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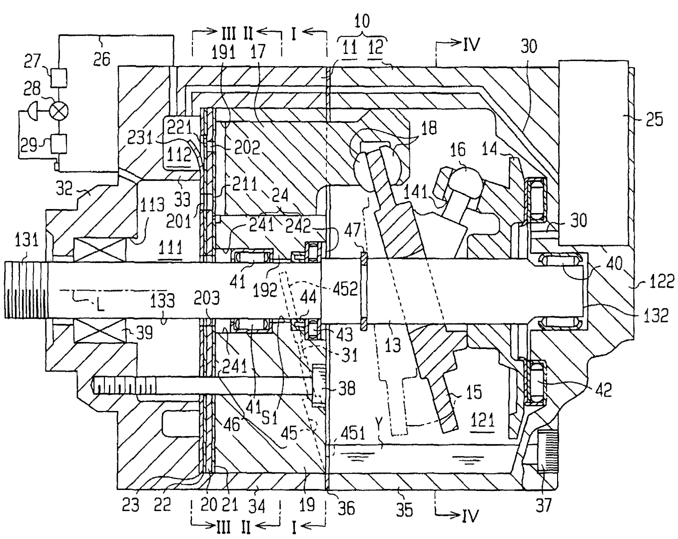
(54) **Structure of channel in variable displacement piston type compressor**

(57) A variable displacement piston type compressor has a housing, a drive shaft, a cam plate and a piston. The drive shaft is rotatably supported by the housing. The drive shaft has a first end and a second end. The first end of the drive shaft extends through the housing. The cylinder block is placed between the first end and the second end. The suction chamber and the discharge chamber are defined near the first end relative to the cylinder block. The crank chamber is defined near the second end relative to the cylinder block. The refrigerant in the crank chamber is bled into the suction pressure region through a bleed passage. Thereby, the inclination angle of the cam plate is controlled. The crank chamber and the suction chamber are connected with each other through the bleed passage. The bleed passage is formed outside of the drive shaft.

FIG. 1A



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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a structure of a channel in a variable displacement piston type compressor.

[0002] Japanese Unexamined Patent Publication No. 10-61548 discloses a variable displacement piston type compressor. In the constitution, a drive shaft is rotatably supported in a front housing and a cylinder block by a radial bearing. Compression reactive force is generated due to the discharge work of a piston. The compression reactive force is received by the front housing at its end wall through the piston, a pair of shoes, a swash plate, a lug plate and a thrust bearing. When the inclination angle of the swash plate is relatively small, the compression reactive force is relatively small. However, the pressure in a crank chamber urges the drive shaft in the direction from the rear side to the front side. Therefore, even when the inclination angle of the swash plate is relatively small, the load acting to the thrust bearing is relatively large.

[0003] To improve the durability of the thrust bearing, the thrust bearing requires lubricating. In the prior art, the crank chamber and a suction chamber are connected with each other by an axial passage formed in the drive shaft. Refrigerant in the crank chamber flows into the suction chamber through the axial passage. At this time, lubricant oil that flows with the refrigerant lubricates the thrust bearing.

[0004] In the above constitution that an axial passage is formed in the drive shaft, however, the diameter of the drive shaft is increased. Thereby, the size of the compressor is increased. In addition, a cost for machining the drive shaft is increased. Thereby, a cost for manufacturing the compressor is increased.

SUMMARY OF THE INVENTION

[0005] The present invention addresses a variable displacement piston type compressor that improves the reliability of a portion requiring lubrication in a crank chamber without increasing the size of the compressor.

[0006] According to the present invention, the present invention has following features. A variable displacement piston type compressor has a housing, a drive shaft, a cam plate and a piston. The housing has a cylinder block including a plurality of cylinder bores. The housing defines a crank chamber, a suction pressure region and a discharge pressure region. The suction pressure region includes a suction chamber. The discharge pressure region includes a discharge chamber. The drive shaft is rotatably supported by the housing. The drive shaft has a first end and a second end. The first end of the drive shaft extends through the housing. The cylinder block is placed between the first end and the second end. The suction chamber and the discharge chamber are defined near the first end relative to the cylinder block. The crank chamber is defined near the second end relative to the cylinder block. The cam plate is inclinably supported by the drive shaft in the crank chamber. The cam plate is integrally rotated with the drive shaft. The piston is accommodated in each cylinder bore. The rotation of the cam plate is converted into the reciprocating movement of the piston in accordance with the inclination angle of the cam plate. Refrigerant in the suction chamber is drawn into the cylinder bores due to the suction work of the piston. The refrigerant in the cylinder bores is discharged into the discharge chamber due to the discharge work of the piston. Refrigerant in the discharge pressure region is supplied into the crank chamber and the refrigerant in the crank chamber is bled into the suction pressure region through a bleed passage for controlling pressure in the crank chamber. Thereby, the inclination angle of the cam plate is controlled. The crank chamber and the suction chamber are connected with each other through the bleed passage. The bleed passage is formed outside of the drive shaft.

[0007] Furthermore, the present invention has following features. A variable displacement piston type compressor has a cylinder block, a piston, a cam plate, a crank chamber, a drive shaft, a suction chamber and a discharge chamber. The cylinder block includes a plurality of cylinder bores. The piston functions so as to compress refrigerant in each cylinder bore. The cam plate is movably connected to the piston for reciprocating the piston. The crank chamber is defined near one end of the cylinder block. The drive shaft has a rotational axis for rotating to drive the cam plate. The drive shaft is urged in a direction of the rotational axis while the piston reciprocates. The suction chamber is defined near the opposite end to the crank chamber relative to the cylinder block. The refrigerant in the suction chamber is drawn into the cylinder bores due to the suction work of the piston. The discharge chamber is defined near the opposite end to the crank chamber relative to the cylinder block. The refrigerant in each cylinder bore is discharged to the discharge chamber due to the discharge work of the piston. The refrigerant in the discharge chamber is supplied into the crank chamber and the refrigerant in the crank chamber is bled into the suction chamber through a bleed passage for controlling pressure in the crank chamber. The bleed passage is formed outside of the drive shaft.

BRIEF DESCRIPTION OF THE DRAWINGS

[0008] 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. 1A is a cross-sectional view of a variable displacement piston type compressor according to a first preferred embodiment of the present invention;

FIG. 1B is a partially enlarged view of the variable displacement piston type compressor of FIG. 1A;

FIG. 2 is a cross-sectional view of the variable displacement piston type compressor as seen at line I-I in FIG. 1A;

FIG. 3 is a cross-sectional view of the variable displacement piston type compressor as seen at line II-II in FIG. 1A;

FIG. 4 is a cross-sectional view of the variable displacement piston type compressor as seen at line III-III in FIG. 1A;

FIG. 5 is a cross-sectional view of the variable displacement piston type compressor as seen at line IV-IV in FIG. 1A;

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

FIG. 7 is a partially enlarged view of a variable displacement piston type compressor according to a third preferred embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0009] A first preferred embodiment according to the present invention in a variable displacement piston type compressor will now be described with reference to FIGs. 1A through 5. 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.

[0010] As shown in FIG. 1A, a front housing 11 and a rear housing 12 constitute a compressor housing 10. An end surface of a circumferential wall 34 of the front housing 11 and an end surface of a circumferential wall 35 of the rear housing 12 are secured to each other through a gasket 36. The front housing 11 and the rear housing 12 are fixed to each other by a plurality of bolts 37.

[0011] Still referring to FIG. 1A, a valve port 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 valve port plate 20 and an 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. The reference numerals in FIG. 4 are applied to substantially the same components 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.

[0012] Referring back to FIG. 1A, a cylinder block 19 is fitted in the front housing 11 so as to secure the suction valve plate 21. The 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 arranged around the drive shaft 13 in the present embodiment as shown in FIGs. 2 and 3. The reference numerals in FIGs. 2 and 3 are applied to substantially the same components 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.

[0013] Referring back to FIG. 1A, the rear housing 12 and the cylinder block 19 define a crank chamber 121. A drive shaft 13 is rotatably supported by radial bearings 40 and 41 respectively in the rear housing 12 and the cylinder block 19. In a shaft hole 24 of the cylinder block 19 a pair of accommodation chambers 241 and 242 is defined at the front and rear sides of a flange 192 of the cylinder block 19. The radial bearing 41 is accommodated in the accommodation chamber 241. The drive shaft 13 extends outside of the compressor housing 10 through the shaft hole 24 of the cylinder block 19 and a shaft hole 113 of the front housing 11. A front end of the drive shaft 13 connects with an external drive source such as a vehicle engine through a power transmission mechanism (not shown). Thereby, the drive shaft 13 is driven by the external drive source. A shaft seal 39 is placed in the shaft hole 113 so as to prevent the refrigerant in the suction chamber 111 from leaking to the outside of the compressor housing 10 along the circumferential surface 113 of the drive shaft 13.

[0014] Still referring to FIG. 1A, a lug plate 14 is secured to the drive shaft 13. A swash plate 15 is supported by the drive shaft 13 so as to slide along a rotational axis of the drive shaft 13 and is inclinable with respect to the axis of the drive shaft 13. As shown in FIG. 5, a pair of guide pins 16 is secured to the swash plate 15. The reference numerals in FIG. 5 are applied to substantially the same components 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 fitted into a pair of guide holes 141 formed in the lug plate 14. The cooperation of the guide holes 141 and the guide pins 16 allow 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 under the condition that the swash plate 15 is slidably supported by the drive shaft 13.

[0015] Referring back to FIG. 1A, a piston 17 is accommodated in a corresponding one of the cylinder bores 191. Each of the pistons 17 is connected to the swash plate 15. The rotational movement of the swash plate 15, which integrally rotates with the drive shaft 13, is converted into the reciprocating movement of the piston 17 through a pair of shoes 18. Thereby, the piston 17 reciprocates in the corresponding cylinder bores 191 frontward and rearward.

[0016] The suction chamber 111 is included in a suction pressure region. While the piston 17 moves from the left side to the right side in FIG. 1A, the refrigerant in the suction chamber 111 pushes away a corresponding suction valve 211 formed on the suction valve plate 21 from a corresponding suction port 201 formed on the valve port plate 20. Thereby, the refrigerant in the suction chamber 111 is drawn into the corresponding cylinder bore 191. While the piston 17 moves from the right side to the left side in FIG. 1A, the refrigerant that has been drawn into the cylinder bore 191 pushes away a corresponding discharge valve 221 formed on the discharge valve plate 22 from a corresponding discharge port 202 formed on the valve port plate 20. Thereby, the refrigerant in the cylinder bore 191 is discharged into the corresponding discharge chamber 112. The discharge chamber 191 is included in a discharge pressure region. The opening degree of the discharge valve 221 is restricted by the abutment of the discharge valve 221 against the retainer 231, which is formed on the retainer plate 23.

[0017] In the present embodiment, the valve port plate 20, the suction valve plate 21, the discharge valve plate 22 and the retainer plate 23 constitute a valve plate assembly. A thrust bearing 42 is interposed between the end wall 122 of the rear housing 12 and the lug plate 14. Compression reactive force is generated due to the discharge work of the piston 17 and is received by the end wall 122 of the rear housing 12 through the piston 17, the shoes 18, the swash plate 15, the guide pins 16, the lug plate 14 and the thrust bearing 42. At this time, the drive shaft 13 is urged in the direction of a central axis L of the drive shaft 13 while the piston 17 reciprocates.

[0018] A supply passage 30 connects with the discharge chamber 112 and the crank chamber 121. The refrigerant in the discharge chamber 112 is sent to the crank chamber 121 through the supply passage 30. In the supply passage 30, an electromagnetic type displacement control valve 25 is placed. The displacement control valve 25 is magnetized and demagnetized by a controller (not shown). The controller controls magnetization and demagnetization of the displacement control valve 25 based on a temperature detected by a temperature detector (not shown) that detects temperature in a vehicle compartment and a target temperature set by a temperature setting device (not shown). The displacement control valve 25 is closed while electricity is supplied to the displacement control valve 25. The displacement control valve 25 is open while the electricity is stopped being supplied to the displacement control valve 25. That is, on one hand the refrigerant in the discharge chamber 112 is sent to the crank chamber 121 while the displacement control valve 25 is demagnetized, and on the other hand the refrigerant in the discharge chamber 112 is not sent to the crank chamber 121 while the displacement control valve 25 is magnetized. Thus, the displacement control valve 25 controls the amount of refrigerant that flows from the discharge chamber 112 to the crank chamber 121.

[0019] In the accommodation chamber 242, a thrust bearing 43 and a shaft seal 44 are placed. The pressure in the crank chamber 121 is applied to the rear end surface 132 of the drive shaft 13. In the case that the sum of the compression reactive forces acting on the pistons 17 in the cylinder bores 191 is smaller than the force resulting from the pressure applied to the rear end surface 132, the force differential between the sum of the compression reactive forces and the force resulting from the pressure applied to the rear end surface 132 is received by the cylinder block 19 through the drive shaft 13 and the thrust bearing 43. The shaft seal 44 prevents the refrigerant in the suction chamber 111 from leaking to the crank chamber 111 along the circumferential surface 133 of the drive shaft 13.

[0020] In the cylinder block 19, a refrigerant passage 45 is formed. The refrigerant passage 45 has a first end 451 and a second end 452. The first end 451 is opened to a bottom of the crank chamber 121. As shown in FIGs. 2 and 3, the second end 452 is opened to a clearance S1 defined between the inner circumference of the flange 192 and the circumferential surface 133 of the drive shaft 13. The second end 452 is formed more upward than the central axis L of the drive shaft 13. A throttled passage 31 is formed in the refrigerant passage 45.

[0021] As shown in FIGs. 1A and 1B, the crank chamber 121 connects with the suction chamber 111 through the refrigerant passage 45, the clearance S1, the gap inside of the radial bearing 41, the accommodation chamber 241, and a shaft hole 203 formed in the valve port plate 20. The refrigerant passage 45, the clearance S1, the radial bearing 41, the accommodation chamber 241, and the shaft hole 203 constitute a bleed passage 46. The refrigerant in the crank chamber 121 flows into the suction chamber 111 through the bleed passage 46 that connects the crank chamber 121 to the suction chamber 111. The circumferential surface 133 of the drive shaft 13 and the bleed passage 46 in the cylinder block 19 (or the refrigerant passage 45 in the cylinder block 16) meet with each other more upward than the central axis L of the drive shaft 13.

[0022] The inclination angle of the swash plate 15 is varied base on the control of the pressure in the crank chamber 121. As the pressure in the crank chamber 121 increases, the inclination angle of the swash plate 15 is decreased relative to the perpendicular plane to the drive shaft 13. In contrast, as the pressure in the crank chamber 121 decreases, the inclination angle of the swash plate 15 is increased relative to the perpendicular plane to the drive shaft 13. As the refrigerant in the discharge chamber 112 is supplied to the crank chamber 121, the pressure in the crank chamber 121 is increased. When the supply of the refrigerant from the discharge chamber 112 to the crank chamber 121 stops, the pressure in the crank chamber 121 is decreased. That is, the inclination angle of the swash plate 15 is controlled by the displacement control valve 25.

[0023] The maximum inclination angle of the swash plate 15 is restricted by the abutment of the swash plate 15 against the lug plate 14. The minimum inclination angle of the swash plate 15 is restricted by the abutment of the swash plate 15 against the circular clip 47.

[0024] The discharge chamber 112 and the suction chamber 111 connect with each other through an external refrigerant circuit 26. The refrigerant that has been flowed into the external refrigerant circuit 26 from the discharge chamber 112 is returned to the suction chamber 111 in the compressor through a condenser 27, an expansion valve 28 and an evaporator 29.

[0025] In the first preferred embodiment of the present invention, the following advantageous effects are obtained.

(1-1) When the opening degree of the displacement control valve 25 is not zero degree, a part of the refrigerant in the discharge chamber 112 flows into the crank chamber 121 through the supply passage 30. In addition, the lubricant oil that flows with the refrigerant flows from the discharge chamber 112 into the crank chamber 121. The refrigerant in the crank chamber 121 flows into the suction chamber 111 through the bleed passage 46. In addition, the lubricant oil that flows with the refrigerant flows from the crank chamber 121 into the suction chamber 111. The bleed passage 46, which is required to control displacement and to lubricate a portion requiring lubrication in the crank chamber 121, is formed outside of the drive shaft 13. In the constitution that the bleed passage 46 is formed outside of the drive shaft 13, the diameter of the drive shaft 13 can be reduced than that of the drive shaft 13 of which a bleed passage 46 is formed inside. The reduction of the diameter of the drive shaft 13 enables the compressor to be compact. Therefore, the constitution that the bleed passage 46 is formed outside of the drive shaft 13 is effective to prevent the compressor from becoming large.

(1-2) The compression reactive force is generated due to the discharge work of the piston 17 in the direction from the front end of the drive shaft 13 to the rear end of the drive shaft 13. When the inclination angle of the swash plate 15 is relatively small, the compression reactive force is relatively small. The pressure in the crank chamber 121 urges the drive shaft 13 in the direction from the rear end of the drive shaft 13 to the front end of the drive shaft 13. Therefore, when the inclination angle of the swash plate 15 is relatively small, the load that is applied to the thrust bearing 42 for receiving the compression reactive force is substantially zero or relatively extremely small. That is, the constitution that the suction chamber 111 and the discharge chamber 112 are separated from the crank chamber 121 through the cylinder block 19 in the direction from the front end of the drive shaft 13 to the rear end of the drive shaft 13 is effective to improve the reliability of the thrust bearing 42, which is a portion requiring lubrication in the crank chamber 121.

(1-3) A part of the bleed passage 46 is formed in the shaft hole 24 of the cylinder block 19. When a gap exists between the circumferential surface of the shaft hole 24 and the circumferential surface 133 of the drive shaft 13, the gap can function as a part of the bleed passage 46. To define the gap between the circumferential surface of the shaft hole 24 and the circumferential surface 133 of the drive shaft 13, the diameter of the shaft hole 24 is formed to be larger than that of the drive shaft 13. The constitution having such a relation between the diameters is easily obtained. Accordingly, the shaft hole 24 of the cylinder block 19 is suitable for forming at least a part of the bleed passage 46.

(1-4) The radial bearing 41 in the bleed passage 46 is lubricated by the lubricant oil that flows with the refrigerant passing through the bleed passage 46. Therefore, the shaft hole 24 provided with the radial bearing 41 is suitable for forming the bleed passage 46.

(1-5) The first end 451 of the refrigerant passage 45 that is a first end of the bleed passage 46 is opened to the bottom of the crank chamber 121. As shown in FIG. 1 A, lubricant oil Y accumulates at the bottom of the crank chamber 121. The lubricant oil Y is sent to the bleed passage 46 since the refrigerant in the crank chamber 121 flows to the suction chamber 111. The constitution that the lubricant oil Y that has accumulated at the bottom of the crank chamber 121 is sent to the suction chamber 111 is effective to raise a lubricating efficiency of portions requiring lubrication such as the radial bearing 41 and the shaft seal 39 other than the portion requiring lubrication in the crank chamber 121.

(1-6) The second end 452 of the refrigerant passage 45 is opened to the clearance S1. The second end 452 is at a higher position than the central axis

L of the drive shaft 13. Therefore, the lubricant oil can accumulate even in the clearance S1 and the accommodation chamber 241. Thereby, the lubricating efficiency of the radial bearing 41 is raised.

[0026] A second preferred embodiment according to the present invention in a variable displacement piston type compressor will now be described with reference to FIG. 6. The same reference numerals of the first preferred embodiment are applied to substantially the same elements in the second preferred embodiment.

[0027] Referring to FIG. 6, a shaft seal 44A is placed in the accommodation chamber 241. The shaft seal 44A prevents the refrigerant in the accommodation chamber 241 from leaking to the suction chamber 111 along the circumferential surface 133 of the drive shaft 13. The accommodation chamber 241 and the suction chamber 111 are connected by a refrigerant passage 48 formed in the cylinder block 19 and a throttled passage 49 formed through the valve port plate 20. A first end 481 of the refrigerant passage 48 is opened relatively upward in the accommodation chamber 241.

[0028] A clearance S2 is defined between an inner circumferential surface of a race 431 of the thrust bearing 43 and the circumferential surface 133 of the drive shaft 13. The refrigerant in the crank chamber 121 flows to the suction chamber 111 through a bleed passage 50 constituted of a gap between the race 431 and a race 432 of the thrust bearing 43, the clearance S2, the clearance S1, a gap in the radial bearing 41, the accommodation chamber 241, the refrigerant passage 48 and the throttled passage 49.

[0029] In the second preferred embodiment, the above-described effects (1-1) through (1-4) of the first preferred embodiment are substantially obtained. In addition, the following effects are obtained.

(2-1) The throttled passage 49 requires a relatively small diameter so as to properly adjust the pressure in the crank chamber 121. It is difficult to form a passage having a relatively small diameter and a relatively large length by drilling. However, since the thickness of the valve port plate 20 is not relatively large, the length of the throttled passage 49 becomes relatively small. Therefore, the valve port plate 20 is suitable for forming the throttled passage 49 having a relatively small diameter.

(2-2) Since the cylinder block 19 does not require a throttled passage having a relatively small diameter, the refrigerant passage 48 having a proper diameter is easily formed by drilling.

[0030] A third preferred embodiment according to the present invention in a variable displacement piston type compressor will now be described with reference to FIG. 7. The same reference numerals of the first preferred embodiment are applied to substantially the same elements in the third preferred embodiment.

[0031] An annular blocking body 51 made of synthetic resin is placed in the accommodation chamber 241. For example, the blocking body 51 is made of polytetrafluoroethylene. The blocking body 51 is slidable respective to the circumferential surface in the accommodation chamber 241, the circumferential surface 133 of the drive shaft 13 and the suction valve plate 21. Grooves 52 and 521 are formed respectively on the outer circumferential surface 512 and the end surface 513 of the blocking body 51. The refrigerant in the crank chamber 121 flows to the suction chamber 111 through the gap in the thrust bearing 43, the clearances S2 and S1, the gap in the thrust bearing 41, the accommodation chamber 241, the grooves 52 and 521, and the shaft hole 203. That is, the pressure in the crank chamber 121 is adjusted by the outflow of the refrigerant in the crank chamber 121 (or the release of the pressure in the crank chamber 121) through the grooves 52 and 521 of the blocking body 51 that are throttling means. The grooves 52 and 521 of the blocking body 51 function as a throttle so as to separate the pressure in the suction chamber 111 from the pressure in the crank chamber 121.

[0032] In the third preferred embodiment, the above-described effects (1-1) through (1-4) of the first preferred embodiment are substantially obtained. In addition, the following effects are obtained.

(3-1) In a sense, the blocking body 51 is placed for leaking the refrigerant properly from the crank chamber 121 to the suction chamber 111. The refrigerant in the crank chamber 121 leaks through a gap between the outer circumferential surface 512 of the blocking body 51 and the circumferential surface of the accommodation chamber 241 and a gap between the inner circumferential surface 511 of the blocking body 51 and the circumferential surface 133 of the drive shaft 13. Therefore, the function for preventing the refrigerant from leaking through the gaps does not require its accuracy. Accordingly, it is not required that the outer circumferential surface 512 of the blocking body 51 is completely close to the circumferential surface of the accommodation chamber 241 and that the inner circumferential surface 511 of the blocking body 51 is completely close to the circumferential surface 133 of the drive shaft 13.

Even when the outside diameter of the blocking body 51 is slightly larger than the diameter of the accommodation chamber 241, the elastic deformation of the blocking body 51 made of synthetic resin enables the blocking body 51 to fit into the accommodation chamber 241. Also, even when the inside diameter of the blocking body 51 is slightly smaller than the diameter of the drive shaft 13, the elastic deformation of the blocking body 51 made of synthetic resin enables the blocking body 51 to fit around the drive shaft 13.

In other words, the blocking body 51 made of synthetic resin permits low accuracy in size. Such a blocking body is manufactured at low cost and is easily manufactured by molding.

(3-2) Although the cross-sectional area for passing the grooves 52 and 521 that are throttled passage is relatively small, the grooves 52 and 521 are easily manufactured by molding.

(3-3) The grooves 52 and 521 are formed respectively on the outer circumferential surface 512 and the end surface 513 of the blocking body 51. The outer circumferential surface 512 and the end surface 513 of the blocking body 51 are easy to form a groove. Therefore, the surface of the blocking body 51 is suitable for forming the grooves 52 and 521.

(3-4) When the outside diameter of the blocking body 51 is much smaller than the diameter of the accommodation chamber 241, the sum of the cross-sectional area for passing a clearance between the circumferential surface of the accommodation chamber 241 and the outer circumferential surface 512 of the blocking body 51 and the cross-sectional area for passing the groove 52 is much larger than the desired value of the cross-sectional area of the throttle. However, the end surface 513 of the blocking body 51 is closed to the suction valve plate 21 due to the pressure differential between the crank chamber 121 and the suction chamber 111. In addition, a part of the groove 521 formed on the end surface 513 faces to the suction valve plate 21. Therefore, the cross-sectional area for passing a clearance between the end surface 513 and the suction valve plate 21 is prescribed to be a desired value of the cross-sectional area for passing the groove 521. Thereby, a satisfactory throttle is ensured.

(3-5) The polytetrafluoroethylene that is excellent in a sliding performance is suitable for a material of the blocking body 51.

[0033] In the present invention, the following embodiments are also practiced.

[0034] Firstly, a bleed passage that linearly extends through the cylinder block 19, the suction valve plate 21, the valve port plate 20 and the discharge valve plate 22 may connect with the crank chamber 121 and the suction chamber 111.

[0035] Secondly, in the second preferred embodiment, the refrigerant passage 48 and the throttled passage 49 may be omitted. In place of the refrigerant passage 48 and the throttled passage 49, a throttled passage may be formed in the shaft seal 44A.

[0036] Thirdly, in the third preferred embodiment, a blocking body 51 made of rubber may be employed. For example, a nitrile-butadiene rubber may be employed.

[0037] The nitrile-butadiene rubber that is excellent in a resistance to deterioration relative to refrigerant and lubricant oil is suitable for a material of the blocking body 51. While the blocking body 51 made of rubber is formed by molding, the elastic deformation of rubber allows rubber to be lower accuracy in size than synthetic resin. Therefore, the blocking body 51 made of rubber is manufactured more easily than a blocking body made of synthetic resin.

[0038] 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.

Claims

1. A variable displacement piston type compressor comprising:

a housing having a cylinder block including a plurality of cylinder bores, the housing defining a crank chamber, a suction pressure region and a discharge pressure region, the suction pressure region including a suction chamber, the discharge pressure region including a discharge chamber;
a drive shaft rotatably supported by the housing, the drive shaft having a first end and a second end, the first end of the drive shaft extending through 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 inclinably supported by the drive shaft in the crank chamber, the cam plate being integrally rotated with the drive shaft;
a piston accommodated in each cylinder bore, the rotation of the cam plate being converted into the reciprocating movement of the piston in accordance with the inclination angle of the cam plate, refrigerant in the suction chamber being drawn into the cylinder bores due to the suction work of the piston, the refrigerant in the cylinder bores being discharged into the discharge chamber due to the discharge work of the piston, refrigerant in the discharge pressure region being supplied into the crank chamber and the refrigerant in the crank chamber being bled into the suction pressure region through a bleed passage for controlling pressure in the crank chamber, thereby the inclination angle of

the cam plate being controlled, the crank chamber and the suction chamber being connected with each other through the bleed passage, the bleed passage being formed outside of the drive shaft.

2. The variable displacement piston type compressor according to claim 1, wherein at least a part of the bleed passage is formed in a shaft hole of the cylinder block through which the drive shaft extends.

3. The variable displacement piston type compressor according to claim 2, wherein the drive shaft is rotatably supported by a radial bearing in the cylinder block, the radial bearing being placed in the bleed passage.

4. The variable displacement piston type compressor according to claim 1, wherein the discharge chamber and the suction chamber are separated from the cylinder bores by a valve plate assembly, a part of the bleed passage extends through the cylinder block and the valve plate assembly, and a part of the bleed passage in the valve plate assembly is a throttled passage.

5. The variable displacement piston type compressor according to claim 1, wherein the upstream end of the bleed passage is opened to the bottom of the crank chamber.

6. The variable displacement piston type compressor according to claim 5, wherein the bleed passage passes through the cylinder block and then reaches the circumferential surface of the drive shaft, and the circumferential surface of the drive shaft and the bleed passage in the cylinder block meet more upward in a vertical direction than the central axis of the drive shaft.

7. The variable displacement piston type compressor according to claim 1, further comprising a throttling means having a blocking member in the bleed passage and a throttled passage formed on the blocking member, the blocking member being made of resin or rubber.

8. The variable displacement piston type compressor according to claim 7, wherein the blocking member is made of polytetrafluoroethylene.

9. The variable displacement piston type compressor according to claim 7, wherein the blocking member is made of nitrile-butadiene rubber.

10. The variable displacement piston type compressor according to claim 1, wherein the cam plate is inclinable swash plate for varying displacement of the compressor.

11. The variable displacement piston type compressor according to claim 1, further comprising lubricant oil that fluids with the refrigerant therein.

12. A variable displacement piston type compressor comprising:

a cylinder block including a plurality of cylinder bores;
a piston for compressing refrigerant in each cylinder bore;
a cam plate movably connected to the piston for reciprocating the piston;
a crank chamber defined near one end of the cylinder block;
a drive shaft having a rotational axis for rotating to drive the cam plate, wherein the drive shaft is urged in a direction of the rotational axis while the piston reciprocates;
a suction chamber defined near the opposite end to the crank chamber relative to the cylinder block, the refrigerant in the suction chamber being drawn into the cylinder bores due to the suction work of the piston;
a discharge chamber defined near the opposite end to the crank chamber relative to the cylinder block, the refrigerant in the cylinder bores being discharged to the discharge chamber due to the discharge work of the piston; and
wherein the refrigerant in the discharge chamber is supplied into the crank chamber and the refrigerant in the crank chamber is bled into the suction chamber through a bleed passage for controlling pressure in the crank chamber, the bleed passage being formed outside of the drive shaft.

13. The variable displacement piston type compressor according to claim 12, wherein at least a part of the bleed passage is formed in a shaft hole of the cylinder block through which the drive shaft extends.

14. The variable displacement piston type compressor according to claim 13, wherein the drive shaft is rotatably supported by a radial bearing in the cylinder block, the radial bearing being placed in the bleed passage.

15. The variable displacement piston type compressor according to claim 12, wherein the discharge chamber and the suction chamber are separated from the cylinder bores by a valve plate assembly, a part of the bleed passage extends through the cylinder block and the valve plate assembly, and a part of the bleed passage in the valve plate assembly is a throttled passage.

16. The variable displacement piston type compressor according to claim 12, wherein the upstream end of the bleed passage is opened to the bottom of the crank chamber.

17. The variable displacement piston type compressor according to claim 16, wherein the bleed passage passes through the cylinder block and then reaches the circumferential surface of the drive shaft, and the circumferential surface of the drive shaft and the bleed passage in the cylinder block meet more upward in a vertical direction than the central axis of the drive shaft.

18. The variable displacement piston type compressor according to claim 12, further comprising a throttling means having a blocking member in the bleed passage and a throttled passage formed on the blocking member, the blocking member being made of resin or rubber.

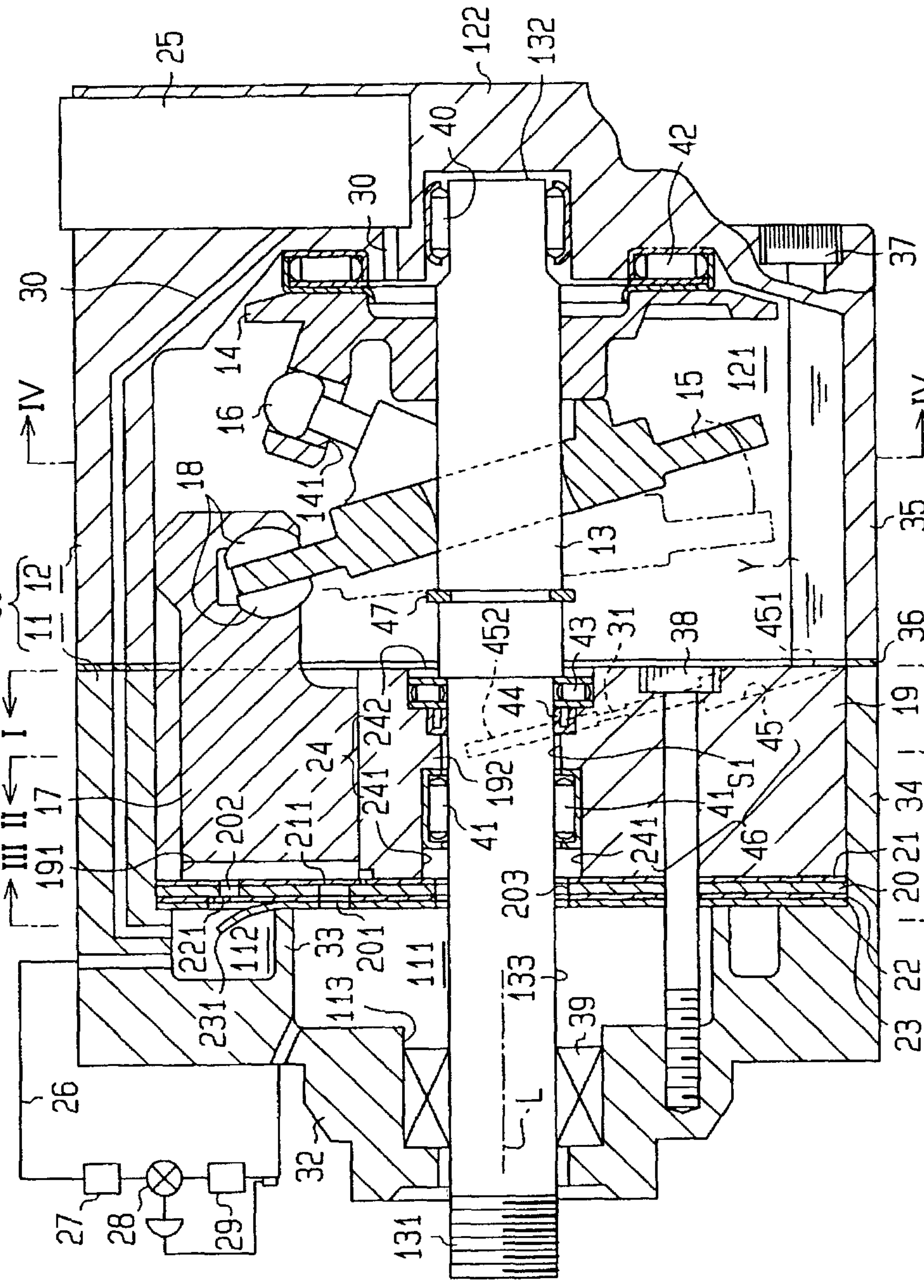

FIG. 1A

FIG. 1B

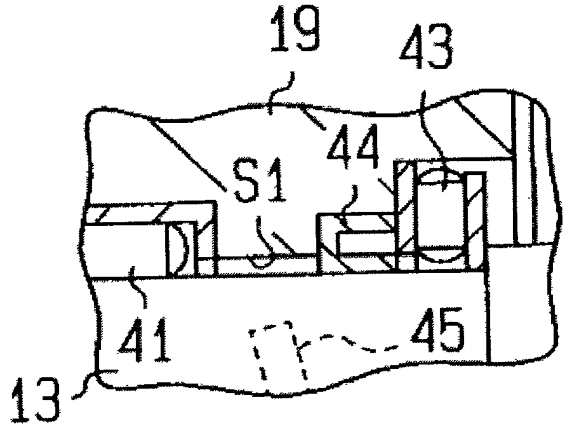

FIG. 2

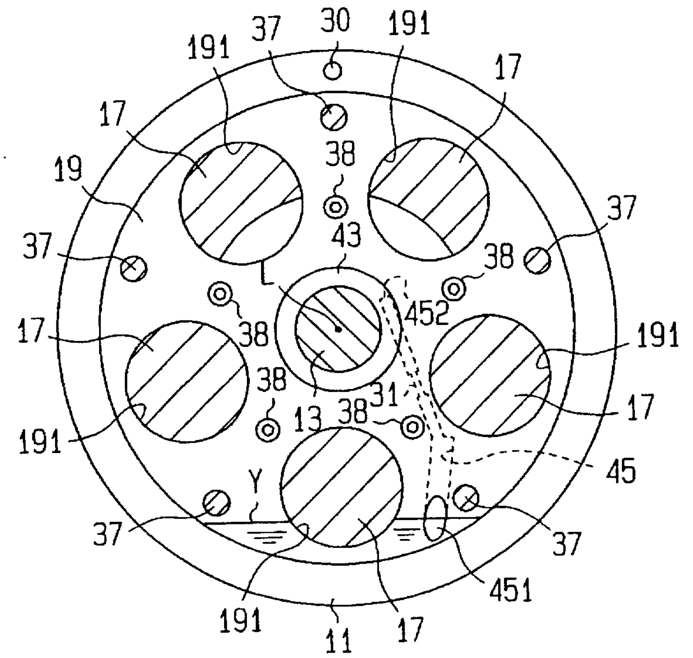
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FIG. 3

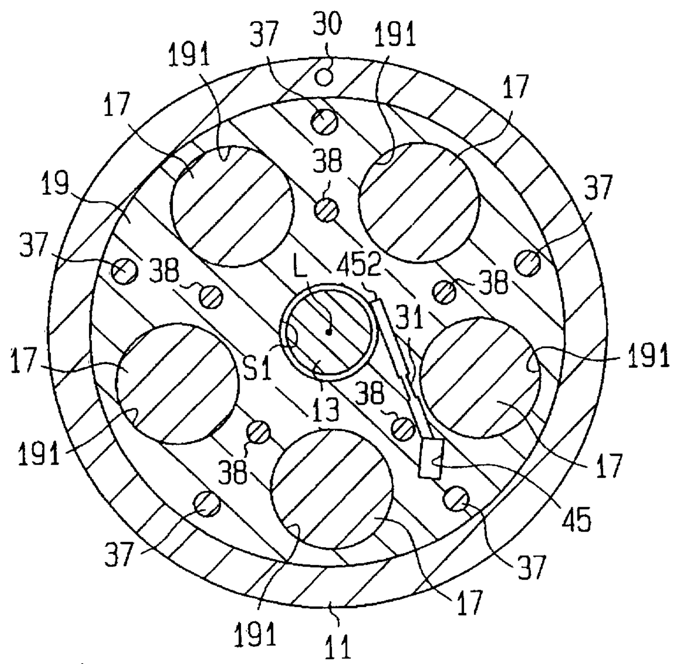
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FIG. 4

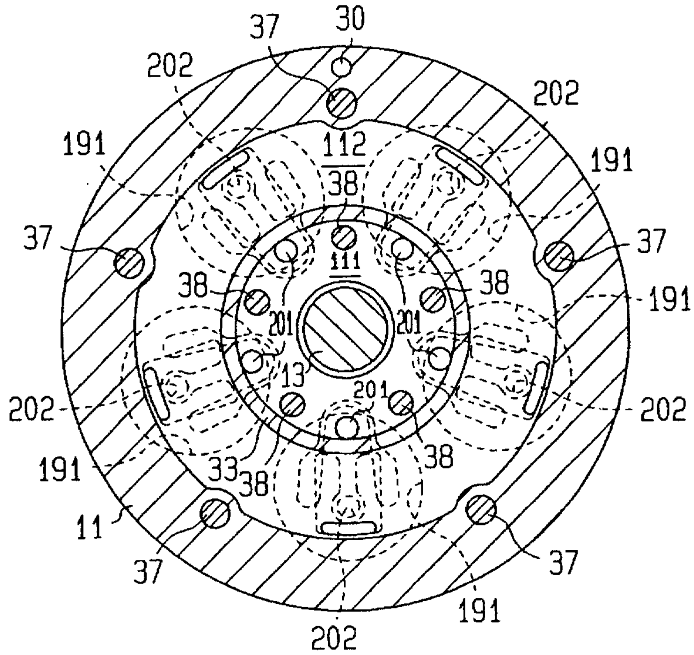
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FIG. 5

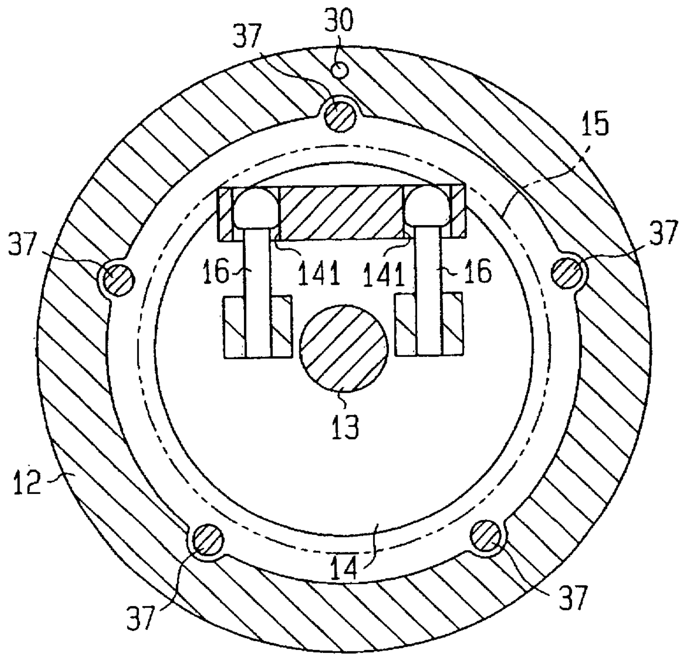
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FIG. 6

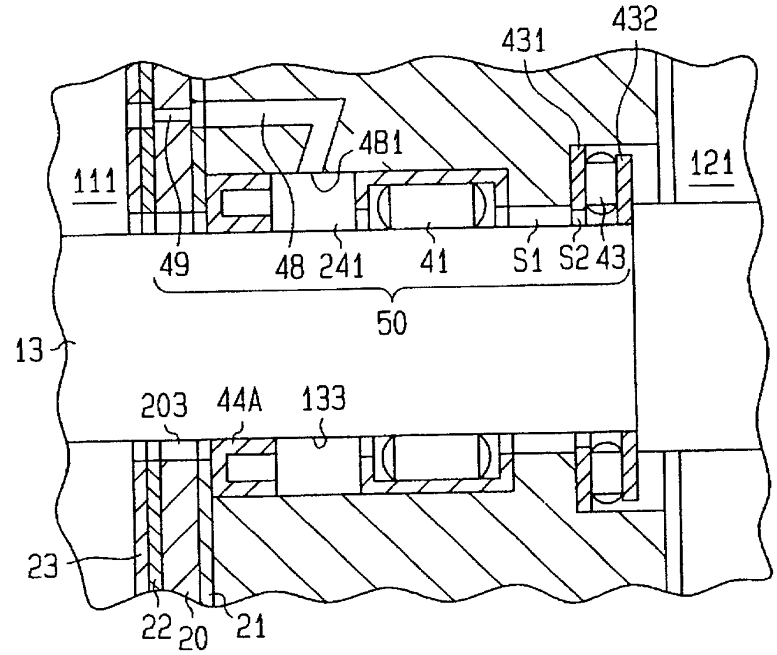

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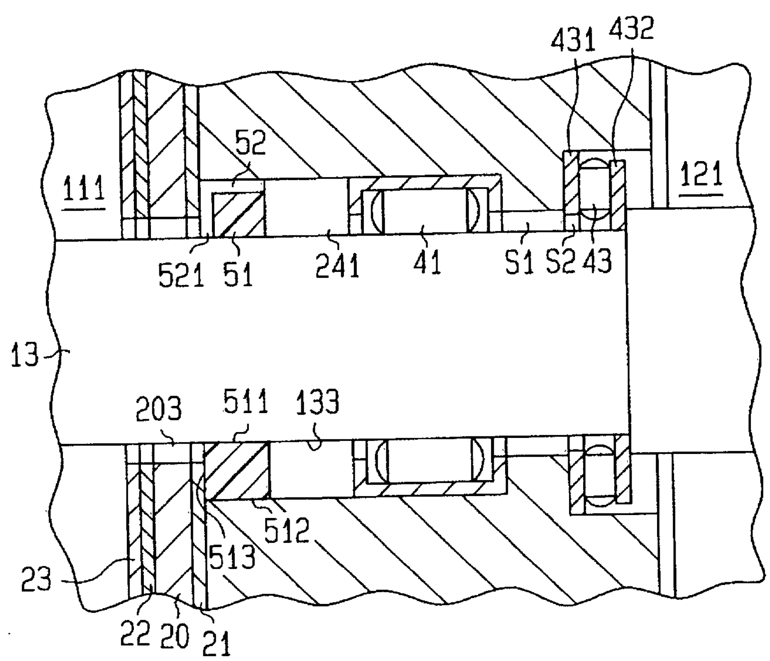
FIG. 7



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