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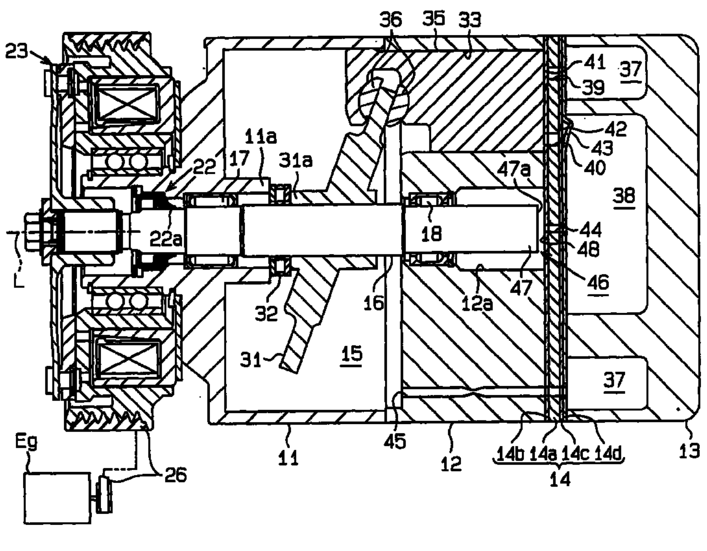
(54) Crankcase pressure control for swash plate compressor

(57) A piston-type compressor includes a crank chamber (15), a drive shaft (16), cylinder bores (33), and a valve plate (14). The drive shaft (16) passes through the crank chamber (15), and the valve plate is fixed to the rear of the cylinder bores (33). The drive shaft (16) includes a valve body (47). The rear end surface (47a) of the valve body (47) and the front surface of the valve plate (14) form a valve mechanism (46). The valve mechanism (46) is located in the pressurizing passage (18, 12a, 44). The valve mechanism (46) adjusts the opening size of the pressurizing passage (18, 12a, 44) in accordance with the axial movement of the drive shaft (16) from a reference position. This maintains an appropriate relationship between the pressure in the cylinder bores (33) and the pressure in the crank chamber (15).

Fig.1



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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a piston-type compressor that compresses refrigerant gas for air-conditioning vehicles.

[0002] Fig. 7 shows a typical piston-type compressor. The compressor of Fig. 7 has a housing 101, which includes a crank chamber 102 and supports a drive shaft 103. A lip seal 104, which is located between the housing 101 and the drive shaft 103, seals the drive shaft 103. A swash plate 105 is fixed to the drive shaft 103 in the crank chamber 102. A thrust bearing 106 is located between the front (left in Fig. 7) end of the swash plate 105 and an inner surface of the housing 101.

[0003] The housing 101 further includes cylinder bores 107, a suction chamber 108, and a discharge chamber 109. A piston 110 is accommodated in each cylinder bore 107. The pistons 110 are coupled to the swash plate 105. A valve plate 111, which is located in the housing 101, includes suction ports 111a, suction valves 111b, discharge ports 111c, and discharge valves 111d. The valve plate 111 separates the cylinder bores 107 from the suction chamber 108 and the discharge chamber 109.

[0004] Rotation of the drive shaft 103 is converted into reciprocation of the pistons 110 by the swash plate 105. When the pistons 110 reciprocate, refrigerant gas is drawn to the cylinder bores 107 from the suction chamber 108 through the corresponding suction ports 111a and suction valves 111b. Then, the refrigerant gas is compressed and discharged to the discharge chamber 109 through the corresponding discharge ports 111c and the discharge valves 111d. This cycle of compression is repeated while the pistons 110 reciprocate.

[0005] The intermittent compression of refrigerant gas applies a thrust load to the drive shaft 103 through the pistons 110 and the swash plate 105. The thrust load is applied frontward (leftward in Fig. 7) in the axial direction of the drive shaft 103. The thrust load is varied by the pressure in the cylinder bores 107. The thrust bearing 106 receives the thrust load between the swash plate 105 and the housing 101. The greater the peak of the thrust load is, the more durable the thrust bearing 106 must be. To improve the durability of the thrust bearing 106, it is necessary to increase the size of the thrust bearing 106, which increases the size of the compressor.

[0006] The load applied to the thrust bearing 106 can be reduced as follows. High-pressure refrigerant gas is supplied to the crank chamber 102 by blowby gas from the cylinder bores 107. The blowby gas is the main source of the gas that increases the pressure in the crank chamber 102. The pressure in the crank chamber 102 applies a rearward (rightward in Fig. 7) thrust load to the drive shaft 103 through the pistons 110 and the swash plate 105. That is, the thrust load based on the pressure in the crank chamber 102 counters the thrust load based on the pressure in the cylinder bores 107. Accordingly, an increase of blowby gas reduces the load applied to the thrust bearing 106.

[0007] The relationship between the pressure in the cylinder bores 107 and the pressure in the crank chamber 102 is continuously varied in accordance with the compressor operation state. The relationship between the two pressures varies, tor example, when the suction pressure changes in accordance with the cooling load, the discharge pressure changes in accordance with the change of condensation capacity in the refrigeration circuit connected to the compressor, and when the rotation speed of the drive shaft 103 changes. Accordingly, the average thrust applied to the thrust bearing 106 continuously varies.

[0008] When the thrust load applied to the thrust bearing 106 is zero, the thrust bearing 106 is retained at a predetermined position. When a frontward thrust load greater than zero is applied to the drive shaft 103, the thrust bearing 106 is pressed frontward. When a rearward thrust load (less than zero) is applied, a rearward force is applied to the drive shaft 103 and the thrust bearing 106. In this way, the drive shaft 103 is moved frontward or rearward in accordance with the magnitude of the thrust load.

[0009] In the compressor of Fig. 7, there is no structure for positively controlling the pressure in the crank chamber 102. Therefore, it is impossible to set and maintain the average thrust load applied to the thrust bearing 106 to zero. When a frontward average thrust load greater than zero is applied, the thrust bearing 106 is pressed against the inner wall of the housing and receives a great load, which shortens the life of the thrust bearing 106. When a rearward average thrust load (less than zero) is applied, the following problems are caused.

(1) The thrust bearing 106 does not limit the movement of the drive shaft 103. Accordingly, the drive shaft 103 moves rearward along the axis L. During the movement, a lip seal 104 may be greatly offset from a predetermined contact position with respect to the drive shaft 103. Since the surface of the drive shaft 103 is usually covered with sludge, the lip seal 104 may contact the sludge, which reduces the effectiveness of the lip seal 104 and permits gas to leak from the crank chamber 102.

(2) When the drive shaft 103 moves axially rearward, the pistons 110, which are coupled to the drive shaft 103 by the swash plate 105, move rearward in the corresponding cylinder bores 107. This moves the top and bottom dead centers of the pistons 110 toward the valve plate 111, which causes the pistons 110 to abut against the valve plate 111. The abutment causes noise and vibration, which

may damage the pistons 110 and the valve plate 111.

[0010] To solve the above problems, there is a need for an urging member such as a spring 112, which urges the drive shaft 103 frontward along the axis L and which is represented by the broken line of Fig. 7. This increases the number of parts and costs. The spring 112 is provided under the assumption that a rearward average thrust load is applied to the thrust bearing 106. A frontward average thrust load further shortens the life of the thrust bearing 106, which complicates the maintenance of the compressor.

SUMMARY OF THE INVENTION

[0011] An objective of the present invention is to provide a piston-type compressor that facilitates the maintenance.

[0012] To achieve the above objective, the present invention provides a piston-type compressor structured as follows. A housing includes a crank chamber and a cylinder bore. A drive shaft passes through the crank chamber and is supported by the housing to rotate. A cam plate is located in the crank chamber and is coupled to the driver shaft. The cam plate integrally rotates with the drive shaft. A piston is located in the cylinder bore. The piston is coupled to the cam plate, and rotation of the drive shaft is converted into reciprocation of the piston by the cam plate, which compresses gas in the cylinder bore. A pressurizing passage connects the crank chamber with a discharge pressure area. A bleed passage connects the crank chamber with a suction pressure area. A valve mechanism regulates the opening size of at least one of the pressurizing passage and the bleed passage in accordance with axial movement of the drive shaft.

[0013] Other aspects and advantages of the present 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

[0014] The features of the present invention that are believed to be novel are set forth with particularity in the appended claims. The invention, together with 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:

Fig. 1 is a cross-sectional view of a piston-type compressor according to a first embodiment of the present invention;

Fig. 2 is a partial enlarged cross-sectional view of the compressor of Fig. 1;

Fig. 3 shows a thrust load applied to a thrust bearing in accordance with the operation of the compressor;

Fig. 4 is a cross-sectional view of a piston-type compressor according to a second embodiment of the present invention;

Fig. 5 is a cross-sectional view of a piston-type compressor according to a third embodiment of the present invention;

Fig. 6 is a cross-sectional view of the compressor of Fig. 5 when the swash plate is the minimum displacement position;

Fig. 7 is a prior art cross-sectional view of a fixed displacement piston-type compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0015] First to third embodiments of the present invention will now be described. In the description of the second and third embodiments, members similar to those of the first embodiment have the same reference numbers and the description will focus on the differences from the first embodiment. In Fig. 1, the left end of the compressor is the front end of the compressor, and the right end the rear end of the compressor.

[0016] As shown in Fig. 1, a piston-type compressor of the present invention has a housing, which includes a front housing member 11, a cylinder block 12, and a rear housing member 13. The front housing member 11 is coupled to the front end of the cylinder block 12. The cylinder block 12 serves as a middle housing member. The rear housing member 13 is coupled to the rear end of the cylinder block 12 through a valve plate 14. The valve plate 14 includes a main plate 14a, a first sub-plate 14b, a second sub-plate 14c, and a retainer plate 14d. The first sub-plate 14b is located on the front surface of the main plate 14a, the second sub-plate 14c is located on the rear surface of the main plate 14a, and the retainer plate 14d is located on the rear surface of the second sub-plate 14c

[0017] A crank chamber 15 is defined by the front housing member 11 and the cylinder block 12. A drive shaft 16 is supported by the front housing member 11 and the cylinder block 12 and passes through the crank chamber 15.

[0018] The front end of the drive shaft 16 is supported by a radial bearing 17. The rear end of the drive shaft 16 is located in a central chamber 12a and is supported by a radial bearing 18, which is a roller bearing. The rear opening of the central chamber 12a is closed by the valve plate 14.

[0019] The front end of the drive shaft 16 extends through the front housing member 11 to the exterior. A lip seal 22 seals the drive shaft 16. The lip seal 22 is located between the front end of the drive shaft 16 and the front housing member 11. The lip seal 22 includes a lip ring 22a. When the lip ring 22a is pressed against the surface of the drive shaft 16, the drive shaft 16 is sealed.

[0020] An electromagnetic friction clutch 23 is located between a vehicle engine Eg, which is an external drive source, and the drive shaft 16. When the clutch 23 is engaged, power of the engine Eg is transmitted to the drive shaft 16 through a belt 26 and the clutch 23. When the clutch 23 is disengaged, the transmission of power from the engine Eg to the drive shaft 16 is stopped.

[0021] A cam plate, which is a swash plate 31 in this embodiment, is located in the crank chamber 15. A boss 31a is integrally formed at the center of the swash plate 31. The boss 31a is fixed to the drive shaft 16. A bearing seat 11a is formed on the inner surface of the front housing member 11 and extends into the crank chamber 15. The bearing seat 11a surrounds the drive shaft 16. A thrust bearing 32 is retained between the front end of the boss 31a and the distal end of the bearing seat 11a.

[0022] Cylinder bores 33 (only one shown) are formed in the cylinder block 12. A single head piston 35 is accommodated in each cylinder bore 33. The front of each cylinder bore 33 is closed by the corresponding piston 35, and the rear of each cylinder bore 33 is closed by the valve plate 14. Each piston 35 is coupled to the periphery of the swash plate 31 through the shoes 36. Rotation of the drive shaft 16 is converted into reciprocation of the pistons 35 by the swash plate 31 and the shoes 36.

[0023] A suction pressure area, which is a suction chamber 37 in this embodiment, is formed in the periphery of the rear housing member 13. A discharge pressure area, which is a discharge chamber 38 in this embodiment, is formed near the center of the rear housing member 13. The suction chamber 37 and the discharge chamber 38 are separated from the cylinder bores 33 by the valve plate 14. Suction ports 39 and discharge ports 40 are formed in the main plate 14a of the valve plate 14 to correspond to the cylinder bores 33. Suction valves 41 are formed in the first sub-plate 14b to correspond to the suction ports 39. Discharge valves 42 are formed in the second sub-plate 14c to correspond to the discharge ports 40. Retainers 43 are formed in the retainer plate 14d to correspond to the discharge valves 42. The retainers 43 determine the maximum opening size of the discharge valves 42.

[0024] When each piston 35 moves from its top dead center to its bottom dead center, refrigerant gas in the suction chamber 37 is drawn to the corresponding cylinder bore 33 through the corresponding suction port 39 and suction valve 41. When each piston 35 moves from its bottom dead center to its top dead center, refrigerant gas in the corresponding cylinder bore 33 is compressed to a certain level and is discharged to the discharge chamber 38 through the corresponding discharge port 40 and discharge valve 42.

[0025] As shown in Fig. 1, the central chamber 12a is connected to the crank chamber 15 through space in the rollers of the radial bearing 18. A through hole 44, which is formed at the center of the valve plate 14, connects the central chamber 12a with the discharge chamber 38. The space of the radial bearing 18, the central chamber 12a, and the through hole 44 form a pressurizing passage. Accordingly, the discharge chamber 38 is connected to the crank chamber 15 through the pressurizing passage. A bleed passage 45 connects the crank chamber 15 with the suction chamber 37. The bleed passage 45 includes a fixed restriction valve.

[0026] A valve mechanism 46 is located in the pressurizing passage. The valve mechanism 46 includes a valve body 47 (and the rear end surface 47a of the valve body 47) and the front surface 48 of the valve plate 14. The valve mechanism 46 adjusts the opening size of the pressurizing passage in accordance with the movement of the drive shaft 16 along the axis L with respect to a reference position. The valve mechanism 46 will now be described in detail.

[0027] As shown in Fig. 2, the rear end of the drive shaft 16 is located in the vicinity of the valve plate 14 in the central chamber 12a. That is, the rear end of the drive shaft 16 is longer than that of the prior art drive shaft 103 of Fig. 7. The rear end of the drive shaft 16 serves as a valve body 47. The first sub-plate 14b is exposed to the central chamber 12a. The rear end surface 47a of the valve body 47 faces the front surface 48 of the valve plate 14 (the first sub-plate 14b). The through hole 44 is located at the center of the front surface 48 (on axis L) and is open at the central chamber 12a. The rear end surface 47a and the front surface 48 form a valve to control the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15.

[0028] As shown in Fig. 2, lubricant films 49 are formed on the rear end surface 47a and on the front surface 48. The lubricant films 49 are made of a solid lubricant including fluororesin such as polytetrafluoroethylene.

[0029] While refrigerant gas is intermittently compressed, a thrust load is applied to the drive shaft 16 through the pistons 35 and the swash plate 31. The thrust load urges the drive shaft frontward in the direction of the axis L. The thrust load is varied by the pressure in the cylinder bores 33. The thrust load is received by the thrust bearing 32 between the swash plate 31 and the front housing member 11.

[0030] High-pressure refrigerant gas is supplied to the crank chamber 15 as blowby gas from the cylinder bores 33 and through the pressurizing passage. Refrigerant gas in the crank chamber 15 continuously flows to the suction chamber 37 through the bleed passage 45.

The relationship between the supply of high-pressure refrigerant gas to the crank chamber 15 and the release of high-pressure refrigerant gas from the crank chamber 15 determines the pressure in the crank chamber 15. The pressure in the crank chamber 15 applies a thrust load to the drive shaft 16 through the pistons 35 and the swash plate 31. The thrust load is applied in the rearward axial direction. That is, the thrust load based on the pressure in the crank chamber 15 counters the thrust load based on the pressure in the cylinder bores 33. Accordingly, the thrust load applied to the thrust bearing 32 based on the pressure in the cylinder bores 33 is reduced by the thrust load based on the pressure in the crank chamber 15.

[0031] As shown in Fig. 2, the valve body 47 adjusts the opening size of the pressurizing passage, that is, the valve body 47 adjusts the pressure in the crank chamber 15, such that the average thrust load applied to the thrust bearing 32 is zero during a predetermined operation of the compressor. In the predetermined operation, the suction pressure, the discharge pressure, and the rotation speed of the drive shaft 16 are respectively set to predetermined values. The rear end surface 47a of the valve body 47 is spaced from the front surface 48 of the valve plate 14 by 0.1-0.5mm. This dimension is exaggerated in the drawings. When the predetermined values change, the pressure in the cylinder bores 33 is quickly varied, and the average thrust load applied to the thrust bearing 32 becomes frontward or rearward.

[0032] As shown by the broken line X1 of Fig. 3, when the average thrust load applied to the thrust bearing 32 is rearward, the drive shaft 16 moves rearward (rightward in Fig. 1). When the drive shaft 16 moves rearward as shown by a broken line of Fig. 2, the valve body 47 is placed closer to the valve plate 14. Accordingly, the space between the rear end surface 47a of the valve body 47 and the front surface 48 of the valve plate 14 is reduced. This reduces the passage area between the central chamber 12a and the through hole 44, that is, this reduces the opening size of the pressurizing passage.

[0033] As the opening size of the pressurizing passage is reduced, the flow rate of high-pressure refrigerant gas supplied from the discharge chamber 38 to the crank chamber 15 decreases. Accordingly, the pressure in the crank chamber 15 is reduced, which reduces the thrust load applied to the drive shaft 16. As a result, the average thrust load applied to the thrust bearing 32 is returned to zero.

[0034] As shown by the broken line X2 of Fig. 3, when a frontward average thrust load that is greater than zero is applied to the thrust bearing 32, the drive shaft 16 moves frontward. As shown by a broken line of Fig. 2, the valve body 47 is separated from the valve plate 14 when the drive shaft 16 moves frontward. Accordingly, the space between the rear end surface 47a of the valve body 47 and the front surface 48 of the valve plate 14 increases, which increases the opening size of the pressurizing passage.

[0035] When the opening size of the pressurizing passage is increased, the supply of refrigerant gas from the discharge chamber 38 to the crank chamber 15 through the pressurizing passage increases. Accordingly, the pressure in the crank chamber 15 increases, which increases the thrust load applied to the drive shaft 16. As a result, the average thrust load applied to the thrust bearing 32 is returned to zero.

[0036] The valve mechanism 46 adjusts the pressure in the crank chamber 15 in accordance with the movement of the drive shaft 16 such that a desirable relationship between the pressure in the cylinder bores 33 and the pressure in the crank chamber 15 is maintained. That is, the valve mechanism 46 operates to nullify the average thrust load applied to the thrust bearing 32. This extends the life of the thrust bearing 32 and prevents power loss.

[0037] The rearward movement of the drive shaft 16 is limited by the abutment of its rear end against the valve plate 14. This prevents the lip seal 22 from being greatly moved from its original position and also prevents the pistons 35 from contacting the valve plate 14.

[0038] If the rear end of the drive shaft 16 is continuously pressed against the valve plate 14, friction occurs between the valve plate 14 and the drive shaft 16, which may damage parts and increase power loss. However, in the present embodiment, when the rear end of the drive shaft 16 contacts the valve plate 14, the pressurizing passage is completely closed. Accordingly, the pressure in the crank chamber 15 is reduced, which returns the average thrust load applied to the thrust bearing 32 to zero. This prevents the rear end of the drive shaft 16 from being pressed against the valve plate 14.

[0039] In contrast, in the prior art compressor of Fig. 7, if the rear end of the drive shaft 103 is extended toward the valve plate 111, as in the first embodiment of the present invention, the lip seal 104 is prevented from moving greatly from its initial position. Also, the pistons 110 are prevented from contacting the valve plate 111. However, the compressor of Fig. 7 does not have the valve mechanism 46, and the pressure in the crank chamber 102 is not positively varied. Accordingly, the rear end of the drive shaft 103 is continuously pressed against the valve plate 111, which causes friction and power loss.

[0040] The valve mechanism 46 opens and closes the pressurizing passage. That is, the valve mechanism 46 uses high-pressure refrigerant gas to adjust the pressure in the crank chamber 15. This increases the speed of changing the pressure in the crank chamber 15 compared to controlling the bleed passage 45 only.

[0041] The valve body 47 is integrally formed with the drive shaft 16. That is, the valve mechanism 46 adjusts the opening size of the pressurizing passage in accordance with the movement of the drive shaft 16.

Accordingly, there is no need for electrical elements such as an electromagnetic portion 53b, a sensor 53, and a computer C, which will be shown in a second embodiment. Therefore, the structure of the valve mechanism 46 is simple.

[0042] The valve body 47 is the extended rear end of the drive shaft 16. Accordingly, compared to forming an independent valve body and coupling the valve body to the drive shaft 16, the number of parts is reduced.

[0043] The lubricant films 49 are formed on the rear end surface 47a and on the front surface 48 of the valve plate 14. This contributes to preventing friction and power loss due to the contact between the drive shaft 16 and the valve plate 14.

[0044] Fig. 4 shows a second embodiment of the present invention. A valve mechanism 51 of the second embodiment is different from that of the first embodiment. A pressurizing passage 54 directly connects the discharge chamber 38 to the crank chamber 15 without passing through the central chamber 12a. In the second embodiment, the drive shaft 16 does not serve as a valve body. The valve mechanism 51 includes an external control valve 52, a sensor 53, and a valve controller. The control valve 52, which is an electromagnetic valve, includes a valve body 52a and an electromagnetic portion 52b. The valve body 52a opens and closes the pressurizing passage 54, and the electromagnetic portion 52b drives the valve body 52a. The sensor 53 is located in the central chamber 12a to face the drive shaft 16. The sensor 53 detects the amount of movement of the drive shaft 16 from a certain reference position. The valve controller, which is a computer C, controls the supply of electric current to the electromagnetic portion 52b, which operates the valve body 52a and adjusts the opening size of the pressurizing passage 54.

[0045] The valve mechanism 51 operates to nullify the average thrust load applied to the thrust bearing 32. Accordingly, the load on the thrust bearing 32 is reduced. Since the valve mechanism 51 opens and closes the pressurizing passage, the pressure in the crank chamber 15 is quickly changed. Further, the valve mechanism 51 adjusts the opening size of the pressurizing passage 54 from the exterior of the pressurizing passage 54. In this embodiment, the location of the pressurizing passage 54 does not depend on the location of the drive shaft 16.

[0046] On behalf of the control valve 52 or in combination with the control valve 52, an external control valve 52c may also be provided, as shown by the imaginary line in Fig. 4. The control valve 52c includes a valve body 52d and an electromagnetic portion 52e. The valve body 52d opens and closes the bleed passage 45, and the electromagnetic portion 52e drives the valve body 52a. As described above, the computer C controls the supply of electric current to the electromagnetic portion 52e, which operates the valve body 52d and adjusts the opening size of the bleed passage 45.

[0047] A third embodiment of the present invention will now be described with reference to Figs. 5 and 6. The third embodiment is different in that the compressor displacement is variable. As shown in Figs. 5 and 6, a lug plate 56 is fixed to the drive shaft 16 in the crank chamber 15. The swash plate 31 is supported by the drive shaft 16. The swash plate 31 inclines with respect to the drive shaft 16 and moves along the surface of the drive shaft 16. A hinge mechanism 57 is located between the swash plate 31 and the lug plate 56. The hinge mechanism 57 permits the swash plate 31 to rotate integrally with the drive shaft 16 and to vary its inclination. The thrust bearing 32 is located between the inner surface of the front housing member 11 and the lug plate 56.

[0048] A snap ring 58 is fixed to the drive shaft 16 between the swash plate 31 and the cylinder block 12. As shown in Fig. 6, the minimum inclination of the swash plate 31 is determined by the abutment of the swash plate 31 against the ring 58.

[0049] A pressure control passage 59 connects the discharge chamber 38 with the crank chamber 15 to control the compressor displacement. A displacement control valve 60, which is an electromagnetic valve, is located in the pressure control passage 59. The displacement control valve 60 varies the opening size of the control passage 59, which adjusts the flow rate of high-pressure refrigerant gas from the discharge chamber 38 to the crank chamber 15. As already mentioned, the relationship between the flow rate of refrigerant gas from the discharge chamber 38 to the crank chamber 15 as blowby gas and through the pressurizing passage and the amount of refrigerant gas released from the crank chamber 15 to the suction chamber 37 through the bleed passage 45 varies the pressure in the crank chamber 15. Accordingly, the difference of the pressure in the crank chamber 15 and the pressure in the cylinder bores 33 is varied, which varies the inclination of the swash plate 31. As a result, the stroke of the pistons 35 is varied, which adjusts the compressor displacement.

[0050] The control valve 60, which is an external valve, is able to control the compressor displacement regardless of the cooling load. For example, when accelerating the vehicle, it is necessary to minimize the displacement and reduce the load applied to the engine Eg. When the cooling load is great (when the compressor is operating at the maximum displacement), the control valve 60 quickly maximizes the opening size of the control passage 59 from the closed position. Then, refrigerant gas in the discharge chamber 38 quickly flows to the crank chamber 15. Since the release of refrigerant gas through the bleed passage 45 is relatively slow, the pressure in the crank chamber increases. This dramatically increases the difference between the pressure in the crank chamber 15 and that in the cylinder bores 33, which minimizes the inclination of the swash plate 31 and the displacement. As a result, the swash plate 31 may be pressed against the ring 58 and may strongly pull the lug plate 56 through the hinge mechanism 57. Accordingly, the drive shaft 16 may move rearward by a significant distance.

[0051] However, as in the first and second embodiments, the rearward movement of the drive shaft 16 is limited by the abutment of the drive shaft against the valve plate 14. This prevents the lip seal 22 from moving significantly from the initial position on the drive shaft 16 and also prevents the pistons 35 from contacting the valve plate 14. When the pressure in the crank chamber 15 is reduced by the valve mechanism 46, the drive shaft 16 is quickly separated from the valve plate 14.

[0052] If the pressurizing passage is completely closed, the pressure in the crank chamber 15 may decrease more than necessary. Then, the drive shaft 16 is separated from the valve plate 14, and the swash plate 31 is separated from the ring 58.

[0053] However, the valve mechanism 46 increases the opening size of the pressurizing passage in accordance with the frontward movement of the drive shaft 16. Accordingly, the reduction of the pressure in the crank chamber 15 is appropriately controlled. Therefore, the swash plate 31 remains inclined at the minimum inclination, which maintains the minimum displacement of the compressor.

[0054] The present invention can further be embodied as follows.

[0055] The valve mechanism 46, 51 may open and close only the bleed passage 45. In this case, when the drive shaft 16 moves rearward, the valve mechanism 46, 51 increases the opening size of the bleed passage 45. When the drive shaft 16 moves frontward, the valve mechanism 46, 51 reduces the opening size of the bleed passage 45.

[0056] The valve mechanism 46, 51 may open and close both the pressurizing passage (18, 12a, 44), 54 and the bleed passage 45.

[0057] The thrust bearing 32 may be a slide bearing instead of a roller bearing. Since the load applied to the thrust bearing 32 is reduced in the present invention, the capacity of the thrust bearing 32 can be reduced. Therefore, though the capacity of the slide bearing is smaller than the roller bearing, the slide bearing can be used without increasing the size of the thrust bearing 32.

[0058] Further, the thrust bearing 32 may be omitted. For example, solid lubricant films may be formed on an inner surface of the front housing member 11 (or the front end surface of the bearing seat 11a) and the swash plate 31 (or the front end surface of the boss 31a, the lug plate 56 in the third embodiment). The solid lubricant films can solve the problem of friction between the front housing member 11 and the swash plate 31 (or the lug plate 56) without the thrust bearing 32.

[0059] In any of the first and second embodiments, the valve mechanism 46, 51 may maintain either a certain frontward or a certain rearward average thrust load applied to the thrust bearing 32.

[0060] It should be apparent to those skilled in the art that the present invention may be embodied in many other specific forms without departing from the spirit or scope of the invention. Therefore, 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.

[0061] A piston-type compressor includes a crank chamber (15), a drive shaft (16), cylinder bores (33), and a valve plate (14). The drive shaft (16) passes through the crank chamber (15), and the valve plate is fixed to the rear of the cylinder bores (33). The drive shaft (16) includes a valve body (47). The rear end surface (47a) of the valve body (47) and the front surface of the valve plate (14) form a valve mechanism (46). The valve mechanism (46) is located in the pressurizing passage (18, 12a, 44). The valve mechanism (46) adjusts the opening size of the pressurizing passage (18, 12a, 44) in accordance with the axial movement of the drive shaft (16) from a reference position. This maintains an appropriate relationship between the pressure in the cylinder bores (33) and the pressure in the crank chamber (15).

Claims

1. A piston-type compressor, wherein a drive shaft (16) is rotatably supported by a housing (11, 12, 13) which includes a crank chamber (15) and a cylinder bore (33) so that the drive shaft (16) passes through the crank chamber (15), wherein a cam plate (31) is located in the crank chamber (15) and is coupled to the drive shaft (16) so that the cam plate (31) integrally rotates with the drive shaft (16), wherein a plurality of pistons (35), which are located in the cylinder bore (33), are coupled to the cam plate (31) and rotation of the drive shaft (16) is converted into reciprocation of the pistons (35), which compresses gas in the cylinder bore (33), wherein pressure in the crank chamber (15) is regulated using a pressurizing passage (18, 12a, 44, 54) which connects the crank chamber (15) with a discharge pressure area (38) and a bleed passage (45) which connects the crank chamber (15) with a suction pressure area (37), the compressor being characterized by

a valve mechanism (46, 54) which regulates the opening size of at least one of the pressurizing passage (18, 12a, 44, 54) and bleed passage (45) in accordance with axial movement of the drive shaft (16).

2. The compressor according to claim 1 characterized in that the compressor further including a valve plate (14), which separates the cylinder bore (33) from the discharge pressure area (38) and the suction pressure area (37), wherein the valve mechanism (46, 51) adjusts the opening size of at least one of the pressurizing passage (18, 12a, 44, 54) and bleed passage (45) to lower the pressure in the crank chamber (15) when the drive shaft (16) and the pistons (35) move toward the valve plate (14), wherein the valve mechanism (46, 51) adjusts the opening size of at least one of the pressurizing passage (18, 12a, 44, 54) and the bleed passage (45) to increase the pressure in the crank chamber (15) when the drive shaft (16) and the pistons (35) move away from the valve plate (14).

3. The compressor according to claim 1 or 2 characterized in that at least the pressurizing passage (18, 12a, 44, 54) is opened or closed.

4. The compressor according to any one of claims 1 to 3 characterized in that the valve mechanism (46) includes a valve body (47), which is integrally coupled to the drive shaft (16).

5. The compressor according to any one of claims 1 to 3 characterized in that the valve mechanism (46) includes a valve body (47), which has a rear end surface (47a) and which is integrally connected to the drive shaft (16), a front surface (48) of the valve plate (14), which faces the rear end surface (48) of the valve body (47), and a hole (44) formed in the front surface (48) of the valve plate (14), wherein the distance between the rear end surface (47a) of the valve body (47) and the front surface (48) of the valve plate (14) determines the flow rate of gas through the hole (44).

6. The compressor according to any one of claims 1 to 3 characterized in that the valve mechanism (41) includes a valve body (47), which is integrally connected to the drive shaft (16), the valve body (47) having a rear end surface (47a), wherein the valve mechanism (41) further includes a front surface (48) of the valve plate (14), which faces the rear end surface (47a) of the valve body (47), wherein a lubricant film (49) is formed on at least one of the rear end surface (47a) of the valve body (47) and the front surface (48) of the valve plate (14).

7. The compressor according to any one of claims 1 to 6 characterized in that the valve mechanism includes a sensor (53) detecting the movement of the drive shaft (16), an external control valve (52, 52c), which is powered by electricity and adjusts the opening size of at least one of the pressurizing passage (18, 12a, 44, 54) and the bleed passage (45), and a valve controller (C) for controlling the external control valve (52, 52c) in accordance with the movement of the drive shaft (16) detected by the sensor (53).

8. The compressor according to any one of claims 1 to 7 characterized in that the cam plate is a swash plate (31).

9. The compressor according to claim 8 characterized in that the compressor further includes a displacement control valve (60) and a displacement control passage (59), wherein the displacement control valve (60) is located in the control passage (60), and the displacement control passage (59) connects the discharge pressure area (38) with the crank chamber (15), wherein the displacement control valve (60) adjusts the opening size of the displacement control passage (59) and varies the pressure in the crank chamber (15), which varies the inclination of the swash plate (31) and the stroke of the pistons (35), which controls the displacement.

Fig.1

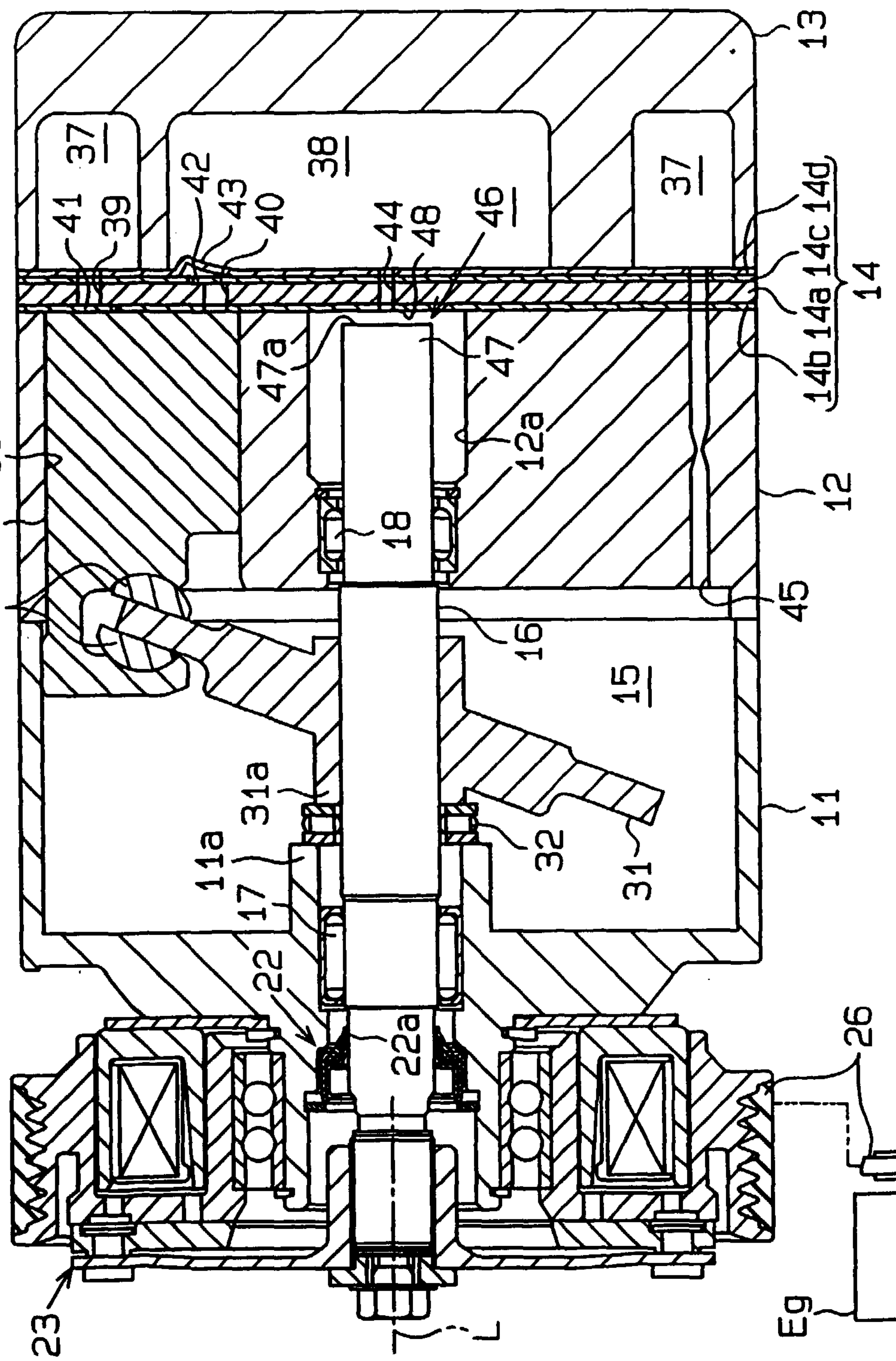

13
37
38
37
41
39
42
43
40
44
48
46
14d
14c
14a
14b
14
47a
47
33
35
12a
12
18
36
16
45
15
11a
31a
17
32
31
11
22
22a
26
23
L
Eg

Fig.2

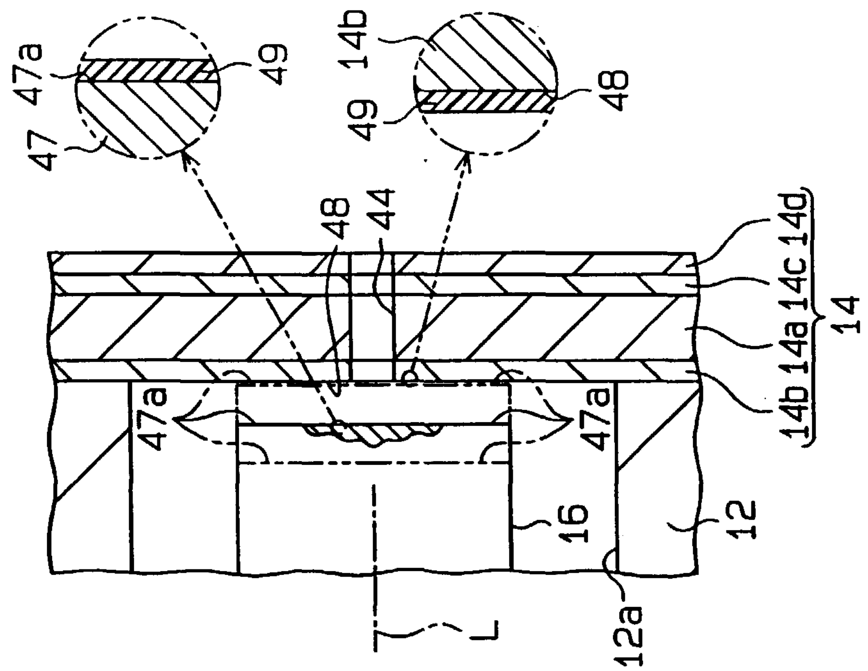

47a
49
47
14b
49
48
48
44
47a
47a
14d
14c
14a
14b
14
16
12
12a
L

Fig.3

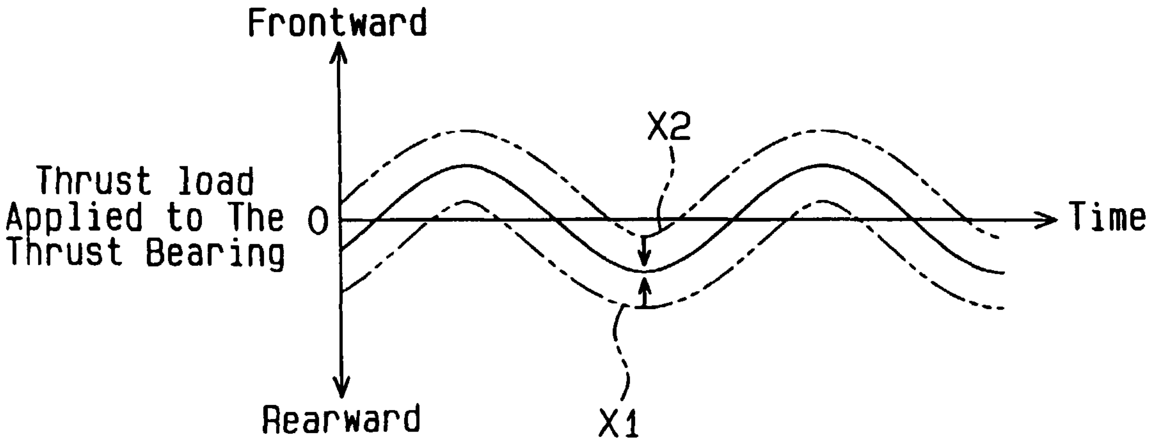

Frontward
Thrust load
Applied to The
Thrust Bearing
0
Time
X2
X1
Rearward

Fig.4

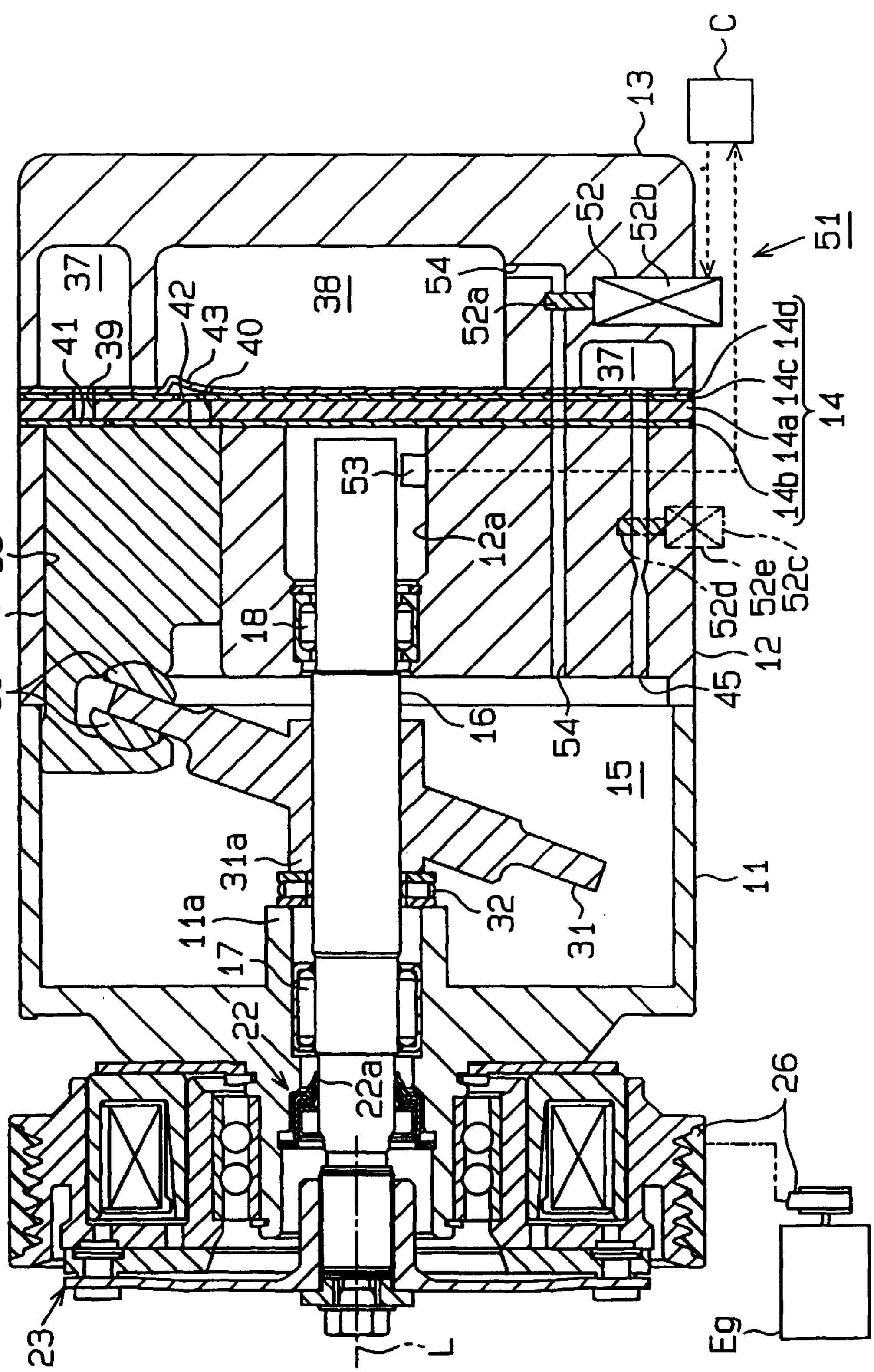
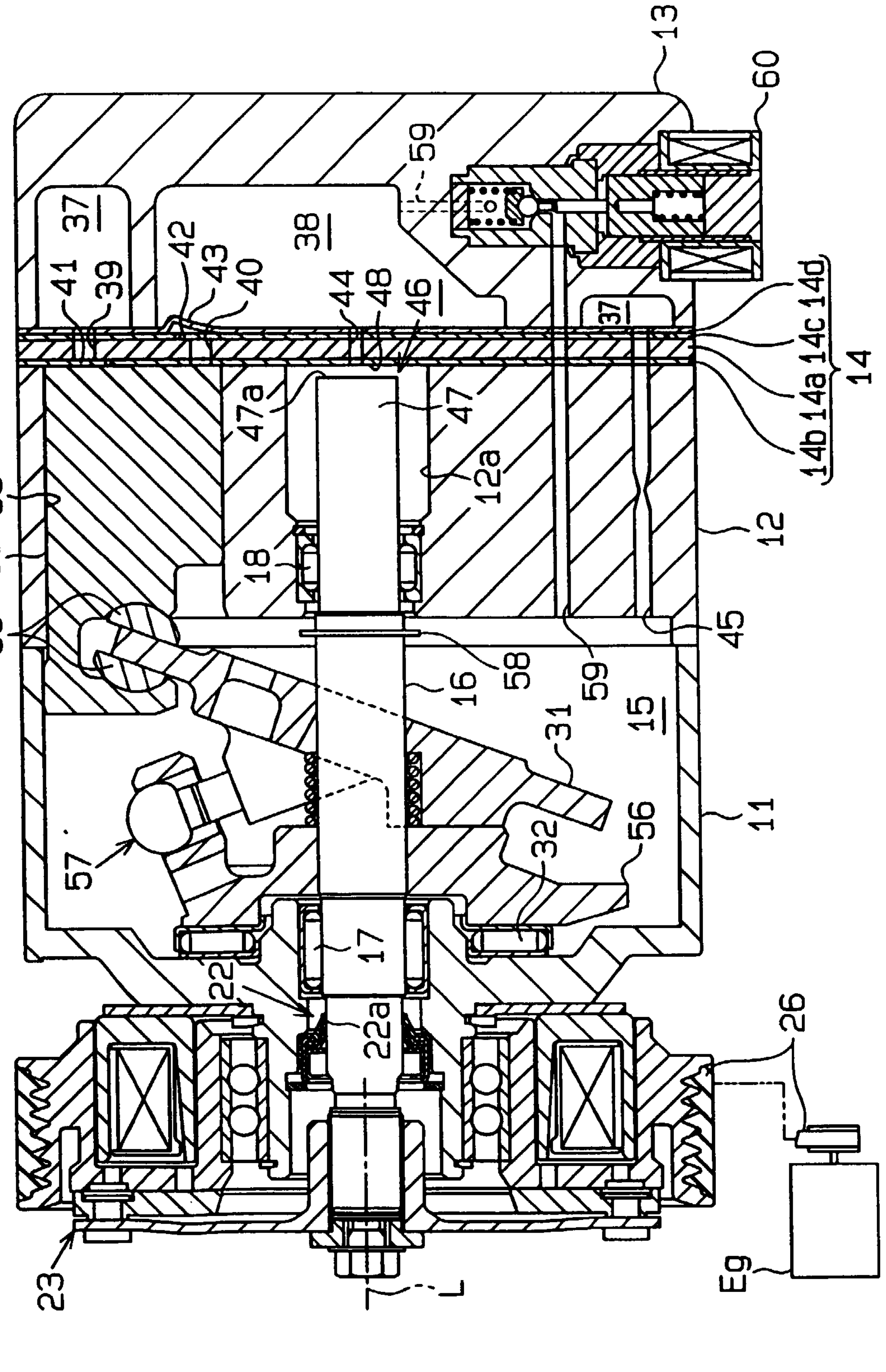

C
13
52
52b
51
54
52a
38
37
42
43
40
41
39
14d
14c
14a
14b
14
53
12a
52d
52e
52c
12
45
33
35
18
36
16
54
15
31a
11a
17
32
31
11
22
22a
26
23
Eg
L

Fig.5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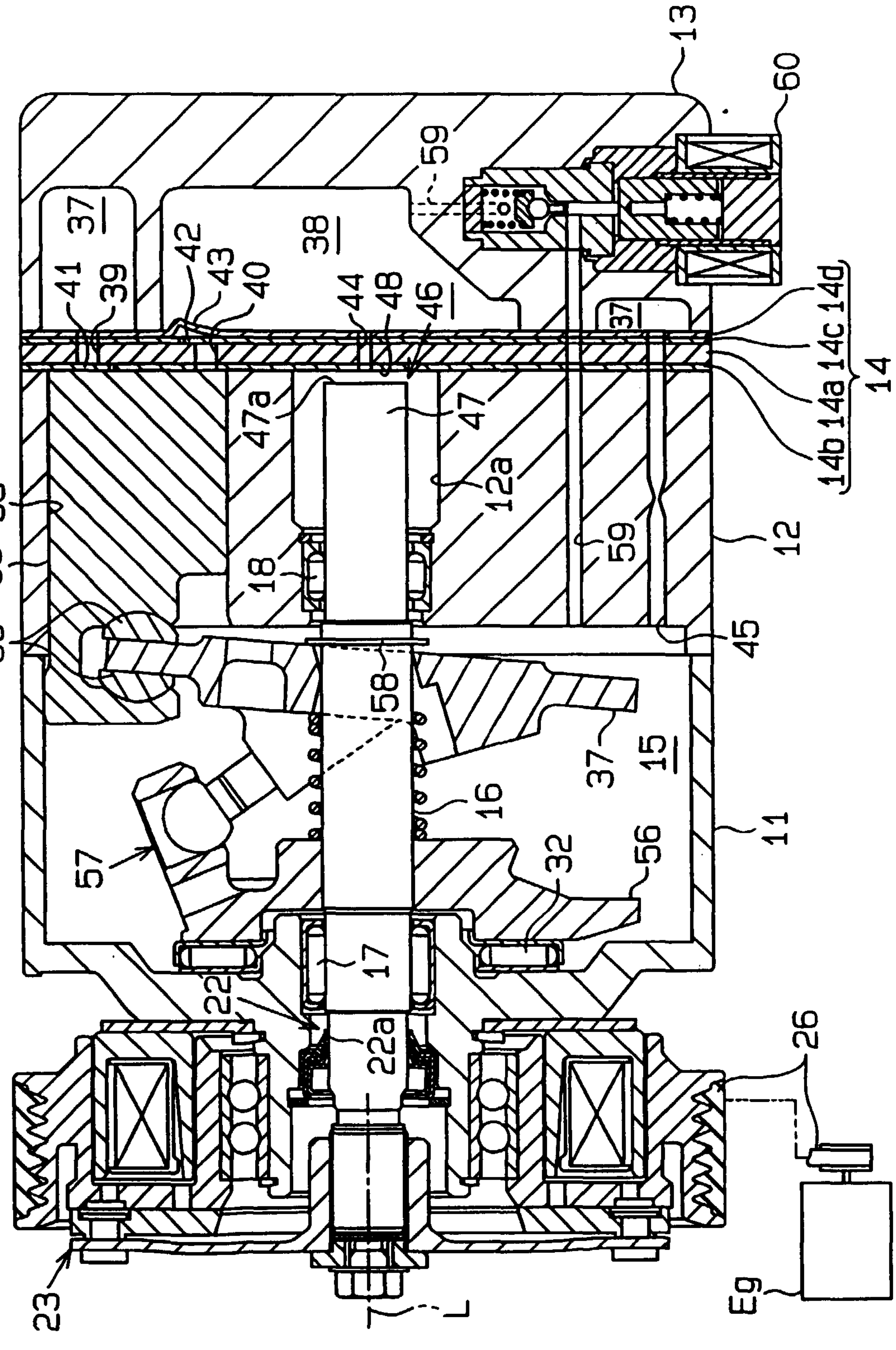
60
59
37
38
41
39
42
43
40
44
48
46
37
14d
14c
14a
14b
14
47a
47
12a
33
35
18
12
36
58
59
45
16
15
31
11
56
32
57
17
22
22a
26
23
L
Eg

Fig.6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44
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46
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14c
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14b
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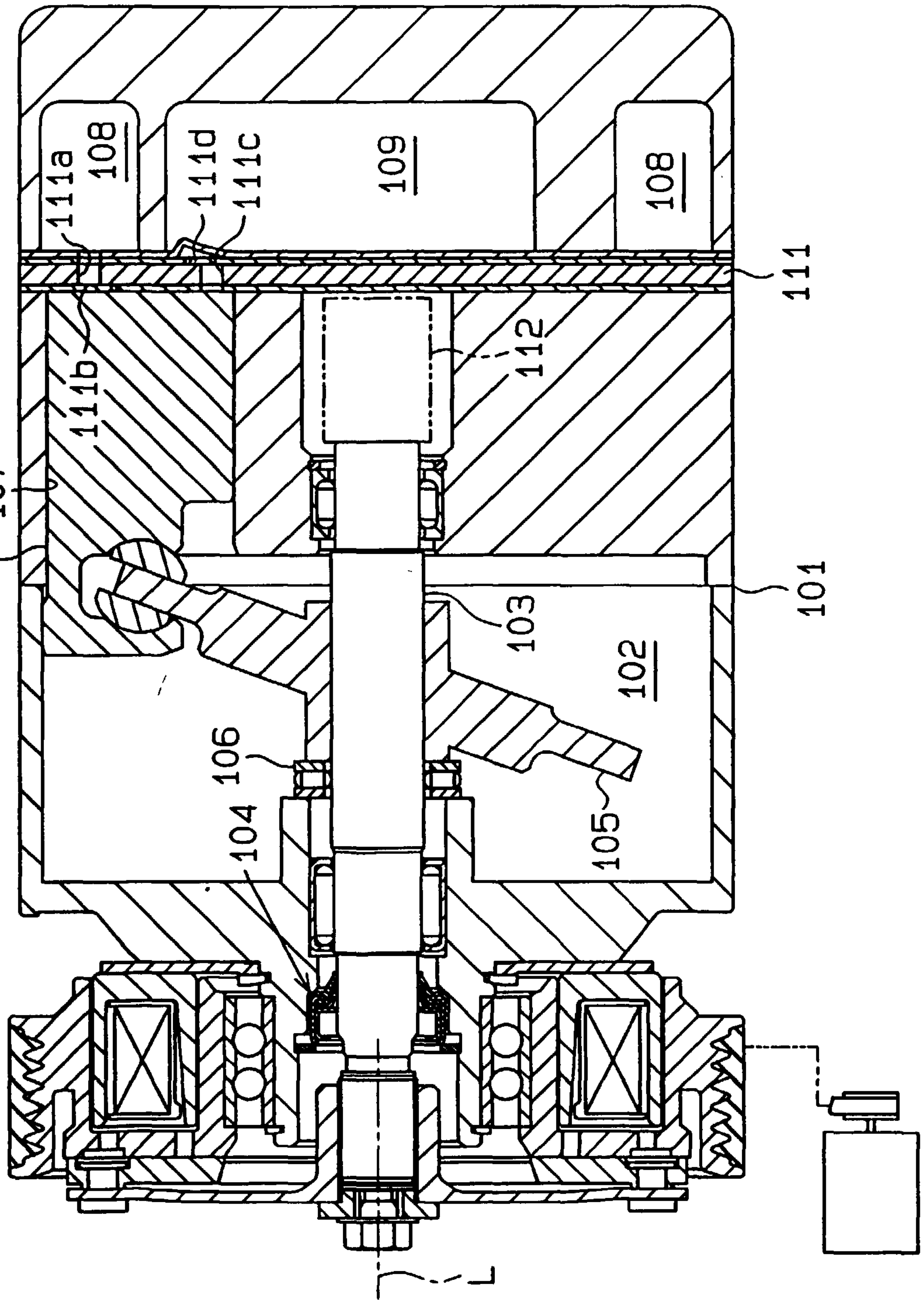
47
12a
33
35
18
59
12
36
45
58
37
15
16
57
32
56
11
17
22
22a
26
23
L
Eg

Fig.7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111d
111c
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111b
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110
101
103
102
106
104
105