between the piston 33 and a spring-receiving portion 36 press-fitted into the second body 12. The spring 37 has a smaller spring force than the spring 20. The piston 33 has an integral extended portion 38 toward the solenoid section 10c and into the first valve element 18. A washer 39 fixed to an end of the extended portion 38, e.g. by swaging, is brought into engagement with a stepped portion formed on the first valve element 18. Thus, when the first control valve 10A is fully closed, the piston 33 is forcibly pulled by the first valve element 18 in valve-opening direction of the second control valve 10B, and hence the second control valve 10B can be held fully open. The outer diameter of the piston 33 is equal to the inner diameter of the first valve seat 17 such that when the piston 33 is engaged with the first valve element 18, the piston 33 is not adversely affected by the discharge pressure Pdl.

[0020] The second control valve 10B has film-like seal rings 40 and 41, formed e.g. of rubber, for sealing clearances between the piston 33 and the second body 12 by the pressures in the ports 14, 15. The seal rings 40 and 41 are arranged between a stepped portion of the second body 12 and the spring-receiving portion 36 and between a stepped portion of the second body 12 and the second valve seat 31, respectively.

[0021] The solenoid section 10c has a fixed core 51 in a central opening of the first body 11 and fitted into an opening of a bottomed sleeve 52 in a manner blocking the opening. The bottomed sleeve 52 contains a plunger 53, a shaft 54 axially extending through the core 51 and rigidly fixed to the plunger 53, an adjustment member 55 disposed on the bottom of the bottomed sleeve 52 for axially plastically deforming the bottom, thereby adjusting spring loads, a spring 56 between the core 51 and the plunger 53, and a spring 57 between the plunger 53 and the adjustment member 55. The shaft 54 is axially movably guided by the core 51 and the plunger 53, with a free end extending into the guide 19. When the solenoid section 10c is energized, the free shaft end abuts at an intercommunicating plate 24 fitted into the guide 19 on a side opposite from the check valve 21, for urging the first valve element 18 in valve-opening direction of the first control valve 10A. The bottomed sleeve 52 is surrounded by a coil 58 and a yoke 59.

[0022] O-rings 61, 62, 63, 64 are provided for sealing between the ports 13, 14, the ports 14, 15, the ports 15, 16, and the port 13 and the atmosphere, respectively, when the control valve 10 is mounted in the compressor.

[0023] When the solenoid section 10C is de-energized, as shown in Fig. 1, while the compressor is operating, the first control valve 10A is forcibly fully closed by the spring 20, and the second control valve 10B is fully open since the piston 33 is pulled in valve-opening direction against the spring 37 by the first valve element 18. All the discharged refrigerant is introduced into the crankcase via the second control valve 10B, and the compressor is in the minimum displacement operation state. The control valve 10 thus can be applied to a variable displacement compressor without an electromagnetic clutch between the compressor and the driving engine.

[0024] When the compressor is started, control current is supplied to the solenoid section 10C. As the control current increases, the plunger 53 is pulled by the core 51, whereby the first valve element 18 is pushed upward by the shaft 54, as viewed in Fig. 1. The piston 33 of the second control valve 10B, engaged with the first valve element 18, is also pushed upward, as viewed in Fig. 1, by the spring 37, until the second valve element 32 is seated on the second valve seat 31 to fully close the second control valve 10B. Since the refrigerant discharged ceases to be introduced into the crankcase, the compressor is now shifting to the maximum displacement operation.

[0025] When the control current is further increased, the first valve element 18 continues to be lifted, but in the second control valve 10B, which has been shifting in the valve-closing direction along with the lift of the first valve element 18, the motion of the piston 33 is stopped by the second valve element 32 being seated on the second valve seat 31. The first valve element 18 is disengaged from the piston 33.

[0026] After that, when the control current is held at a predetermined value, the first valve element 18 is stopped at a position where the urging force of the solenoid section and the force of the spring 20 are balanced. This position does not change until the value of the control current is changed. When the first valve element 18 is stopped, the first control valve 10A is set to a predetermined passage cross-sectional area with respect to the refrigerant passage. Refrigerant at discharge pressure Pdh, introduced into port 13, flows through the refrigerant passage having the predetermined passage cross-sectional area to port 14 (discharge pressure Pdl). A predetermined differential pressure (Pdh - Pdl = ΔP) is generated across the first control valve 10A. Since the discharge pressure Pdh2 at port 15 of the second control valve 10B is approximately equal to the discharge pressure Pdh at port 13 of the first control valve 10A, the differential pressure ΔP generated across the first control valve 10A can be sensed by the piston 33.

[0027] Up to this time point in time, the compressor has been operating toward its maximum displacement operation, so that the discharge pressure Pdh presently increases, and the flow rate through the first control valve 10A also increases.

[0028] When the flow rate becomes larger than a predetermined value, an urging force in valve-opening direction, caused by application of the differential pressure (Pdh2 - Pdl $\approx$ ΔP) to the piston 33, overcomes the spring 37. The second valve element 32 moves away from the second valve seat 31 to open the second control valve 10B. Discharged refrigerant is introduced into the crankcase and starts controlling the variable displacement of the compressor.

[0029] After that, when the flow rate through the first control valve 10A increases to increase the differential pressure ΔP across the first control valve 10A, the piston 33 senses the change in the differential pressure to further open the second control valve 10B. This controls the compressor in a direction of decreasing the displacement. When the flow rate through the first control valve 10A decreases to decrease the differential pressure ΔP across the first control valve 10A, the piston 33 senses the change in the differential pressure to shift the second control valve 10B in valve-closing direction. This controls the compressor in a direction of increasing the displacement.

[0030] At this time, although in the second control valve 10B, discharge pressure Pdh2 from the port 15, is about to leak into the port 14 via the clearance between the piston 33 and the second body 12, the leakage of refrigerant is blocked by the seal ring 41 deformed by the discharge pressure Pdh2. Further, although the discharge pressure Pdh from the port 13, leaks into the guide 19 via a portion where the guide 19 and the first body 11 are in sliding contact, and further into the port 14 via the check valve 21, this slight leakage is negligible since the leakage does not adversely affect the operations of the control valve 10 and the compressor.

[0031] The second control valve 10B controls the flow rate to the crankcase such that the differential pressure Δ P is held constant. The control valve 10 controls the compressor such that refrigerant is discharged at the flow rate corresponding to the value of the control current supplied to the solenoid section 10C.

[0032] When control current to the solenoid section 10C is suddenly switched off the solenoid force ceases, whereby the first control valve 10A is instantaneously fully closed by the spring 20. The first valve element 18 pulls the piston 33 downward, as viewed in Fig. 1, to fully open the second control valve 10B forcibly. The compressor shifts to the minimum displacement operation. Now, the discharge pressure Pdh quickly decreases, but the discharge pressure Pdl at the refrigerant compressor outlet port only progressively decreases since the first valve element 18 is in the fully-closed state. Therefore, sometimes the discharge pressure Pdl becomes higher than the discharge pressure Pdh immediately after switch-off of the control current. In this case, the first control valve 10A is held fully-closed since the discharge pressure Pdl acts on the first valve element 18 and the check valve 21 in valve closing directions, while the second control vale 10B is fully-open without being actuated by the discharge pressure Pdl since the piston 33 and the first valve element 18 have the same pressure-receiving areas. Leakage from the port 14 to the port 15 via the clearance between the second body 12 and the piston 33 is blocked by the seal ring 40.

[0033] Whenever the discharge pressure Pdl at the refrigerant outlet port side becomes equal to or higher than the discharge pressure Pdh, the first control valve 10A can be held in the fully-closed state, to act similarly to the check valve which conventionally has to be provided in the refrigerant outlet port. The second control valve 10B is fully-open, such that the compressor positively shifts to minimum displacement operation.

[0034] The control valve 70 in Fig. 3 (second embodiment) differs from the first embodiment in that the check valve 21 of Figs 1, 2 is omitted.

[0035] In a first control valve 70A the guide 19 connected to the first valve element 18 comprises a hollow cylindrical portion having a closed end, and an integral sliding portion, and a radially outwardly extending flange at an open end of the hollow cylindrical portion. An outer peripheral surface of the flange slides on an inner wall surface of the first body 11. The shaft 54 of a solenoid section 70C extends into the hollow cylindrical portion for abutment at the closed end. The guide 19 has an intercommunicating hole 19a formed in a side wall such that pressure in the solenoid section 70C always equals the discharge pressure Pdh upstream of the first control valve 70A.

[0036] When the solenoid section 70C is de-energized (Fig. 3), the first control valve 70A is fully closed by spring 20, and a second control valve 70B is fully open since the first valve element 18 forcibly pulls the piston 33 that senses a differential pressure across the first control valve 70A, in valve-opening direction. The discharge chamber and the crankcase communicate via the second control valve 70B. The compressor is in the minimum displacement operation state.

[0037] When control current is supplied to the solenoid section 70C, as the control current increases, the shaft 54 of the solenoid section 70C lifts the first valve element 18, whereby the first control valve 70A starts to open, while the second control valve 70B starts to shift in an interlocked manner in valve-closing direction. After the second control valve 70B is closed, the compressor is placed in the maximum displacement operation state, the first control valve 70A assumes a lift position corresponding to the value of the control current, and defines a passage cross-sectional area corresponding to the control current, without being interlocked with the second control valve 70B.

[0038] When the compressor shifts to the maximum displacement operation state, the flow rate through the first control valve 70A increases. When the differential pressure across the first control valve 70A becomes equal to or larger than a predetermined value, the piston 33 that senses the differential pressure acts to open the second control valve 70B to make the displacement of the compressor variable.

[0039] If the solenoid section 70C is de-energized when the control valve 70 is in the control state, the first control valve 70A is fully closed instantaneously by the spring 20, and the second control valve 70B is constrained and forcibly fully opened during transition of the first control valve 70A to the fully-closed state. This sharply decreases the discharge pressure Pdh in the discharge chamber while progressively decreasing the discharge pressure Pdl at the refrigerant outlet port, to thereby invert the relationship between the discharge pressure Pdh and the discharge pressure Pdl. However, the first valve element 18 and the integrated piston 33 have the same pressure-receiving area, and hence are insensitive to the discharge pressure Pdl, so that even if the discharge pressure Pdl exceeds the discharge pressure Pdh, both the fully-closed state of the first control valve 70A and the fully-open state of the second control valve 70B are maintained.

[0040] The control valve 80 in Fig. 4 (third embodiment) differs from the of the first and second embodiments by a simplified construction, namely by forming the two ports 13 and 15 into a common port.

[0041] A first control valve 80A has the first valve element 18 upstream of the first valve seat 17. The first valve element 18 has an axial through hole and a hollow cylindrical portion 81 extending via a valve hole is rigidly press-fitted into the through hole. The hollow cylindrical portion 81 is integral with a piston 82 axially slidably disposed within the first body 11. A hollow part of the hollow cylindrical portion 81 extends into the piston 82 such that the hollow part communicates with a side opposite to the side where the hollow cylindrical portion 81 is formed, and the hollow part extending through the piston 82 forms a refrigerant passage 83 through which the discharge pressure Pdh is introduced into the solenoid section 80C. The piston 82 is urged in valve-closing direction by the spring 20 between the piston 82 and an end face of the second body 12. The piston 82 has a large outer diameter on the side toward a second control valve 80B, such that the first control valve 80A defines a valve structure in which when the force of the spring 20 exceeds the force of the solenoid section 80C, the clearance between the first body 11 and the piston 82 is sealed.

[0042] The axial central port 16 in the second body 12 forms by an inner open end a second valve seat port. The movable second valve element 32, is integral with a hollow cylindrical body 84 axially slidably disposed within the second body 12. The hollow cylindrical body 84 has the same outer diameter as the piston 82 of the first control valve 80A, has a plurality of intercommunicating holes, and has the first valve seat 17 rigidly press-fitted into the inside. The spring 37 between the first valve seat 17 and the piston 82 urges the hollow cylindrical body 84 in valve-closing direction of the second control valve 80B.

[0043] When the solenoid section 80C is de-energized (Fig. 4), the first control valve 80A is fully closed by the spring 37 since the first valve seat 17 abuts at the first valve element 18, while in the second control valve 80B, the second valve element 32 is in a fully-open position. The discharge chamber and the crankcase communicate via the second control valve 80B. The compressor is in the minimum displacement operation state.

[0044] Then, when control current is supplied to the solenoid section 80C, as the control current increases, the shaft 54 of the solenoid section 80C pushes the piston 82 of the first control valve 80A upward. The movable parts of the first and second control valves 80A, 80B move in unison in valve-closing direction of the second control valve 80B. After the second valve element 32 is seated on the second valve seat the compressor shifts to the maximum displacement operation state. Then, when the piston 82 is pushed in valve-closing direction of the second control valve 80B, the first valve element 18 is progressively lifted from the first valve seat 17 to progressively open the first control valve 80A. Subsequently, the first valve element 18 stops at a lift position corresponding to the value of the control current. The first control valve 80A defines a passage cross-sectional area corresponding to the control current.

[0045] When the compressor shifts to the maximum displacement operation state, the flow rate through the first control valve 80A increases and generates a differential pressure which is received by the cross-sectional areas of the first valve seat 17 and the hollow cylindrical body 84 forming a differential pressure-sensing section. When the differential pressure becomes equal to or larger than a predetermined value, the first valve seat 17 and the hollow cylindrical body 84, open the second control valve 80B to make the displacement of the compressor variable.

[0046] Now, when the discharge pressure Pdh increases while the compressor is being controlled at a predetermined displacement, the second control valve 80 operates in valve-opening direction to control the capacity in a decreasing direction. When the discharge pressure Pdh decreases, the second control valve 80B operates in valve-closing direction to control the capacity in increasing direction. At this time, although the first control valve 80A as well responds in accordance with the change of the discharge pressure Pdh, the displacement of the compressor is determined in dependence on the control balance between the first and second control valves 80A, 80B, an d is substantially set to a predetermined displacement.

[0047] If the solenoid section 80C is suddenly de-energized while the control valve 80 is in the control state, the first control valve 80A is fully closed instantaneously by the spring 20, and the second control valve 80B is constrained and forcibly fully opened during transition of the first control valve 80A to the fully-closed state. This sharply decreases the discharge pressure Pdh while progressively decreasing the discharge pressure Pdl in the refrigerant outlet port, so that the discharge pressure Pdl presently becomes higher than the discharge pressure Pdh. However, the pressure-receiving area at which the first valve set 17, the first valve element 18, and the hollow cylindrical body 84, which are integrally engaged with each other, receive the discharge pressure Pdl in upward direction is the same as the pressure-receiving area at which the piston 82 receives the discharge pressure Pdl in downward direction so that the control valve 80 has a structure which is operatively insensitive to the discharge pressure Pdl. Accordingly, even if the discharge pressure Pdl exceeds the discharge pressure Pdh, the fully-closed state of the first control valve 80A and the fully-open state of the second control valve 80B are maintained.

[0048] The control valve 90 in Fig. 5 (fourth embodiment) differs from the third embodiment by a higher response speed when returning again to the control state after suddenly transitioning from the control state to the stopped state.

[0049] The valve element 18 and the integrated hollow cylindrical portion 81 define the linearly axial refrigerant passage 83. The hollow cylindrical portion 81 fixed to the piston 82 extends through the piston 82 such that an open end of the refrigerant passage 83 on the solenoid section can be opened and closed by an end face of the shaft 54. The control valve 90 thus has a valve structure in which when the solenoid section 90C is de-energized (Fig. 5), the port 13 (discharge pressure Pdh) and the solenoid section 90C communicate with each other, whereas when the solenoid section 90C is energized, the communication between the port 13 and the inside of the solenoid section 90C is blocked.

[0050] The basic operation of the control valve 90 of Fig. 5 is the same as that of the third embodiment. However, when the solenoid section 90C is energized immediately after the control valve 90 has suddenly shifted or transitioned from the control state to the stopped state, to more rapidly return again to the control state, the shaft 54 will lift the first valve element 18 while the shaft end face closes the open end of the hollow cylindrical portion 81. When the first valve element 18 is lifted even by a slight degree, the piston 82 as well is lifted accordingly, to open the clearance between the piston 82 and the first body 11. Refrigerant at still high discharge pressure Pdl enters a space between the piston 82 and the solenoid section 90C. This causes an urging force on the piston actuating a first control valve 90A in valve-opening direction 2, and helps the solenoid section 90C to open the first control valve 90A. This shortens the time period required for fully opening the second control valve 90B. The control valve 90 returns to its original control state sooner.

In the control valve 100 of Fig. 6 (fifth embodiment) the structure of the embodiment is combined with the first embodiment. The ports 13 and 15 are formed independently and further the fourth embodiment is improved in that the pressure Pc acts on the second control valve 90B in valve-opening direction.

[0051] In the control valve 100 of Fig. 6, the second valve seat 31 at the port 15 for the discharge pressure Pdh2 is formed on the foremost end of the second body 12. The second valve element 32 is axially movably guided in the second body 12. A lower end of the second valve element 32 extends into a chamber of the port 13 (discharge pressure Pdh). An engaging portion 101 held by a closing portion of the hollow cylindrical body 84 of the first control valve 100A is rigidly press-fitted on the lower end of the second valve element 32. The spring 35 between the engaging portion 101 and the first valve element 18 urges the second valve element 32 in valve-closing direction. An upper end face of the engaging portion 101 5 is tapered to close the clearance between the second valve element 32 and the second body 12 when the second valve element 32 is seated on the second valve seat 31.

[0052] A second control valve 100B is configured such that both discharge pressures Pdh, Pdh2, which are approximately equal, are applied to axially opposite ends of the second valve element 32, respectively. Thus the control valve 100 performs control operation in response to a differential pressure between the pressures as applied to the first valve seat 17 and the hollow cylindrical body 84 from axially opposite sides, without being adversely affected by the pressure Pc.

[0053] In the control valve 110 of Fig. 7 (sixth embodiment) the first control valve 110A and the solenoid section 110C are very similar to the first embodiment but the second control valve 110B has a different construction.

[0054] In the second control valve 110B the differential pressure-sensing section is formed by a valve element-holding portion 111 that holds the second valve element 32 and a bellows 112. Axially opposite ends of the bellows 112 are tightly connected to an upper end of the valve element-holding portion 111 and to an upper end of the spring-receiving portion 36 such that the bellows 112 can expand and contract axially. In this differential pressure-sensing section, similarly to the piston 33 of the second control valve 10B in the first embodiment, the valve element-holding portion 111 causes the second valve element 32 to axially move according to a differential pressure between the discharge pressures Pdh, 2 Pdl, whereby the valve lift of the second control valve 1108 can be adjusted. Since the bellows 112 defines a partition between the ports 15, 14 it is possible to completely prevent leakage by the differential pressure between the pressures Pdh2 and Pdl, without the seal rings 40 and 41.

[0055] The control valve 120 of Fig. 8 (seventh embodiment) differs from the second embodiment by a simpler construction. In a first control valve 120A both ends of the first valve element 18 and the guide 19 slidingly contact the inner walls of the first body 11 and the second body 12. The guide 19 has a communication hole 121 in a top portion close to the location where the shaft 54 of the solenoid section 120C abuts.

[0056] A second control valve 120B includes the piston 33 that senses the differential pressure between the discharge pressure Pdh2 at the port 15 and the discharge pressure Pdl at the port 14. A shaft 122 is fixed to the piston 33. One end of the shaft 122 forms the second valve element 32 of the second control valve 120B, while the other end forms an engaging portion for the first valve element 18 when when the solenoid section 120C is de-energized to cooperate with the first valve element 18 to form a valve element of a valve that opens and closes the axial refrigerant passage of the first valve element 18 and the guide 19. The spring 37 urging the piston 33 in valve-closing direction of the second control valve 120B is disposed between the seal ring 40 and the first valve element 18 of the first control valve 120A. Although the spring 37 should urge the piston 33 relative to the second body 12, the spring is arranged to urge the piston 33 relative to the first valve element 18 so as to reduce a needed spring load and to simplify the construction.

[0057] When the solenoid section 120C is de-energized, the spring 20 pushes the first valve element 18 downward to close the first control valve 120A. The first valve element 18 pulls the shaft 122 downward to fully open the second control valve 120B. Since the first valve element 18 and the engaging portion of the shaft 122 are tightly engaged with each other, the refrigerant passage through the first valve element 18 and the guide 19 is closed. Discharge pressure Pdh leaks through where the guide 19 and the first body 11 are in sliding contact, so that pressure in the solenoid section 120C becomes close to the discharge pressure Pdh.

[0058] When the solenoid section 120C is energized, the shaft 54 pushes the first valve element 18 via the guide 19 upward to open the first control valve 120A. The piston 33 as well is pushed upward by the spring 37 interlocked with the pushing action for the first valve element 18. When the second valve element 32 is seated, the second control valve 120B is fully closed. When then the first valve element 18 is further pushed upward, the shaft 122 first tightly engaged with the first valve element 18 looses contact with the first valve element 18, so that the refrigerant passage through the first valve element 18 and the guide 19 is opened. The solenoid section

120C communicates with the port 14 to equalise the pressures.

[0059] When supply of the control current to the solenoid section 120C is suddenly stopped from the control state in which the first valve element 18 is lifted to a predetermined value by the control current supplied to the solenoid section 120C, the first control valve 120A is fully closed, and the second control valve 120B is fully opened. The refrigerant passage through the first valve element 18 and the guide 19 is closed. This causes the compressor to shift to the minimum displacement operation state, whereby the discharge pressures Pdh and Pdh2 on the discharge chamber side are sharply decreased, and the discharge pressure Pdl in the solenoid section 120C leaks into the port 13 through where the guide 19 and the first body 11 are in sliding contact, to make the discharge pressure Pdl close to the sharply decreased discharge pressures. The discharge pressures Pdh and Pdh2, which have been sharply decreased to become approximately equal to each other, are applied to the axially opposite ends of the movable part of the guide 19, the first valve element 18, and the piston 33, which are made integral with each other, and therefore the fully-closed state of the first control valve 120A and the fully-open state of the second control valve 120B are maintained almost only by the load of the spring 20.

[0060] The control valve 130 of Fig. 9 (eighth embodiment) is constructed by using lower-cost component parts in place of the high-cost bellows 112 used in the the sixth embodiment, and in place of high-cost cut parts used in the seventh embodiment.

[0061] Almost all component elements of a first control valve 130A and a second control valve 130B are pressed parts formed by pressing pipes, and the pressed parts are assembled by press-fitting or swaging.

[0062] In the first control valve 130A, a first body 131 having an inwardly bent end is fixed to a solenoid section 130C by swaging the foremost end of the core 51 protruding from the yoke 59. A second body 132 and a third body 133 having one end forming the first valve seat 17 are rigidly press-fitted into the first body 131. The third body 133 axially guides the here bell-shaped guide 19. A bell-shaped shaft-receiving portion 134 having the communication hole 121 is press-fitted into the guide 19. The guide 19 carries the fixed first valve element and is urged in valve-closing direction by the spring 20 between a flange end and a protrusion formed on the inside of the body 133.

[0063] The second body 132 has an open end containing a diaphragm 135, e.g. made of polyimide, sealing between the port 15 (discharge pressure Pdh2) and the port 14 (discharge pressure Pdl). The diaphragm 135 senses the differential pressure between the discharge pressures Pdh2 and Pdl, and has an outer peripheral portion sandwiched between first and second rings 136, 137 A central diaphragm portion is sandwiched between a centre disk 138 and a flange portion 139. The first and second rings 136, 137 are fixed to the second body 132 together with a fourth body 140 by swaging open ends of the second body 132, sandwiching the diaphragm 135 therebetween. The centre disk 138 and the flange portion 139 are fixed to each other sandwiching the diaphragm 135 therebetween, by press-fitting a shaft 141 into a central portion of the centre disk 138 and the hollow cylindrical second valve element 32 of the second control valve 130B, integrally formed with the flange portion 139. It should be noted that portions of the first and second rings 136 and 137, sandwiching the outer peripheral portion and the central portion of the diaphragm 135, are configured such that the inner diameter of the first ring 136 is larger than the inner diameter of the second ring 137, and the outer diameter of the centre disk 138 is larger than the outer diameter of the flange portion 139. The first ring 136 has a stepped portion, and a portion thereof forms a stopper 142 restricting the displacement of the diaphragm 135.

[0064] A cup-shaped fifth body 143 is press-fitted into the fourth body 140. The fifth body 143 has a valve hole of the second control valve 130B formed in the centre of the bottom. An opening of the valve hole forms the port 16 leading to the crankcase. The shaft 141 extends through the valve hole of the second control valve 130B, and a spring-receiving portion 144 is externally fitted on a foremost end of the shaft 141. Interposed between the bottom of the fifth body 143 and the spring-receiving portion 144 is the spring 37 which urges the second valve element 32 in valve-closing direction.

[0065] When the solenoid section 130C is de-energized, the spring 20 pushes the guide 19 downward to close the first control valve 130A, and at the same time the guide 19 pulls the shaft 141 downward until the centre disk 138 is brought into abutment with the stopper 142, to fully open the second control valve 130B. Since then the guide 19 and the shaft 141 are tightly engaged with each other, and discharge pressure Pdh leaks into the first body 131 through where the guide 19 and the third body 133 are in sliding contact, so that pressure in the solenoid section 130C is close to the discharge pressure Pdh.

[0066] When the solenoid section 130C is energized, the shaft 54 pushes the first valve element 18 via the guide 19 upward to open the first control valve 130A. The shaft 141 as well is pulled upward by the spring 37, interlocked with the upward pushing action of the first valve element 18. As soon as the second valve element 32 is seated, the second control valve 130B is fully closed. When then the first valve element 18 is further pushed upward, the shaft 141 first tightly engaged with the guide 19 loses the sealing engagement with the guide 19, so that the solenoid section 130C communicates via a hole in the centre of the guide 19, through which the shaft 141, and the communication hole 121 of the shaft-receiving portion 134 with the port 14. Pressure in the solenoid section 130C becomes equal to the discharge pressure Pdl.

[0067] After that, when control current is held at the predetermined value, the first valve element 18 stops at a position where the urging force of the solenoid section 130C corresponding to the predetermined current value and the spring 20 are balanced. The first valve element 18 is lifted from the first valve seat 17 and is stopped. The refrigerant passage of the first control valve 130A has a predetermined passage cross-sectional area, so that the discharge pressure Pdh, from the port 13, passes through the refrigerant passage the discharge pressure Pdl is discharged from the port 14 into the compressor refrigerant outlet port. Then a predetermined differential pressure ΔP is generated across the first control valve 130A. Since the discharge pressure Pdh2 approximately equals the discharge pressure Pdh, the differential pressure ΔP is sensed by the diaphragm 135, which drives the second valve element 32 which in turn controls the flow rate via the second control valve 130B to the crankcase. The compressor discharge flow rate corresponds to the value of the control current.

[0068] When the control current is suddenly switched off from the control state in which the first valve element 18 is lifted to a predetermined degree by the control current, the first control valve 130A is fully closed, and the second control valve 130B is fully opened. The hole in the guide 19 is blocked. The compressor shifts to the minimum displacement operation state, whereby the discharge pressures Pdh and Pdh2 on the discharge chamber side are sharply decreased. The discharge pressure Pdl in the solenoid section 130C leaks into the port 13 through where the guide 19 and the first body 11 are in sliding contact, to make the discharge pressure Pdl close to the discharge sharply decreased pressures Pdh. The progressively decreased discharge pressure Pdl is applied to the guide 19 from the port 14 side whereas the sharply decreased discharge pressures Pdh is applied to the inside of the guide 19. Hence the differential pressure and the spring 20 maintain the first control valve 130A closed and the second control valve 130B fully open.

[0069] With regard to the differential pressure-sensing section, it is known that the effective pressure-receiving area of the diaphragm 135 varies according to the displacement stroke of the diaphragm. As shown in Fig. 10A, the effective pressure-receiving area of the diaphragm 135 depends on the area of a circle a diameter (effective diameter b) of which is the distance between the centres of curvature circles a of respective corrugated portions. Here, when the pressure P1 applied from above becomes higher than the pressure P2 applied from below a central portion of the diaphragm 135 is displaced downward (Fig. 10B). Since an inner peripheral portion of each corrugated portion is also displaced together with the central portion, the curvature of the corrugated portion is increased, and the centre of the curvature moves inward, whereby the effective diameter becomes an effective diameter b1 smaller than the effective diameter b. This action decreases the effective pressure-receiving area. Here, (Fig. 10C), when the pressure P2 exceeds the pressure P1 in a state in which the central portion of the diaphragm 135 is displaced, the corrugated portion alone expands to swell toward the side of the pressure P1, which causes the centre of the curvature to move outward such that the effective diameter becomes an effective diameter b2 larger than the effective diameter b. This action increases the effective pressure-receiving area.

[0070] This situation corresponds to the case where the control current is suddenly switched off to make the discharge pressure Pdl downstream of the first control valve 130A higher than the discharge pressure Pdh2 upstream. The second valve element 32 moving in unison with the diaphragm 135 is caused by the differential pressure between the discharge pressures Pdl and Pdh2 to act in the valve-closing direction. More specifically, when the solenoid section 130C is de-energized, the second control valve 130B is fully opened by the spring 20, but immediately after that, when the discharge pressure Pdl exceeds the discharge pressure Pdh2, the diaphragm 135 which is responsive to the differential pressure acts on the second control valve 130B in valve-closing direction. Therefore, particularly when the value of the differential pressure is large, it would be impossible to maintain the fully-closed state of the first control valve 130A and the fully-open state of the second control valve 130B.

[0071] In contrast, the control valve 130 is configured such that when the discharge pressure Pdl exceeds the discharge pressure Pdh2, the force that acts on the second control valve 130B in valve-closing direction is inhibited from increasing. To this end, it is only required that an effective diameter c2 of the diaphragm 135 obtained when the discharge pressure Pdl is higher than the discharge pressure Pdh2 is made smaller than an effective diameter c1 of the diaphragm 135 obtained when the discharge pressure Pdh2 is higher than the discharge pressure Pdl. This is realized, as shown in FIGS. 11A and 11B, by making the respective inner diameters of the first and second rings 136 and 137 different, and by making the outer diameters of the centre disk 138 and the flange portion 139 different. More specifically, the inner diameter of the stepped portion of the first ring 136 is larger than the inner diameter of the second ring 137, and the outer diameter of the centre disk 138 is larger than that of the flange portion 139. However, the effective diameter c1 is equal to the inner diameter of the first valve seat of the first control valve 130A such that when the first control valve 130A is closed, the diaphragm 135 has the same pressure-receiving area as the first valve element 18 for the discharge pressure Pdl. It should be noted that the distance between the inner periphery of the first ring 136 and the outer periphery of the centre disk 138 is equal to the distance between the inner periphery of the second ring 137 and the outer periphery of the flange portion 139.

[0072] As a result, when Pdl < Pdh2 holds, as shown in FIG. 11A, the corrugated portion of the diaphragm 135 is defined by the centre disk 138 having a larger outer diameter and the first ring 136 having a larger inner diameter, and the effective pressure-receiving area of the diaphragm 135 at this time is determined by the effective diameter c1. On the other hand, when Pdl > Pdh2 holds, as shown in FIG. 11B. the corrugated portion of the diaphragm 135 is defined by the flange portion 139 having a smaller outer diameter and the second ring 137 having a smaller inner diameter, and the effective pressure-receiving area of the diaphragm 135 at this time is determined by the effective diameter c2. As described above, when de-energization of the solenoid section 130C causes transition from the state in which the discharge pressure Pdl is lower than the discharge pressure Pdh2 to the state in which the discharge pressure Pdl is higher than the discharge pressure Pdh2, the effective pressure-receiving area of the diaphragm 135 is changed to be smaller. This reduces the force acting by the differential pressure on the second control valve 130B in valve-closing direction. A stopping operation of the automotive air conditioner may then be performed smoothly.

Claims

1. A control valve for a variable displacement compressor, including a first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A) that controls a first refrigerant flow rate between a compressor discharge chamber and a refrigerant outlet port, a second control valve (10B, 70B, 80B, 90B, 100B, 110B, 1208, 130B) that controls a second refrigerant flow rate between the discharge chamber and a compressor crankcase based on a differential pressure across said first control valve, to change the compressor displacement, and to thereby control the first flow rate to be constant, and a solenoid section (10C, 70C, 80C, 90C, 100C, 110C, 120C, 130C) for setting the first flow rate,
characterised in that the control valve is designed to be insensitive to pressure on a downstream side of the first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A), that with the solenoid section (10C, 70C, 80C, 90C, 100C, 110C, 120C, 130C) in a non-energized state the first control valve is held in a fully closed state, and the second control valve (10B, 70B, 80B, 90B, 100B, 110B, 120B, 130B) is held in a fully open state, and that the second control valve (10B, 70B, 80B, 90B, 100B, 110B, 120B, 130B) is forcibly held in the fully open state even when the pressure on the downstream side of the first control valve is equal to or higher than pressure on an upstream side of the first control valve.

2. Control valve according to claim 1, **characterised in that** the first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A) has a first valve element (18) the lift position thereof in relation to a first valve seat (17) is set according to an urging force generated by the solenoid section (10C, 70C, 80C, 90C, 100C, 110C, 120C, 130C), while the first valve element (18) is urged in valve-closing direction against the urging force of the solenoid section (10C, 70C, 80C, 90C, 100C, 110C, 120C, 130C);
at the second control valve (10B, 70B, 80B, 90B, 100B, 110B, 120B, 130B) has a differential pressure-sensing section with the same pressure-receiving area as a pressure-receiving area of a part of the first valve element (18), which section receives the pressure on the downstream side of the first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A) when the first valve element (18) is in a closed position on the first valve seat (17), for sensing a differential pressure between the pressures on the upstream and downstream sides of the first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A), and a second valve element (32) that is held by said differential pressure-sensing section; and that
upon transition of said solenoid section (10C, 70C, 80C, 90C, 100C, 110C, 120C, 130C) to a de-energised state from a control state in which said solenoid section sets said first control valve (10A, 70A, 80A, 90A, 100A, 110A, 120A, 130A), the first valve element (18) being shifted to a fully closed position by the urging force which engages with the differential pressure-sensing section which has been away from said first valve element (18) during this control state, to shift the second valve element (32) to a fully open position.

3. Control valve according to claim 2, **characterised in that** the second control valve (10B, 70B, 110B, 130B), the first control valve (10A, 70A, 110A, 130A), and said solenoid section (10C, 70C, 110C, 130C) follow one another along the same axis,
that the first control valve (10A, 70A, 110A, 130A) has a first solenoid section side port (13) for introducing refrigerant from the discharge chamber, a second port (14) on a side toward said second control valve (10B, 70B, 110B, 130B), for discharging refrigerant into the refrigerant outlet port, the first valve seat (17) between said first and second ports (13, 14) with the first valve element (18) movably disposed on a downstream side of the first valve seat (17), and that
the second control valve (10B, 70B, 110B, 130B) for introducing refrigerant from the discharge chamber has a third port (15) separated from said second port (14) by said differential pressure-sensing section, a fourth axial port (16) opposite from said first control valve (10A, 70A, 110A, 130A) for discharging refrigerant into the crankcase, the differential pressure-sensing section being urged in valve-closing direction, a second valve seat (31) in the fourth port (16), the second valve element (32) being disposed upstream of the second valve seat (31) and being mov-

ably held by the differential pressure-sensing section in a manner movable relative to the second valve seat (31).

4. Control valve according to claim 3, **characterised in that** said first control valve (10A, 70A) has an axially slidable guide (19) within a space of the first valve element (18) communicating with the first port (13) and connected to the first valve element (18) via a valve hole, for guiding axial motions of the first valve element (18), and a spring (20) between the guide (19) and the first valve seat (17), for urging the first valve element (18) in valve-closing direction.

5. Control valve according to claim 4, **characterised in that** the guide (19) has a same pressure-receiving area as the pressure-receiving area of the part of the first valve element (18), which receives the pressure upstream of the first control valve (10A, 70A) when the first valve element (18) is in the closed position, and receives pressure upstream of the first control valve (10A, 70A) in valve-closing direction.

6. Control valve according to claim 5, **characterised in that** the first valve element (18) is slidable along the axis of the first control valve (10A, 70A) within a space communicating with the second port (14).

7. Control valve according to claim 5, **characterised in that** the first valve element (18) and the guide (19) commonly define an axial refrigerant passage and contain a check valve (21) for closing the refrigerant passage when the pressure in the refrigerant passage on a second port side has become higher than the pressure on a solenoid section side.

8. Control valve according to claim 6, **characterised in that** the first valve element (18) and the guide (19) commonly define an axial refrigerant passage, and that a valve is provided that engages at the differential pressure-sensing section to close the refrigerant passage when said first valve element (18) forcibly shifts the second valve element (32) fully open.

9. Control valve according to claim 4, **characterised in that** the guide (19) is provided with an intercommunicating hole (19a) for equalising the pressure in the solenoid section (70C) and the upstream pressure of the first control valve (70A).

10. Control valve according to claim 3, **characterised in that** the second control valve (10B, 70B) has a piston (33) as the differential pressure-sensing section for receiving pressure from the second port (18) and pressure from the third port (15) at axially opposite ends to operate according to the differential pressure between these pressures, and that a spring (37) urges the piston (33) in valve closing direction.

11. Control valve according to claim 10, **characterised in that** the spring (37) is disposed between the piston (38) and the first valve element (18).

12. Control valve according to claim 10, **characterised in that** the second control valve (10B, 70B) has a film-like seal ring (40, 41) on at least one of open ends of a clearance formed between the piston (33) and a body (12) that axially movably holds the piston (33), which clearance opens toward said second and third ports (14, 15), for sealing the clearance by the pressure from the second or third ports (14, 15).

13. Control valve according to claim 12, **characterised in that** the second control valve (10B, 110B, 70B) has a valve element base portion-accommodating portion formed in an end face of said piston (33), opposed to said second valve seat (31), and that a base portion of the second valve element (32) is captured in the valve element base portion-accommodating portion in a state urged in valve-closing direction.

14. Control valve according to claim 3, **characterised in that** the second control valve (10B) has a bellows (112) with axially opposite ends tightly connected to the differential pressure-sensing section holding the second valve element (32), and a body (36) that axially movably accommodates the differential pressure-sensing section, and that the bellows (112) is capable of axially extending and contracting while sealing the third port (13) from said second port (14).

15. Control valve according to claim 3, **characterised in that** the second control valve (130B) has a diaphragm (135) as the differential pressure-sensing section, that the diaphragm (135) is disposed between the second and third ports (14, 15) for receiving pressures from the second and third ports at axially opposite surfaces to cause the second valve element (32) to operate by the differential pressure between these pressures.

16. Control valve according to claim 15, **characterised in that** the diaphragm (135) is tightly connected to a body in a state in which an outer diaphragm periphery is sandwiched between first and second rings (136, 137), a central portion being sandwiched between a centre disk (138) and a flange portion (139) of the hollow cylindrical second valve element (31), the diaphragm (135) being fixed to a shaft (141) axially extending through the diaphragm (135), the centre disk (18) and the flange portion (139).

17. Control valve according to claim 16, **characterised in that** the diaphragm (135) is configured such that its pressure-receiving area for receiving pressure when the pressure in the third port (15) is higher than

the pressure in the second port (14) equals a pressure-receiving area of the first valve element (18) for receiving the pressure from the second port (14) when the first control valve (130A) is closed.

18. Control valve according to claim 17, **characterised in that** the diaphragm (135) is configured such that the inner diameter of the second ring (137) is smaller than the inner diameter of the first ring (136), that the an outer diameter of the flange portion (139) is smaller than the outer diameter of the centre disk (138), and that the pressure-receiving area of the diaphragm (115) for receiving pressure when the pressure in the second port (14) is higher than the pressure in the third port (15) is smaller than the pressure-receiving area for receiving pressure when the pressure from the third port (15) is higher than the pressure from the second port (14).

19. Control valve according to claim 2, **characterised in that** the second control valve (80B, 90B), the first control valve (80A, 90A), and the solenoid section (80C, 90C) follow one another along a common axis, and the the first control valve (80A, 90A) has a first port (14) on a solenoid section side for discharging refrigerant into the refrigerant outlet port, that a second port (13) formed on a second control valve side for introducing refrigerant from the discharge chamber, that a first valve seat (17) is disposed between the first and second ports, that the first valve element (18) is axially movably disposed upstream of the first valve seat (17), and that the second control valve (80B, 90B) has a third axial port (16) opposite from the first control valve (80A, 90A), for discharging refrigerant introduced into said second port into the crankcase, that a hollow cylindrical body (84) has the first valve seat (17) fixed inside and is axially movably disposed in a state urged in valve-closing direction and defines the differential pressure-sensing section, and that a second valve element (32) for opening and closing the third port (16) is integrally formed with said hollow cylindrical body (84).

20. Control valve according to claim 19, **characterised in that** the first control valve (80A, 90A) includes a piston (82) slidably disposed along an axis of the first valve element (18) within a space communicating with the first port (14), and connected to the first valve element (18) via a valve hole, for guiding axial motions of the first valve element (18), and that a spring (20) is disposed between the piston (82) and a body accommodating the piston, for urging said first valve element (18) in valve-closing direction.

21. Control valve according to claim 20, **characterised in that** the piston (82) has the same outer diameter as the hollow cylindrical body (84), to thereby inhibit the piston (82) from sensing the downstream side pressure of the first control valve (80A, 90A) when the first valve element (18) has closed.

22. Control valve according to claim 20, **characterised in that** the first valve element (18) and the piston (82) commonly define an axial refrigerant passage for causing pressure from the second port (13) to be received by a piston end face facing the solenoid section (80C, 90C).

23. Control valve according to claim 22, **characterised in that** when a shaft (54) of the solenoid section (90C) urges the first valve element (18), the refrigerant passage is closed by said shaft (54).

24. Control valve according to claim 2, **characterised in that** the second control valve (10B), the first control valve (100A), and the solenoid section (100C) are arranged along the same axis, that the first control valve (100A) has a first port (14) on a solenoid section side for discharging refrigerant into the refrigerant outlet port, a second port (13) on a second control valve side for introducing refrigerant from the discharge chamber, a first valve seat (17) between the first and second ports, the first valve element (18) being arranged upstream of the first valve seat, and that the second control valve (100B) has a third port (16) for discharging refrigerant into the crankcase, a fourth axial port (15) on a side opposite from the first control valve (100A) for introducing refrigerant from the discharge chamber, a hollow cylindrical body (84) disposed within a space communicating with the second port (13) and having the first valve seat (17) rigidly fixed in a state urged in valve-closing direction and forming the differential pressure-sensing section, a second valve seat (31) in the fourth port (15), and a second movable valve element (32) that is disposed such that one end thereof is opposed to the second valve seat (31), and the other end thereof is urged in valve closing direction with respect to said first valve element (18), for thereby being engaged with said hollow cylindrical body (84).

25. Control valve according to claim 24, **characterised in that** the first control valve (100A) includes an axially movable piston (82) within a space communicating with the first port (14), and connected to the first valve element (18) via a valve hole, for guiding axial motions of the first valve element (18), that a spring (20) is disposed between the piston (82) and a body (12) accommodating the piston (82), for urging the first valve element (18) in valve-closing direction, and that said first valve element (18) has an axial refrigerant passage (83) which is closed by a shaft (54) of the solenoid section (100C) when the shaft (54) actuates the first valve element (18).

26. Control valve according to claim 25, **characterised**

in that the piston (82) is configured such that a portion thereof receiving the pressure from the first port (14) in valve-closing direction has the same outer diameter as the hollow cylindrical body (84) to thereby inhibit the piston (82) from sensing the downstream pressure of the first control valve (100A) when the first valve element (18) has closed.

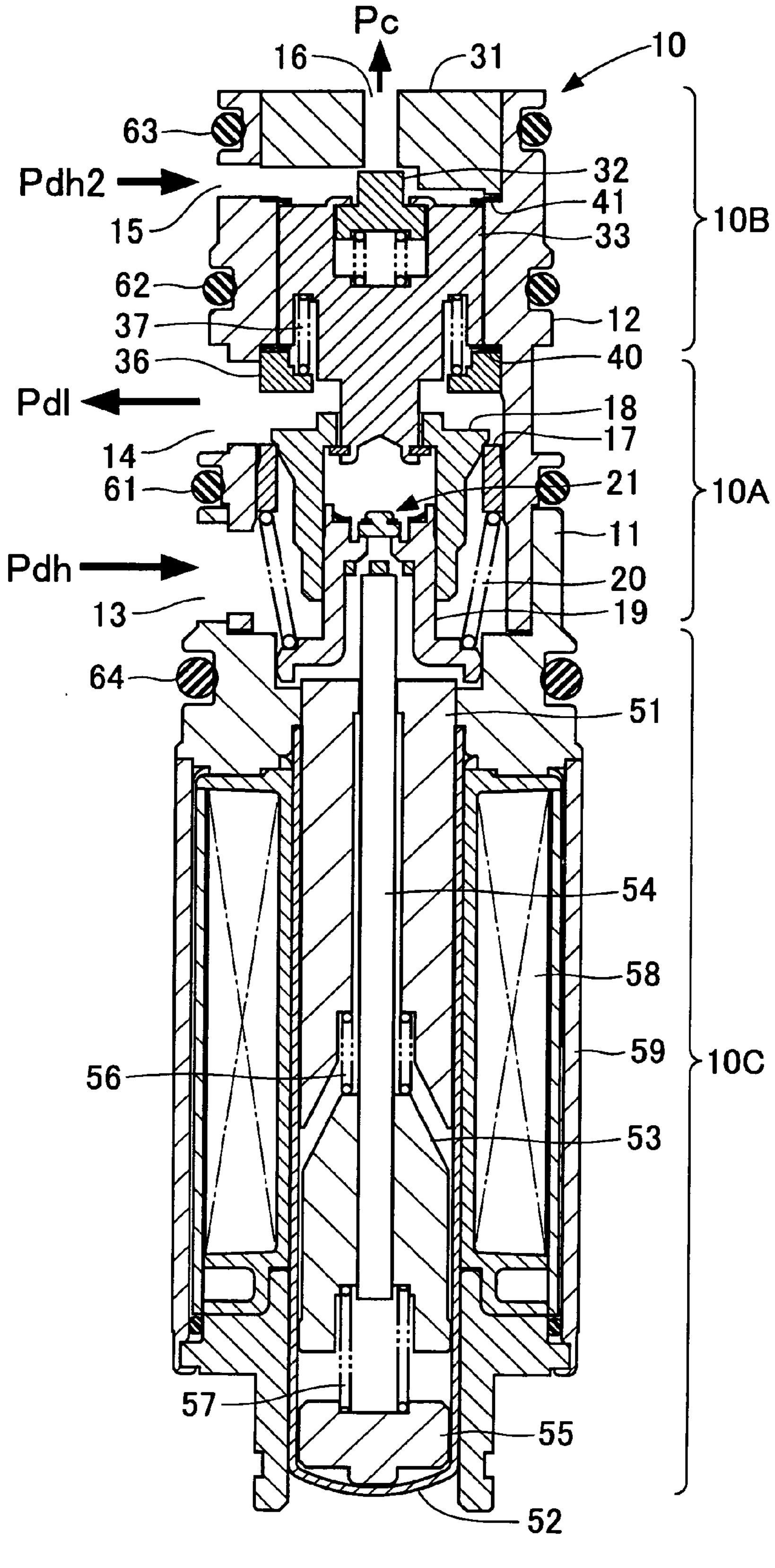
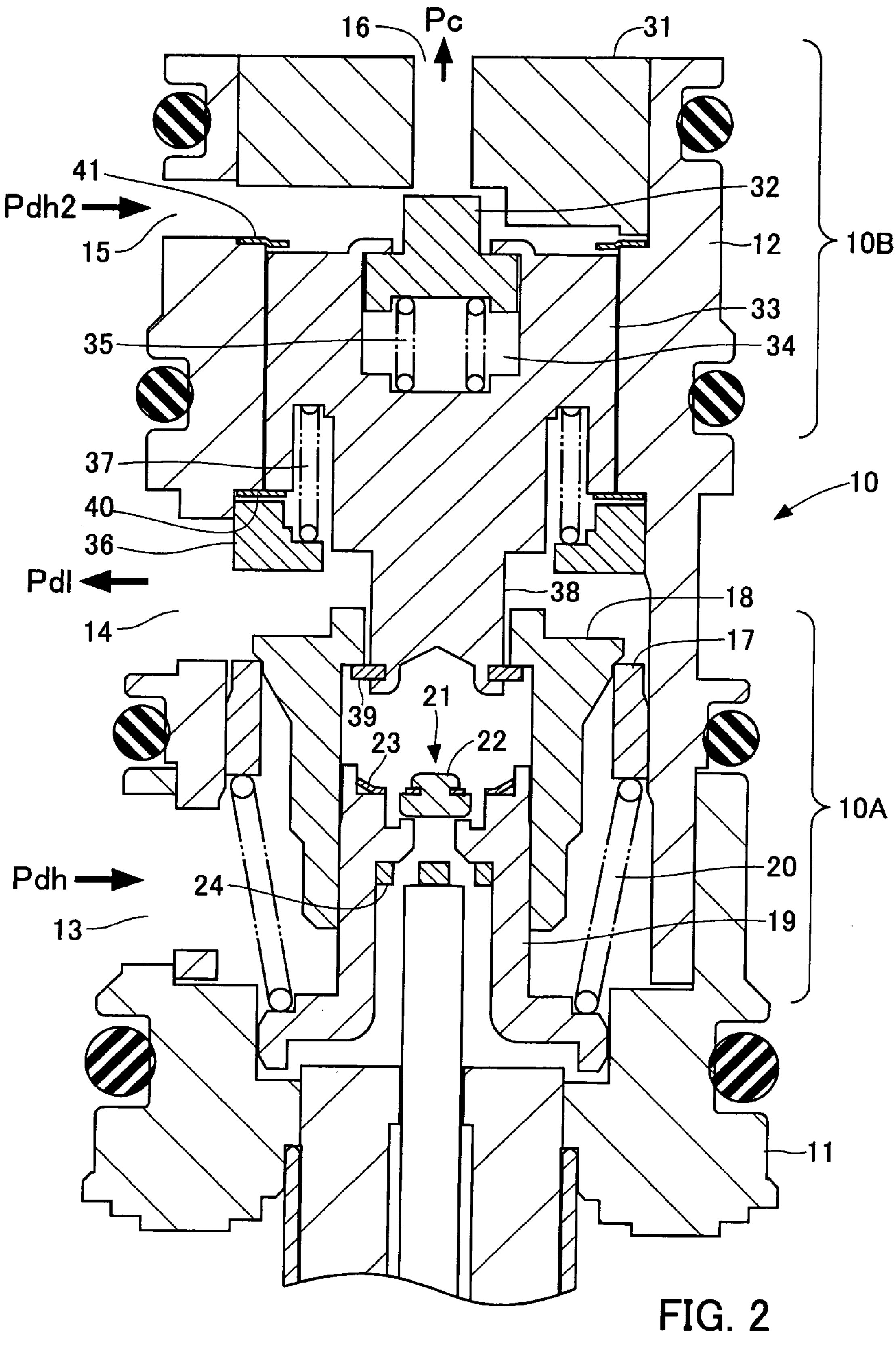

Pc
16
31
10
63
Pdh2
32
41
15
33
10B
62
37
12
36
40
Pdl
18
14
17
61
21
10A
11
Pdh
20
13
19
64
51
54
58
56
59
10C
53
57
55
52

FIG. 1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Pdh2
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15
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10B
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35
34
37
10
40
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Pdl
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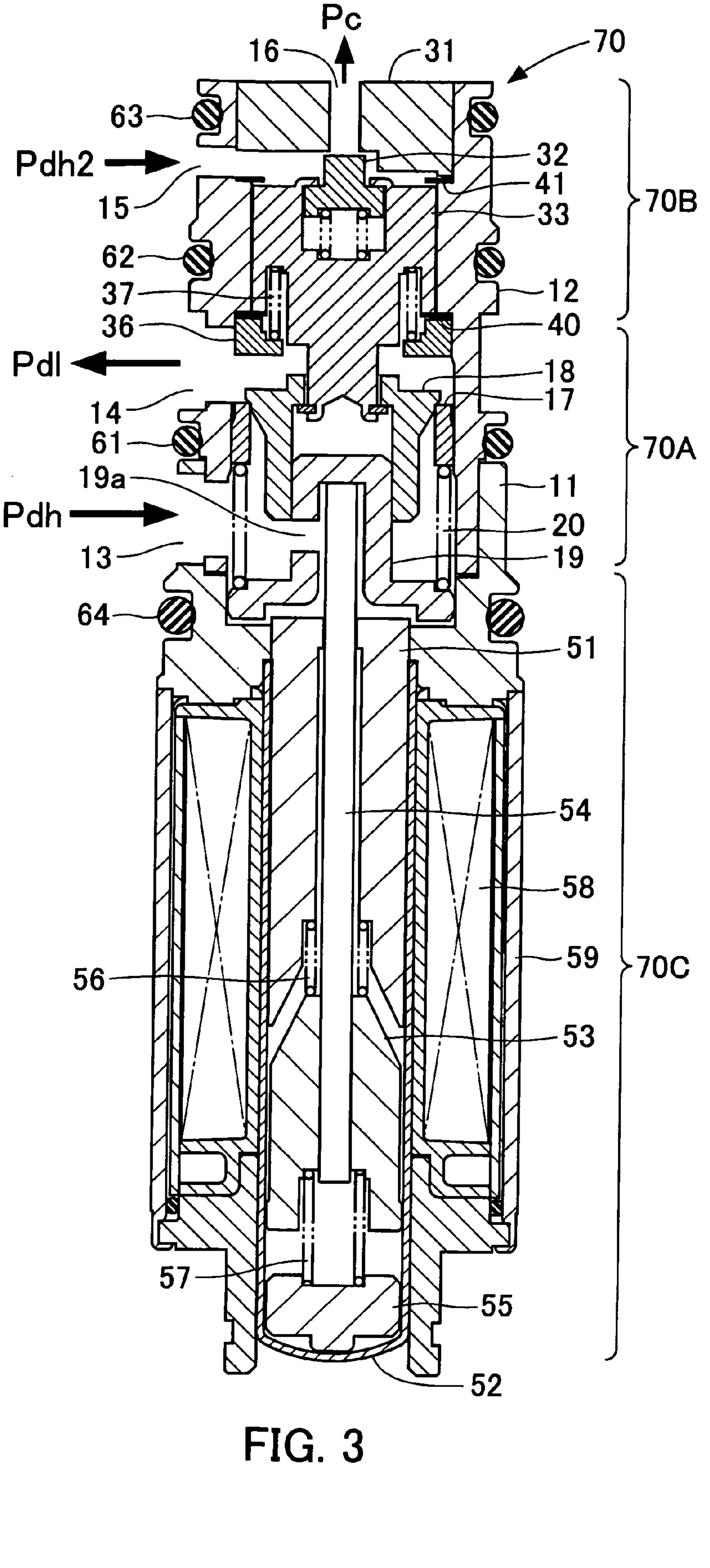
18
14
17
39
21
23
22
10A
Pdh
24
20
13
19
11

FIG. 2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63
Pdh2
32
41
15
33
70B
62
37
12
36
40
Pdl
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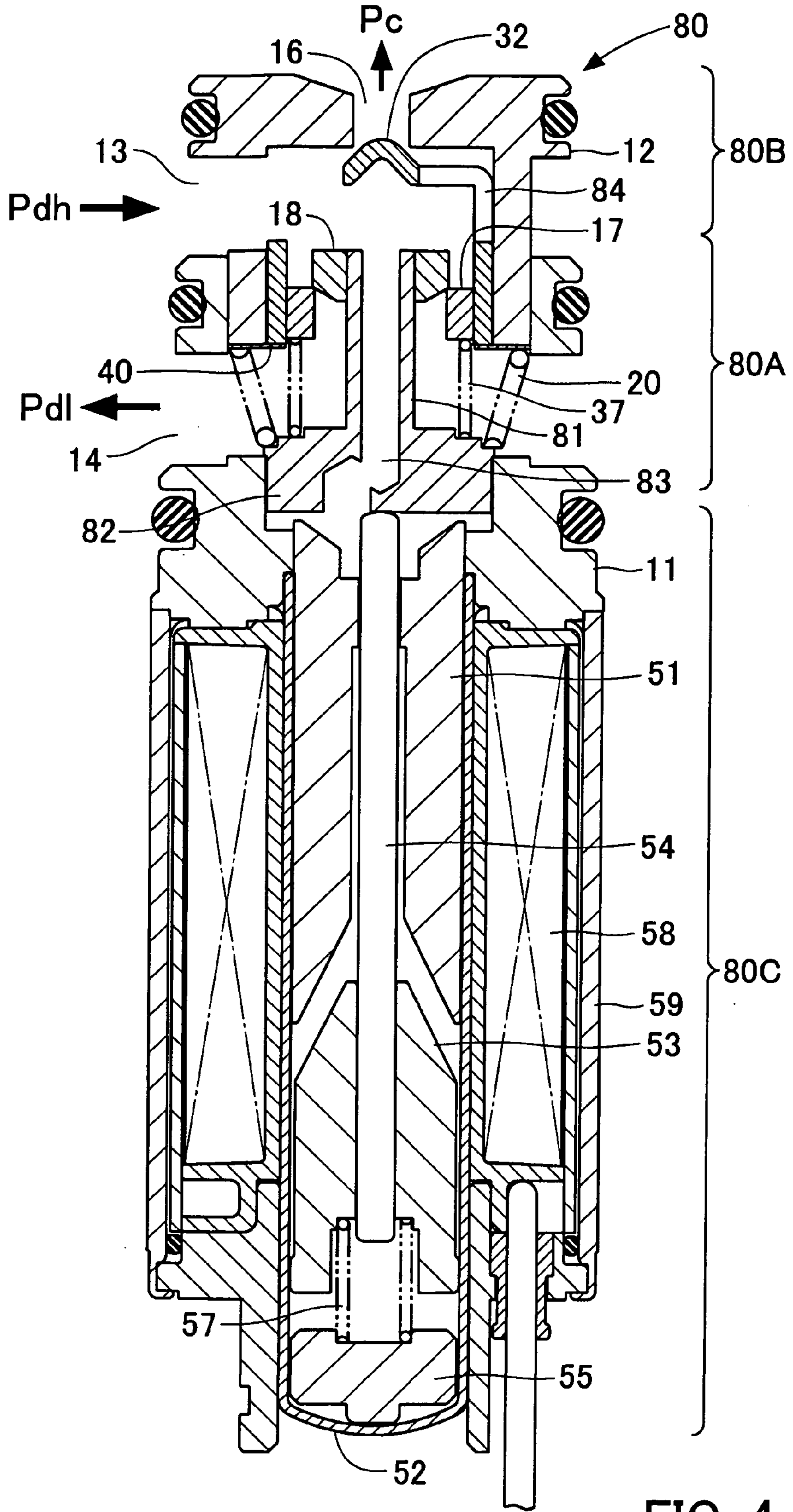
14
17
61
70A
19a
11
Pdh
20
13
19
64
51
54
58
59
56
70C
53
57
55
52

FIG. 3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32
80
13
12
80B
Pdh
84
18
17
40
20
80A
Pdl
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81
14
83
82
11
51
54
58
80C
59
53
57
55
52

FIG. 4

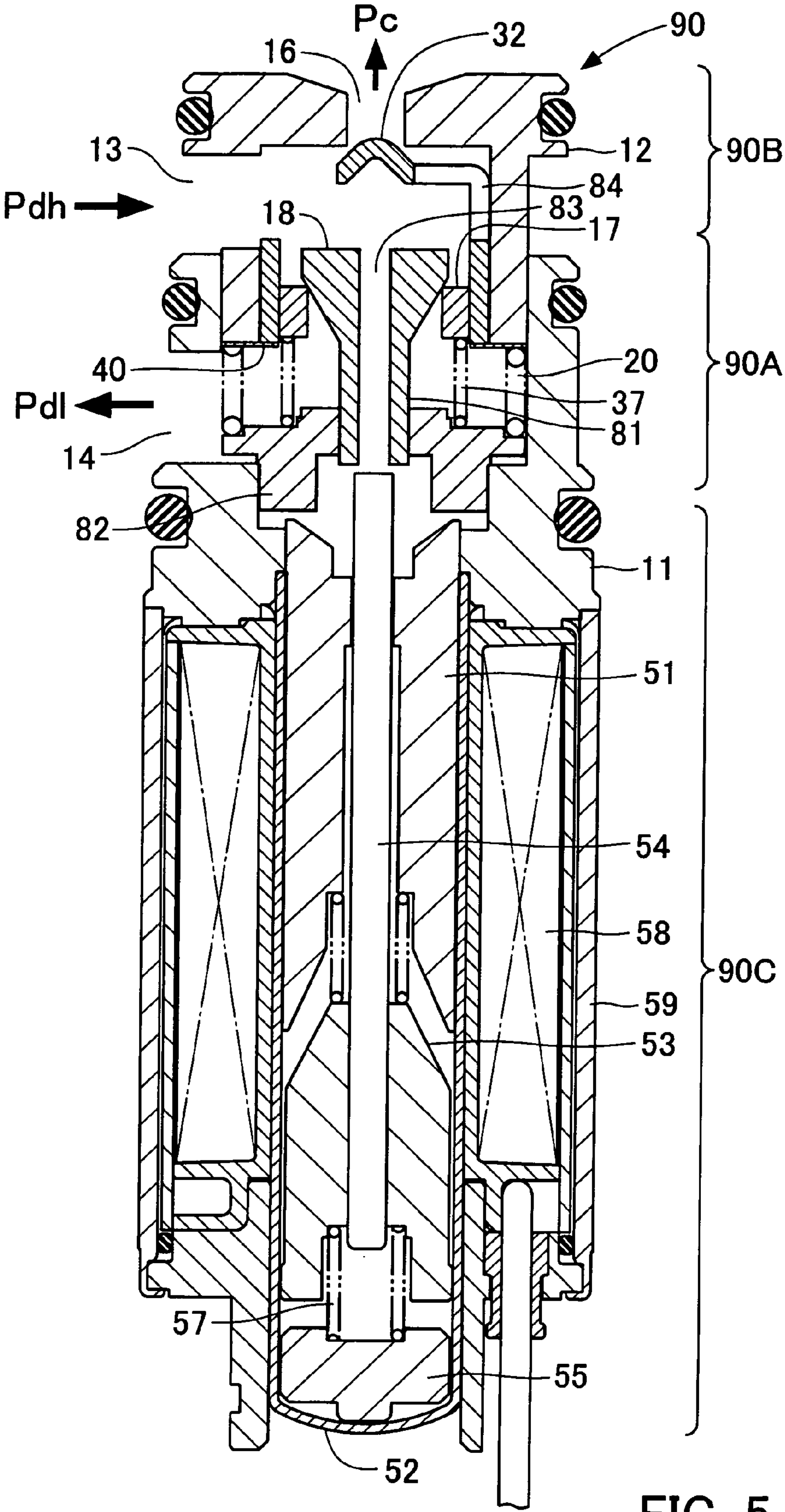
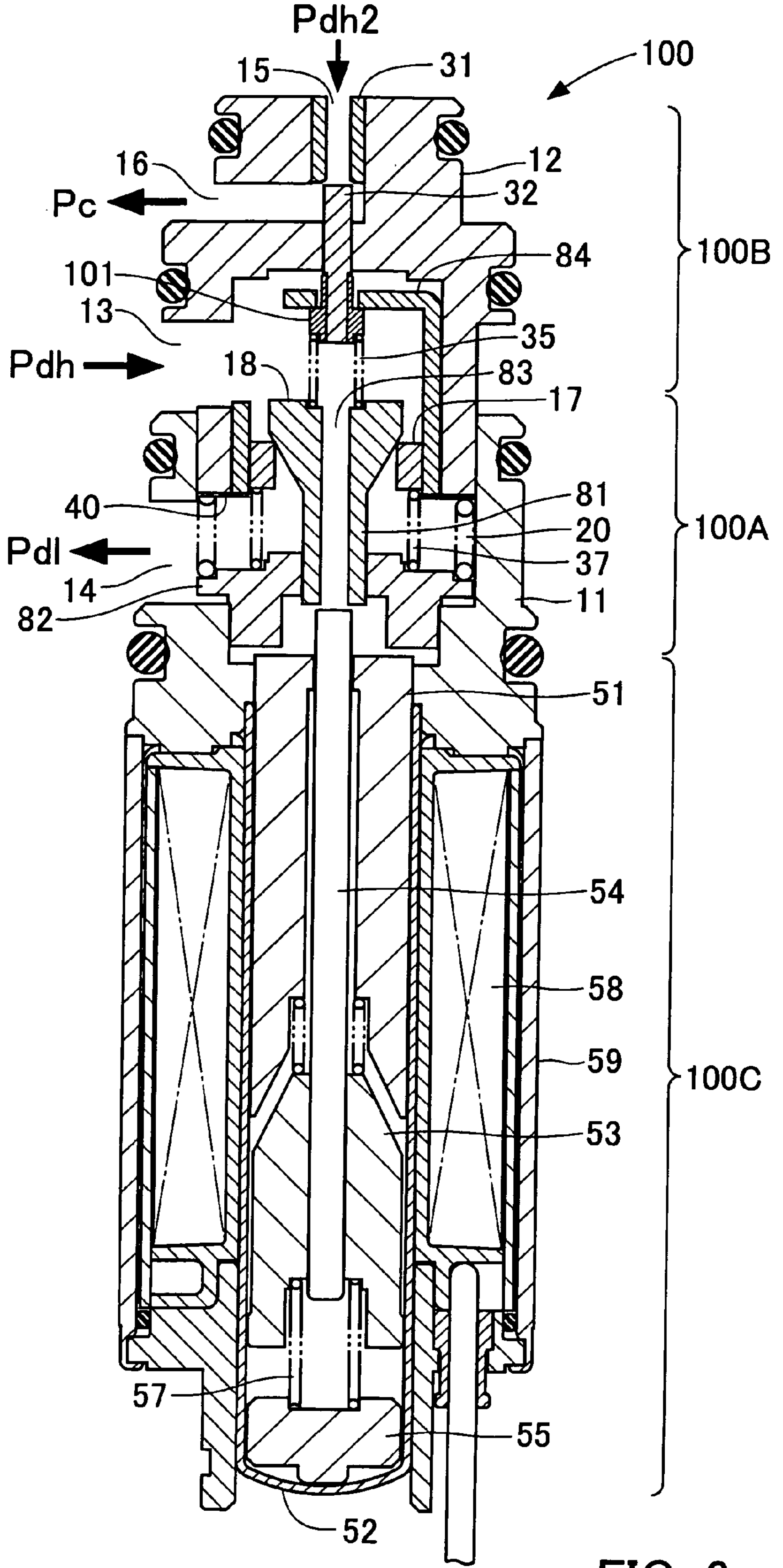
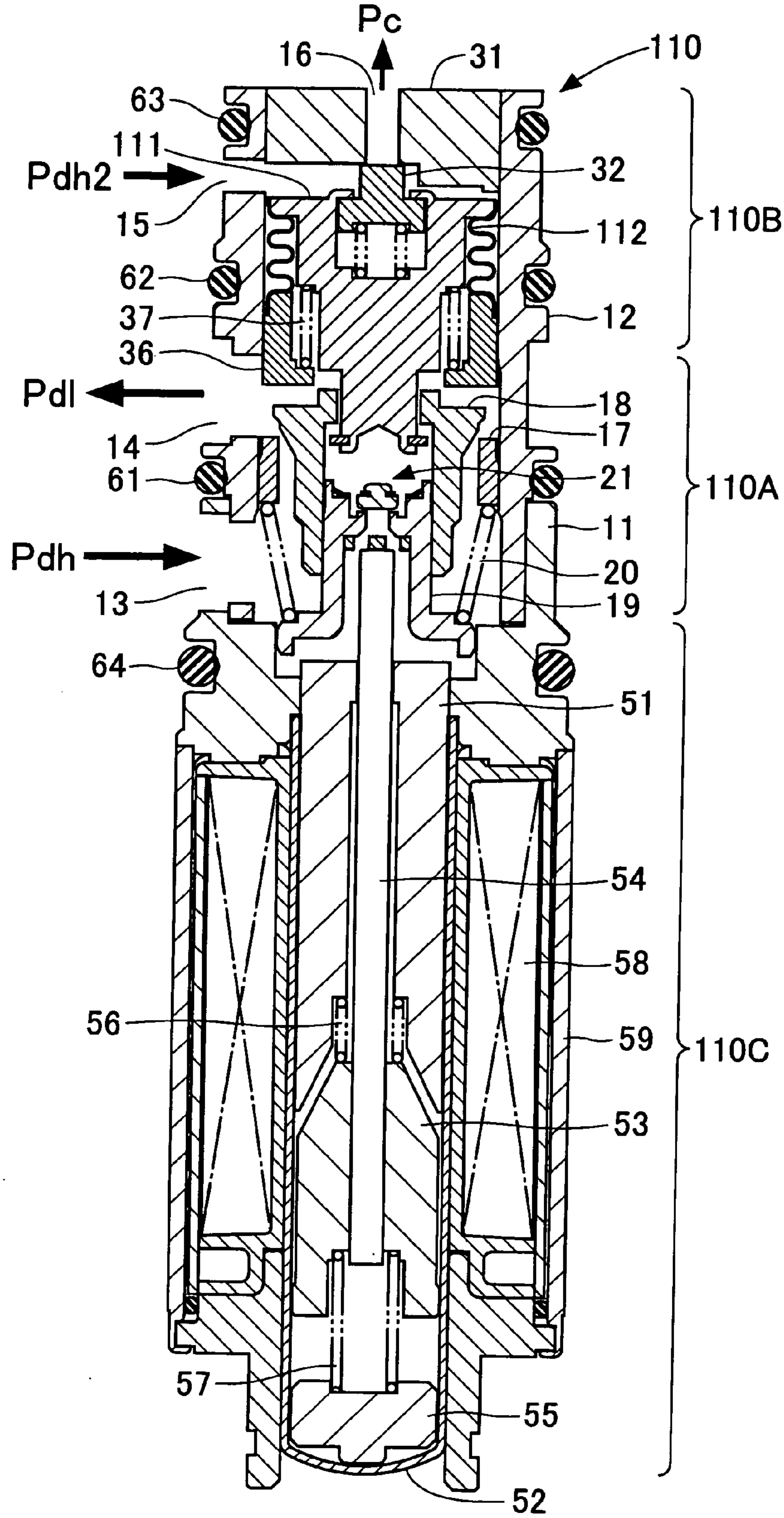

Pc
32
90
16
13
12
90B
Pdh
84
83
18
17
40
20
90A
Pdl
37
81
14
82
11
51
54
58
90C
59
53
57
55
52

FIG. 5



Pdh2
100
15
31
16
12
Pc
32
84
101
100B
13
35
Pdh
18
83
17
40
81
20
100A
Pdl
37
14
11
82
51
54
58
59
100C
53
57
55
52

FIG. 6



Pc
16
31
110
63
111
Pdh2
32
15
112
110B
62
37
12
36
Pdl
18
14
17
61
21
110A
11
Pdh
20
13
19
64
51
54
58
56
59
110C
53
57
55
52

FIG. 7

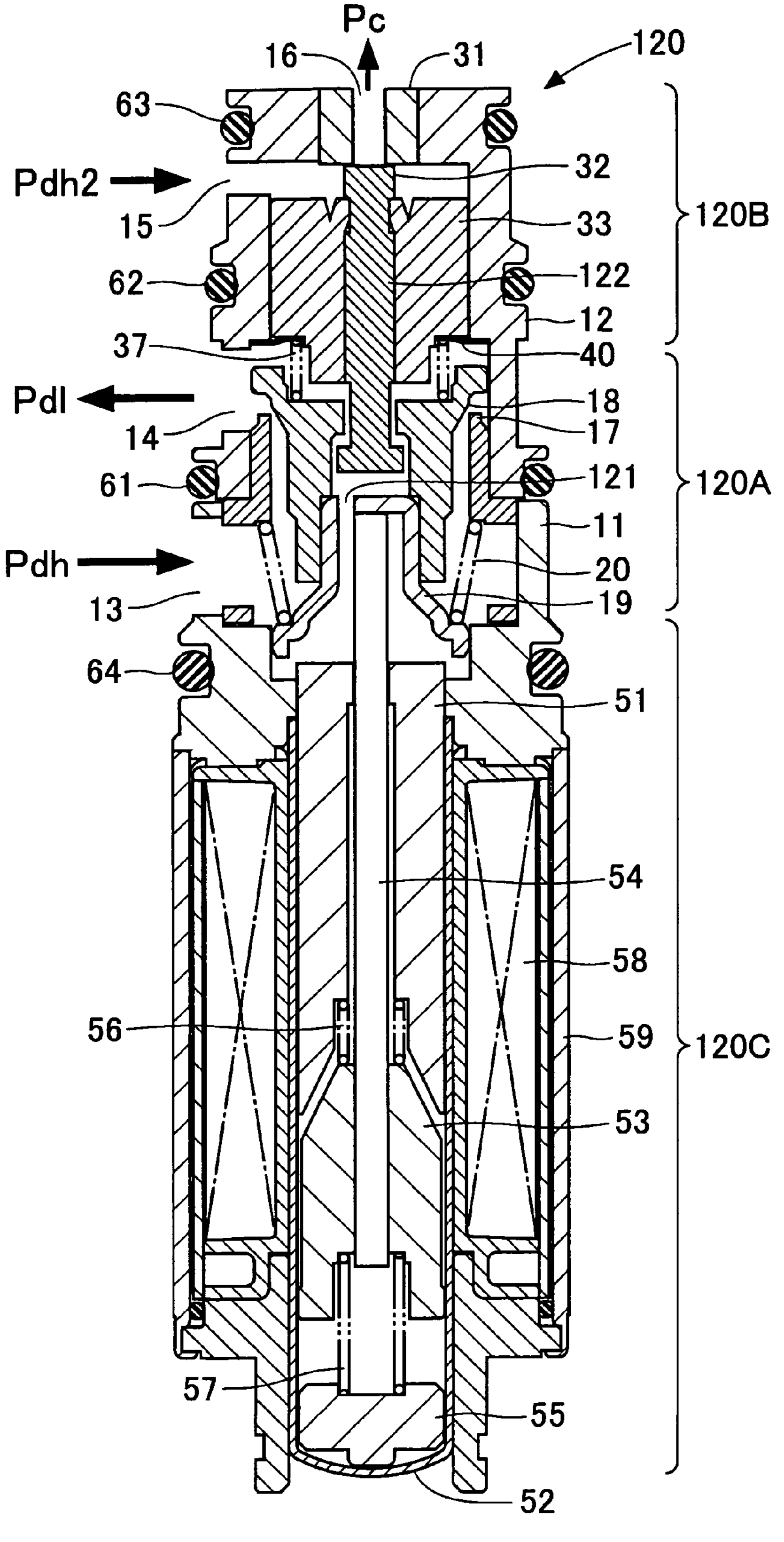

Pc
16
31
120
63
Pdh2
32
15
33
120B
62
122
12
37
40
Pdl
18
14
17
61
121
120A
11
Pdh
20
13
19
64
51
54
58
56
59
120C
53
57
55
52

FIG. 8

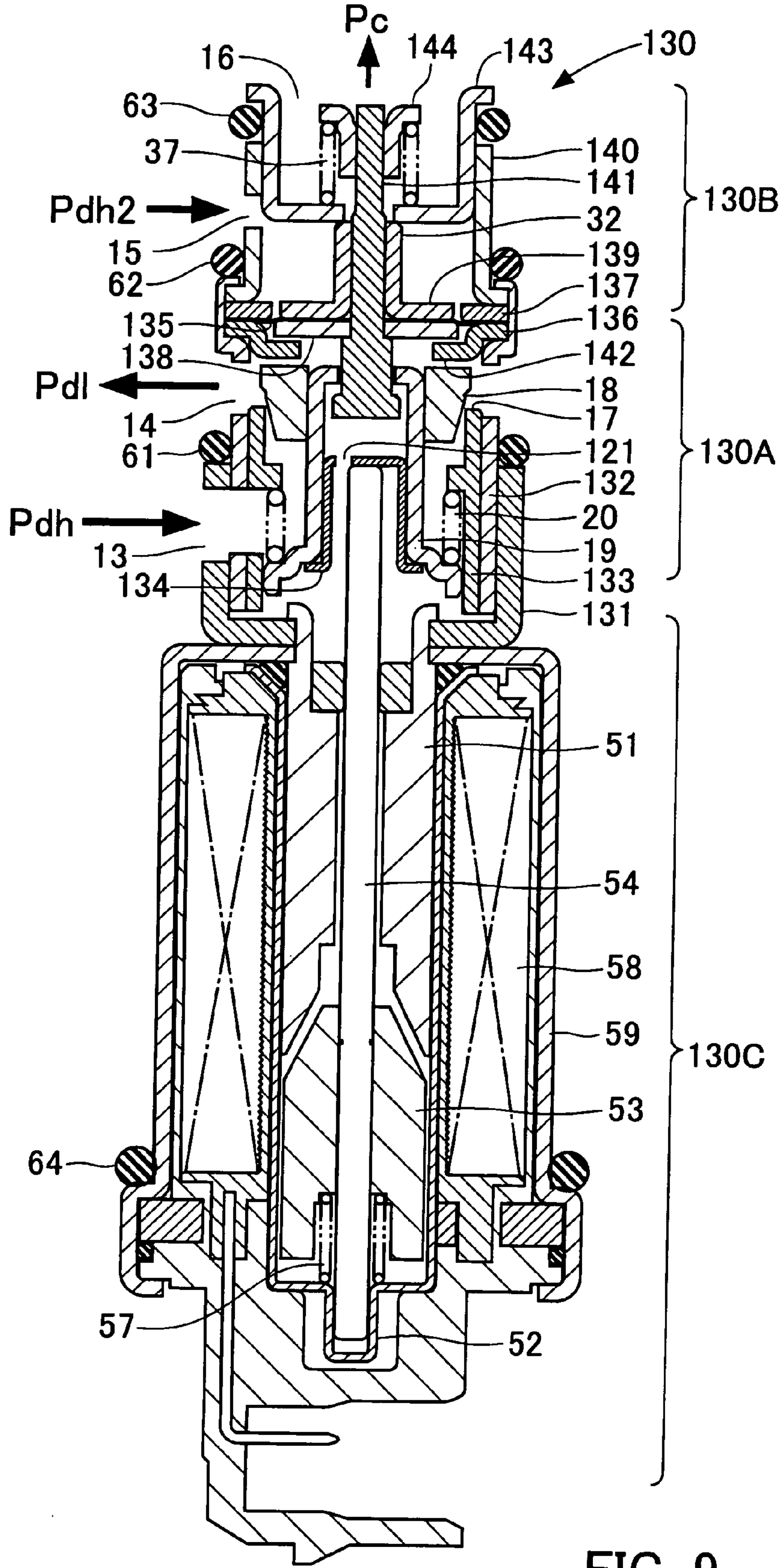

Pc
16
144
143
130
63
37
140
141
130B
Pdh2
32
15
139
62
137
135
136
138
142
Pdl
18
14
17
61
121
130A
132
Pdh
20
13
19
134
133
131
51
54
58
59
130C
53
64
57
52

FIG. 9

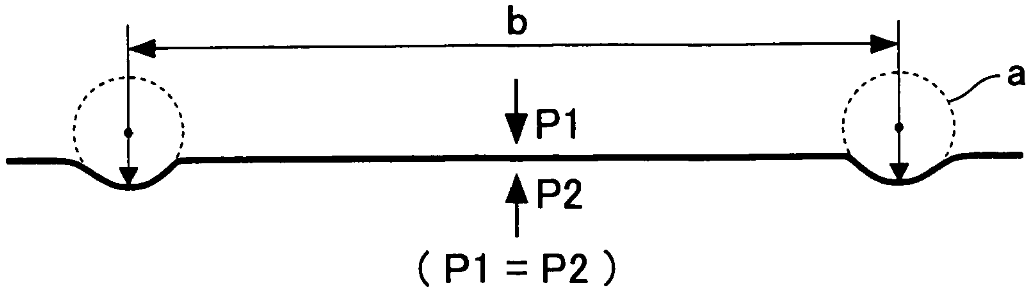


FIG. 10A

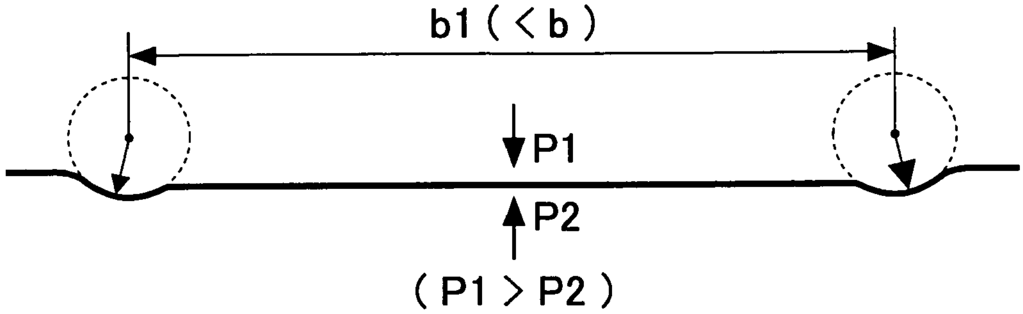


FIG. 10B

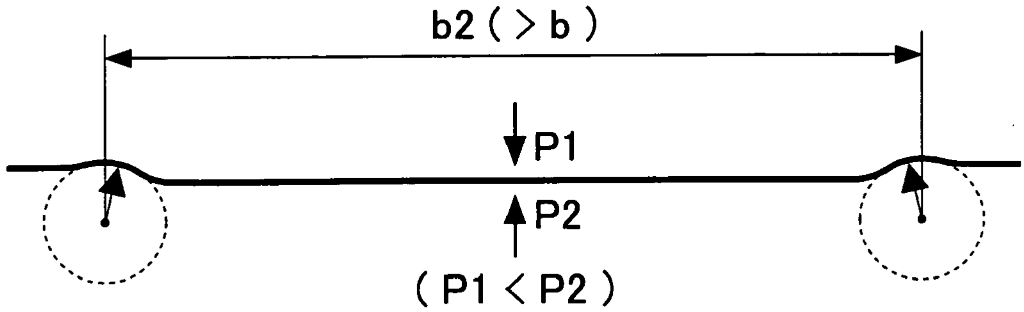


FIG. 10C

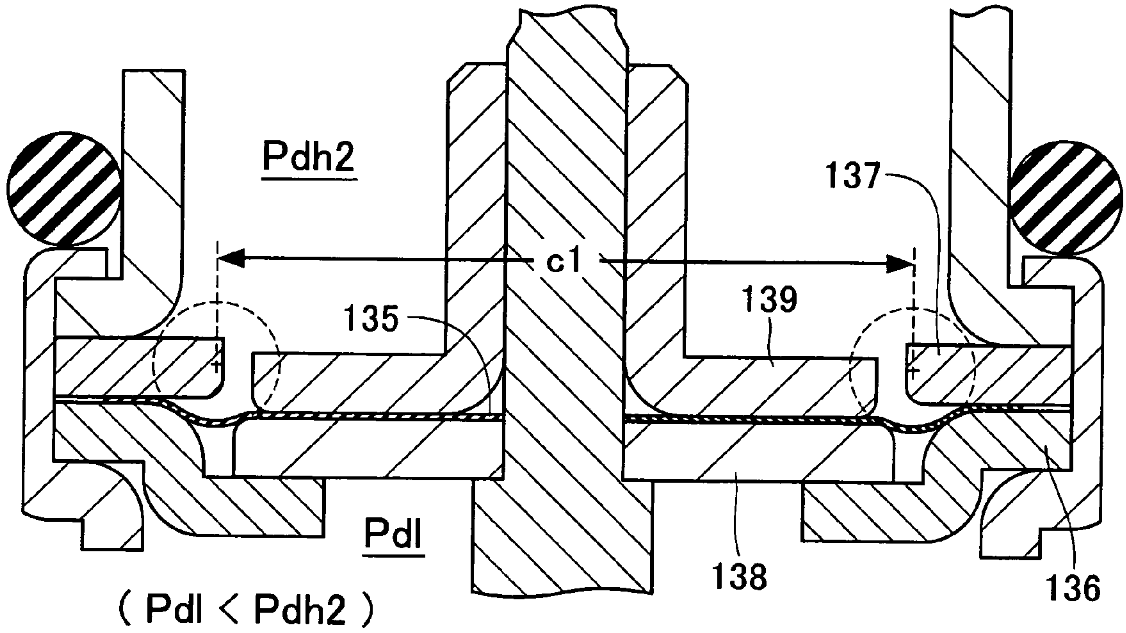


FIG. 11A

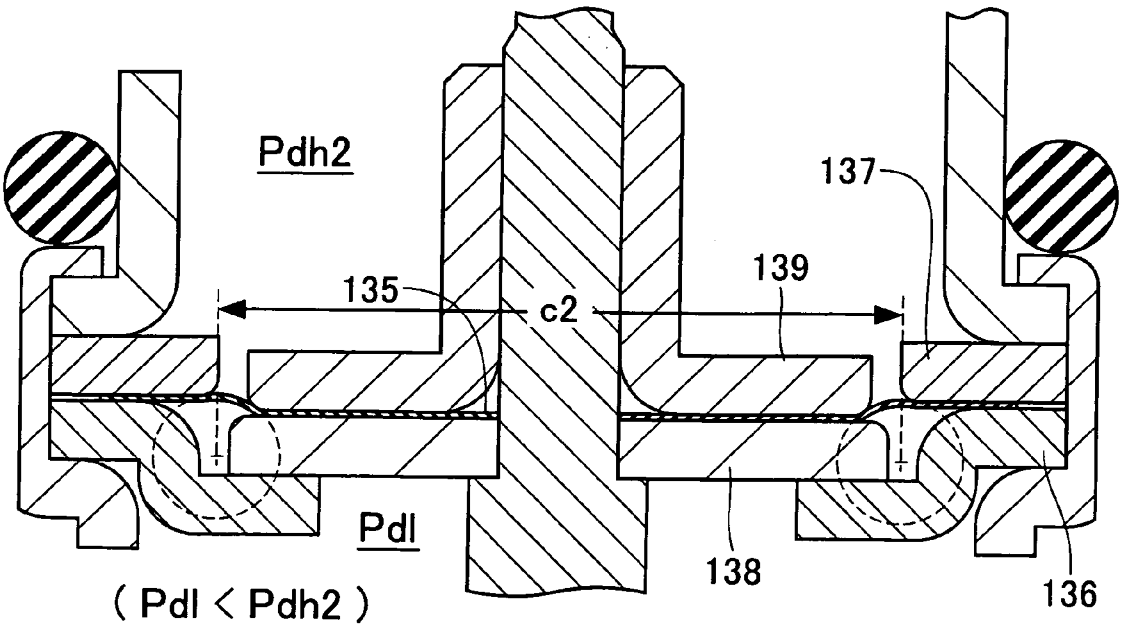


FIG. 11B

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2001107854 A **[0003] [0004]**
- JP 2004116349 A **[0005]**