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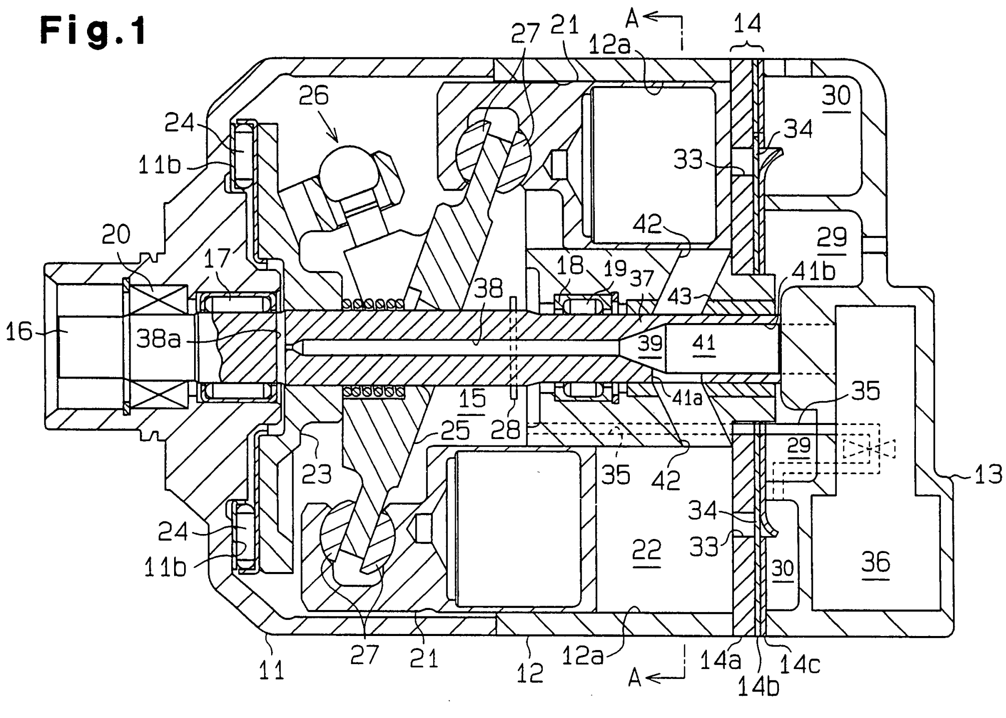
(54) **Swash plate compressor valve**

(57) A shaft (16) and a rotary valve (37), which are formed integrally, are made of an iron-based metal while a cylinder block (12) is made of an aluminum-based metal. A sleeve (43) forms a slide surface between the cylinder block (12) and the rotary valve (37) when the shaft (16) and the rotary valve (37) rotate together. The sleeve (43) has a coefficient of thermal expansion closer to those of the shaft (16) and the rotary valve (37) than that of the cylinder block (12). This structure prevents an increase in the clearance between the housing and the rotary valve, due to the increased temperature at the time the shaft rotates at a high speed, and prevents gas leakage and a reduction in sealability.

Fig.1



Description

[0001] The present invention relates to a bearing structure of a rotary valve in a swash plate type compressor suitable for air-conditioning of a vehicle.

[0002] A reciprocative compressor as disclosed in Japanese Unexamined Patent Publication No. Hei 6-137265 is known. This compressor has a cylinder block having a plurality of cylinder bores around the axial center, a shaft inserted into an axial hole of the cylinder block, a plurality of pistons which are coupled to a swash plate in a crank chamber that operates together with the shaft and reciprocate in the respective cylinder bores, and a housing which has a suction chamber communicatable to the axial hole of the cylinder block and a discharge chamber formed in an outward area of the suction chamber and closes end faces of the cylinder block. This type of compressor has communication passages formed between the respective cylinder bores and the axial hole of the cylinder block, and a rotary valve coupled to the shaft in such a way as to be rotatable in synchronism with the shaft. The rotary valve has a suction passage for sequentially connecting the communication passages of the individual cylinder bores in a suction stroke to the suction chamber. The shaft is made of an iron-based metal, and the rotary valve of an aluminum-based metal. An engage hole is bored in one end portion of the rotary valve. Attached to the engage hole is a steel liner that has a base plate portion, which abuts on one end of the shaft, and extruding pieces, which are split from the base plate portion through selectively bending and are fitted in the engage hole. The split opening portion of the liner is fitted over an engage shaft protruding from the shaft end.

[0003] Because the shaft and the rotary valve in such a compressor are formed of different members, however, the compressor has a larger number of components. To reduce the number of the components, the shaft and the rotary valve may be formed integrally. From the viewpoint of securing the strength, the shaft is often made of an iron-based metal having rigidity. In a case where the shaft and the rotary valve may be formed integrally, therefore, the rotary valve is likely to be made of an iron-based metal. Generally, the housing is made of an aluminum-based metal to become lighter. As the shaft rotates at a high speed, therefore, the temperature of the slide surface between the housing and rotary valve, which are made of different metals, rises and the clearance between the housing and the rotary valve increases due to the difference between their coefficients of thermal expansion. The increased clearance leads to gas leakage and a reduction in sealability, which would lower the performance of the compressor.

[0004] Accordingly, it is an object of the present invention to provide a swash plate type compressor which prevents an increase in the clearance between the housing and the rotary valve when the shaft rotates at a high speed.

[0005] A swash plate type compressor includes a housing having a cylinder block. The cylinder block has a plurality of cylinder bores around a shaft. The shaft is rotatably supported in the housing. An suction pressure area is formed in the housing. A plurality of pistons are respectively inserted into the cylinder bores and reciprocate in the respective cylinder bores via a swash plate in accordance with rotation of the shaft to thereby perform a suction stroke for taking a refrigerant gas in the suction pressure area into a compression chamber formed in each cylinder bore. The swash plate type compressor comprises communication passages, a rotary valve, and a sleeve. The communication passages is formed in the cylinder block in such a way as to communicate with the cylinder bores, respectively. The rotary valve is formed integral with the shaft. The rotary valve has a suction passage for connecting the communication passage of each cylinder bore in the suction stroke to the suction pressure area. The sleeve is provided on the rotary valve in the cylinder block. The sleeve has a coefficient/of thermal expansion closer to that of the shaft than that of the cylinder block.

[0006] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

[0007] The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a schematic cross-sectional view of a compressor according to one embodiment of the invention taken along the line B-B in Fig. 2;

Fig. 2 is a cross-sectional view of the compressor taken along the line A-A in Fig. 1; and

Fig. 3 is a partly enlarged cross-sectional view showing the details of the compressor in Fig. 1.

[0008] One embodiment of the present invention as embodied into a swash plate type compressor which is used in a vehicular air-conditioning system will be described below with reference to Figs. 1 to 3.

[0009] As shown in Fig. 1, a front housing 11 is connected to the front end of a cylinder block 12. A rear housing 13 is connected to the rear end of the cylinder block 12 via a valve plate assembly 14. The front housing 11, the cylinder block 12 and the rear housing 13 are fastened by through bolts 11a (see Fig. 2) and constitute the housing of the compressor. The front housing 11, the cylinder block 12 and the rear housing 13 are made of an aluminum-based metal. The coefficient of thermal expansion of aluminum is about 19 to 23 x 10^{-6}/°C. Note that the lefthand side in Fig. 1 shows the frontward of the compressor, and the right-hand side shows the rearward thereof.

[0010] The valve plate assembly 14 includes a main plate 14a, a sub plate 14b stacked over the rear surface of the main plate 14a and a retainer plate 14c stacked over the rear surface of the sub plate 14b. The valve plate assembly 14 is connected to the cylinder block 12 at the front surface of the main plate 14a.

[0011] A crank chamber 15 is defined between the front housing 11 and the cylinder block 12. A shaft 16 is rotatably supported between the front housing 11 and the cylinder block 12 in such a way as to pass through the crank chamber 15. The front end portion of the shaft 16 is supported on the front housing 11 via a radial bearing 17. A retaining hole 18 is formed nearly in the center of the cylinder block 12. The rear end portion of the shaft 16 is supported on a radial bearing 19 which is provided in the retaining hole 18. A shaft seal 20 is provided at the front end portion of the shaft 16. The shaft 16 is made of an iron-based metal. The coefficient of thermal expansion of iron is about 10 to 12 x 10^{-6}/°C.

[0012] A plurality of cylinder bores 12a (only two shown in Fig. 1) are formed in the cylinder block 12 in such a way as to surround the shaft 16 at equiangles and equal distances. One-headed pistons 21 are retained in a reciprocative manner in the respective cylinder bores 12a. The front opening of each cylinder bore 12a is closed by the front surface of the associated piston 21, and the rear opening of that cylinder bore 12a is closed by the front surface of the valve plate assembly 14. A compression chamber 22 is defined in each cylinder bore 12a and its volume varies in accordance with the reciprocation of the associated piston 21.

[0013] A lug plate 23 is fixed to the shaft 16 in the crank chamber 15 in such a way as to be rotatable together with the shaft 16. The lug plate 23 is abuttable on an inner wall surface 11b of the front housing 11 via a thrust bearing 24. The inner wall surface 11b supports the axial weight originated from the compression repulsive force of each piston 21 and restricts the forward slide movement of the shaft 16.

[0014] A swash plate 25 is provided in the crank chamber 15 with the shaft 16 put through a through hole formed in the swash plate 25. A hinge mechanism 26 is positioned between the lug plate 23 and the swash plate 25. The hinge coupling with the lug plate 23 via the hinge mechanism 26 and the support on the shaft 16 allow the swash plate 25 to be rotatable in synchronism with the lug plate 23 and the shaft 16, be slidable in the axial direction of the shaft 16 and be tiltable with respect to the shaft 16. The lug plate 23 and the hinge mechanism 26 constitute a variable displacement mechanism.

[0015] Each piston 21 is engaged with the peripheral portion of the swash plate 25 via a shoe 27. The rotation of the shaft 16 is converted into the reciprocating motion of the pistons 21 via the swash plate 25 and the shoes 27.

[0016] The lug plate 23, the swash plate 25, the hinge mechanism 26 and the shoes 27 constitute a crank mechanism which converts the rotational movement of the shaft 16 to a compression action to compress the refrigerant gas in the compression chamber 22.

[0017] A restricting portion 28 is provided on the shaft 16 between the swash plate 25 and the cylinder block 12. The restricting portion 28 is a ring-like member secured to the outer surface of the shaft 16. The minimum inclination angle of the swash plate 25 is defined by the abutment on the restricting portion 28, while the maximum inclination angle of the swash plate 25 is defined by the abutment on the lug plate 23.

[0018] As shown in Fig. 1, a suction chamber 29 and a discharge chamber 30 are defined in the rear housing 13. Discharge ports 33 and discharge valves 34 which open and close the respective ports 33 are formed in the valve plate assembly 14 in association with the respective cylinder bores 12a. Each cylinder bore 12a communicates with the discharge chamber 30 via the respective discharge port 33. The suction chamber 29 is connected to the discharge chamber 30 via an external refrigeration circuit (not show).

[0019] An supply passage 35 which connects the crank chamber 15 to the discharge chamber 30 is formed in the cylinder block 12 and the rear housing 13. Disposed in the supply passage 35 is a control valve 36 which constitutes the variable displacement mechanism. The control valve 36 is a known solenoid valve. The control valve 36 provides a valve chamber in the supply passage 35. The angle of opening of the control valve 36 is adjustable by the amount of the excitation current of the solenoid. The control valve 36 also serves as a restrictor. Therefore, the supply passage 35 is closed by the excitation of the solenoid and is released by the deexcitation of the solenoid.

[0020] The rear end portion of the shaft 16 forms a rotary valve 37. The shaft 16 is integral with the rotary valve 37 so that as the shaft 16 rotates, the rotary valve 37 rotates together with the shaft 16. The shaft 16 and the rotary valve 37 are made of the same iron-based metal. A circulation passage 38 is formed in the shaft 16 and the rotary valve 37. An oil separator 39, which separates oil from the refrigerant gas, is provided at the rear end portion of the circulation passage 38, i.e., nearly the center portion of the rotary valve 37. Coating is applied to the surfaces of the shaft 16 and the rotary valve 37.

[0021] An inlet 38a of the circulation passage 38 is formed in the rearward of the radial bearing 17. The rear end portion of the circulation passage 38 is widened by the oil separator 39 and forms a communication chamber 41b. The rear end of the communication chamber 41b is connected to the suction chamber 29 in such a way that the refrigerant gas flows there. Accordingly, the circulation passage 38 constitutes a bleed passage which connects the crank chamber 15 to the suction chamber 29.

[0022] The inner surface of the oil separator 39 is inclined in such a way that the inside diameter of the oil separator 39 becomes larger toward the rear end, which is the downstream side to the flow of the refrigerant gas from the crank chamber 15 to the suction chamber 29, from the distal end which is the upstream side. The diameter of the oil separator 39 is the largest at the rear end.

[0023] A communication hole 41a which communicates with the circulation passage 38 from the side is formed in the rotary valve 37, as shown in Fig. 1. As the rotary valve 37 rotates in the direction of the arrow in Fig. 2 in accordance with the rotation of the shaft 16, communication passages 42 of the cylinder bores 12a communicate with the communication hole 41a. The communication hole 41a and the communication chamber 41b constitute a suction passage 41.

[0024] The suction passage 41 is provided on a rearer end side (the downstream side or right-hand side in Fig. 1) to the shaft 16 than the oil separator 39. One end of the communication passage 42 communicates with the associated cylinder bore 12a and the other end of the passage 42 is located in a position corresponding to the suction passage 41 (communication hole 41a). When the rotary valve 37 rotates, the communication passage 42 of the cylinder bore 12a in a suction stroke communicates with the suction passage 41, while the communication passage 42 of the cylinder bore 12a in a discharge stroke does not communicate with the suction passage 41. At this time, the slide surface (sealed portion) between the rotary valve 37 and the cylinder block 12 is sealed in an air-tight manner.

[0025] The slide surface between the rotary valve 37 and the cylinder block 12 is formed by a sleeve 43. The sleeve 43 is fitted in the cylinder block 12 by casting or press fitting. The sleeve 43 is made of an iron-based metal which has a coefficient of thermal expansion closer to those of the shaft 16 and the rotary valve 37.

[0026] The action of the compressor with the above-described structure will be discussed below.

As the shaft 16 rotates, the swash plate 25 rotates together with the shaft 16 via the lug plate 23 and the hinge mechanism 26. The rotational motion of the swash plate 25 is converted to the reciprocating motion of the pistons 21 via the shoes 27. As this driving continues, the suction, compression and discharge of the refrigerant are repeated one after another in the compression chamber 22. The refrigerant is supplied to the suction chamber 29 from the external refrigeration circuit, is fed into the compression chamber 22 (suction stroke), is compressed by the movement of the associated piston 21 (compression stroke) and is discharged to the discharge chamber 30 via the associated discharge port 33 (discharge stroke). The discharged refrigerant is fed out to the external refrigeration circuit via a discharge passage.

[0027] Then, a control apparatus (not shown) adjusts the degree of opening of the control valve 36 or the degree of opening of the supply passage 35 in accordance with the refrigerant load, thereby changing the state of communication of the discharge chamber 30 with the crank chamber 15.

[0028] When the refrigerant load is large, the degree of opening of the supply passage 35 is reduced, thereby decreasing the flow rate of the refrigerant gas to be supplied to the crank chamber 15 from the discharge chamber 30. As the amount of the refrigerant gas to be supplied to the crank chamber 15 is reduced, the pressure in the crank chamber 15 gradually drops due to the escape of the refrigerant gas to the suction chamber 29 via the circulation passage 38 or the like. As a result, the difference between the pressure in the crank chamber 15 and the pressure in each cylinder bore 12a via the associated piston 21 becomes smaller, so that the swash plate 25 is displaced in the direction of increasing the inclination angle (leftward in Fig. 1). Therefore, the amount of the stroke of the piston 21 increases, thus making the discharge volume greater.

[0029] When the refrigerant load becomes smaller, on the other hand, the degree of opening of the control valve 36 is increased, thereby increasing the flow rate of the refrigerant gas to be supplied to the crank chamber 15 from the discharge chamber 30. When the amount of the refrigerant gas to be supplied to the crank chamber 15 exceeds the escape amount of the refrigerant gas to the suction chamber 29 via the circulation passage 38, the pressure in the crank chamber 15 gradually rises. Consequently, the difference between the pressure in the crank chamber 15 and the pressure in each cylinder bore 12a via the associated piston 21 becomes larger, so that the swash plate 25 is displaced in the direction of decreasing the inclination angle (rightward in Fig. 1). The amount of the stroke of the piston 21 therefore decreases, thus reducing the discharge volume.

[0030] The refrigerant gas which is fed toward the suction chamber 29 via the circulation passage 38 is whirled in accordance with the rotation of the oil separator 39. This causes the centrifugal separation of the oil from the refrigerant gas. The separated oil is discharged out of the oil separator 39 by the centrifugal force or the like based on the rotation of the oil separator 39. The discharged oil is supplied between the rotary valve 37 and the cylinder block 12 and between the piston 21 and the associated cylinder bore 12a via the suction passage 41 and the associated communication passage 42.

[0031] Part of the refrigerant gas, from which the oil has been separated in the oil separator 39, is supplied to the suction chamber 29 via the communication chamber 41b. The refrigerant gas supplied to the suction chamber 29 (this gas has a small amount of oil mixed therein) is discharged to the external refrigeration circuit via the associated compression chamber 22 and the discharge chamber 30.

[0032] As the shaft 16 and the rotary valve 37 rotate together, the refrigerant gas in the suction chamber 29 is sucked into each cylinder bore 12a via the suction passage 41 of the shaft 16 and the communication passage 42 of that bore 12a in the suction stroke. Because the suction of the refrigerant gas continues in each cylinder bore 12a smoothly and stably, a pressure loss becomes extremely small.

[0033] The sleeve 43 serves as a rotary valve receiving portion of the cylinder block 12. The sleeve 43 is formed by casting or press fitting in the cylinder block 12. When the shaft 16 rotates at a high speed, the rotary valve 37 slides with respect to the sleeve 43, raising the temperature of the slide surface therebetween. Since the rotary valve 37 and the sleeve 43 are both made of an iron-based metal and their coefficients of thermal expansion are almost equal to each other, the clearance between the rotary valve 37 and the sleeve 43 can be prevented from increasing.

[0034] The above-mentioned embodiment have the following advantages.

[0035] The rotary valve 37 and the shaft 16 are integrally formed of an iron-based metal and the sleeve 43 is made of an iron-based metal whose coefficient of thermal expansion is closer to that of the shaft 16 (and the rotary valve 37). This can reduce the number of components and prevents an increase in the clearance between the slide surfaces of the rotary valve 37 and the sleeve 43, which would be caused by a temperature rise at the time the shaft 16 rotates at a high speed. This prevents gas leakage from the clearance and a reduction in the performance of the compressor. The sleeve 43 maintains the sealability between the rotary valve 37 and the cylinder block 12 over a long period of time. It is therefore possible to smoothly rotate the rotary valve 37 and suppress sliding noise of the rotary valve 37.

[0036] The rotary valve 37 and the sleeve 43 are made of an iron-based metal, which is excellent in rigidity over an aluminum-based metal. This can ensure a high strength.

[0037] Coating is applied to the surfaces of the shaft 16 and the rotary valve 37. The coating can prevent burning of the shaft 16 and the rotary valve 37 when they rotate together.

[0038] The control valve 36 is provided in the supply passage 35. The control valve 36 can control the pressure in the crank chamber 15 by using the high pressure in the discharge chamber 30. Thus, discharge volume can be controlled with high accuracy.

[0039] The inner surface of the oil separator 39 is inclined in such a way that the inside diameter of the oil separator 39 becomes larger from the upstream side toward the downstream of the flow of the refrigerant gas with respect to the flow of the refrigerant gas. This facilitates the oil adhered to the inner surface of the oil separator 39 to be discharged outside from the downstream of the oil separator 39 by the centrifugal force at the time the shaft 16 rotates.

[0040] It should be apparent to those skilled in the art that the present invention may be embodied in many other specific forms without departing from the spirit or scope of the invention. Particularly, it should be understood that the invention may be embodied in the following forms.

[0041] The iron-based metal for the sleeve can be any metal as long as its coefficient of thermal expansion is close to that of the iron-based metal for the shaft. For example, such iron-based metals available include gray cast iron (11 to 12 x 10^{-6}/°C), ductile iron (11 to 12 x 10^{-6}/°C) and Ni-resist D-3 (8 to 9 x 10^{-6}/°C). In this case, similar advantages to those obtained when iron is used are also obtained.

[0042] Since the coefficient of thermal expansion of aluminum is approximately 19 to 23 x 10^{-6}/°C and the coefficient of thermal expansion of iron is approximately 10 to 12 x 10^{-6}/°C, the coefficient of thermal expansion of the iron-based metal for the sleeve can lie in a range of approximately 7 to 15 x 10^{-6}/°C.

[0043] The sleeve may be made of any material other than a metal, as long as its coefficient of thermal expansion is close to that of the iron-based metal for the shaft. That is, a resin or ceramics may be used in place of a metal. Of ceramics, alumina which has a coefficient of thermal expansion of 6 to 8 x 10^{-6}/°C and zirconia which has a coefficient of thermal expansion of 9 to 11 x 10^{-6}/°C can be used, for example. Those of various kinds of engineering plastics whose coefficients of thermal expansion are near 10 to 13 x 10^{-6}/°C may be used. In this case, advantages similar to those mentioned above are also obtained.

[0044] The suction chamber 29, which is provided in the rear housing 13, may be omitted. In this case, the refrigerant is led directly into the communication chamber 41b, which constitutes an suction pressure area.

[0045] The radial bearing 19 may be omitted. The shaft 16 may be supported by the sleeve 43 only.

[0046] The compressor may be a wobble type variable displacement compressor.

[0047] The compressor may be a double-headed piston type compressor.

[0048] Therefore, the present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.

[0049] A shaft (16) and a rotary valve (37), which are formed integrally, are made of an iron-based metal while a cylinder block (12) is made of an aluminum-based metal. A sleeve (43) forms a slide surface between the cylinder block (12) and the rotary valve (37) when the shaft (16) and the rotary valve (37) rotate together. The sleeve (43) has a coefficient of thermal expansion closer to those of the shaft (16) and the rotary valve (37) than that of the cylinder block (12). This structure prevents an increase in the clearance between the housing and the rotary valve, due to the increased temperature at the time the shaft rotates at a high speed, and prevents gas leakage and a reduction in sealability.

Claims

1. A swash plate type compressor including a housing having a cylinder block (12), the cylinder block (12) having a plurality of cylinder bores (12a) around a shaft (16), the shaft (16) being rotatably supported in the housing, an suction pressure area being formed in the housing, wherein a plurality of pistons (21) which are respectively inserted into the cylinder bores (12a) and reciprocate in the respective cylinder bores (12a) via a swash plate (25) in accordance with rotation of the shaft (16) to thereby perform a suction stroke for taking a refrigerant gas in the suction pressure area into a compression chamber (22) formed in each cylinder bore (12a), the swash plate type compressor comprising:

   communication passages (42) formed in the cylinder block in such a way as to communicate with the cylinder bores (12a), respectively; and a rotary valve (37) formed integral with the shaft (16), wherein the rotary valve (37) has a suction passage (41) for connecting the communication passage (42) of each cylinder bore (12a) in the suction stroke to the suction pressure area;

   characterized in that the swash plate type compressor comprises
   a sleeve (43) provided on the rotary valve (37) in the cylinder block, wherein the sleeve (43) has a coefficient of thermal expansion closer to that of the shaft (16) than that of the cylinder block (12).

2. The swash plate type compressor according to claim 1, **characterized in that** the suction pressure area includes a suction chamber (29), a discharge chamber (30) is further formed in the housing, each of the pistons performs a suction stroke for taking the refrigerant gas in the suction chamber (29) into a compression chamber (22) formed in each cylinder bore (12a), a compression stroke for compressing the refrigerant gas and a discharge stroke for discharging the refrigerant gas, sucked into the compression chamber (22), into the discharge chamber (30), and the suction passage (41) connects the communication passage (42) of each cylinder bore (12a) in the suction stroke to the suction chamber (29).

3. The swash plate type compressor according to claim 1 or 2, **characterized in that** the sleeve (43) forms a slide surface to the rotary valve (37).

4. The swash plate type compressor according to any one of claims 1 to 3, **characterized in that** the shaft (16) and the rotary valve (37) are made of an iron-based metal and the cylinder block (12) is made of an aluminum-based metal.

5. The swash plate type compressor according to any one of claims 1 to 4, **characterized in that** the sleeve (43) is made of an iron-based metal.

6. The swash plate type compressor according to any one of claims 1 to 4, **characterized in that** the sleeve (43) is made of a resin or ceramics.

7. The swash plate type compressor according to any one of claims 1 to 3, **characterized in that** the shaft (16), the rotary valve (37) and the sleeve (43) are made of an iron-based metal.

8. The swash plate type compressor according to claim 5 or 7, **characterized in that** the coefficient of thermal expansion of the sleeve (43) ranges about 7 to 15 x 10^{-6}/°C .

9. The swash plate type compressor according to claim 1, **characterized in that** the sleeve (43) is cast or press-fitted in the cylinder block.

Fig.1

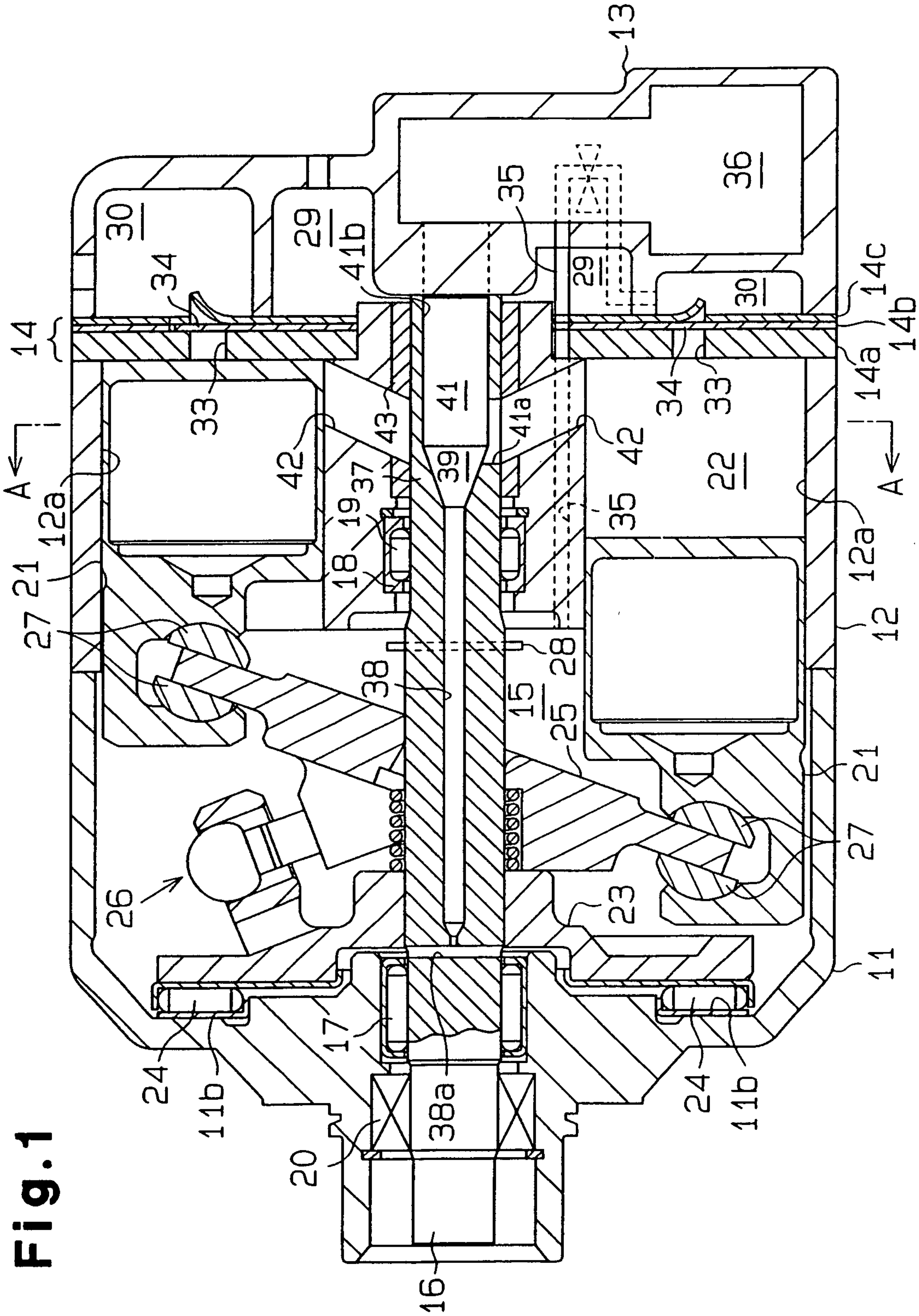

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Fig.2

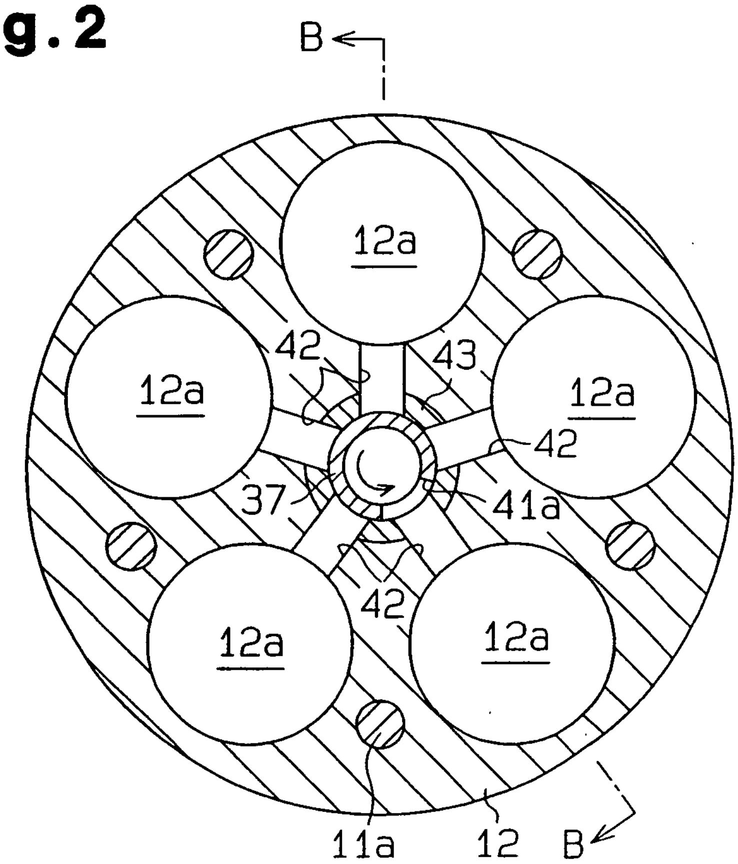
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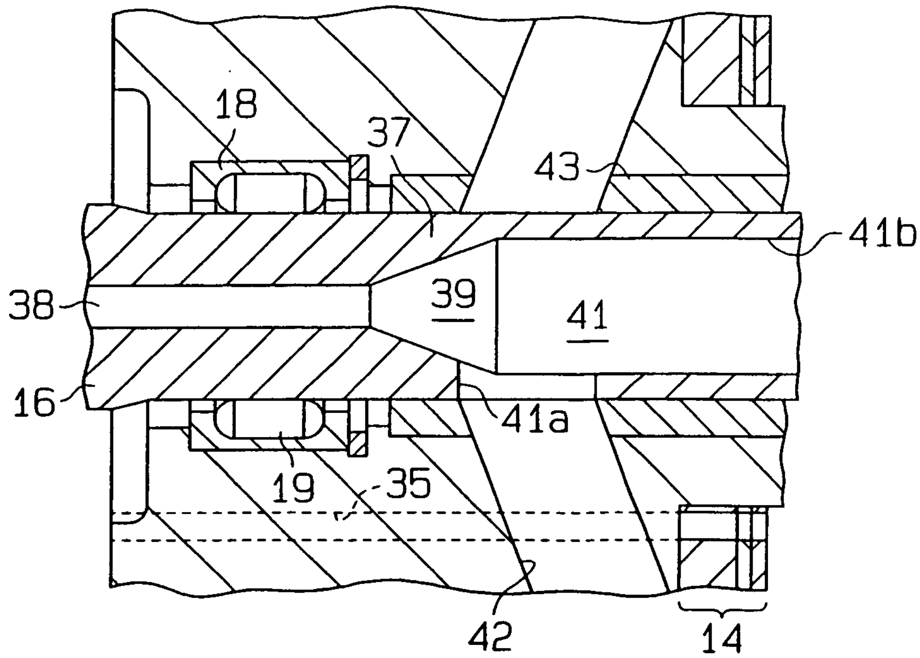
Fig.3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