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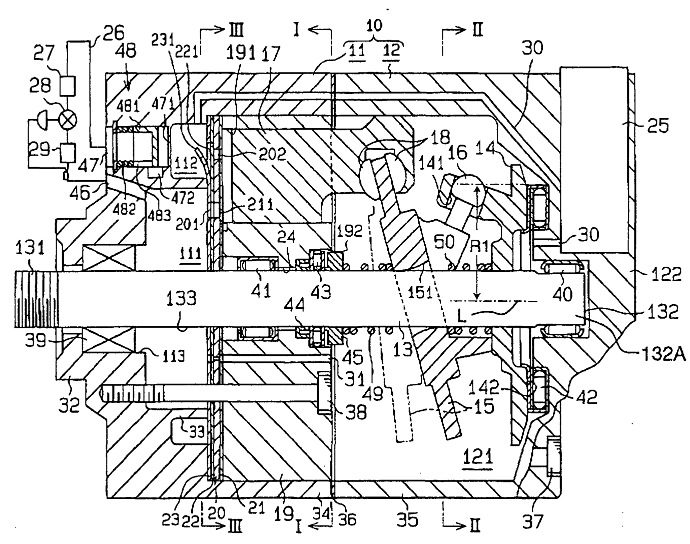
(54) **Piston type variable displacement compressor**

(57) A piston type variable displacement compressor has a housing, a drive shaft, a first thrust regulating mechanism including a first thrust bearing and a second thrust regulating mechanism including a regulating member and a second thrust bearing. The drive shaft has a first end and a second end. The first end projects from the housing. The first thrust regulating mechanism regulates a drive shaft to move in the direction from the first end to the second end. The first thrust regulating mechanism allows the drive shaft to rotate. The first thrust bearing has a first radius. The second thrust regulating mechanism regulates the drive shaft to move in a direction from the second end to the first end. The second thrust regulating mechanism allows the drive shaft to rotate. The second thrust bearing has a second radius. The second radius is smaller than the first radius.

FIG. 1A



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Description

BACKGROUND OF THE INVENTION

[0001] This application is related to European Patent Application No. 01110253.0. The present invention generally relates to a piston type variable displacement compressor.

[0002] As disclosed in Japanese Unexamined Patent Publication No. 2001-20858, in a piston type variable displacement compressor, a drive shaft is rotatably supported by two radial bearings in a front housing and a cylinder block. Compression reactive force is generated in the process of compressing refrigerant by a piston. The compression reactive force reaches an end wall of the front housing via the piston, a shoe, a swash plate, a lug plate and a thrust bearing that is placed at the front side of the front housing. The compression reactive force increases as the inclination angle of the swash plate increases. Therefore, the thrust bearing with a relatively large radius is used for receiving the relatively large compression reactive force since the thrust bearing with its load capacity increases with the radius.

[0003] Although the compression reactive force reduces as the inclination angle of the swash plate reduces, the pressure in a crank chamber urges the drive shaft toward a front end of the drive shaft in the front housing. Therefore, the load to the thrust bearing that is placed at the front side of the front housing cannot be ignored even when the inclination angle of the swash plate is relatively small. Furthermore, as the radius of the thrust bearing becomes large, the rolling speed of the roller in the thrust bearing increases. When the load resulting from the pressure in the crank chamber is applied to the large-radius thrust bearing, power loss becomes substantially large even when the inclination angle of the swash plate is relatively small.

[0004] The present invention addresses the above mention problem associated with the power loss in a state that the inclination angle of the swash plate is relatively small.

SUMMARY OF THE INVENTION

[0005] In accordance with the present invention, a piston type variable displacement compressor has a housing. The housing includes a cylinder block having a plurality of cylinder bores and defines a crank chamber, a suction pressure region including a suction chamber and a discharge pressure region including a discharge chamber. A drive shaft is supported in the housing and has a first end and a second end. The first end protrudes from the housing. The cylinder block is placed between the first end and the second end. The suction chamber and the discharge chamber are defined on the first end side relative to the cylinder block. The crank chamber is defined on the second end side relative to the cylinder block. A cam plate is slidably supported by the drive shaft and is inclinable with respect to the drive shaft. The cam plate is rotated by the rotation of the drive shaft. A plurality of pistons is accommodated in the cylinder bores. Each piston is coupled to the cam plate. The rotation of the cam plate is converted into the reciprocating movement of the pistons. In accordance with the reciprocating movement of the pistons, refrigerant is introduced from the suction chamber into the cylinder bores where it is compressed, and the compressed refrigerant is discharged from the cylinder bores to the discharge chamber. The discharge pressure region is connected to the crank chamber via a passage, and the crank chamber is connected to the suction pressure region via another passage. The refrigerant flows from the discharge pressure region to the crank chamber and flows from the crank chamber to the suction pressure region. Pressure in the crank chamber is varied by adjusting an opening degree of one of the passages. An inclination angle of the cam plate is varied by adjusting pressure in the crank chamber. A first thrust regulating mechanism includes a first thrust bearing and regulates the drive shaft to move in the direction from the first end of the drive shaft to the second end of the drive shaft. The first thrust regulating mechanism allows the drive shaft to rotate. The first thrust bearing has a first radius. A second thrust regulating mechanism includes a regulating member and a second thrust bearing and regulates the drive shaft to move in the direction from the second end to the first end. The second thrust regulating mechanism allows the drive shaft to rotate. The regulating member is provided on the drive shaft. The second thrust bearing has a second radius. The second radius is smaller than the first radius.

[0006] The present invention is also applicable to a compressor. The compressor includes a piston. The piston compresses refrigerant in a cylinder bore. A cam plate is movably connected to the piston for reciprocating the piston. A shaft has a rotational axis and rotates the cam plate. The shaft experiences a movement along the rotational axis while the piston reciprocates. The compressor has a thrust regulating mechanism and a second thrust regulating mechanism. The first thrust regulating mechanism regulates the movement of the shaft in a first direction along the rotational axis. The second thrust regulating mechanism regulates the movement of the shaft in a second direction along the rotational axis. The second thrust regulating mechanism includes a regulating ring and a second thrust bearing. The regulating mechanism is fixedly placed on the shaft and has a second thrust bearing contact surface. The second thrust bearing is fixedly placed near the shaft and in contact with the second thrust bearing contact surface to regulate the movement of the shaft in the second direction along the rotational axis.

BRIEF DESCRIPTION OF THE DRAWINGS

[0007] 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 A is a longitudinal cross-sectional view of a variable displacement compressor of a first preferred embodiment according to the present invention;

FIG. 1B is a partially enlarged cross-sectional view of the variable displacement compressor of the first preferred embodiment according to the present invention;

FIG. 2 is a cross-sectional view of the variable displacement compressor taken along the line I - I in FIG. 1;

FIG. 3 is a cross-sectional view of the variable displacement compressor taken along the line II - II in FIG. 1;

FIG. 4 is a cross-sectional view of the variable displacement compressor taken along the line III-III in FIG. 1;

FIG. 5 is a diagram showing a resultant spring characteristic from a restoring spring and a spring for decreasing inclination angle in the first preferred embodiment according to the present invention;

FIG. 6 is a partially enlarged cross-sectional view of a variable displacement compressor of a second preferred embodiment according to the present invention; and

FIG. 7 is a longitudinal cross-sectional view of a variable displacement compressor of a third preferred embodiment according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0008] A first preferred embodiment according to the present invention in a clutchless piston type variable displacement compressor will now be described by generally referring to FIGs. 1 A through 7. Referring particularly to FIG. 1A, the left side and the right side of the drawing respectively correspond to the front side and the rear side of the compressor. Carbon dioxide is used as refrigerant. As shown in FIG. 1A, a housing 10 of the piston type variable displacement compressor is constituted of a front housing 11 and a rear housing 12. An end surface of a circumferential wall 34 of the front housing 11 is connected to an end surface of a circumferential wall 35 of the rear housing 12 by a gasket 36. The front housing 11 is fixed to the rear housing 12 by a plurality of bolts 37.

[0009] A valve plate 20, a suction valve plate 21, a discharge valve plate 22 and a retainer plate 23 are fitted in the front housing 11. A suction chamber 111 and a discharge chamber 112 are defined between the retainer plate 23 and the front end wall 32 of the front housing 11. As shown in FIG. 4, the suction chamber 111 is separated from the discharge chamber 112 by a separation wall 33 and is surrounded by the discharge chamber 112. Each reference numeral in FIG. 4 refers to a substantially identical element having the same number in FIG. 1A, and the corresponding description will be provided later with respect to FIG. 1A if it has not yet been provided. If the description has been previously given, it will not be reiterated.

[0010] Still referring to FIG. 1A, a cylinder block 19 is fitted in the front housing 11 so as to contact the suction valve plate 21. The front end wall 32 of the front housing 11 is screwed by a plurality of screws 38 through the cylinder block 19. Thereby, the cylinder block 19 is fixed to the front housing 11. The cylinder block 19 has a plurality of cylinder bores 191. Although only one cylinder bore 191 is shown in FIG. 1A, five cylinder bores are radially arranged around the drive shaft 13 in the present embodiment as shown in FIGs. 2 and 4. Each reference numeral in FIGs. 2 and 4 refers to a substantially identical element having the same number in FIG. 1A, and the corresponding description will be provided later with respect to FIG. 1A if it has not yet been provided. If the description has been previously given, it will not be reiterated.

[0011] The rear housing 12 and the cylinder block 11 define a crank chamber 121. A drive shaft 13 is rotatably supported by radial bearings 40 and 41 in the rear housing 12 and the cylinder block 19. The radial bearings 40 and 41 receive the radial load from the drive shaft 13. The drive shaft 13 protrudes from the front end of the housing 10 through a shaft hole 24 of the cylinder block 19 and a shaft hole 113 of the front housing 11. A front end 131 of the drive shaft 13, which protrudes from the front end of the housing 10, is connected to an external drive source such as a vehicular engine via a power transmitting mechanism, which is not shown in the drawings. Driving force is transmitted from the external drive source to the drive shaft 13. A shaft seal 39 is placed in the shaft hole 113 and prevents the refrigerant in the suction chamber 111 from leaking to the outside of the housing 10 along the circumferential surface 133 of the drive shaft 13.

[0012] A lug plate 14 is secured to the drive shaft 13. The drive shaft 13 is inserted through a shaft hole 151 of a swash plate 15. The swash plate 15 is supported by the drive shaft 13 so as to slide along a central axis L of the drive shaft 13 and is inclinable with respect to the central axis L of the drive shaft 13. The central axis L functions as a rotational axis of the drive shaft 13. As shown in FIG. 3, a pair of guide pins 16 extends from the swash plate 15 and includes both a shaft portion and a head portion. Each reference numeral in FIG. 3 refers to a substantially identical element having the same number in FIG. 1A, and the corresponding description will be provided later with respect to FIG. 1A if it has not yet been provided. If the description has been previously given, it will not be reiterated. The guide pins 16 are respectively slidably inserted into guide holes 141 formed in the lug plate 14. The cooperation of the guide holes 141 and guide pins 16 allows the swash plate 15 to incline with respect to the axis of the drive shaft 13 and to rotate integrally with the drive shaft 13. The inclination of the swash plate 15 is guided by the slidable movement of the guide pins 16 in the corresponding guide holes 141. The swash plate 15 is slidably supported by the drive shaft 13.

[0013] Referring back to FIG. 1A, a piston 17 is accommodated in a corresponding one of cylinder bores 191. Each of the pistons 17 is coupled to the swash plate 15. The rotating movement of the swash plate 15, which rotates integrally with the drive shaft 13, is converted into the reciprocating movement of the piston 17 through a pair of shoes 18. The piston 17 reciprocates in the corresponding cylinder bore 191.

[0014] The suction chamber 111 is included in a suction pressure region, and the discharge chamber 112 is included in a discharge pressure region. As each piston 17 moves from the left side to the right side in FIG. 1A, the refrigerant in the suction chamber 111 is drawn into the corresponding cylinder bore 191 through a corresponding suction port 201 in the valve plate 20 and a corresponding suction valve 211 in the suction valve plate 21. As each piston 28 moves from the right side to the left side in FIG. 1A, the refrigerant in the cylinder bore 191 is compressed and is discharged to the discharge chamber 112 through a corresponding discharge port 202 in the valve plate 20 and a corresponding discharge valve 221 in the discharge valve plate 22. An opening degree of each discharge valve 221 is restricted by the contact of the discharge valve 221 against a corresponding retainer 231, which is formed on the retainer plate 23.

[0015] A first thrust bearing 42 has a circular form and is interposed between the end wall 122 of the rear housing 12 and the lug plate 14. Compression reactive force is generated in the process of compressing the refrigerant by the pistons 17 and is received by the end wall 122 of the rear housing 12 via the pistons 17, the shoes 18, the swash plate 15, the guide pins 16, the lug plate 14 and the first thrust bearing 42. A first thrust regulating mechanism includes the guide pins 16, the lug plate 14 and the first thrust bearing 42. The first thrust regulating mechanism regulates the movement of the drive shaft 13 in the direction from the front end 131 of the drive shaft 13 to a rear end 132A of the drive shaft 13 while it allows the drive shaft 13 to rotate.

[0016] The discharge chamber 112 is connected to the crank chamber 121 via a supply passage 30. The refrigerant in the discharge chamber 112 flows to the crank chamber 121 via the supply passage 30. An electromagnetic displacement control valve 25 is interposed in the supply passage 30. A controller controls magnetization and de-magnetization of the displacement control valve 25 based on a target temperature determined by a temperature setting device and a detected temperature by a temperature sensor that detects a temperature in a vehicle compartment. The controller, the temperature setting device and the temperature sensor are not shown in the drawings. The displacement control valve 25 is open when the displacement control valve 25 is de-magnetized. The displacement control valve 25 is closed when the displacement control valve 25 is magnetized. When the displacement control valve 25 is de-magnetized, the refrigerant in the discharge chamber 112 flows to the crank chamber 121. Thus, the displacement control valve 25 controls the amount of refrigerant that flows from the discharge chamber 112 to the crank chamber 121. The crank chamber 121 is connected to the suction chamber 111 via a bleed passage 31 having a throttled portion. The refrigerant in the crank chamber 121 flows to the suction chamber 111 via the bleed passage 31.

[0017] An annular recess 192 is formed in the inner circumferential surface of the shaft hole 24 at the rear side of the cylinder block 19. A second thrust bearing 43 is placed in the recess 192 in a circular form as shown in FIG. 2. As also shown in FIG. 2, a hypothetical minimal circle C touches upon a plurality of the cylinder bores 191, and the center of the circle C is at the central axis L of the drive shaft 13. Viewing in the direction of the central axis L of the drive shaft 13, the recess 192 is formed inside the circle C. Namely, the radius of the second thrust bearing 43 is smaller than that of the circle C. In other words, the radius of the second thrust bearing 43 is smaller than the distance between the central axis of the drive shaft 13 and each of the cylinder bores 191. The radius of the second thrust bearing 43 is also smaller than that of the first thrust bearing 42. The radius of the first thrust bearing 42 is defined as R1 as shown in FIG. 1A while that of the second thrust bearing is defined as R2 as shown in FIG. 2. The radii R1 and R2 respectively denote the distances between the central axis L of the drive shaft 13 and the outer circumferential surfaces of the first and second thrust bearings 42 and 43. The radius R2 is smaller than the radius R1.

[0018] As shown in FIG. 1B, an annular gap 193 and an annular chamber 194 are formed at the bottom of the recess 192. Each reference numeral in FIG. 1B refers to a substantially identical element having the same number in FIG. 1A, and the corresponding description will be provided later with respect to FIG. 1A if it has not yet been provided. If the description has been previously given, it will not be reiterated. The annular gap 193 and the annular chamber 194 are defined between a ring race 431 of the second thrust bearing 43 and the cylinder block 19. The ring race 431 of the second thrust bearing 43 is in contact with the outer peripheral surface of the recess 192.

[0019] Referring to FIG. 1A and 1B, a shaft seal 44 is placed in the chamber 194. The shaft seal 44 prevents the refrigerant in the crank chamber 121 from leaking to the suction chamber 111 along the circumferential surface 133 of the drive shaft 13.

[0020] A regulating ring 45 is fixedly fitted around the drive shaft 13. An annular protrusion 451 is formed on the end surface of the regulating ring 45 adjacent to the second thrust bearing 43. The annular protrusion 451 is formed on the inner peripheral surface of the regulating ring 45 and is in contact with the inner peripheral surface of the ring race 432. A small amount of preload is applied to the first thrust bearing 42 and the second thrust bearing 43. The preload is generated by sandwiching the second thrust bearing 43 between the recess 192 and the annular protrusion 451.

[0021] The rear end surface 132 of the drive shaft 13 experiences the pressure in the crank chamber 121. When the sum of the compression reactive force from each of the cylinder bores 191 and the force resulting from the atmospheric pressure applied to the surface of the front end 131 of the drive shaft 13 is smaller than the force resulting from the pressure in the crank chamber 121 applied to the rear end surface 132, the force differential between the sum and the force applied to the rear end surface 132 is received by the cylinder block 19 through the regulating ring 45 and the second thrust bearing 43.

[0022] A second thrust regulating mechanism includes the regulating ring 45, the second thrust bearing 43 and the cylinder block 19. The second thrust regulating mechanism regulates the movement of the drive shaft 13 in the direction from the rear end 132A to the front end 131 while it allows the drive shaft 13 to rotate.

[0023] A restoring spring 49 is placed between the swash plate 15 and the regulating ring 45, and a spring 50 for decreasing inclination angle is placed between the swash plate 15 and the lug plate 14. A straight line E in FIG. 5 shows a resultant spring characteristic from the restoring spring 49 and the spring 50.

[0024] The inclination angle of the swash plate 15 varies in accordance with the pressure in the crank chamber 121. As the pressure in the crank chamber 121 increases, the inclination angle of the swash plate 15 decreases. In contrast, as the pressure in the crank chamber 121 decreases, the inclination angle of the swash plate 15 increases. As the refrigerant in the discharge chamber 112 flows to the crank chamber 121, the pressure in the crank chamber 121 increases. When the supply of the refrigerant from the discharge chamber 112 to the crank chamber 121 stops, the pressure in the crank chamber 121 decreases. Namely, the inclination angle of the swash plate 15 is controlled by the displacement control valve 25. The maximum inclination angle of the swash plate 15 is restricted by the contact of the swash plate 15 at the lug plate 14.

[0025] The refrigerant is introduced into the suction chamber 111 through an inlet 46 and is discharged from the discharge chamber 112 to an outlet 47. The inlet 46 is connected to the outlet 47 via an external refrigerant circuit 26. A condenser 27, an expansion valve 28 and an evaporator 29 are placed in the external refrigerant circuit 26. A check valve 48 is interposed in the outlet 47.

[0026] A valve body 481 of the check valve 48 is urged by a spring 482 in the direction to shut a valve hole 471. When the body valve 481 is at the position as shown in FIG.1A, the refrigerant in the discharge chamber 112 outflows to the external refrigerant circuit 26 via the valve hole 471, a detour 472, an opening 483 formed in the valve body 481, and the inside of the valve body 481. When the valve body 481 shuts the valve hole 471, the refrigerant in the discharge chamber 112 does not outflow to the external refrigerant circuit 26.

[0027] The displacement control valve 25 controls suction pressure to be a target suction pressure in accordance with the value of an electric current supplied to the displacement control valve 25. As the value of the electric current supplied to the displacement control valve 25 increases, the opening degree of the displacement control valve 25 decreases and the amount of refrigerant supplied from the discharge chamber 112 to the crank chamber 121 also decreases. Since the refrigerant in the crank chamber 121 outflows to the suction chamber 111 through the bleed passage 31, the pressure in the crank chamber 121 falls. Therefore, the inclination angle of the swash plate 15 increases, and the amount of refrigerant discharged from the compressor increases. The increased discharged refrigerant from the compressor causes the suction pressure to decrease. On the other hand, as the value of the electric current supplied to the displacement control valve 25 decreases, the opening degree of the displacement control valve 25 increases and the amount of refrigerant supplied from the discharge chamber 112 to the crank chamber 121 increases. Then, the pressure in the crank chamber 121 increases, and the inclination angle of the swash plate 15 decreases. Therefore, the amount of refrigerant discharged from the compressor decreases. The decreased discharged refrigerant from the compressor causes the suction pressure to increase.

[0028] When the value of the electric current supplied to the displacement control valve 25 becomes zero, the opening degree of the displacement control valve 25 reaches the maximum and the inclination angle of the swash plate 15 becomes minimum. Discharge pressure is relatively low at this time. In the above discharged state, the spring constant of the spring 482 is determined in a such manner that the force resulting from the upstream pressure beyond the check valve 48 in the outlet 47 is less than the sum of the force resulting from the downstream pressure below the check valve 48 and the force of the spring 482. Therefore, when the inclination angle of the swash plate 15 becomes minimum, the valve body 481 shuts the valve hole 471 and the circulation of the refrigerant in the external refrigerant circuit 26 stops. When the circulation of the refrigerant stops, the thermal load on the compressor is substantially reduced to zero. Namely, air-conditioning is stopped.

[0029] The minimum inclination angle of the swash plate 15 is slightly larger than zero degree. Therefore, even when the inclination angle of the swash plate 15 is minimum, the refrigerant is still discharged from each of the cylinder bores 191 to the discharge chamber 112. The refrigerant flows from the discharge chamber 112 into the crank chamber 121 via the supply passage 30. Then, the refrigerant flows from the crank chamber 121 to the suction chamber 111 via the bleed passage 31. The refrigerant in the suction chamber 111 is introduced into each of the cylinder bores 191. The refrigerant in the cylinder bores 191 is compressed and then discharged into the discharge chamber 112. Namely, when the inclination angle of the swash plate 15 is minimum, there is a continuous path that passes through the discharge chamber 112, the supply passage 30, the crank chamber 121, the bleed passage 31, the suction chamber 111 and the cylinder bore 191 in the compressor. The pressures in the discharge chamber 112, the crank chamber 121 and the suction chamber 111 are different from each other. Therefore, the refrigerant circulates in the above continuous path under different pressure levels, and the inside of the compressor is lubricated by lubricating oil contained in the refrigerant.

[0030] When the increased electric current is supplied to the displacement control valve 25, the displacement control valve 25 decreases its opening and the pressure in the crank chamber 121 decreases. Therefore, the inclination angle of the swash plate 15 increases from the minimum inclination angle. As the inclination angle of the swash plate 15 increases, the discharge pressure increases, and the force resulting from the upstream pressure beyond the check valve 48 in the outlet 47 becomes larger than the sum of the force resulting from the downstream pressure below the check valve 48 and the force of the spring 482. Therefore, when the inclination angle of the swash plate 15 is more than the minimum inclination angle, the valve body 481 opens the valve hole 471, and the refrigerant outflows to the external refrigerant circuit 26.

[0031] When an engine stops, the operation of the compressor stops. That is, the rotation of the drive shaft 13 stops, and the displacement control valve 25 is demagnetized. The inclination angle of the swash plate 15 becomes minimum due to the de-magnetization of the displacement control valve 25. Then, the pressure in various chambers of the compressor substantially becomes equal. Therefore, when the pressure in the discharge chamber 112, the crank chamber 121 and the suction chamber 111 is substantially equal, the inclination angle of the swash plate 15 becomes larger than the minimum inclination angle due to the urging force of the restoring spring 49. The spring characteristics of the restoring spring 49 and the spring 50 is determined in a such manner that the inclination angle of the swash plate 15 is slightly lager than the minimum inclination angle in a state where the pressure in the discharge chamber 112, the crank chamber 121 and the suction chamber 111 is substantially equal while the drive shaft 13 does not rotate. At this time, the urging forces of the restoring spring 49 and the spring 50 are neutralized. The inclination angle of the swash plate 15, which is slightly larger than the minimum inclination angle, is called an initial inclination angle. It is determined that the initial inclination angle of the swash plate 15 is slightly larger than the inclination angle that at least causes the compressor to restore the displacement. In a state that the inclination angle of the swash plate 15 is the initial inclination angle while the swash plate 15 is rotating, if the opening degree of the displacement control valve 25 becomes zero, as the pressure in the crank chamber 121 increases, the inclination angle of the swash plate 15 steadily increases even if assuming that the restoring spring 49 and the spring 50 are not placed. When the electric current is initially supplied to the displacement control valve 25 to close the displacement control valve 25, the pressure in the crank chamber 121 decreases and the urging force of the spring 49 increases the inclination angle of the swash plate 15 at least to the inclination angle that causes the compressor to restore the displacement. When the swash plate 15 rotates and the displacement control valve 25 is de-magnetized to open the displacement control valve 25 to its maximum, there is pressure differential between in the crank chamber 121 and in the suction chamber 111. Thereby, the inclination angle of the swash plate 15 changes from the initial inclination angle to the minimum inclination angle due to the above pressure differential.

[0032] According to the first preferred embodiment in the clutchless compressor which controls the displacement as mentioned above, following advantageous effects are obtained. (1-1) In a piston type variable displacement compressor as disclosed in Japanese Unexamined Patent Publication No. 2001-20858, load, which is larger than the compression reactive force generated in compressing the refrigerant by the piston and the force resulting from the pressure in the crank chamber, is applied to the thrust bearing placed at the front side of the front housing. Therefore, even when the inclination angle of the swash plate is relatively small, the swash plate rotates in a state where the load resulting from the pressure in the crank chamber is applied to the thrust bearing with its relatively large radius. Accordingly, the power loss is also relatively large even when the inclination angle of the swash plate is relatively small.

[0033] In the preferred embodiment, the pressure in the crank chamber 121 urges the drive shaft 13 in the direction from the rear end 132A of the drive shaft 13 to the front end 131 of the drive shaft 13. Namely, the force resulting from the pressure in the crank chamber 121 is not applied to the first thrust bearing 42 as a load. Therefore, when the inclination angle of the swash plate 15 is relatively small, the load at the first thrust bearing 42 due to receiving the compression reactive force is substantially zero or relatively small. The load applied to the first thrust bearing 42 is smaller than the load applied to the thrust bearing in the prior art such as disclosed in Japanese Unexamined Patent Publication No. 2001-20858. The reduced load applied to the first thrust bearing 42 reduces the power loss during the compressor operation. Therefore, the power loss is reduced when the inclination angle of the swash plate 15 is relatively small.

[0034] The atmospheric pressure is applied to the surface of the front end 131 of the drive shaft 13, and the pressure in the crank chamber 121 is larger than the atmospheric pressure. There is a pressure differential between the pressure in the crank chamber 121 and the atmospheric pressure on the drive shaft 13. The drive shaft 13 is urged in the direction from its rear end 132A to its front end 131 due to the above pressure differential. When the urging force resulting from the pressure differential between the pressure in the crank chamber 121 and the atmospheric pressure is larger than the compression reactive force, the load resulting from the difference between the urging force and the compression reactive force is applied to the second thrust bearing 43. Since the resulted load is not large, the load capacity of the second thrust bearing 43 does not need to be large and the radius of the second thrust bearing 43 is substantially small. Utilizing the second thrust bearing with its relatively small radius is effective for reducing the power loss.

(1-2) In the first preferred embodiment, the suction chamber 111 and the discharge chamber 112 are defined near the front end 131 while the crank chamber 121 is defined near the rear end 132A. Also, the cylinder block is placed between the suction chamber 111 and the crank chamber 121. The drive shaft 13 protrudes from the front end of the housing 10. Therefore, the compression reactive force and the force resulting from the atmospheric pressure are applied to the drive shaft 13 in the direction from the front end 131 to the rear end 132A. The rear end surface 132 of the drive shaft 13 experiences the pressure in the crank chamber 121. The force resulting from the pressure in the crank chamber 121 is applied to the drive shaft 13 in the opposite direction with respect to the compression reactive force. The three forces work in the opposite directions to cancel the force with each other. Therefore, when the inclination angle of the swash plate 15 is relatively small, the load applied to the first thrust bearing 42 for receiving the compression reactive force is substantially zero or considerably small due to the above structure of the compressor. Consequently, the durability of the first thrust bearing 42, which needs lubricating, is substantially improved.

(1-3) Each piston 17 reciprocates in the corresponding cylinder bore 191 in the direction from the crank chamber 121 to the discharge chamber 112. The second thrust bearing 43 receives the load resulting from the pressure in the crank chamber 121, which is applied to the rear end surface 132 of the drive shaft 13. The radius of the second thrust bearing 43 is smaller than that of the circle C that is inscribed in a plurality of the cylinder bores 191. Therefore, the second thrust bearing 43 is placed in the crank chamber 121 in a such manner that the second thrust bearing 43 does not interfere with the pistons 17.The relatively spacious crank chamber 121 provides an appropriate space for the second thrust bearing 43.

(1-4) The second thrust bearing 43 receives the load applied to the drive shaft 13 in the direction from the rear end 132A to the front end 131 through the regulating ring 45 as a regulating member. The second thrust bearing 43 is inserted between the recess192 and the regulating ring 45. The movement of the regulating ring 45 in the direction from the rear end 132A to the front end 131 is restricted by the contact of the regulating ring 45 against the second thrust bearing 43. The movement of the regulating ring 45 in the direction from the front end 131 to the rear end 132A is restricted by the contact of the lug plate 14, which is secured to the drive shaft 13, against the first thrust bearing 42. Namely, the movement of the drive shaft 13, the regulating ring 45 and the lug plate 14 is restricted along the axial direction of the drive shaft 13. The restriction of the movement of the drive shaft 13 by the regulating ring 45 and the lug plate 14 substantially prevents the position of the top dead center of the pistons 17 from fluctuating. Thereby, the position of the dead center of the piston 17 is substantially stable.

(1-5) The regulating ring 45 is fixedly fitted around the drive shaft 13 and is suitable as a regulating member that is in contact with the second thrust bearing 43.

(1-6) The radius of the shaft hole 151 of the swash plate 15 is only slightly larger than that of the drive shaft 13. After the lug plate 14 and the regulating ring 45 are secured to the drive shaft 13, the drive shaft 13 can not be inserted through the shaft hole 151 of the swash plate 15 or the swash plate 15 can not be placed between the lug plate 14 and the regulating plate 45. However, after the lug plate 14 is secured to the drive shaft 13, a contact surface 142 of the lug plate 14, which is shown in FIG. 1A, is machined so as to raise the accuracy of the dimension of the contact surface 142 of the lug plate 14 with respect to the first thrust bearing 42. Therefore, the drive shaft 13 need to be inserted through the shaft hole 151 of the swash plate 15 before the regulating ring 45 is secured to the drive shaft 13. Namely, if the drive shaft 13 and the regulating ring 45 are formed integrally as a unit, the radius of the drive shaft 13 between the regulating ring 45 and the lug plate 14 needs to be the same as that of the regulating ring 45. The structure of the drive shaft 13 causes the radius of the drive shaft 13 to become relatively large. Utilizing the drive shaft 13 with its relatively large radius causes the radius of the housing 10 of the compressor to become relatively large, thereby the size of the compressor becomes large. Furthermore, since the weight of the drive shaft 13 become relatively heavy, the power loss increases. Therefore, the radius of the drive shaft 13 can be relatively small due to the separation of the regulating ring 45 from the drive shaft 13.

(1-7) In the piston type compressor without a clutch, driving force is directly transmitted from the external drive source to the drive shaft 13. In other words, the drive shaft 13 rotates while the external drive source is running. Therefore, the minimal inclination angle of the swash plate 15 needs to be as small as possible for reducing the power loss when the air-conditioning is stopped. Since the power loss needs to be reduced in the clutchless piston type compressor when the inclination angle of the swash plate 15 is relatively small, the present invention is preferably applied to the clutchless piston type compressor.

(1-8) Average pressure in the compressor with carbon dioxide as the refrigerant is several ten times higher than that in the compressor with fluoro series. The load resulting from the pressure applied to the rear end surface 132 of the drive shaft 13 increases due to the relatively high pressure in the compressor. Therefore, in comparison with the use of fluoro series, the power loss becomes larger when carbon dioxide is used as the refrigerant, and the inclination angle of the swash plate 15 is relatively small. The compressor with carbon dioxide is suitable for the present invention to reduce the power loss.

(1-9) The spring characteristics of the restoring spring 49 and the spring 50 for decreasing inclination angle need to be precisely determined so as to cause the compressor to steadily restore the displacement. For this reason, the positions of fixed ends of the restoring spring 49 and the spring 50 respectively need to be restricted in a such manner that the restoring spring 49 and the spring 50 do not move in an axial direction of the drive shaft 13. The position of the fixed end of the spring the spring 50 is restricted by the contact of the spring 50 against the lug plate 14. The position of the fixed end of the restoring spring 49 is restricted by the contact of the restoring spring 49 against the regulating ring 45. The regulating ring 45, which transmits the load to the second thrust bearing 43 in the direction from the rear end 132A to the front end 131, also has a role to regulate the spring characteristics of the restoring spring 49 and the spring 50 so as to cause the compressor to steadily restore the displacement.

[0035] A second preferred embodiment will be described by referring to FIG. 6. The same reference numerals denote the substantially identical elements as those in the first preferred embodiment. A large diameter portion 134 of the drive shaft 13 is in slide contact with the shaft hole 151 of the swash plate 15 and has a maximal radius of the drive shaft 13. A step 135 between the large diameter portion134 and a small diameter portion 136 is in contact with the ring race 432 of the second thrust bearing 43, and the restoring spring 49 is interposed between the ring race 432 and the swash plate 15. The second thrust bearing 43 receives load transmitted from the drive shaft 13 in the direction from the rear end 132A to the front end 131 via the step 135. According to the second preferred embodiment, substantially the same advantageous effects are obtained as mentioned in paragraph (1-1) through (1-4), (1-7) and (1-8) according to the first preferred embodiment.

[0036] A third preferred embodiment will be described by referring to FIG. 7. The same reference numerals denote the substantially identical elements as those in the first preferred embodiment. A regulating ring 45A fixedly is fitted around the drive shaft 15 in the suction chamber 111. A second thrust bearing 43A and a thrust receiving ring 52 are interposed between the front end wall 32 of the front housing 11 and the regulating ring 45A. The restoring spring 49 is interposed between the swash plate 15 and a circlip 51 that is fitted around the drive shaft 13. The load applied to the drive shaft 13 in the direction from the rear end 132A to the front end 131 is received by the front end wall 32 of the front housing 11 through the regulating ring 45A, the second thrust bearing 43A and the thrust receiving ring 52. According to the third preferred embodiment, substantially the same advantageous effects are obtained as mentioned in paragraph (1-1) through (1-4), (1-7) and (1-8) according to the first preferred embodiment.

[0037] The present invention is not limited to the above-mentioned embodiments but may be modified into the other examples. The present invention is applied to a piston type variable displacement compressor with a clutch.

[0038] 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 of the appended claims.

[0039] A piston type variable displacement compressor has a housing, a drive shaft, a first thrust regulating mechanism including a first thrust bearing and a second thrust regulating mechanism including a regulating member and a second thrust bearing. The drive shaft has a first end and a second end. The first end projects from the housing. The first thrust regulating mechanism regulates a drive shaft to move in the direction from the first end to the second end. The first thrust regulating mechanism allows the drive shaft to rotate. The first thrust bearing has a first radius. The second thrust regulating mechanism regulates the drive shaft to move in a direction from the second end to the first end. The second thrust regulating mechanism allows the drive shaft to rotate. The second thrust bearing has a second radius. The second radius is smaller than the first radius.

Claims

1. A piston type variable displacement compressor comprising:

   a housing including a cylinder block having a plurality of cylinder bores, the housing defining a crank chamber, a suction pressure region including a suction chamber and a discharge pressure region including a discharge chamber;
   a drive shaft supported by the housing, the drive shaft having a first end and a second end, the first end of the drive shaft projecting from the housing, the cylinder block being placed between the first end and the second end, the suction chamber and the discharge chamber being defined near the first end relative to the cylinder block, the crank chamber being defined near the second end relative to the cylinder block;
   a cam plate slidably supported by the drive shaft, the cam plate being inclinable with respect to the drive shaft, the cam plate being rotated by the rotation of the drive shaft;
   a plurality of pistons accommodated in the cylinder bores, each of the pistons being coupled to the cam plate, the rotation of the cam plate being converted into the reciprocating movement of the pistons, in accordance with the reciprocating movement of the pistons, refrigerant being introduced from the suction chamber into the cylinder bores, being compressed and being discharged from the cylinder bores to the discharge chamber, the discharge pressure region being connected to the crank chamber via a passage, the crank chamber being connected to the suction pressure region via another passage, the refrigerant flowing from the discharge pressure region to the crank chamber and flowing from the crank chamber to the suction pressure region, pressure in the crank chamber being varied by adjusting an opening degree of one of the passages, an inclination angle of the cam plate being varied by adjusting the pressure in the crank chamber;
   a first thrust regulating mechanism including a first thrust bearing having a first radius, the first thrust regulating mechanism regulating the drive shaft to move in the direction from the first end to the second end, the first thrust regulating mechanism allowing the drive shaft to rotate; and
   a second thrust regulating mechanism including a regulating member provided on the drive shaft and a second thrust bearing having a second radius, the second thrust regulating mechanism regulating the drive shaft to move in the direction from the second end to the first end, the second thrust regulating mechanism allowing the drive shaft to rotate, the second radius being smaller than the first radius.

2. The piston type variable displacement compressor according to claim 1, wherein the drive shaft has a central axis, the cylinder bores are radially arranged around the drive shaft, and the radius of the second thrust bearing is smaller than that of a hypothetical minimal circle which tangentially connects the cylinder bores and whose center is at the central axis of the drive shaft.

3. The piston type variable displacement compressor according to claim 1, wherein the first thrust regulating mechanism further comprises:

   a guide pin located on the cam plate; and
   a lug plate fixedly placed on the drive shaft, the lug plate being connected to the guide pin.

4. The piston type variable displacement compressor according to claim 1, wherein the regulating member is a separate body with respect to the drive shaft.

5. The piston type variable displacement compressor according to claim 4, wherein the regulating member is a regulating ring.

6. The piston type variable displacement compressor according to claim 5, wherein the regulating ring contacts the second thrust bearing.

7. The piston type variable displacement compressor according to claim 1, wherein the housing receives load transmitted from the drive shaft to the second thrust bearing in the direction from the second end to the first end of the drive shaft via the second thrust bearing.

8. The piston type variable displacement compressor according to claim 7, wherein the cylinder block receives load transmitted from the drive shaft to the second thrust bearing in the direction from the second end to the first end of the drive shaft via the second thrust bearing.

9. The piston type variable displacement compressor according to claim 1, wherein the first thrust regulating mechanism is placed in the crank chamber.

10. The piston type variable displacement compressor according to claim 1, wherein the second thrust regulating mechanism is placed in the crank chamber.

11. The piston type variable displacement compressor according to claim 1, wherein the second thrust regulating mechanism is placed in the suction chamber.

12. The piston type variable displacement compressor according to claim 1, wherein the drive shaft has a large diameter portion and a small diameter portion, and the regulating member is a step formed between the large diameter portion and the small diameter portion.

13. The piston type variable displacement compressor according to claim 1, wherein the piston type variable displacement compressor is a clutchless type compressor.

14. The piston type variable displacement compressor according to claim 1, wherein the refrigerant is carbon dioxide.

15. The piston type variable displacement compressor according to claim 1, wherein compression reactive force is generated in compressing the refrigerant by the pistons, the sum of the compression reactive force and the force resulting from an atmospheric pressure is applied to the drive shaft in the direction from the first end of the drive shaft to the second end, the force resulting from the pressure in the crank chamber is applied to the drive shaft in the direction from the second end to the first end of the drive shaft.

16. The piston type variable displacement compressor according to claim 1, wherein load is applied to the second thrust regulating mechanism when the inclination of the swash plate is relatively small.

17. The piston type variable displacement compressor according to claim 1, wherein the cam plate is a swash plate.

18. A compressor comprising:

a piston for compressing refrigerant in a cylinder bore;
a cam plate movably connected to the piston for reciprocating the piston;
a shaft having a rotational axis for rotating to drive the cam plate, the shaft experiencing a movement along the rotational axis while the piston reciprocates;
a first thrust regulating mechanism for regulating the movement of the shaft in a first direction along the rotational axis; and
a second thrust regulating mechanism for regulating the movement of the shaft in a second direction along the rotational axis, the second thrust regulating mechanism including a regulating ring fixedly placed on the shaft and having a second thrust bearing contact surface, the second thrust regulating mechanism further including a second thrust bearing fixedly placed near the shaft and in contact with the second thrust bearing contact surface to regulate the movement of the shaft in the second direction along the rotational axis.

19. The compressor according to claim 18, wherein the first thrust regulating mechanism further comprises:

a guide pin located on the cam plate;
a lug plate fixedly placed on the shaft for receiving the guide pin, the lug plate having a first thrust bearing contact surface; and
a first thrust bearing fixedly placed near the shaft and in contact with the first thrust bearing contact surface to regulating the movement of the shaft in the first direction along the rotational axis.

20. The compressor according to claim 18, wherein the cam plate is an inclinable swash plate for varying displacement of the compressor.

21. The compressor according to claim 18 further comprising a crank chamber and a suction chamber.

22. The compressor according to claim 21, wherein the first thrust regulating mechanism and the second thrust regulating mechanism are located in the crank chamber.

23. The compressor according to claim 21, wherein the first thrust regulating mechanism is located in the crank chamber while the second thrust regulating mechanism is located in the suction chamber.

24. The compressor according to claim 18, wherein the first thrust regulating mechanism further comprises:

a guide pin located on the cam plate;

a lug plate fixedly placed on the shaft for receiving the guide pin, the lug plate having a first thrust bearing contact surface; and
a first thrust bearing fixedly placed around the shaft in a circular shape having a first radius, the first thrust bearing in contact with the first thrust bearing contact surface to regulate the movement of the shaft in the first direction along the rotational axis.

25. The compressor according to claim 24, wherein the second thrust bearing is fixedly placed around the shaft in a circular shape having a second radius, the second radius is smaller than the first radius.

26. The compressor according to claim 25, wherein a plurality of the pistons and a plurality of the cylinder bores are placed in concentric manner around the shaft, the second radius is smaller than that of a hypothetical minimal circle that tangentially touches every one of the cylinder bores.

27. The compressor according to claim 18, wherein the shaft has a uniform radius.

28. The compressor according to claim 18, wherein the shaft has at least two radii along the rotational axis.

29. The compressor according to claim 18, wherein the refrigerant is carbon dioxide.

FIG. 1A

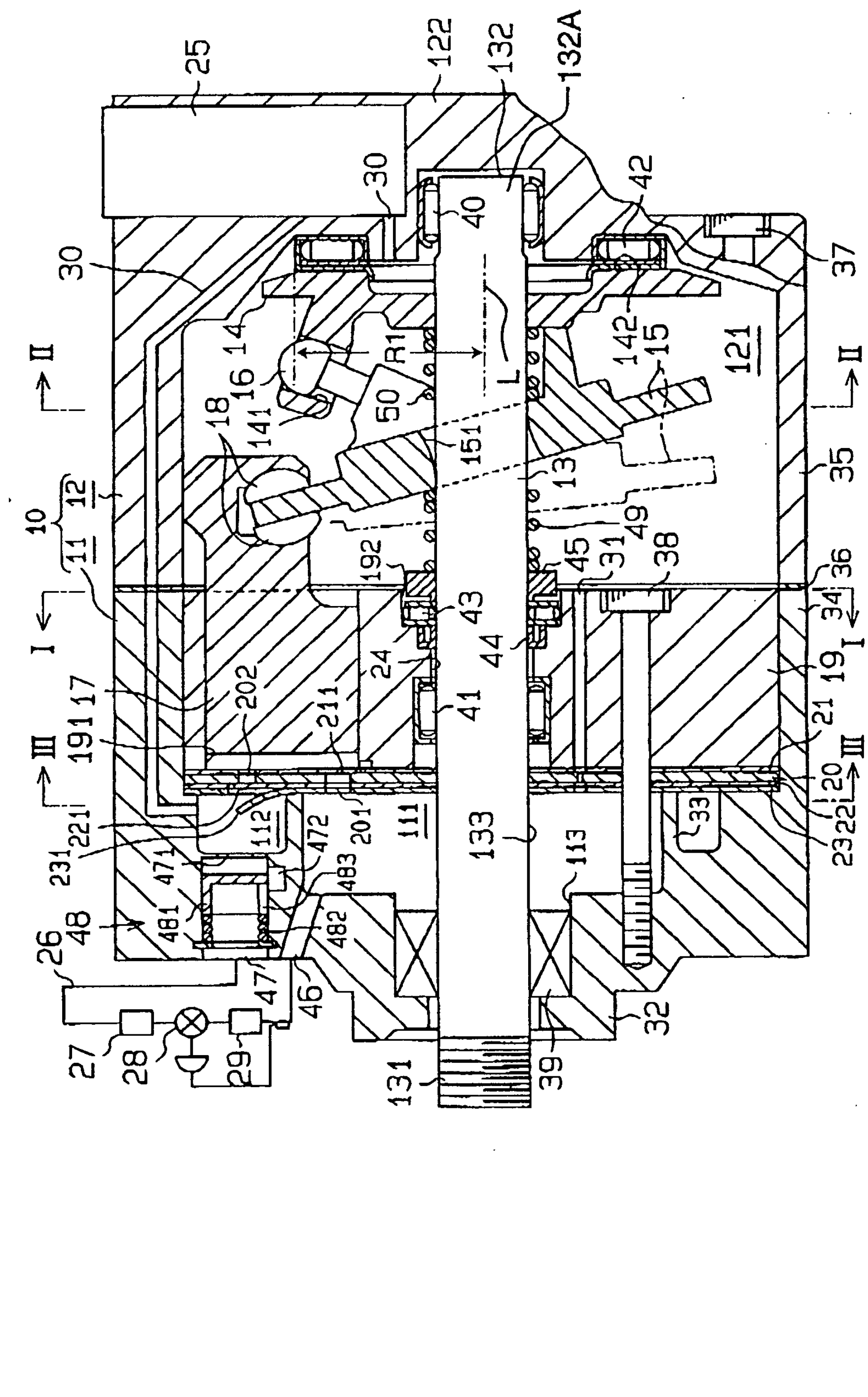
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FIG. 1B

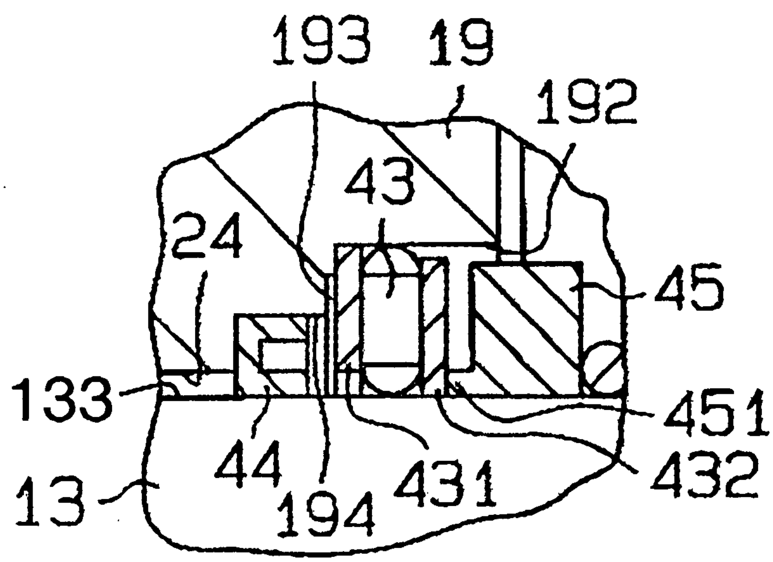

FIG. 2

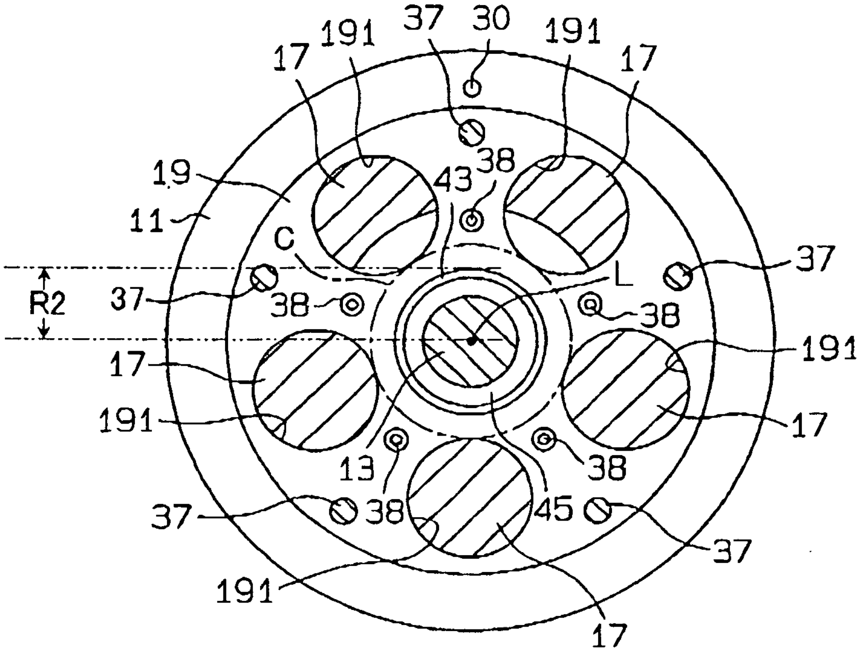
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FIG. 3

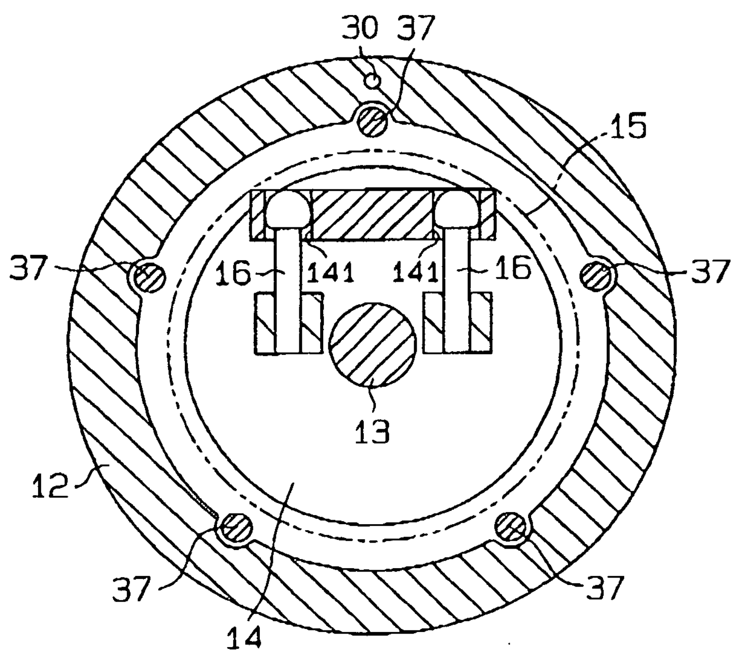
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FIG. 4

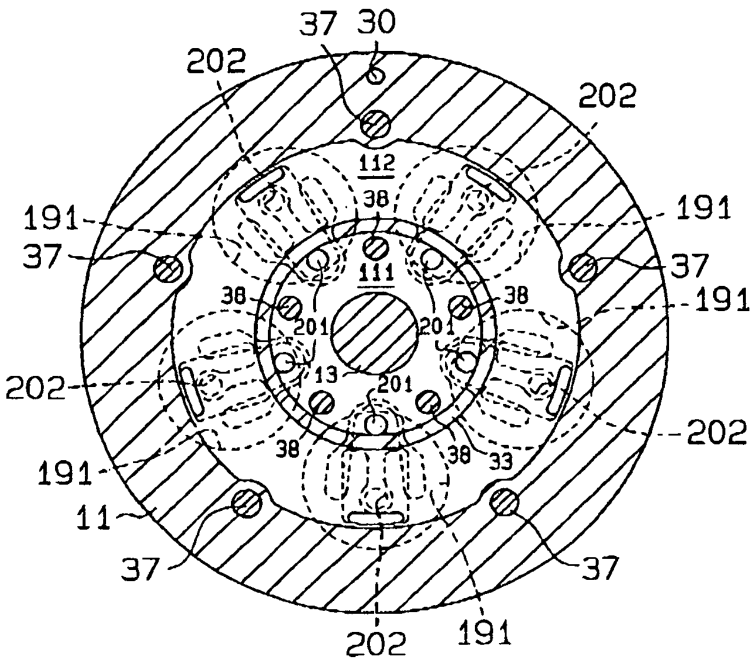
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FIG. 5

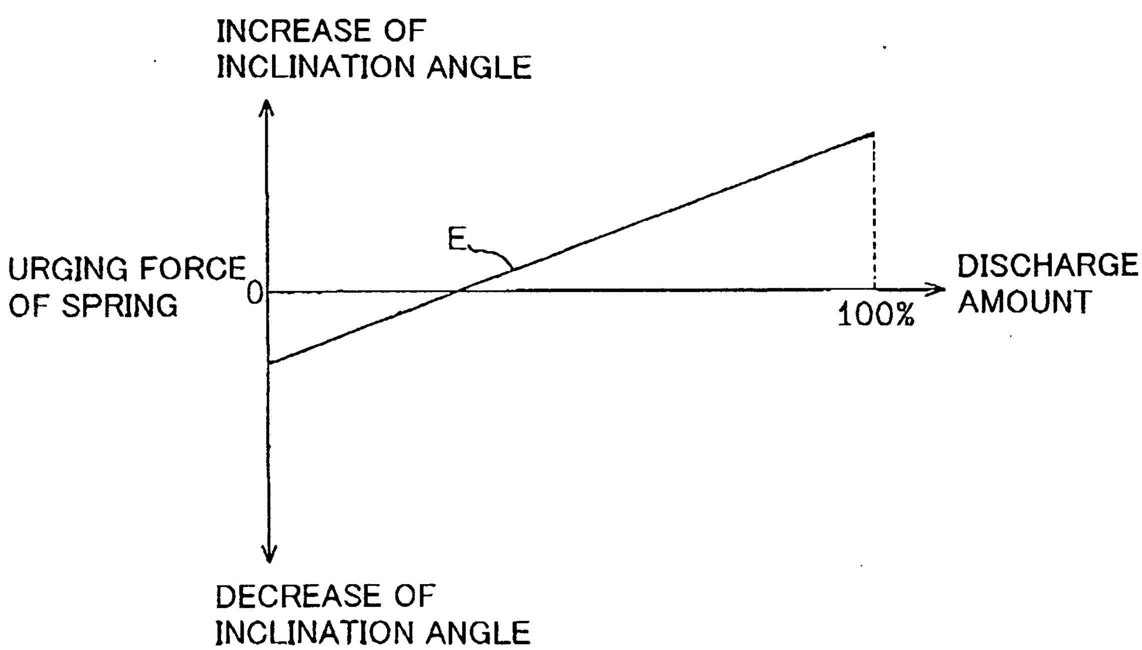
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FIG. 6

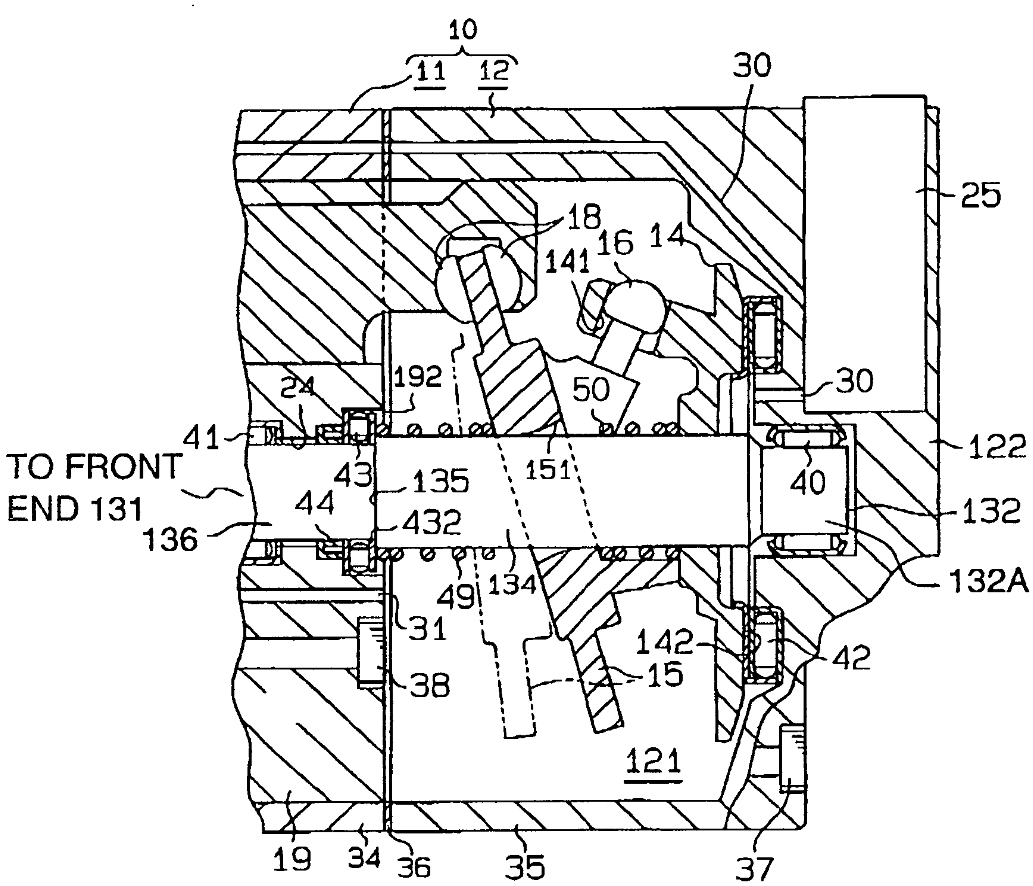
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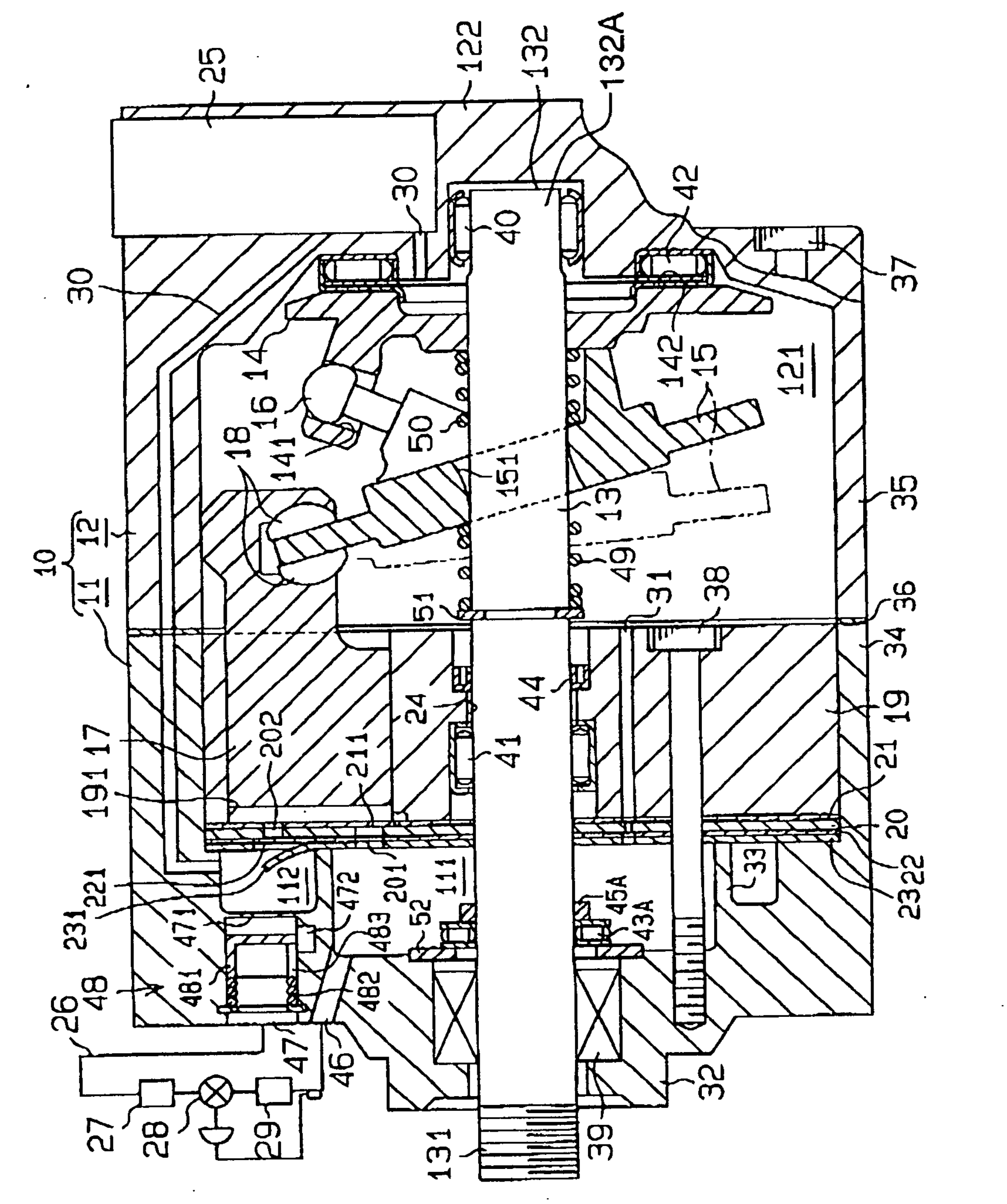
FIG. 7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