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(11) **EP 1 024 286 A2**

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
02.08.2000 Bulletin 2000/31

(51) Int. Cl.⁷: **F04B 27/18**

(21) Application number: **00101797.9**

(22) Date of filing: **28.01.2000**

(84) Designated Contracting States:
**AT BE CH CY DE DK ES FI FR GB GR IE IT LI LU
MC NL PT SE**
Designated Extension States:
AL LT LV MK RO SI

(30) Priority: **29.01.1999 JP 2144099**

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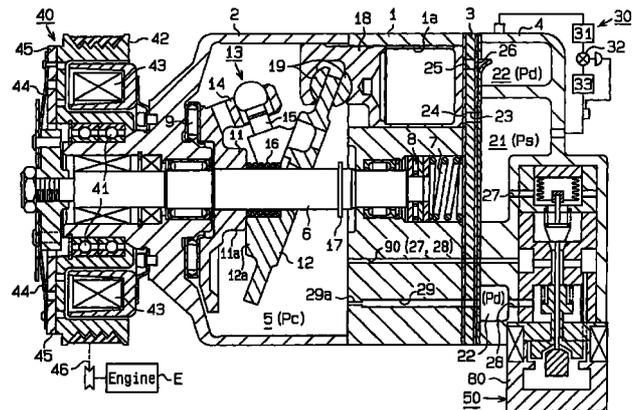
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(54) **Control valve for variable displacement compressor**

(57) A displacement control valve adjusts the pressure in a crank chamber (5) of a compressor to vary the displacement of the compressor. The control valve has a first valve mechanism (60), a second valve mechanism (70) and a solenoid mechanism (80). The solenoid mechanism (80) independently actuates the first and second valve mechanisms (60, 70). The first valve mechanism (60) selectively opens and closes a supply passage (28) of the compressor. The second valve mechanism (70) adjusts the flow rate of gas through a bleed passage (27) of the compressor. The second valve mechanism (70) includes a second valve opening (72), a second valve element (73), a bellows (76) and a second plunger (83). The second valve element (73) and the bellows (76) are accommodated in a single pressure chamber (71). The pressure of a suction chamber is introduced to the pressure chamber (71). The control valve has a relatively simple structure and is thus inexpensive and compact.

Fig.1



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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to control valves for variable displacement compressors, which vary the inclination angle of a swash plate by changing the internal pressure of a crank chamber to change the displacement.

[0002] In a typical variable displacement compressor, the inclination angle of a swash plate varies according to the internal pressure of a crank chamber to change the displacement of the compressor. Outlet control is one method for controlling the internal pressure of the crank chamber. Using outlet control, a high-pressure coolant gas having a pressure corresponding to the discharge pressure is constantly supplied at a predetermined flow rate to the crank chamber, while the flow rate of gas from the crank chamber is controlled by a control valve. Thus, the control valve controls the internal pressure of the crank chamber (crank pressure P_c). With the outlet control method, it is difficult to rapidly increase the crank pressure, making it difficult to quickly and responsively change the inclination angle of the swash plate, or the displacement. To overcome this problem, control valves have been proposed that have an outlet control valve mechanism and an inlet control valve mechanism. Such control valves are disclosed, for example, in Japanese Unexamined Patent Publication Nos. 5-99136 and 10-103249.

[0003] The control valve disclosed in Japanese Unexamined Patent Publication No. 5-99136 is provided with a first valve element that selectively opens and closes a supply passage, which connects a discharge chamber and a crank chamber, and a second valve element that selectively opens and closes a bleed passage, which connects the crank chamber and a suction chamber. A rod connected to the two valve elements is electromagnetically driven by a solenoid. The first valve element and the second valve element are completely separated, and the first and second valve elements do not open simultaneously. In the control valve in this publication, the rod is connected to the second valve element to be movable relative to the second valve element. Accordingly, even after the second valve element is seated, the seal between the second valve element and the rod tends to be inadequate, making it difficult to avoid gas leakage from the crank chamber to the suction chamber.

[0004] The control valve disclosed in Japanese Unexamined Patent Publication No. 10-103249 is provided with a first valve element that selectively opens and closes a supply passage, which connects a discharge chamber to a crank chamber, and a second valve element that adjusts the size of an opening of a bleed passage, which connects the crank chamber to a suction chamber. A first rod and a second rod extend from the first valve element and the second valve ele-

ment respectively. The second rod is fitted to the first rod to be movable relative to the first rod, so that the first and second valve elements can be operated independently. The rods are connected to respective plungers, and the plungers are electromagnetically driven by a common coil. With this control valve, unlike the control valve disclosed in Japanese Unexamined Patent Publication No. 5-99136, there is no problem of gas leakage through the clearance between the first valve element and the second rod as long as the valves are seated on their corresponding valve seats.

[0005] However, Japanese Unexamined Patent Publication No. 10-103249 merely proposes a control valve for a specific clutchless, swash plate-type, variable displacement compressor. That is, the compressor of this publication has a shutter near one end of the drive shaft for shutting off the suction chamber from a suction passage, which communicates with an external refrigerant circuit. The shutter stops the flow of coolant gas in the external refrigerant circuit. When the shutter blocks the suction passage from the suction chamber, two regions with different suction pressures (i.e., the suction passage and the suction chamber) are defined.

[0006] The control valve to be used with the compressor of publication No. 10-103249 has a pressure detecting chamber, to which the pressure in the suction passage is introduced, and a second valve chamber, to which the pressure in the suction chamber is introduced. A bellows is located in the pressure detecting chamber, while the second valve element is located in the second valve chamber. The pressure detecting chamber and the second valve chamber are isolated from each other by a diaphragm. Accordingly, the suction passage and the suction chamber are not connected to each other through the internal space of the control valve. The bellows and the second valve element are connected by a pressure detecting rod. The bellows moves the second valve element depending on the suction pressure introduced from the suction passage to the pressure detecting chamber.

[0007] The pressure detecting rod must be located to pass through the diaphragm between the pressure detecting chamber and the second valve chamber. In order to ensure separation between the pressure detecting chamber and the second valve chamber, high accuracy machining is required so that no clearance exists between the pressure detecting rod and the diaphragm. This makes the manufacture of such a control valve difficult and increases the cost significantly.

SUMMARY OF THE INVENTION

[0008] It is an object of the present invention to provide a simple and inexpensive control valve for a variable displacement compressor.

[0009] To achieve the above object, the present invention provides a control valve for a variable displacement compressor. The control valve adjusts the

pressure in a crank chamber of the compressor to vary the compressor displacement. The compressor includes a suction pressure zone, the pressure of which is a suction pressure, a discharge pressure zone, the pressure of which is a discharge pressure, a bleed passage for connecting the crank chamber to the suction pressure zone, and a supply passage for connecting the crank chamber to the discharge pressure zone. The control valve comprises a housing, a first valve mechanism retained in the housing to selectively open and close the supply passage, a second valve mechanism retained in the housing to adjust the flow rate of gas released from the crank chamber to the suction pressure zone through the bleed passage, and a solenoid mechanism retained in the housing to independently actuate the first valve mechanism and the second valve mechanism. The first valve mechanism includes a first valve opening defined in the housing. The first valve opening forms part of the supply passage. A first valve element selectively opens and closes the first valve opening. A first plunger is connected to the first valve element. The second valve mechanism includes a second valve opening defined in the housing. The second valve opening forms part of the bleed passage. A second valve element adjusts the opening size of the second valve opening. A pressure sensing member moves the second valve element in accordance with the suction pressure. A second plunger is connected to the second valve element. The solenoid mechanism includes a coil. Current supplied to the coil produces an electromagnetic force for independently biasing the first and second plungers in accordance with the level of the current. A pressure chamber is defined in the housing to accommodate the second valve element and the pressure sensing member. The pressure chamber is exposed to the suction pressure.

[0010] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0011] The invention together with the objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Figure 1 is a cross-sectional view of a variable displacement compressor according to one embodiment of the present invention;

Figure 2 is a cross-sectional view showing a displacement control valve incorporated in the compressor shown in Figure 1;

Figure 3 is a cross-sectional view showing a part of the displacement control valve according to another embodiment; and

Figure 4 is a cross-sectional view showing a part of the displacement control valve according to another embodiment.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0012] The present invention will be described by way of example referring to Figures 1 and 2.

[0013] As shown in Figure 1, a swash plate compressor is provided with a cylinder block 1, a front housing member 2 joined with the front end of the cylinder block 1, and a rear housing member 4 joined with the rear end of the cylinder block 1 through a valve plate 3. The cylinder block 1, the front housing 2, the valve plate 3 and the rear housing member 4 are held together with a plurality of through bolts (not shown) to form a compressor housing.

[0014] A crank chamber 5 is defined between the cylinder block 1 and the front housing member 2. A drive shaft 6 is supported by the cylinder block 1 and the front housing member 2 with a plurality of radial bearings.

[0015] A bore housing a helical spring 7 and a thrust bearing 8 is formed substantially at the center of the cylinder block 1. In the crank chamber 5, a rotating support 11 is fixed on the drive shaft 6 to rotate integrally with the drive shaft 6. Another thrust bearing 9 is located between the rotating support 11 and an inner wall of the front housing member 2. The drive shaft 6 is urged toward the front housing member 2 by the spring 7.

[0016] The drive shaft 6 is connected through an electromagnetic clutch 40 to an engine E, which serves as an external power source. The electromagnetic clutch 40 includes a pulley 42, an annular solenoid coil 43 and an armature 45. The pulley 42 is supported by a bearing 41 at the front end of the front housing member 2. The armature 45 is connected to the drive shaft 6 by a leaf spring 44.

[0017] When electric current is supplied to the coil 43, electromagnetic attraction is produced between the armature 45 and the pulley 42, which causes the armature 45 to engage the pulley 42, as shown in Figure 1. Consequently, the driving power of the engine E is transmitted to the drive shaft 6 through a transmission belt 46, the pulley 42, the armature 45 and the leaf spring 44. When the supply of electric current to the coil 43 is interrupted, the armature 45 is separated from the pulley 42 by the force of the leaf spring 44 to interrupt the power transmission. Thus, the driving power of the engine E can be transmitted to the drive shaft 6 selectively by controlling the supply of electric current to the coil 43.

[0018] A swash plate 12, or drive plate, is housed in

the crank chamber 5. The drive shaft 6 passes through a hole formed at the center of the swash plate 12. The swash plate 12 is connected through a hinge mechanism 13 to the rotating support 11 and the drive shaft 6. The hinge mechanism 13 consists essentially of a supporting arm 14 provided on the rotating support 11, and a guide pin 15 with a spherical head. The supporting arm 14 has a generally cylindrical socket for supporting the spherical head. The hinge mechanism 13 causes the swash plate 12 and the drive shaft 6 to rotate together and permits the swash plate 12 to move in the axial direction of the drive shaft 6, along the surface of the drive shaft 6, and to incline with respect to the axis of the drive shaft 6.

[0019] A coil spring 16 is fitted to the drive shaft 6 between the rotating support 11 and the swash plate 12. The spring 16 urges the swash plate 12 in a direction to decrease the angle of inclination of the swash plate 12 (as measured with respect to a plane perpendicular to the axis of the drive shaft 6). A snap ring 17 is fixed on the drive shaft 6 between the swash plate 12 and the cylinder block 1. The snap ring 17 restricts the movement of the swash plate 12 in the direction of the rear housing member 4 to determine the minimum inclination angle of the swash plate 12. The minimum inclination angle is, for example, 3 to 5 degrees. The maximum inclination angle of the swash plate 12 is determined by the abutment of a counterweight 12a formed on the swash plate 12 against a restricting section 11a of the rotating support 11.

[0020] A plurality of cylinder bores 1a (only one bore is shown) are defined in the cylinder block 1. The cylinder bores 1a are arranged at predetermined angular intervals along a circle drawn about the axis of the drive shaft 6. A single-headed piston 18 is located in each cylinder bore 1a. Each piston 18 is connected to the swash plate 12 through a pair of shoes 19.

[0021] A suction chamber 21, the pressure of which is referred to as the suction pressure P_s , and a discharge chamber 22, the pressure of which is referred to as the discharge pressure P_d , are defined in the rear housing member 4. The valve plate 3 contains a suction port 23, a suction valve 24, a discharge port 25 and a discharge valve 26 for each cylinder bore 1a. The suction chamber 21 communicates with each cylinder bore 1a through the suction port 23. Each cylinder bore 1a communicates with the discharge chamber 22 through the discharge port 25.

[0022] In the swash plate type compressor shown in Figure 1, the drive shaft 6 is rotated when the engine E is driven, and the swash plate 12 rotates with the rotation of the shaft 6. The rotational movement of the swash plate 12 is converted through the shoes 19 into reciprocating movement of the pistons 18. This reciprocating movement compresses a coolant gas drawn from the suction chamber 21, through the valve plate 3, into each cylinder bore 1a. Compressed coolant gas is exhausted from each cylinder bore 1a into the dis-

charge chamber 22.

[0023] The inclination angle of the swash plate 12 is determined according to various moments applied to the swash plate 12. The moments include a rotational moment, which is based on the centrifugal force of the rotating swash plate 12, a spring force moment, which is based on the force of the spring 16, and a gas pressure moment, which is based on the net force applied to each piston 18 by gas pressure. The rotational moment acts on the swash plate 12 to increase the inclination angle. The gas pressure moment depends on the reactive force of compression acting upon the pistons 18 during the compressing strokes, the internal pressure of the cylinder bore 1a acting upon the pistons 18 during the suction strokes, and the pressure of the crank chamber 5 (crank pressure P_c). The gas pressure moment acts on the swash plate 12 to decrease the inclination angle.

[0024] In this embodiment, when the crank pressure P_c is maintained at a relatively high level, the sum of the gas pressure moment and the spring force moment is greater than the rotational moment. Accordingly, the swash plate 12 tends to shift to the minimum inclination angle position. The sum of the moment based on the gas pressure and the moment based on the spring force is balanced with the moment of the rotational movement by adjusting the crank pressure P_c . Thus, the inclination of the swash plate 12 can be set at a desired angle between the minimum inclination angle position and the maximum inclination angle position. The stroke of each piston 18, or the discharge displacement of the compressor, is adjusted according to the inclination angle of the swash plate 12.

[0025] As shown in Figures 1 and 2, the mechanism for controlling the crank pressure P_c consists essentially of a displacement control valve 50 and a plurality of passages 27, 28 and 29. That is, the compressor housing is provided with a bleed passage 27 connecting the crank chamber 5 to the suction chamber 21, and a supply passage 28 connecting the crank chamber 5 to the discharge chamber 22. The control valve 50 is located in the bleed passage 27 and the supply passage 28. The valve 50 can regulate the flow in these passages 27,28 independently. The passage between the control valve 50 and the crank chamber 5 serves both as part of the bleed passage 27 and part of the supply passage 28 and is therefore referred to as a common passage 90.

[0026] The portion of the bleed passage 27 connecting the control valve 50 to the suction chamber 21 functions also as a pressure detecting passage for applying the suction pressure P_s to the control valve 50. Further, the compressor housing is provided with, in addition to the supply passage 28, an auxiliary supply passage 29 connecting the crank chamber 5 to the discharge chamber 22. The auxiliary supply passage 29 is provided with a fixed restriction 29a.

[0027] The discharge chamber 22 and the suction chamber 21 are connected to each other through an

external refrigerant circuit 30. The external refrigerant circuit 30 forms, together with the compressor, a cooling circuit of an air conditioner. The external refrigerant circuit 30 is provided with a condenser 31, a thermostatic expansion valve 32 and an evaporator 33. The valve position of the expansion valve 32 is feedback-controlled based on the temperature detected by a temperature detecting cylinder located at the outlet of the evaporator 33. The outlet temperature of the evaporator 33 reflects the thermal load applied to the refrigerator circuit. The expansion valve 32 supplies an appropriate amount of coolant to the evaporator 33 depending on the thermal load applied to the refrigerator circuit. This adjusts the flow rate of the coolant in the external refrigerant circuit 30.

[0028] As shown in Figure 2, a temperature sensor 34 is located adjacent to the evaporator 33. The temperature sensor 34 detects the temperature of the evaporator 33 and outputs a signal indicating the detection result to a controller C. The controller C is a computer that performs overall control of heating and cooling for a vehicle passenger compartment. A cabin temperature sensor 35 for detecting the temperature in the passenger compartment, a cabin temperature setter 36 for setting the target temperature in the passenger compartment, an actuating switch 37 and an engine revolution speed sensor 38 are connected to the inlet side of the controller C along with the temperature sensor 34. A drive circuit 39A for controlling the supply of electric current to the solenoid coil 43 of the electromagnetic clutch 40 and another drive circuit 39B for controlling the supply of electric current to a coil 85 (to be described later) of the control valve 50 are connected to the output side of the controller C.

[0029] The controller C controls the electromagnetic clutch 40 and the control valve 50 based on various information including the temperature of the evaporator 33 detected by the temperature sensor 34, the temperature detected by the cabin temperature sensor 35, the target temperature set by the cabin temperature setter 36, the position of the switch 37, and the engine speed from the engine revolution speed sensor 38. The controller C computes the value of electric current to be supplied to the coil 85 of the control valve 50 based on the received information and instructs the drive circuit 39B accordingly.

[0030] As shown in Figure 2, the control valve 50 is provided with a first valve mechanism 60 for selectively opening and closing the supply passage 28, a second valve mechanism 70 for adjusting the opening size of the bleed passage 27, and a solenoid mechanism 80 connected to the two valve mechanisms 60 and 70. The three mechanisms 60,70,80 are incorporated into a valve housing 51. The first valve mechanism 60 and the second valve mechanism 70 can be operated independently as will be described later.

[0031] The first valve mechanism 60 is provided with a first valve chamber 61 and an axial passage 62

defined in the valve housing 51. The first valve chamber 61 communicates with the discharge chamber 22 through a supply port 52 and the supply passage 28. The pressure of the discharge chamber 22 (discharge pressure P_d) is applied to the first valve chamber 61. A linear passage, or the axial passage 62 is connected with the crank chamber 5 through a common port 53 and the common passage 90, and the crank pressure P_c is applied to the axial passage 62. The portion of the axial passage 62 opening to the first valve chamber 61 constitutes a first valve opening 63. The first valve chamber 61, the axial passage 62 and the first valve opening 63 form a part of the supply passage 28.

[0032] A first valve element 64 is located in the first valve chamber 61 to move in the axial direction of the control valve 50. The first valve element 64 opens and closes the first valve opening 63. The first valve element 64 is connected to a first plunger 82 through a first rod 65. The first plunger 82 is located in a solenoid chamber 81 located adjacent to the first valve chamber 61. An opening spring 66 is located between the first valve element 64 and the inner wall of the first valve chamber 61. The opening spring 66 urges the first valve element 64 away from the first valve opening 63, so the first valve mechanism 60 is normally open. The first valve element 64, the first rod 65 and the first plunger 82 constitute an integral body, and the integral body has an axial hole 67. The cross-sectional area S_1 of the first rod 65 is substantially equal to the cross-sectional area S_2 of the first valve opening 63.

[0033] The second valve mechanism 70 is provided with a pressure chamber, or a second valve chamber 71, defined above and adjacent to the axial passage 62 in the valve housing 51. The second valve chamber 71 functions as a pressure detecting chamber. The second valve chamber 71 communicates with the suction chamber 21 through a port 54 and the downstream part of the bleed passage (pressure detecting passage) 27. Accordingly, the pressure of the suction chamber 21 (suction pressure P_s) is applied to the second valve chamber 71.

[0034] The second valve chamber 71 has an annular spring seat 55 extending from the inner circumferential wall of the valve housing 51. The spring seat 55 divides the second valve chamber 71 into an upper region and a lower region. However, these two regions communicate with each other through a center hole in the spring seat 55, and they have the same pressure. The portion of the axial passage 62 opening to the second valve chamber 71 constitutes a tapered second valve opening 72. The axial passage 62, the second valve chamber 71 and the second valve opening 72 form part of the bleed passage 27.

[0035] A second valve element 73 is located in the second valve chamber 71 to be movable in the axial direction of the control valve 50. The second valve element 73 varies the area of the second valve opening 72 that is available for gas flow. The lower end of the sec-

ond valve element 73 is connected to a second plunger 83 through a second rod 74. The second plunger 83 is housed in the solenoid chamber 81. The second rod 74 extends through the axial passage 62 and the first valve chamber 61 and into the solenoid chamber 81. The second rod 74 is fitted in the axial hole 67 of the first rod 65. A closing spring 75 is located between the second valve element 73 and the spring seat 55. The closing spring 75 urges the second valve element 73 toward the second valve opening 72 to normally close the second valve opening 72.

[0036] A bellows 76, which serves as a pressure sensing member, is also located in the second valve chamber 71. One end (the upper end in the drawings) of the bellows 76 is fixed to the wall of the valve chamber 71, and a connecting cylinder 77 is fixed to the other end. The second valve element 73 is fixed to a pressure detecting rod 78, and the upper end of the pressure detecting rod 78 is inserted into the connecting cylinder 77. The pressure detecting rod 78 is not fixed to the connecting cylinder 77 and is movable relative to the connecting cylinder 77. The bellows 76 is connected to the second valve element 73 and moves toward and away from the second valve element 73. The bellows 76 expands and contracts depending on the suction pressure P_s , which is applied to the second valve chamber 71, which causes the second valve element 73 to change the effective size of the second valve opening 72.

[0037] The solenoid mechanism 80 includes a solenoid chamber 81 defined in the valve housing 51. When the control valve 50 is attached to the rear housing member 4 of the compressor, an annular chamber 56 is defined between the valve housing 51 and an inner surface of the rear housing member 4 at a position corresponding to the location of the common port 53. The valve housing 51 contains a pressure application passage 57 formed to connect the annular chamber 56 and the solenoid chamber 81. The crank pressure P_c is applied through the annular chamber 56 and the pressure application passage 57 to the solenoid chamber 81.

[0038] A fixed iron core 84 is located between the solenoid chamber 81 and the first valve chamber 61. The solenoid chamber 81 contains the first plunger 82 and the second plunger 83. A coil 85 is wound around the fixed iron core 84 to surround the plungers 82 and 83. Energization of the coil 85 is controlled by the controller C. The electromagnetic force generated by energization of the coil 85 urges the plungers 82 and 83 toward the fixed iron core 84 against the forces of the springs 66 and 75, respectively. If an electric current having a certain level is supplied to the coil 85, the force of the first plunger 82 toward the fixed iron core 84 overcomes the force of the opening spring 66 to fully close the first valve mechanism 60. On the contrary, if no electric current is supplied to the coil 85, the first valve mechanism 60 is fully opened. Accordingly, the first

valve mechanism 60 is opened and closed selectively by external control. The second valve mechanism 70 adjusts the effective size of the valve opening 72 depending on the level of electric current supplied to the coil 85 and the suction pressure P_s .

[0039] The operation of the variable displacement compressor of Figs 1 and 2 will now be described.

[0040] When the actuating switch 37 is turned off, the electromagnetic clutch 40 is disengaged, and the compressor is inoperative. At this time, no electric current is supplied to the coil 84 of the control valve 50, and thus no electromagnetic force is applied to the plungers 82 and 83. Accordingly, in the first valve mechanism 60, the first valve opening 63 is opened fully by the opening spring 66, and in the second valve mechanism 70, the second valve opening 72 is closed by the closing spring 75. If the inoperative state of the compressor continues for a relatively long time, the internal pressures of the chambers 5, 21 and 22 in the compressor are equalized, and the swash plate 12 is maintained at the minimum inclination by the spring 16.

[0041] If the cabin temperature detected by the cabin temperature sensor 35 exceeds the temperature preset by the cabin temperature setter 36 when the actuating switch 37 is turned on, the controller C energizes the solenoid coil 43 of the electromagnetic clutch 40. Thus, the engine E drives the compressor. Simultaneously, the controller C energizes the coil 85 of the control valve 50. This causes the fixed iron core 84 to electromagnetically attract the first plunger 82, and the first valve element 64 closes the first valve opening 63 against the force of the opening spring 66 (see Figure 2) and fully closes the supply passage 28.

[0042] An electromagnetic force corresponding to the level of electric current is generated between the first plunger 82 and the second plunger 83 by the energization of the coil 85. This electromagnetic attractive force is transmitted through the second rod 74 to the second valve element 73, which increases the effective size of the second valve opening 72 against the force of the closing spring 75. As long as the coil 85 is magnetized, a connection is established among the second plunger 83, the second valve element 73 and the bellows 76. The force of the bellows 76 depends on the fluctuation of the suction pressure P_s , which is applied to the second valve chamber 71, and affects the position of the second valve element 73. In other words, the second valve mechanism 70 determines the effective size of the valve opening 72 based on the balance of the electromagnetic force applied to the second plunger 83, the force of the closing spring 75, and the force of the bellows 76, which reflects the suction pressure P_s . Thus, the first valve element 64 and the second valve element 73 operate independently even when the coil 85 is energized.

[0043] As the thermal load increases, the outlet pressure of the evaporator 33 (i.e., suction pressure P_s) increases gradually, and the difference between the

temperature detected by the cabin temperature sensor 35 and the temperature preset by the cabin temperature setter 36 also increases. When this occurs, the controller C controls the value of electric current supplied to the coil 85 based on the detected cabin temperature and the preset cabin temperature. More specifically, the supply of electric current is increased as the detected cabin temperature increases, and the force applied to the second valve element 73 is increased in the direction of increasing the effective size of the second valve opening 72. This has the effect of reducing the suction pressure P_s . That is, the increase in the electric current level supplied to the coil 85 causes second valve mechanism 70 to decrease the suction pressure P_s . In other words, the second valve mechanism 70 determines a target value of the suction pressure P_s depending on the level of electric current supplied to the coil 85.

[0044] The greater the effective size of the second valve opening 72, the greater the flow rate of gas from the crank chamber 5 through the bleed passage 27 to the suction chamber 21. Meanwhile, gas cannot flow from the discharge chamber 22 through the supply passage 28 into the crank chamber 5, since the first valve mechanism 60 is closed. Therefore, the crank pressure P_c drops. Further, when there is a great thermal load, the pressure of the gas drawn into the cylinder bores 1a, or suction pressure P_s , is relatively high, and the difference between the internal pressure of the cylinder bores 1a and the crank pressure P_c becomes small. Accordingly, the inclination angle of the swash plate 12 increases to increase the discharge displacement of the compressor. As a result, the suction pressure P_s is lowered gradually.

[0045] When the effective size of the second valve opening 72 is maximized, a maximum amount of gas flows from the crank chamber 5 through the bleed passage 27 to the suction chamber 21. The crank pressure P_c then falls to approximately the pressure in the suction chamber 21 (suction pressure P_s), and the inclination of the swash plate 21 reaches the maximum angle, which results in the maximum discharge displacement. In the maximum discharge displacement state, fluctuation in the condensation capacity in the condenser 31 can increase the pressure in the discharge chamber 22 (discharge pressure P_d) greatly. In this state, the relatively high discharge pressure P_d is applied to the first valve chamber 61 and the first valve element 64.

[0046] However, the cross-sectional area S_1 of the first rod 65 connecting the first valve element 64 to the first plunger 82 is substantially equal to the cross-sectional area S_2 of the first valve opening 63. Therefore, the pressure-receiving areas on both ends of the first valve element 64 are almost equal when the first valve element 64 closes the first valve opening 63. Consequently, the forces acting on the first valve element 64 in its moving direction substantially offset each other, which allows the first valve element to be smoothly operated without being affected by the discharge pres-

sure P_d and the crank pressure P_c .

[0047] As the thermal load decreases, the outlet pressure of the evaporator 33 is lowered gradually, and the difference between the temperature detected by the cabin temperature sensor 35 and the temperature preset by the cabin temperature setter 36 becomes small. Therefore, the controller C controls the level of the electric current supplied to the coil 85 to optimize the discharge capacity of the compressor for the thermal load. More specifically, the level of the electric current is reduced as the detected cabin temperature is lowered and the force of the second valve element 73 in the direction to increase the effective size of the second valve opening 72 is reduced. This has the effect of increasing the suction pressure P_s . That is, the reduction in the level of the electric current supplied to the coil 85 causes the second valve mechanism 70 to maintain the suction pressure P_s at a higher value.

[0048] The smaller the effective size of the second valve opening 72, the smaller the flow rate of gas from the crank chamber 5 through the bleed passage 27 to the suction chamber 21, which increases the crank pressure P_c . Further, when there is a small thermal load, the pressure of the gas drawn into the cylinder bore 1a, or the suction pressure P_s , is relatively low, and the difference between the internal pressure of the cylinder bore 1a and the crank pressure P_c becomes great. Accordingly, the inclination angle of the swash plate 12 decreases, which reduces the discharge displacement. As a result, the suction pressure P_s is increased gradually.

[0049] When the thermal load is approximately nil, the temperature of the evaporator 33 drops gradually toward the temperature at which frosting occurs. If the temperature detected by the temperature sensor 34 drops to a preset level (a temperature at which frosting can occur in the evaporator 33) or lower, the controller C interrupts the supply of electric current to the coil 85. This causes the electromagnetic attraction between the fixed iron core 84 and the first plunger and that between the first plunger 82 and the second plunger 83 to disappear. Accordingly, the first valve mechanism 60 fully opens the supply passage 28 under the force of the opening spring 66, while the second valve mechanism 70 closes the bleed passage 27 under the force of the closing spring 75. As a result, the high-pressure gas in the discharge chamber 22 is supplied in a large amount through the supply passage 28 into the crank chamber 5 to increase the crank pressure P_c . Consequently, the swash plate 12 shifts to the minimum inclination angle position, which minimizes the cooling capacity of the air conditioner.

[0050] When the actuating switch 37 is turned off, the controller C interrupts energization of the coil 85 to shift the swash plate 12 to the minimum inclination angle position.

[0051] The pressure detecting rod 78 and the connecting cylinder 77 of the bellows are not fixed and are

thus able to move relative to each other. The closing spring 75 constantly urges the second valve element 73 away from the bellows 76. When the bellows 76 contracts, due to an increase in the suction pressure P_s , while the solenoid mechanism 80 is de-energized, the connecting cylinder 77 moves relative to the pressure detecting rod. Accordingly, the upward movement of the bellows 76 is not transmitted to the second valve element 73. Therefore, even if the suction pressure P_s increases when the solenoid mechanism 80 is de-energized, the second valve mechanism 70 keeps the second valve opening 72 closed.

[0052] If the temperature of the passenger compartment (thermal load) increases when the actuating switch 37 is turned on and the swash plate 12 is at the minimum inclination position, the temperature of the passenger compartment detected by the cabin temperature sensor 35 exceeds the temperature preset by the cabin temperature setter 36. In response, the controller C outputs a command to energize the solenoid mechanism 80 according to the change in the temperature. With the activation of the solenoid mechanism 80, the first valve mechanism 60 closes the supply passage 28, while the second valve mechanism 70 opens the bleed passage 27 to reduce the crank pressure P_c gradually and to increase the angle of inclination of the swash plate 12.

[0053] As described above, the actions of the first valve mechanism 60 and those of the second valve mechanism 70 are controlled by supply and interruption of electric current to the coil 85 of the solenoid mechanism 80. In particular, the second valve mechanism 70 can vary the target value of the suction pressure P_s by controlling the level of the electric current supplied to the coil 85. Thus, the controller can change the displacement by changing the inclination angle of the swash plate 12 so that the actual suction pressure P_s will approach the target value.

[0054] This embodiment has the following effects.

[0055] The displacement control valve 50 includes the first valve mechanism 60 for selectively opening and closing the supply passage 28 and the second valve mechanism 70 for adjusting the opening size of the bleed passage 27 in one valve housing 51. This miniaturizes, simplifies, and reduces the cost of the mechanism for controlling the displacement of the compressor compared to the employment of two independent valve mechanisms. Miniaturizing the displacement control mechanism results in a smaller compressor.

[0056] Since a single coil 85 is used with the first plunger 82 of the first valve mechanism 60 and the second plunger 83 of the second valve mechanism 70, the structure of the control valve 50 is simplified.

[0057] The second rod 74 connecting the second valve element 73 and the second plunger 83 passes through the integral body, which includes the first valve element 64, the plunger 82 and the first rod 65. This reduces the size of the control valve 50 in the axial

direction.

[0058] The solenoid mechanism 80 is located at one end of the valve housing 51. As shown in Figure 1, when the control valve 50 is incorporated into the rear housing member 4 of the compressor, part of the solenoid mechanism 80 is exposed. This facilitates the connection of wiring to the coil 85.

[0059] The second valve element 73 and the bellows 76 are located in a single chamber 71, to which the suction pressure P_s is applied. This reduces the number of chambers to be formed in the control valve 50 and simplifies the control valve 50. Further, if a second valve element and a bellows were located in different chambers and connected with a pressure detecting rod, as in the control valve disclosed in Japanese Unexamined Patent Publication No. 10-103249, high-accuracy machining would be required to avoid a clearance between the diaphragm located between these two chambers and the pressure detecting rod, which would pass through the diaphragm. This construction is disadvantageous in terms of ease of manufacture and cost. However, the control valve 50 of this embodiment, which has both the second valve element 73 and the bellows 76 in the same chamber 71, overcomes the problems inherent in the control valve disclosed in Japanese Unexamined Patent Publication No. 10-103249.

[0060] In the outlet control method where the inclination angle of the swash plate 12 is adjusted by controlling the amount of gas exhausted from the crank chamber 5, high-pressure coolant gas must be supplied constantly to the crank chamber 5 to change the inclination angle of the swash plate 12 responsively. The variable displacement compressor shown in Figure 1 has the auxiliary supply passage 29 connecting the discharge chamber 22 to the crank chamber 5, and the auxiliary supply passage 29 has the fixed restriction 29a. Accordingly, when the supply passage 28 is closed by the first valve mechanism 60 and the opening of the bleed passage 27 is regulated by the second valve mechanism 70, a predetermined amount of coolant gas is supplied constantly from the discharge chamber 22 to the crank chamber 5 through the auxiliary supply passage 29 to the crank chamber 5. Thus, the crank pressure P_c is maintained constantly at a predetermined value or higher, and thus the swash plate 12 (or the discharge displacement of the compressor) responds rapidly to changes.

[0061] The embodiment of Figs 1 and 2 may be modified as follows.

[0062] The second valve chamber 71, or the pressure detecting chamber 71, may be constructed as shown in Figures 3 or 4. In Figure 3, the length of the pressure detecting rod 78 is shorter than in Figure 2. Further, the closing spring 75 is located between the wall of the second valve chamber 72 and a spring seat 73a formed at the head of the second valve element 73. Here, the pressure detecting rod 78 is loosely fitted in the connecting cylinder 77, as in the control valve 50

shown in Figure 2.

[0063] In Figure 4, the closing spring 75 is located between the wall of the valve chamber 71 and the upper end of the bellows 76. Further, the second valve element 73 is fixed to the other end of the bellows 76. If the construction of the second valve chamber 71 in the control valve 50 shown in Figure 2 is changed as shown in Figure 3 or 4, the operation of the control valve 50 is the same.

[0064] The arrangement of the second rod 74 and that of the first rod 65 in the control valve shown in Figure 2 may be reversed. That is, the second rod 74 may be formed to have tubular structure, and the first rod 65 may be fitted inside the second rod 74 such that the rods are movable relative to each other.

[0065] The pressure detector employed in the second valve mechanism 70 is limited to the bellows but may be, for example, a diaphragm.

[0066] In the embodiments shown in Figures 2 and 3, the pressure detecting rod 78 and the connecting cylinder 77 may be fixed to each other.

[0067] The displacement control valve 50 may be applied to a clutchless variable displacement compressor (constantly driven compressor).

[0068] The present examples and embodiments are to be considered as illustrative and not restrictive, and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.

[0069] A displacement control valve adjusts the pressure in a crank chamber (5) of a compressor to vary the displacement of the compressor. The control valve has a first valve mechanism (60), a second valve mechanism (70) and a solenoid mechanism (80). The solenoid mechanism (80) independently actuates the first and second valve mechanisms (60, 70). The first valve mechanism (60) selectively opens and closes a supply passage (28) of the compressor. The second valve mechanism (70) adjusts the flow rate of gas through a bleed passage (27) of the compressor. The second valve mechanism (70) includes a second valve opening (72), a second valve element (73), a bellows (76) and a second plunger (83). The second valve element (73) and the bellows (76) are accommodated in a single pressure chamber (71). The pressure of a suction chamber is introduced to the pressure chamber (71). The control valve has a relatively simple structure and is thus inexpensive and compact.

Claims

1. A control valve for a variable displacement compressor, wherein the control valve adjusts the pressure in a crank chamber (5) of the compressor to vary the compressor displacement, wherein the compressor includes a suction pressure zone (21), the pressure of which is a suction pressure, a discharge pressure zone (22), the pressure of which is

a discharge pressure, a bleed passage (27) for connecting the crank chamber (5) to the suction pressure zone (21), and a supply passage (28) for connecting the crank chamber (5) to the discharge pressure zone (22), the control valve comprising:

a housing (51);
a first valve mechanism (60) retained in the housing (51) to selectively open and close the supply passage (28), wherein the first valve mechanism (60) includes:

a first valve opening (63) defined in the housing (51), the first valve opening (63) forming part of the supply passage (28);
a first valve element (64) for selectively opening and closing the first valve opening (63); and
a first plunger (82) connected to the first valve element (64);

a second valve mechanism (70) retained in the housing (51) to adjust the flow rate of gas released from the crank chamber (5) to the suction pressure zone (21) through the bleed passage (27), wherein the second valve mechanism (70) includes:

a second valve opening (72) defined in the housing (51), the second valve opening (72) forming part of the bleed passage (27);
a second valve element (73) for adjusting the opening size of the second valve opening (72);
a pressure sensing member (76) for moving the second valve element (73) in accordance with the suction pressure; and
a second plunger (83) connected to the second valve element (73), wherein the first valve mechanism (60) and the second valve mechanism (70) operate independently;

a solenoid mechanism (80) retained in the housing (51) to independently actuate the first valve mechanism (60) and the second valve mechanism (70), wherein the solenoid mechanism (80) includes a coil (85), and current supplied to the coil (85) produces an electromagnetic force for independently biasing the first and second plungers (82, 83) in accordance with the level of the current, the control valve being **characterized by**:

a pressure chamber (71) defined in the housing (51) to accommodate the second valve element (73) and the pressure sensing member (76), wherein the pressure chamber (71) is

exposed to the suction pressure.

2. The control valve according to claim 1 **characterized in that** the first valve mechanism (60) includes an opening spring (66) for urging the first valve element (64) away from the first valve opening (63), and the second valve mechanism (70) includes a closing spring (75) for urging the second valve element (73) toward the second valve opening (72), and wherein, when current is supplied to the coil (85), the first plunger (82) causes the first valve element (64) to close the first valve opening (63) against the force of the opening spring (66) and the second plunger (83) urges the second valve element (73) away from the second valve opening (72) by a force corresponding to the level of the current.
3. The control valve according to claims 1 or 2 **characterized in that** the pressure chamber (71) is located in the bleed passage (27) and the second valve opening (72) opens to the pressure chamber (71).
4. The control valve according to any one of claims 1 to 3 **characterized by** a first rod (64) coupling the first valve element (64) with the first plunger (82) and a second rod (74) coupling the second valve element (73) with the second plunger (83), wherein one of the first rod (64) and the second rod (74) is fitted in the other, such that the first and second rods (65, 74) are axially movable relative to one another.
5. The control valve according to claim 4 **characterized in that** the first and second valve openings (63, 72) are located at the ends of a linear passage (62) defined in the housing (51), and wherein the second rod (74) is fitted in the first rod (64) and extends through the linear passage (62).
6. The control valve according to any one of claims 1 to 5 **characterized in that** the second valve element (73) is coupled to the pressure sensing member (76) such that the second valve element (73) is movable relative to the pressure sensing member (76).
7. The control valve according to any one of claims 1 to 3 **characterized by** a first valve chamber (61) located in the supply passage (28) to accommodate the first valve element (64) and a first rod (64) that connects the first valve element (64) with the first plunger (82), wherein the first valve opening (63) opens to the first valve chamber (61) and is located at an opposite side of the first valve element (64) from the first rod (64), and wherein the cross sectional area of the first rod (64) is substantially equal to the cross sectional area of the first valve opening

(63).

8. The control valve according to claim 7 **characterized in that** the first valve opening (63) is connected to the crank chamber (5) by the supply passage (28) and wherein the first valve chamber (61) is connected to the discharge pressure zone (22) by the supply passage (28).
9. The control valve according to claim 8 **characterized by** a solenoid chamber (81) to accommodate the first and second plungers (82, 83) and a pressure application passage (57) for applying the pressure in the crank chamber (5) to the solenoid chamber (81).
10. A compressor having the control valve according to any one of claims 1 to 9.
11. The compressor according to claim 10 **characterized by** an auxiliary supply passage (29) connecting the crank chamber (5) with the discharge pressure zone (22), wherein the auxiliary supply passage (29) includes a fixed restriction (29a).
12. The compressor according to claims 10 or 11 **characterized in that** a downstream part of the supply passage (28) also serves as an upstream part of the bleed passage (27).
13. The compressor according to any one of claims 10 to 12 **characterized in that** the compressor is coupled to an external power source (E), and a clutch (40) is located between the external power source (E) and the compressor to selectively engage the external power source (E) with the compressor.

Fig. 1

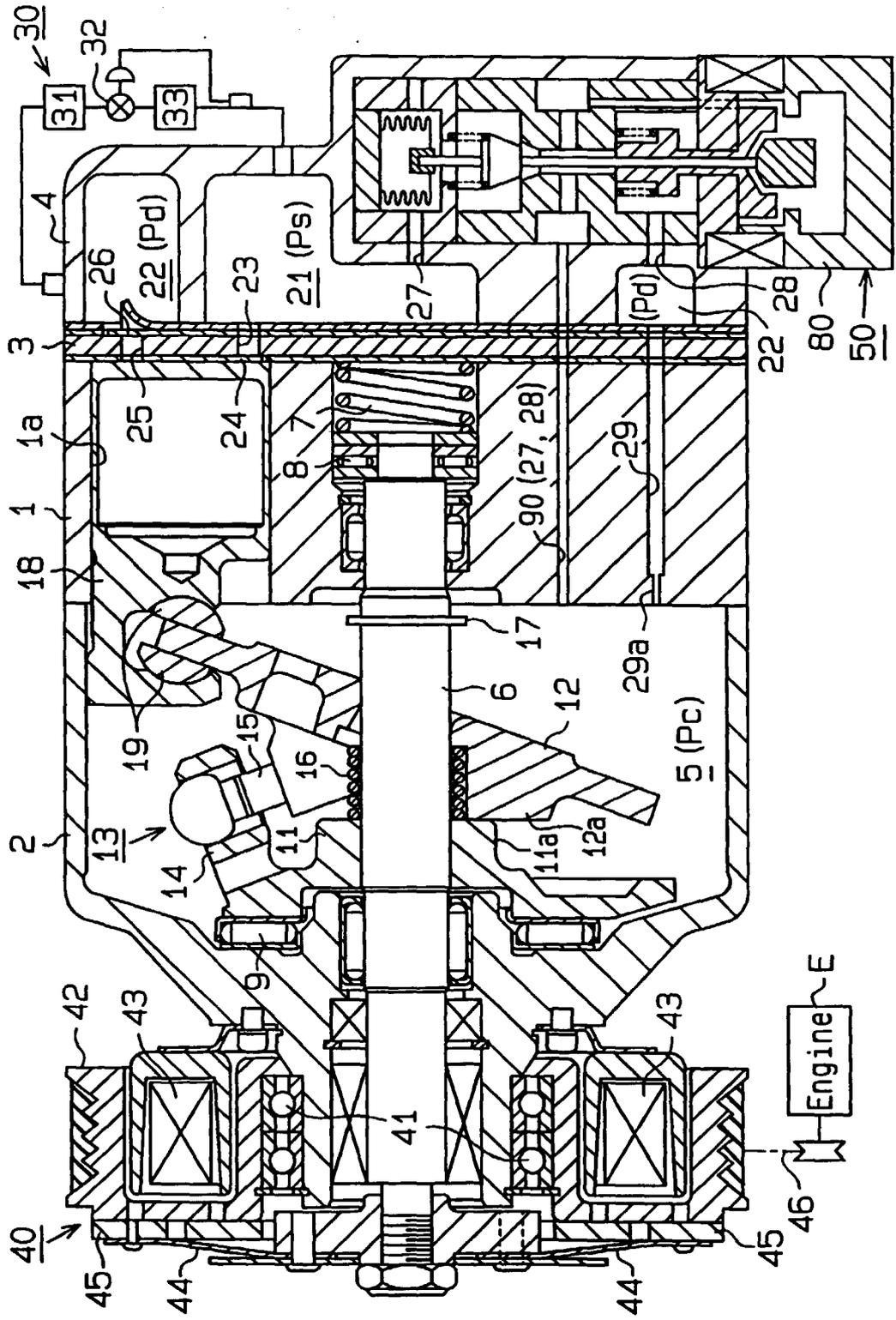


Fig. 2

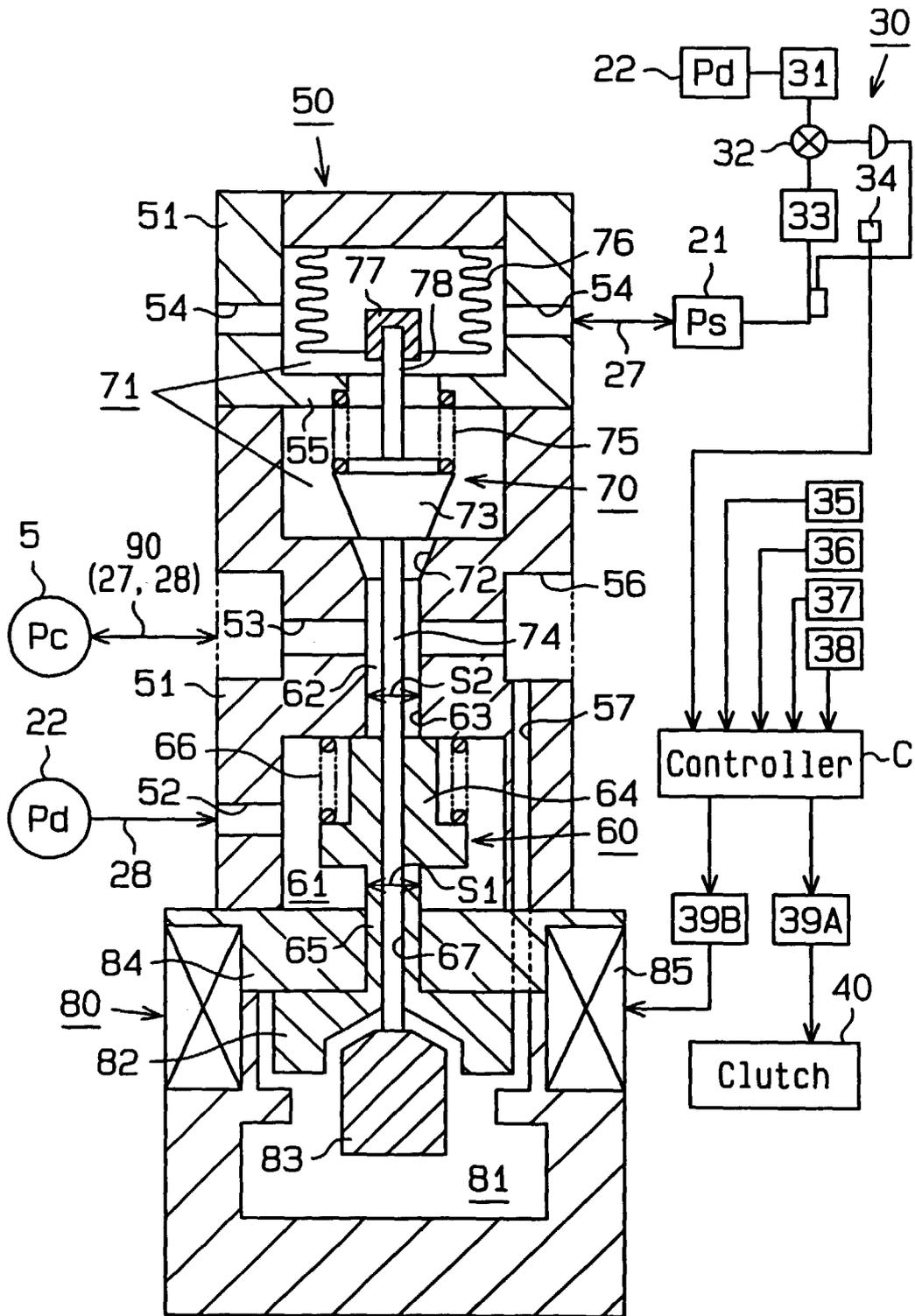


Fig.3

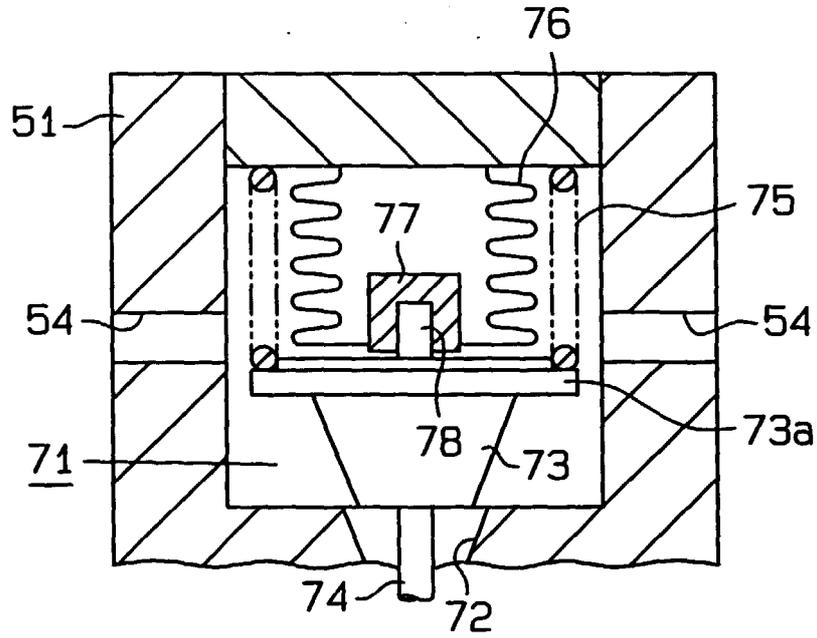


Fig.4

