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(54) **ROTARY COMPRESSOR**

ROTATIONSVERDICHTER

COMPRESSEUR ROTATIF

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EP 2 589 810 B1

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Description

[Technical Field]

[0001] The present invention relates to a rotary compressor used for apparatuses such as an air conditioner, a freezer, an air blower and a hot water supply apparatus.

[Background Technique]

[0002] A rotary type compressor which sucks gas refrigerant evaporated by an evaporator and compresses the sucked gas refrigerant is used for an apparatus such as an air conditioner. A rotary compressor is known as one of such rotary compressors (see patent document 1 for example).

[0003] Fig. 15 is a sectional view of an essential portion showing one example of the rotary compressor.

[0004] An electric motor 2 and a compressor mechanism 3 are connected to each other through a crankshaft 31 and they are accommodated in a hermetic container 1. An oil reservoir 6 is formed in a bottom in the hermetic container 1. The compressor mechanism 3 includes a cylinder 30 which forms a cylindrical inner space, a piston 32 disposed in the inner space of the cylinder 30, an end plate 34 of an upper bearing 34a which closes an upper end surface of the cylinder 30, an end plate 35 of a lower bearing 35a which closes a lower end surface of the cylinder 30, and a vane 33 which partitions an interior of a compression chamber 39 into a low pressure portion and a high pressure portion.

[0005] The compression chamber 39 is made up of the inner space of the cylinder 30, the piston 32 and the end plates 34 and 35. The crankshaft 31 is supported by the upper bearing 34a and the lower bearing 35a. An eccentric core 31a is formed on the crankshaft 31. The eccentric core 31a is disposed between the end plates 34 and 35. The piston 32 is fitted over the eccentric core 31a. The vane 33 reciprocates in a slot provided in the cylinder 30. A tip end of the vane 33 comes into contact with an outer periphery of the piston 32 under pressure, and the vane 33 follows the eccentric rotation of the piston 32 and reciprocates, thereby partitioning the interior of the compression chamber 39 into the low pressure portion and the high pressure portion.

[0006] An oil hole 41 is formed in the crankshaft 31 along its axis. Oil (lubricant oil) in the oil reservoir 6 is supplied to the oil hole 41. A wall of the crankshaft 31 is provided with oil supply holes 42 and 43 which are in communication with the oil hole 41. The oil supply hole 42 is formed in a wall corresponding to the upper bearing 34a and the oil supply hole 43 is formed in a wall corresponding to the lower bearing 35a. An oil groove (not shown) and an oil supply hole (not shown) which is in communication with the oil hole 41 are formed in the wall of the eccentric core 31a.

[0007] A suction port 40 through which low pressure gas is sucked is formed in the cylinder 30. The suction

port 40 is in communication with a low pressure portion (suction chamber) in the compression chamber 39. A discharge port 38 is formed in the upper bearing 34a and high pressure gas compressed in the compression chamber 39 is discharged through the discharge port 38. The discharge port 38 is in communication with the high pressure portion in the compression chamber 39. The discharge port 38 is formed as a hole which is circular as viewed from above, and the discharge port 38 penetrates the upper bearing 34a. The discharge port 38 is provided at its upper surface with a discharge valve 36. The discharge valve 36 is opened when it receives a pressure greater than a predetermined value. The discharge valve 36 is covered with a cup muffler 37.

[0008] The low pressure portion (suction chamber) of the compression chamber 39 is gradually enlarged after sliding portions between the piston 32 and the cylinder 30 pass through the suction port 40, and the low pressure portion sucks gas from the suction port 40. On the other hand, when the sliding portions between the piston 32 and the cylinder 30 approach the discharge port 38, the high pressure portion of the compression chamber 39 is gradually reduced in size, and when the high pressure portion is compressed to a value greater than a predetermined pressure, the discharge valve 36 opens and gas flows out from the discharge port 38. Gas which flows out from the discharge port 38 is discharged into the hermetic container 1 through the cup muffler 37.

[0009] An upper space in the piston is formed by the eccentric core 31a of the crankshaft 31, the end plate 34 of the upper bearing 34a and an inner peripheral surface of the piston 32. A lower space in the piston is formed by the eccentric core 31a of the crankshaft 31, the end plate 35 of the lower bearing 35a and the inner peripheral surface of the piston 32. Oil in the oil hole 41 leaks from the oil supply hole 42 into the upper space in the piston, and oil in the oil hole 41 leaks from the oil supply hole 43 into the lower space in the piston. Pressures in the upper space and the lower space in the piston are always higher than a pressure in the compression chamber 39.

[0010] A height of the cylinder 30 must be set slightly higher than the piston 32 so that the piston 32 can slide in the cylinder 30. As a result, a gap is generated between upper and lower end surfaces of the piston 32 and the end plates 34 and 35. Therefore, oil leaks from the upper space and the lower space in the piston into the compression chamber 39 through this gap. To enhance the efficiency, it is necessary to suppress this leakage and maintain the reliability.

[0011] Here, a method of preventing oil from leaking from the upper space and the lower space in the piston into the compression chamber 39 will be explained using Figs. 10 to 14.

[0012] To simplify the explanation, Figs. 10 to 14 show a state where the crankshaft 31 is omitted. Figs. 10 to 14 are schematic diagrams showing a relation of a gap between the piston 32 and the upper and lower end plates 34 and 35 (in the drawings, the vertical direction is ex-

aggregated and an actual size is about a few tens of μm).

[0013] As shown in Fig. 10, upper and lower ends of the inner peripheral surface of the piston 32 are chamfered, and the chamfered portions of the upper and the lower ends are substantially the same.

[0014] When the chamfered portions of the upper and lower ends of the inner peripheral surface of the piston 32 are set equal to each other, the piston 32 slides with respect to the lower end plate 35 due to a weight of the piston 32 of its own. Hence, a gap of about a few tens of μm is created between the upper end surface of the piston 32 and the upper end plate 34, and oil leaks into the compression chamber 39 through this gap.

[0015] A first technique for enhancing the efficiency is to set a difference between upper and lower chamfered portions of the piston 32 to $B-A>0$ as shown in Fig. 11.

[0016] If the difference between the chamfered portions is set to $(B-A>0)$ which cancels the weight of the piston 32 of its own is provided and an upward force is generated, the piston 32 floats.

[0017] Since gas leakage is generally proportional to cube of a gap, if upper and lower gaps of the piston 32 are unevenly distributed, an amount of gas leakage becomes greater as compared with a case where the upper and lower gaps of the piston 32 are evenly distributed. Hence, it is possible to suppress the amounts of gas and oil leaking into the suction chamber through the gaps in the upper and lower end surfaces of the piston 32, and efficiency is enhanced.

[0018] A second technique for enhancing the efficiency is to reduce, in size, a gap between the piston 32 and the upper and lower end plates 34 and 35 to a few tens of μm as shown in Fig. 12.

[0019] By reducing the gap, in size, it is possible to suppress the leakage and enhance the efficiency.

[0020] However, since behavior of the piston 32 is unstable during the actual operation, if the gap is reduced in size, problems of mirror wear and seizing are prone to be generated especially on the lower end plate 35.

[0021] Hence, as shown in Fig. 13, it is necessary to increase, in size, the chamfered portion of the lower end of the piston 32, increase an oil-retaining area between the piston 32 and the lower end plate 35, enhance the cooling effect and enhance the reliability.

[0022] However, if only the lower chamfered portion of the piston 32 is increased, an upward force is generated due the pressure difference. Therefore, the upper end plate 34 is strongly slides.

[0023] Hence, it is necessary to increase, in size, both the upper and lower chamfered portions of the piston 32 as shown in Fig. 14 so that a large upward force is not generated on the piston 32.

[Prior Art Document]

[Patent Document]

[0024] [Patent Document 1] Japanese Patent Applica-

tion Laid-open No.H8-61276 The document EP 1 640 614 is considered as being the closest prior art and discloses all the features of the preamble of claim 1.

5 [Summary of the Invention]

[Problem to be Solved by the Invention]

[0025] However, there are two problems in the above-described techniques for enhancing the efficiency and the reliability of the rotary compressor.

[0026] A first problem is that when a difference of the upper and lower chamfered portions is adjusted to cancel the weight of the piston 32 of its own as shown in Fig. 11, a value of $B-A$ becomes 0.1 or less and when productivity is to be enhanced, it is extremely difficult to manage sizes.

[0027] A second problem is that when the upper and lower gaps of the piston 32 are reduced in size, it is necessary to increase both the upper and lower chamfered portions of the piston 32, but high pressure gas returns to the suction chamber and the efficiency is deteriorated if a sealing length between the inner surface of the piston 32 and the discharge port 38 is not secured as shown in the patent document 1 and thus, the upper chamfered portion can not be increased in size so much. After all, since only the lower chamfered portion having substantially the same size as that of the upper chamfered portion can be set, the reliability can not largely be enhanced.

[0028] However, if the upper end chamfered portion of the piston 32 is increased in size, since the discharge port 38 and the inner surface of the piston 32 are brought into communication with each other, the high pressure gas returns to the suction and the efficiency is deteriorated. Hence, since it is necessary to secure the sealing length between the inner surface of the piston 32 and the discharge port 38, only the lower chamfered portion having substantially the same size as that of the upper chamfered portion can be set, the reliability can not largely be enhanced.

[0029] The present invention is accomplished to solve such problems, and it is an object of the invention to enhance the productivity, suppress the leakage through upper and lower end surfaces of a piston, suppress the wear and seizing of end plates, and enhance the reliability.

[Means for Solving the Problems]

[0030] To achieve the above object, the present invention provide a rotary compressor comprising a cylinder, an eccentric core of a shaft disposed in the cylinder, a piston fitted into the eccentric core, a vane which follows eccentric rotation of the piston and which reciprocates in a slot formed in the cylinder, and two end plates which close upper and lower end surfaces of the cylinder, characterized in that a second area surrounded by a lower inner surface angle portion which is formed on a lower

end surface of the piston and the end plate which closes the lower end surface of the cylinder is set greater than a first area surrounded by an upper inner surface angle portion which is formed on an upper end surface of the piston and the end plate which closes the upper end surface of the cylinder, and an angle formed between the lower end surface of the piston and the lower inner surface angle portion is set smaller than an angle formed between the upper end surface of the piston and the upper inner surface angle portion.

[Effect of the Invention]

[0031] According to the above-described configuration, a pressure drop when oil flows into the compression chamber from the lower inner surface angle portion becomes large on the lower end surface of the piston. Therefore, even if the second area on the side of the lower inner surface angle portion is increased, a large force is not generated upwardly. Hence, the oil retention amount between the lower end surface of the piston and the end plate which closes the lower end surface of the cylinder is increased, a sealing length between the piston and the discharge port can also be secured and thus, it is possible to enhance the reliability and the efficiency. Further, it is possible to increase a range of tolerance of the area surrounded by the upper and lower end plates and the upper and lower inner surface angle portions of the piston such that the weight of the piston of its own is canceled, the productivity is enhanced and the efficiency is also enhanced.

[Brief Description of the Drawings]

[0032]

Fig. 1 is a vertical sectional view of a rotary compressor according to a first embodiment of the present invention;
 Fig. 2 is an enlarged sectional view of a compressor mechanism of the rotary compressor;
 Fig. 3 is a sectional view of a piston of the rotary compressor;
 Fig. 4 is a diagram showing a distribution of pressures applied to upper and lower portions of the piston of the rotary compressor;
 Fig. 5 is a schematic diagram showing a positional relation between a discharge port and a gap between the piston and upper and lower end plates when upper and lower gaps of the piston of the rotary compressor are reduced in size;
 Fig. 6 is a schematic diagram showing a positional relation between a discharge port and a gap between the piston and upper and lower end plates of the rotary compressor when a force which cancels a weight of the piston of its own is generated;
 Fig. 7 is a sectional view showing the piston of the rotary compressor according to the first embodiment

of the invention;

Fig. 8 is a sectional view of a rotary compressor having another configuration;

Fig. 9 is a sectional view of a rotary compressor having another configuration;

Fig. 10 is a schematic diagram showing a positional relation between a discharge port and a gap between a piston and upper and lower end plates of a general rotary compressor;

Fig. 11 is a schematic diagram showing a positional relation between the discharge port and the gap between the piston and the upper and lower end plates of the general rotary compressor when a force which cancels a weight of the piston is generated.

Fig. 12 is a schematic diagram showing a positional relation between the discharge port and the gap between the piston and the upper and lower end plates when upper and lower gaps of the piston of the general rotary compressor are reduced in size;

Fig. 13 is a schematic diagram showing a positional relation between the discharge port and the gap between the piston and the upper and lower end plates when a lower chamfered portion of the piston of the general rotary compressor is increased;

Fig. 14 is a schematic diagram showing a positional relation between the discharge port and the gap between the piston and the upper and lower end plates when upper and lower chamfered portions of the piston of the general rotary compressor are increased; and

Fig. 15 is a sectional view of a conventional rotary compressor.

[Explanation of Symbols]

[0033]

1	hermetic container
2	electric motor
3	compressor mechanism
22	stator
24	rotor
30	cylinder
31	crankshaft
31a	eccentric core
32	piston
33	vane
34a	upper bearing
34	end plate
35a	lower bearing
35	end plate
36	discharge valve
37	cup muffler
38	discharge port
39	compression chamber
40	suction port
41	oil hole
42	oil supply hole

43 oil supply hole
 44 oil supply hole
 45 oil groove
 46 space
 47 space

[Mode for Carrying Out the Invention]

[0034] According to the first aspect of the invention, a second area surrounded by a lower inner surface angle portion which is formed on a lower end surface of the piston and the end plate which closes the lower end surface of the cylinder is set greater than a first area surrounded by an upper inner surface angle portion which is formed on an upper end surface of the piston and the end plate which closes the upper end surface of the cylinder, and an angle formed between the lower end surface of the piston and the lower inner surface angle portion is set smaller than an angle formed between the upper end surface of the piston and the upper inner surface angle portion.

[0035] According to this configuration, a pressure drop is increased on the lower end surface of the piston when oil flows into the compression chamber from the inner surface of the piston. Hence, even if the second area surrounded by the lower inner surface angle portion of the piston is increased, a large force is not upwardly applied to the piston. Hence, even if the upper and lower gaps of the piston are reduced in size to enhance the efficiency, it is possible to increase the oil retention amount of the lower end plate, and since the sealing length between the upper inner surface angle portion of the piston and the discharge port can be secured, high reliability can be maintained.

[0036] According to a second aspect, in the rotary compressor of the first aspect, the upper inner surface angle portion is formed by chamfering, and the lower inner surface angle portion is formed by spot facing.

[0037] According to this configuration, it is possible to visually determine which is an upper side and which is a lower side at the time of the assembling operation, it is possible to reduce the efficiency deterioration and loss of cost caused by a mistake between the upper side and the lower side.

[0038] According to a third aspect, in the rotary compressor of the second aspect, the angle between the upper end surface of the piston and the upper inner surface angle portion is in a range of 132° to 138°.

[0039] According to this configuration, if the upper and lower sides of the piston are chamfered when a force for canceling the weight of the piston of its own, B-A is about 0.1 mm, but if the spot facing is formed only in the lower side of the piston, a range of tolerance can be increased to such a value that B-A becomes about 0.4 to 0.8 mm, and productivity is enhanced.

[0040] According to a fourth aspect, in the rotary compressor of any one of first to third aspects, the first area and the second area are set such that a weight of the

piston of its own is canceled.

[0041] According to this configuration, the piston floats and the two gaps between the upper and lower end surfaces of the piston and the end plates are equalized. Since gas leakage is generally proportional to cube of a gap, if upper and lower gaps of the piston are unevenly distributed, an amount of gas leakage becomes greater as compared with a case where the upper and lower gaps of the piston are evenly distributed. Hence, since the amounts of gas and oil leaking into the suction chamber through the gaps in the upper and lower end surfaces of the piston are suppressed, the compression loss can be reduced, the same effect as that when the upper and lower gaps are reduced in size even if the upper and lower gaps are not reduced in size, and the reliability is further enhanced as compared with a case where the gaps are reduced in size and the efficiency is further enhanced.

[0042] According to a fifth aspect, in the rotary compressor of any one of the first to fourth aspects, CO₂ which is a high pressure refrigerant is used as working fluid. According to this, even if CO₂ has a large pressure difference, a sliding loss and a leakage loss, it is possible to more effectively enhance the efficiency.

[0043] According to a sixth aspect, in the rotary compressor of any one of the first to fifth aspects, a single refrigerant including hydrofluoroolefin having double bond between carbon and carbon as a basic component or a mixture refrigerant including this single refrigerant is used as working fluid.

[0044] This refrigerant has such properties that the refrigerant can easily be decomposed at a high temperature, but it is possible to more effectively enhance the reliability of the compressor while suppressing high temperature decomposition of the refrigerant by reducing the leakage loss and the sliding loss. This refrigerant does not destroy ozone and has low global warming potential and this refrigerant can contribute to a configuration of an earth-friendly air-conditioning cycle.

[0045] Embodiments of the present invention will be described with reference to the drawings. The invention is not limited to the embodiments.

(First Embodiment)

[0046] Fig. 1 is a vertical sectional view of a rotary compressor according to a first embodiment of the invention. Fig. 2 is an enlarged diagram of a compressor mechanism. Constituent members which are the same as those explained using Fig. 15 are designated with the same symbols, and explanation thereof will be omitted.

[0047] An oil groove 45 and an oil supply hole 44 which is in communication with the oil hole 41 are formed in a wall of an eccentric core 31a of the crankshaft 31.

[0048] The eccentric core 31a of the crankshaft 31, an end plate 34 of an upper bearing 34a and an inner peripheral surface of a piston 32 form an upper space 46 in the piston. The eccentric core 31a of the crankshaft

31, an end plate 35 of a lower bearing 35a and the inner peripheral surface of the piston 32 form a lower space 47 in the piston. Oil in the oil hole 41 leaks from the oil supply hole 42 into the upper space 46 in the piston, and oil in the oil hole 41 leaks from the oil supply hole 43 into the lower space 47 in the piston. Pressures in the upper space 46 in the piston and the lower space 47 in the piston are substantially always higher than a pressure in a compression chamber 39.

[0049] A height of the cylinder 30 must be set slightly higher than that of the piston 32 so that the piston 32 can slide in the cylinder 30. As a result, a gap is generated between an end surface of the piston 32 and the end plate 34 of the upper bearing 34a, and a gap is generated between the end surface of the piston 32 and the end plate 35 of the lower bearing 35a. Hence, oil leaks from the upper space 46 and the lower space 47 into the compression chamber 39 in the piston through these gaps.

[0050] Operations and effect of the rotary compressor having the above-described configuration will be explained below.

[0051] As shown in Fig. 3, according to the first embodiment, a second area 32b surrounded by the end plate 35 and a lower inner surface angle portion formed on a lower end surface of the piston 32 is set greater than a first area 32a surrounded by the end plate 34 and an upper inner surface angle portion formed on an upper end surface of the piston 32.

[0052] According to the first embodiment, an angle D formed between a lower end surface of the piston 32 and the lower inner surface angle portion is set smaller than an angle C formed between an upper end surface of the piston 32 and the upper inner surface angle portion.

[0053] According to the first embodiment, the above-described configuration enhances efficiency and reliability.

[0054] Fig. 4 shows a distribution of a pressure applied to the piston 32 of the first embodiment.

[0055] As shown in Fig. 4, a high pressure is equally distributed to the upper side of the piston 32 on the inner surface side, and pressures from the high pressure to an intermediate pressure are straightly distributed to the upper side of the piston 32 on the end surface sides.

[0056] A high pressure is equally distributed to a lower side of the piston 32, but pressures from an intermediate high pressure (lower than high pressure) to the intermediate pressure are straightly distributed to the lower side of the piston 32 on the side of the end surface sides. That is, since the angle D formed between the lower end surface of the piston 32 and the lower inner surface angle portion is set smaller than the angle C on the lower side of the piston 32, flow of oil is deteriorated, and a pressure drop is generated. Hence, even if a width B of the lower side of the piston 32 is increased as shown in Fig. 5, such a large force is not generated upward.

[0057] Therefore, an oil retention amount at the lower end plate 35 is increased, a sealing length L between the piston 32 and the discharge port 38 can be secured and

therefore, it is possible to enhance the reliability and the efficiency. Further, since A and B are set such that the weight of the piston 32 of its own is canceled, the same effect as that when the upper and lower gaps are reduced in size even if the piston 32 floats and the upper and lower gaps are not reduced in size as shown in Fig. 6.

(Second Embodiment)

[0058] Fig. 7 is a sectional view showing a piston of a rotary compressor according to a second embodiment of the invention. Since other structure is the same as that of the first embodiment, explanation thereof will be omitted.

[0059] As shown in Fig. 7, according to the second embodiment of the invention, the second area 32b surrounded by the end plate 35 and the lower inner surface angle portion formed on the lower end surface of the piston 32 is set greater than the first area 32a surrounded by the end plate 34 and the upper inner surface angle portion formed on the upper end surface of the piston 32.

[0060] According to the second embodiment, the angle D formed between the lower end surface of the piston 32 and the lower inner surface angle portion is set smaller than the angle C formed between the upper end surface of the piston 32 and the upper inner surface angle portion.

[0061] In the second embodiment, the upper inner surface angle portion is formed by chamfering the upper end surface and the upper inner surface of the piston 32, and the lower inner surface angle portion is formed by spot facing the lower end surface and the lower inner surface of the piston 32.

[0062] The angle C between the upper end surface of the piston 32 and the upper inner surface angle portion is preferably in a range of 132° to 138°, and more preferably 135°.

[0063] By spot facing the lower inner surface angle portion of the piston 32, the angle D between the lower end surface of the piston 32 and the lower inner surface angle portion becomes 90°.

[0064] In this embodiment, the upper inner surface angle portion is formed by chamfering, and the lower inner surface angle portion is formed by spot facing. According to this, it is possible to visually determine or distinguish the upper side and the lower side from each other at the time of an assembling operation, and it is possible to reduce the efficiency deterioration and loss of cost caused by error between the upper side and the lower side of the piston 32. When a force canceling the weight of the piston 32 of its own is generated, if the upper and lower sides of the piston 32 are chamfered, B-A is about 0.1 mm, but if the spot facing is formed only in the lower side of the piston 32, a range of tolerance can be increased to such a value that B-A becomes about 0.4 to 0.8 mm, and productivity is enhanced.

[0065] Fig. 8 shows a rotary compressor having a configuration different from that of the first embodiment of the invention. The same configurations as those of the

first embodiment are designated with the same symbols, and explanation thereof will be omitted.

[0066] The rotary compressor shown in Fig. 8 includes a vane 133 which is coupled to an outer periphery of a piston 132 in a projecting form and which distinguishes a low pressure side and a high pressure side of the compression chamber 39 from each other, and a rocking bush 130 which supports the vane 133 such that it can rock and move forward and backward.

[0067] The same configurations as those of the first embodiment and the second embodiment can also be applied to the rotary compressor shown in Fig. 8, and the same effect can be obtained.

[0068] Fig. 9 shows a rotary compressor having another configuration. The same configurations as those of the first embodiment are designated with the same symbols, and explanation thereof will be omitted.

[0069] The rotary compressor shown in Fig. 9 includes a piston 232 and a vane 233 whose tip end is rockably connected to the piston 232.

[0070] The same configurations as those of the first embodiment and the second embodiment can also be applied to the rotary compressor shown in Fig. 9, and the same effect can be obtained.

[0071] If CO₂ is used as working fluid, even if a pressure difference is large and influence of a leakage loss and a sliding loss are large, it is possible to reduce the leakage of fluid at the upper and lower end surfaces of the piston 32, and a force strongly pressing the piston 32 downward can be avoided and thus, it is possible to more effectively enhance the efficiency.

[0072] Further, if a single refrigerant including hydrofluoroolefin having double bond between carbon and carbon as a basic component or a mixture refrigerant including this single refrigerant is used as working fluid, it is possible to suppress the problem of properties which are peculiar to the refrigerant of this kind. That is, this refrigerant is prone to be decomposed at high temperature and is unstable. On the other hand, according to the rotary compressor of the invention, since the lubricating performance of the end surface of the piston 32 and the sliding portion of the end plate 35 is enhanced, it is possible to efficiently suppress the temperature rise at the sliding portion, to prevent decomposition of the refrigerant, and to more effectively enhance the reliability.

[0073] As the working fluid, it is possible to use mixture refrigerant in which tetrafluoropropene (HFO1234yf or HFO1234ze) or trifluoropropene (HFO1243zf) is used as hydrofluoroolefin and difluoromethane (HFC32) is used as hydrofluorocarbon.

[0074] Alternatively, as the working fluid, it is possible to use mixture refrigerant in which tetrafluoropropene (HFO1234yf or HFO1234ze) or trifluoropropene (HFO1243zf) is as hydrofluoroolefin and pentafluoroethane (HFC125) is used as hydrofluorocarbon.

[0075] Alternatively, as the working fluid, it is possible to use mixture refrigerant including three components in which tetrafluoropropene (HFO1234yf or HFO1234ze)

or trifluoropropene (HFO1243zf) is used as hydrofluoroolefin and pentafluoroethane (HFC125) or difluoromethane (HFC32) is used as hydrofluorocarbon.

[0076] Although the embodiments are described based on the single piston type rotary compressor having one cylinder, the rotary compressor may include a plurality of cylinders.

[Industrial Applicability]

[0077] According to the rotary compressor of the invention, it is possible to suppress deterioration of reliability such as wear and seizing, reduce the leakage loss and sliding loss at the same time, and enhance the efficiency of the compressor. According to this, the invention can be applied to an air conditioner and a heat pump hot water supply apparatus using natural refrigerant CO₂ in addition to an air conditioner compressor using HFC-based refrigerant or HCFC-based refrigerant.

Claims

1. A rotary compressor comprising a cylinder (30), an eccentric core (31a) of a shaft (31) disposed in the cylinder (30), a piston (32) fitted into the eccentric core (31a), a vane (133) which follows eccentric rotation of the piston (32) and which reciprocates in a slot formed in the cylinder (30), and two end plates (34, 35) which close upper and lower end surfaces of the cylinder (30), a lower inner surface angle portion (32d) of which outer peripheral circle is diameter B is formed on an inner peripheral end portion of a lower end surface of the piston (32), an upper inner surface angle portion (32c) of which outer peripheral circle is a diameter A is formed on an inner peripheral end portion of an upper end surface of the piston (32), a second area (32b) surrounded by the outer peripheral circle of the diameter B opposite to the end plate (35) which closes the lower end surface of the cylinder (30) is set greater than a first area (32a) surrounded by the outer peripheral circle of the diameter A opposited to the end plate (34) which closes the upper end surface of the cylinder (30), **characterized in that** an angle (D) formed toward between the lower end surface of the piston (32) and the lower inner surface angle portion (32d) is set smaller than an angle (C) formed between the upper end surface of the piston (32) and the upper inner surface angle portion (32c).
2. The rotary compressor according to claim 1, **characterized in that** the upper inner surface angle portion is formed by chamfering, and the lower inner surface angle portion is formed by spot facing.
3. The rotary compressor according to claim 2, **characterized in that** the angle (C) between the up-

per end surface of the piston (32) and the upper inner surface angle portion is in a range of 132° to 138°.

4. The rotary compressor according to any one of claims 1 to 3,
characterized in that the first area and the second area are set such that a weight of the piston (32) of its own is cancelled.
5. The rotary compressor according to any one of claims 1 to 4,
characterized in that CO₂ which is a high pressure refrigerant is used as working fluid.
6. The rotary compressor according to any one of claims 1 to 5,
characterized in that a single refrigerant including hydrofluoroolefin having double bond between carbon and carbon as a basic component or a mixture refrigerant including this single refrigerant is used as working fluid.

Patentansprüche

1. Rotationskompressor, enthaltend einen Zylinder (30), einen exzentrischen Kern (31a) einer in dem Zylinder (30) angeordneten Welle (31), einen in den exzentrischen Kern (31a) eingesetzten Kolben (32), einen Trennschieber (133), der einer exzentrischen Rotation des Kolbens (32) folgt und der sich in einem in dem Zylinder (30) ausgebildeten Schlitz hin- und herbewegt, und zwei Endplatten (34, 35), die die oberen und unteren Endflächen des Zylinders (30) verschließen, wobei ein unterer Innenflächenwinkelabschnitt (32d), dessen äußerer Umfangskreis einen Durchmesser B besitzt, auf einem inneren Umfangsendabschnitt einer unteren Endfläche des Kolbens (32) ausgebildet ist, wobei ein oberer Innenflächenwinkelabschnitt (32c), dessen äußerer Umfangskreis einen Durchmesser A besitzt, auf einem inneren Umfangsendabschnitt einer oberen Endfläche des Kolbens (32) ausgebildet ist, wobei ein zweiter Bereich (32b), der von dem äußeren Umfangskreis des Durchmessers B umgeben ist, der der Endplatte (35) gegenüber liegt, die die untere Endfläche des Zylinders (30) verschließt, größer eingestellt ist als ein erster Bereich (32a), der von dem äußeren Umfangskreis des Durchmessers A umgeben ist, der der Endplatte (34) gegenüber liegt, die die obere Endfläche des Zylinders (30) verschließt,
dadurch gekennzeichnet, dass ein Winkel (D), der in Richtung zwischen der unteren Endfläche des Kolbens (32) und dem unteren Innenflächenwinkelabschnitt (32d) ausgebildet ist, kleiner eingestellt ist als ein Winkel (C), der zwischen der oberen Endfläche des Kolbens (32) und dem oberen Innenflächenwinkelabschnitt (32c) gebildet ist.

2. Rotationskompressor nach Anspruch 1,
dadurch gekennzeichnet, dass der obere Innenflächenwinkelabschnitt durch Abschrägen, und der untere Innenflächenwinkelabschnitt durch Plansenken gebildet ist.
3. Rotationskompressor nach Anspruch 2,
dadurch gekennzeichnet, dass der Winkel (C) zwischen der oberen Endfläche des Kolbens (32) und dem oberen Innenflächenwinkelabschnitt in einem Bereich von 132° bis 138° liegt.
4. Rotationskompressor nach einem der Ansprüche 1 bis 3,
dadurch gekennzeichnet, dass der erste Bereich und der zweite Bereich so eingestellt sind, dass ein Eigengewicht des Kolbens (32) aufgehoben wird.
5. Rotationskompressor nach einem der Ansprüche 1 bis 4,
dadurch gekennzeichnet, dass CO₂, das ein Hochdruckkältemittel ist, als Arbeitsfluid verwendet wird.
6. Rotationskompressor nach einem der Ansprüche 1 bis 5,
dadurch gekennzeichnet, dass ein einzelnes Kältemittel, das Hydrofluoroolefin mit Doppelbindung zwischen Kohlenstoff und Kohlenstoff als Basiskomponente enthält, oder ein Mischkältemittel, das dieses einzelne Kältemittel enthält, als Arbeitsfluid verwendet wird.

Revendications

1. Compresseur rotatif comprenant un cylindre (30), un noyau excentrique (31a) d'un arbre (31) disposé dans le cylindre (30), un piston (32) inséré dans le noyau excentrique (31a), une aube (133) qui suit la rotation excentrique du piston (32) et qui effectue un mouvement de va-et-vient dans une fente formée dans le cylindre (30), et deux plaques d'extrémité (34, 35) qui ferment des surfaces d'extrémité supérieure et inférieure du cylindre (30), une partie d'angle de surface interne inférieure (32d) dont le cercle périphérique externe est un diamètre B formée sur une partie d'extrémité périphérique interne d'une surface d'extrémité inférieure du piston (32), une partie d'angle de surface interne supérieure (32c) dont le cercle périphérique externe est un diamètre A formée sur une partie d'extrémité périphérique interne d'une surface d'extrémité supérieure du piston (32), une deuxième zone (32b) entourée par le cercle périphérique externe du diamètre B opposé à la plaque d'extrémité (35) qui ferme la surface d'extrémité inférieure du cylindre (30) définie plus grande qu'une première zone (32a) entourée par le cercle périphé-

rique externe du diamètre A opposé à la plaque d'extrémité (34) qui ferme la surface d'extrémité supérieure du cylindre (30),

caractérisé en ce qu'un angle (D) formé entre la surface d'extrémité inférieure du piston (32) et la partie d'angle de surface interne inférieure (32d) est défini plus petit qu'un angle (C) formé entre la surface d'extrémité supérieure du piston (32) et la partie d'angle de surface interne supérieure (32c).

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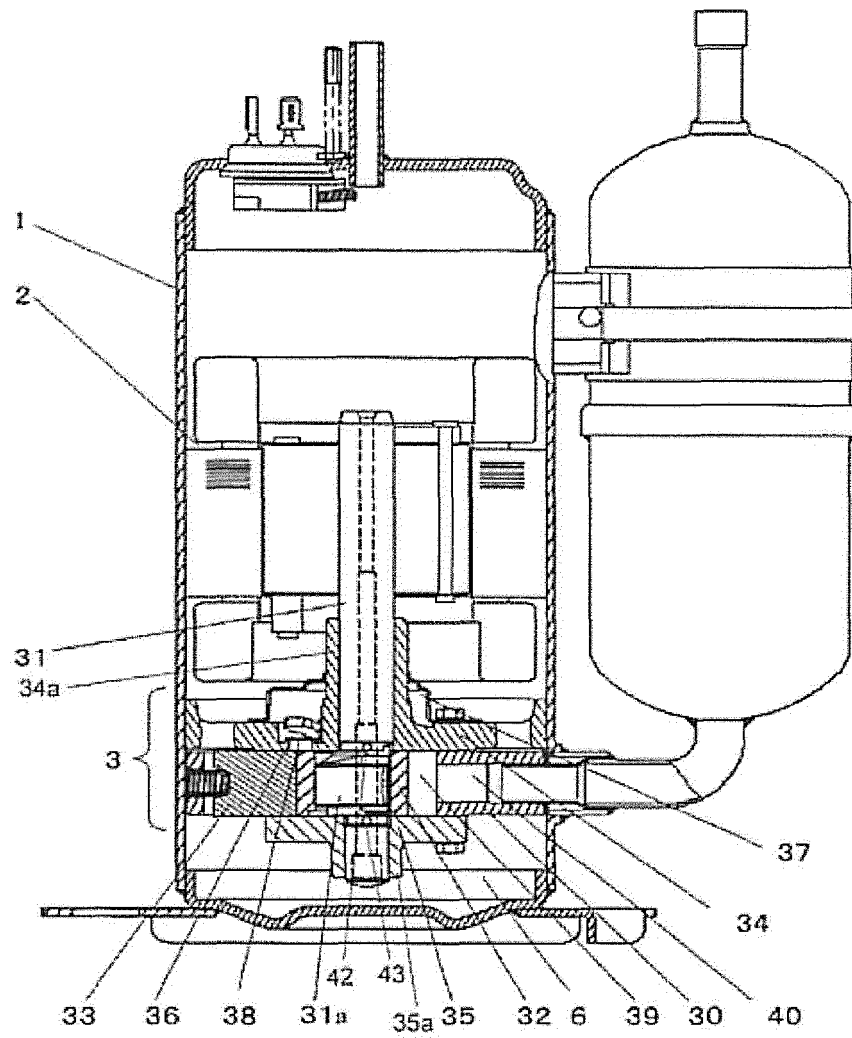
2. Compresseur rotatif selon la revendication 1, **caractérisé en ce que** la partie d'angle de surface interne supérieure est formée par chanfreinage, et la partie d'angle de surface interne inférieure est formée par lamage. 15
3. Compresseur rotatif selon la revendication 2, **caractérisé en ce que** l'angle (C) entre la surface d'extrémité supérieure du piston (32) et la partie d'angle de surface interne supérieure est dans une plage de 132° à 138°. 20
4. Compresseur rotatif selon l'une quelconque des revendications 1 à 3, **caractérisé en ce que** la première zone et la deuxième zone sont définies de telle sorte qu'un poids du piston (32) lui-même est annulé. 25
5. Compresseur rotatif selon l'une quelconque des revendications 1 à 4, **caractérisé en ce que** le CO₂ qui est un fluide frigorigène à haute pression est utilisé comme fluide de travail. 30
6. Compresseur rotatif selon l'une quelconque des revendications 1 à 5, **caractérisé en ce qu'un** fluide frigorigène unique comprenant une hydrofluorooléfine ayant une double liaison entre carbone et carbone comme composant de base ou un fluide frigorigène de mélange comprenant ce fluide frigorigène unique est utilisé comme fluide de travail. 35
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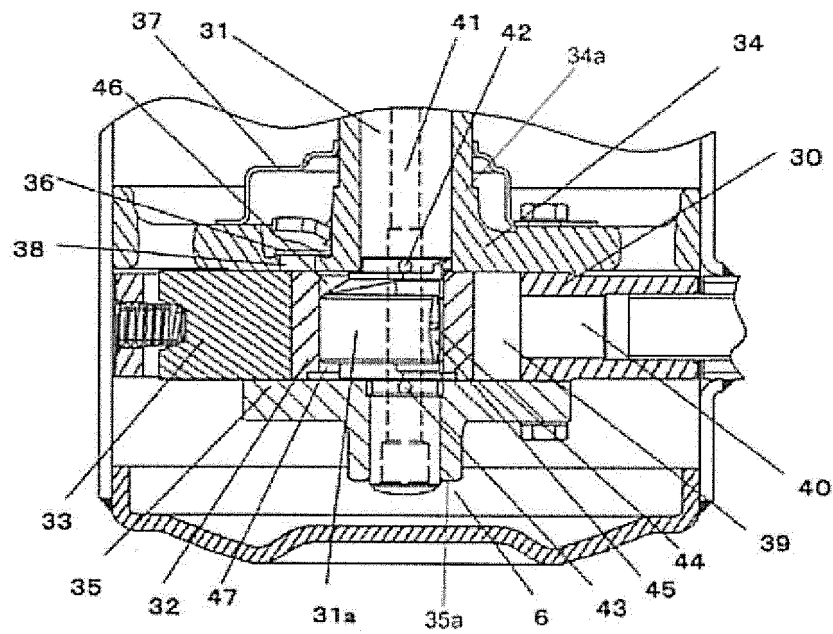
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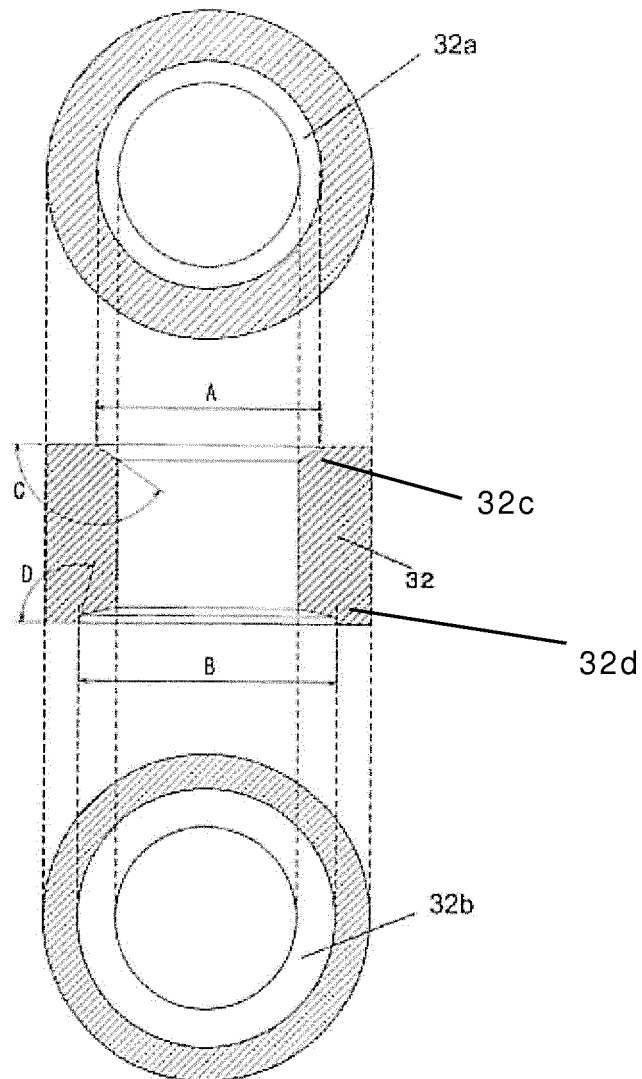
[Fig. 1]



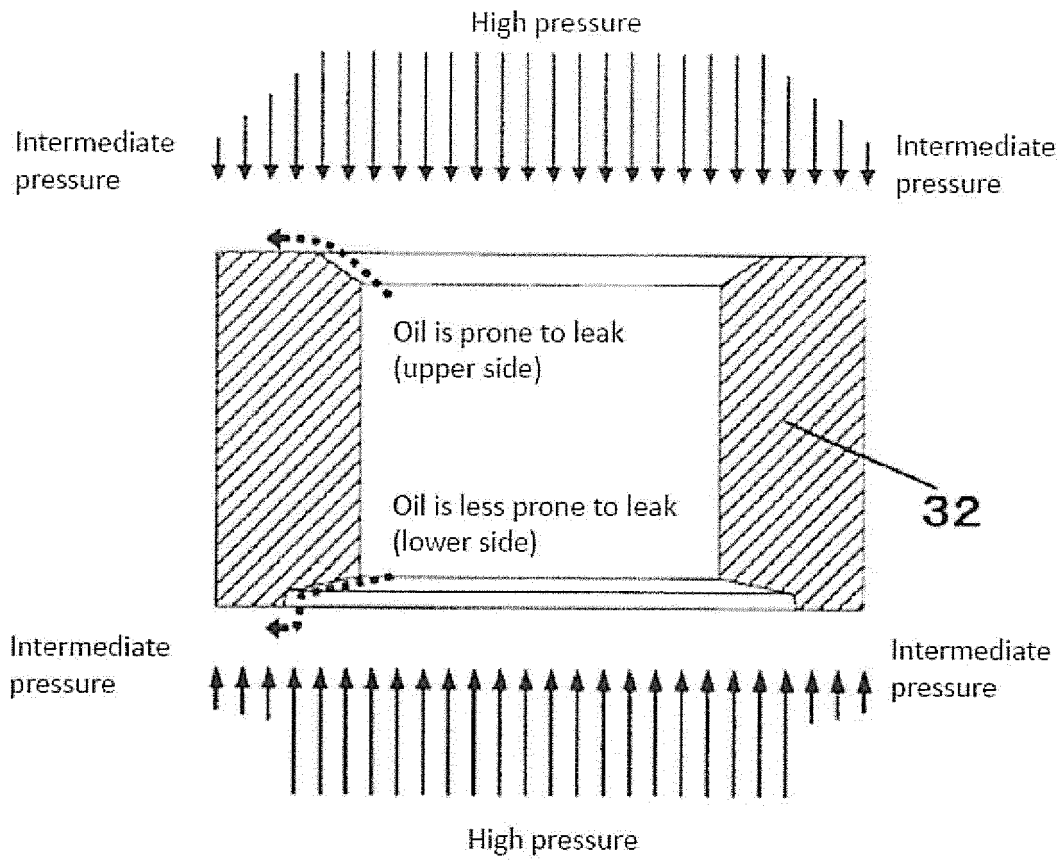
[Fig. 2]



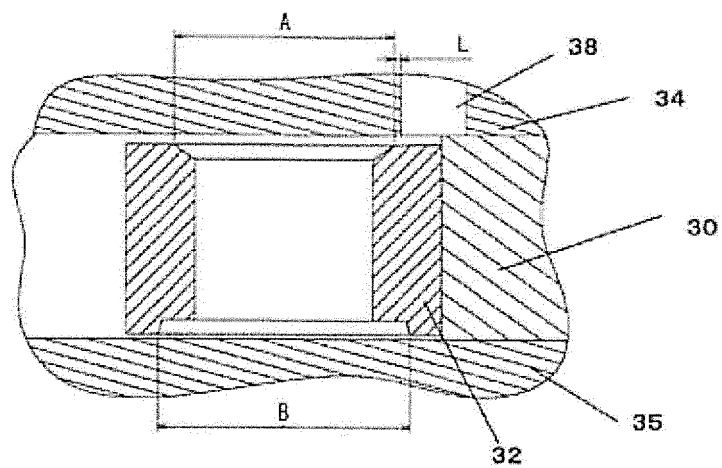
[Fig. 3]



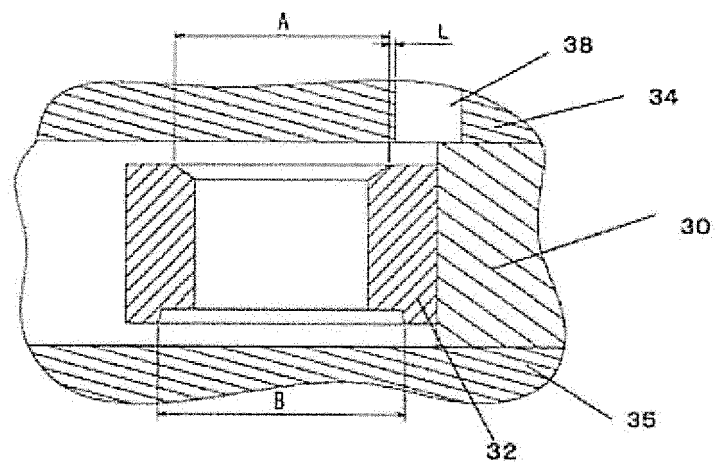
[Fig. 4]



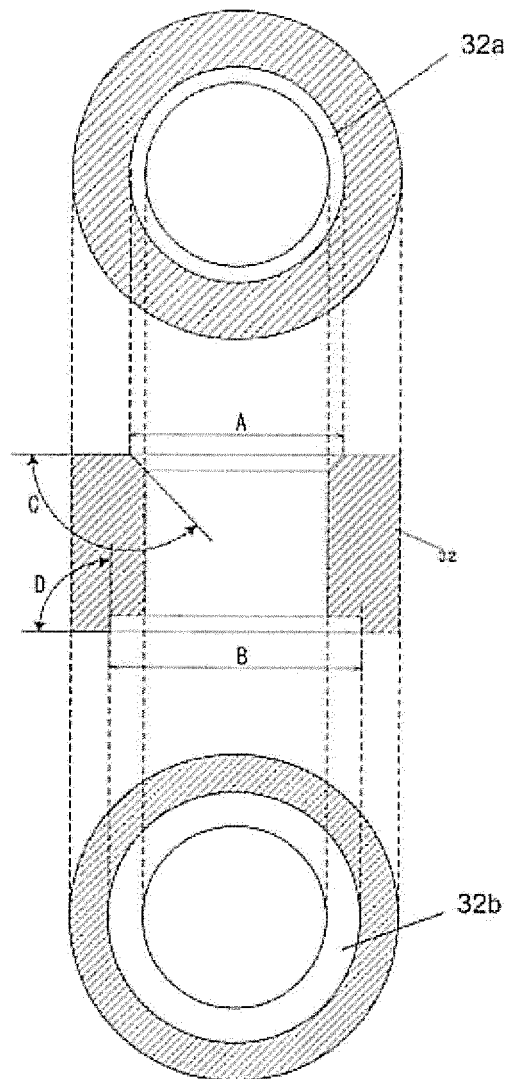
[Fig. 5]



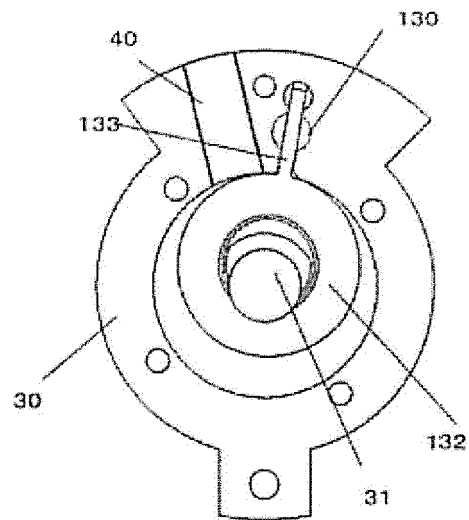
[Fig. 6]



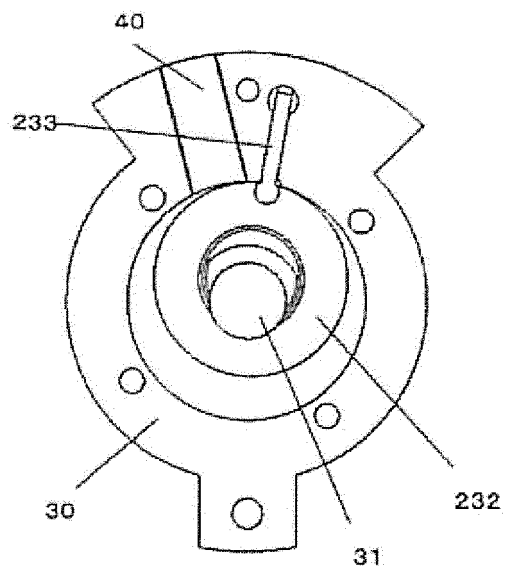
[Fig. 7]



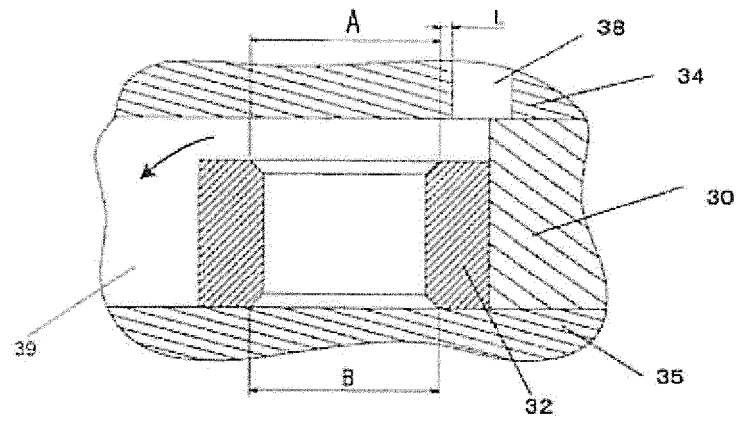
[Fig. 8]



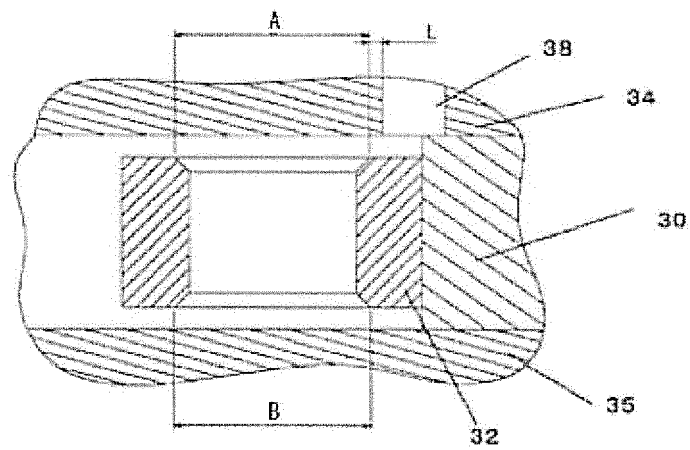
[Fig. 9]



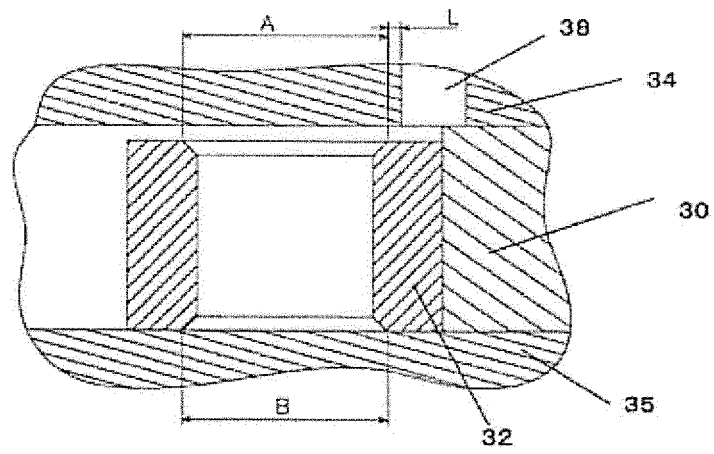
[Fig. 10]



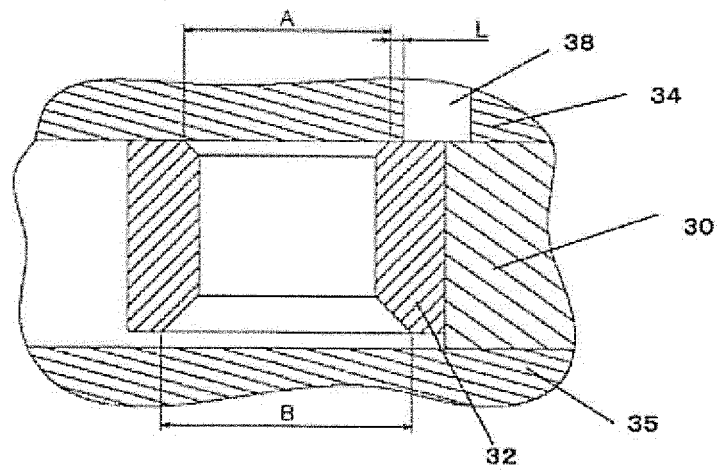
[Fig. 11]



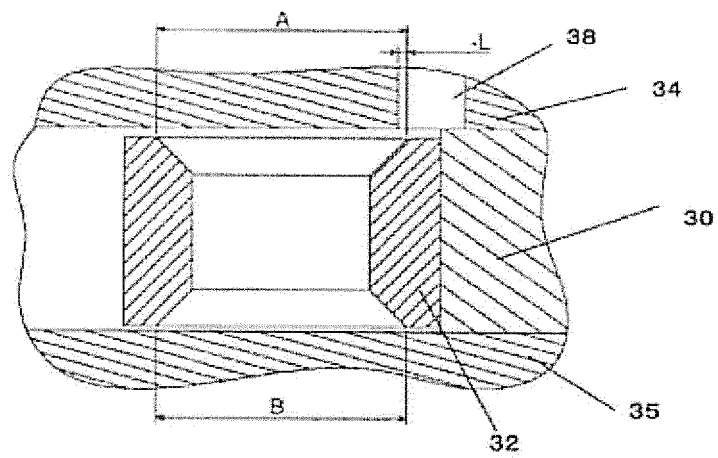
[Fig. 12]



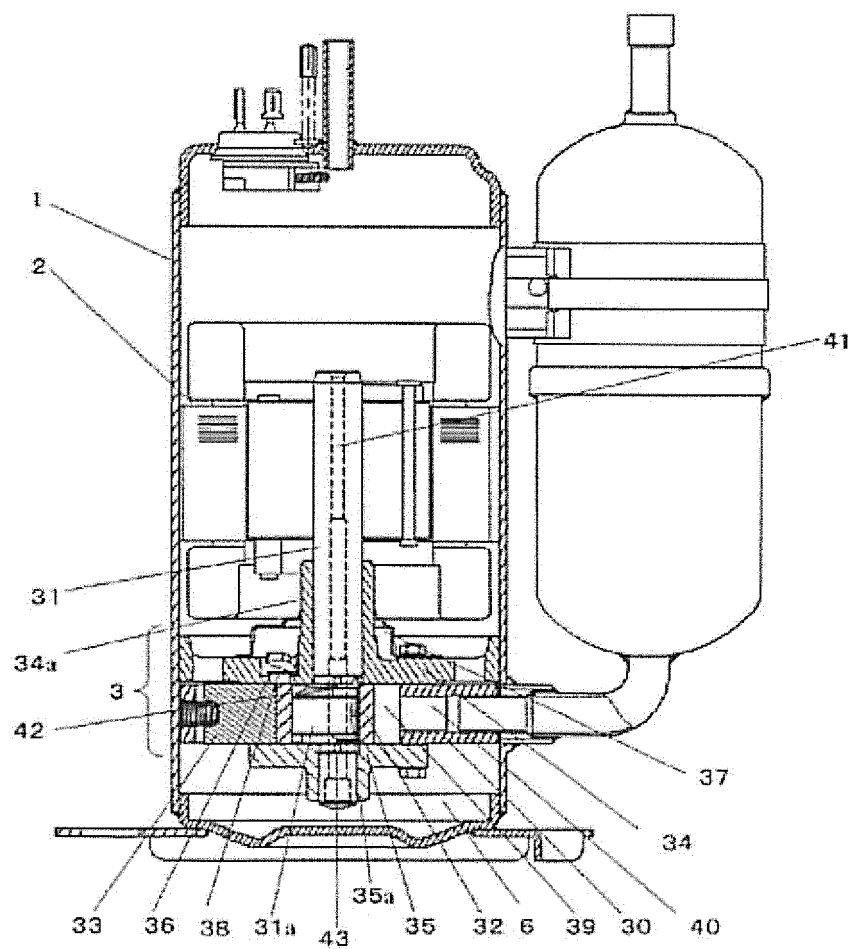
[Fig. 13]



[Fig. 14]



[Fig. 15]



REFERENCES CITED IN THE DESCRIPTION

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