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(54) REFRIGERANT COMPRESSOR AND REFRIGERATION APPARATUS USING SAME

(57) In a refrigerant compressor including a sealed container in which refrigerating machine oil is stored, a shaft part that includes a main shaft and an eccentric shaft, or a bearing part that includes a main bearing and an eccentric bearing, is provided with tapered portions, the tapered portions being respectively formed on one end side and another end side of the shaft part or the bearing part in an axial direction of the bearing part such that a diameter of the shaft part or the bearing part changes from an outer side toward a central side in a longitudinal direction of a crankshaft, each tapered portion allowing the shaft part and the bearing part to come into line contact with each other in a state where an axis of the shaft part is tilted relative to an axis of the bearing part. A ratio C / D, which is a ratio of a clearance C between the shaft part and the bearing part to a diameter D of the shaft part, is set to a value within a range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}. A taper depth d_B of each tapered portion is set to a value not less than 2.0×10^{-3} mm. In a combination of the shaft part and the bearing part corresponding thereto, a ratio G / D, which is a ratio of a maximum gap G to the diameter D of the shaft part, is set to a value not greater than 4.0×10^{-3}. The maximum gap G is a sum of the clearance C and a total value of the taper depths d_B.

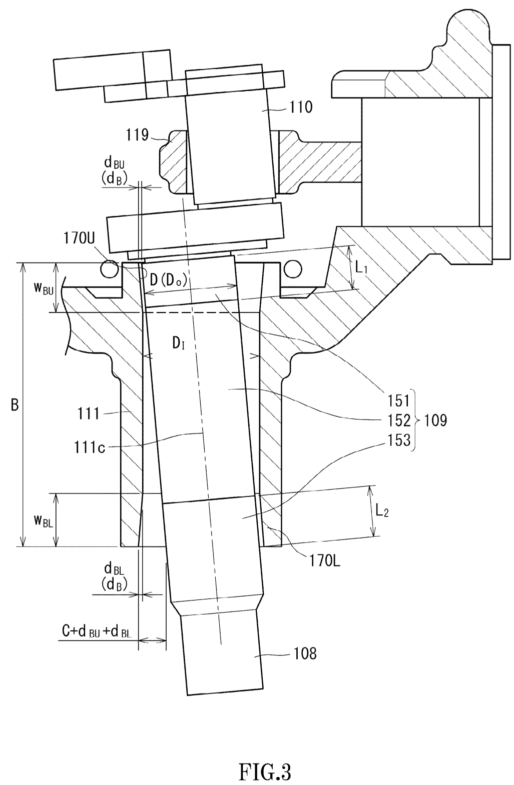

FIG.3

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Description

Technical Field

[0001] The present invention relates to a refrigerant compressor for use in, for example, a refrigerator or an air conditioner, and also to a refrigeration apparatus using the refrigerant compressor.

Background Art

[0002] FIG. 12 shows a schematic sectional view of a conventional refrigerant compressor 1. For example, the conventional refrigerant compressor 1 includes a sealed container 11, in which a compression element 6 and an electric element 5 are accommodated. The compression element 6 includes a crankshaft 7 and a piston 15. The piston 15 is connected to an eccentric shaft 9 of the crankshaft 7. The electric element 5, which rotates the crankshaft 7, includes a stator 3 and a rotor 4. A main shaft 8 of the crankshaft 7 is pivotally supported by a main bearing 14. Refrigerating machine oil 2 is fed to sliding portions in the refrigerant compressor 1.

[0003] During the refrigerant compressor 1 being driven, the crankshaft 7 is rotated together with the rotor 4 of the electric element 5 by electric power supplied from the outside, and eccentric motion of the eccentric shaft 9 causes the piston 15 to make reciprocating motion in a cylinder bore 12 via a connecting rod 17 and a piston pin 16. The piston 15 compresses, in a compression chamber 13, refrigerant gas that is supplied into the sealed container 11 via a suction tube 20 from the outside. In accordance with the rotation of the crankshaft 7, the refrigerating machine oil 2 is fed to the sliding portions from an oil-feeding pump 10 to lubricate the sliding portions and seal between the piston 15 and the cylinder bore 12.

[0004] In recent years, from the viewpoint of global environment conservation, the development of a high-efficient refrigerant compressor has been conducted for the purpose of reducing the use of fossil fuels. For example, a refrigerant compressor as disclosed in Patent Literature 1 has been developed, in which wear of a sliding portion of, for example, the crankshaft is prevented by forming an insoluble coating on the surface of the sliding portion.

[0005] Specifically, in the example shown in FIG. 12, the crankshaft 7 is supported at one end thereof by the main bearing 14. During the suction of the refrigerant gas into and the compression of the refrigerant gas in the sealed container 11, a load applied to the crankshaft 7 in the radial direction varies at least ten times its minimum value. Such load variation causes run-out of the crankshaft 7, in which the crankshaft 7 rotates in a state where the axis of the crankshaft 7 is tilted relative to the axis of the main bearing 14. As a result, lubrication at both ends of the main bearing 14 in the axial direction becomes relatively insufficient. In this respect, an insoluble coating, such as a phosphatic coating, is formed on the surface of the main shaft 8 of the crankshaft 7, and thereby abnormal wear due to direct metal contact between the main shaft 8 and the main bearing 14 is suppressed.

Citation List

Patent Literature

[0006] PTL 1: Japanese Laid-Open Patent Application Publication No. H07-238885

Summary of Invention

Technical Problem

[0007] However, in recent years, there is a demand for a refrigerant compressor with further improved efficiency. For example, various design changes have been taken into consideration, such as expanding the variable speed rotation range at rotating portions of the refrigerant compressor, adopting low-viscosity refrigerating machine oil, and reducing the area of the sliding portions. In a case where such design changes have been made, even if an insoluble coating is formed on the surface of the sliding portions, the coating becomes scraped particularly at, for example, both ends of the main shaft of the crankshaft in the axial direction, at which it is difficult to keep the lubricated state. Consequently, there is a risk that the wear of the sliding portions progresses, which causes reduction in the durability and reliability of the refrigerant compressor.

[0008] In view of the above, an object of the present invention is to provide a refrigerant compressor that makes it possible to achieve improvement in efficiency while preventing reduction in durability and reliability by preventing wear of sliding portions, and to provide a refrigeration apparatus in which the refrigerant compressor is used.

Solution to Problem

[0009] In order to solve the above-described problems, a refrigerant compressor according to one aspect of the present invention includes: a sealed container in which refrigerating machine oil is stored; an electric element accommodated in the sealed container and driven by electric power supplied from outside; and a compression element accommodated in the sealed container and covered with the refrigerating machine oil, the compression element being driven by the electric element to compress refrigerant gas supplied from outside. The compression element includes: a crankshaft including a main shaft and an eccentric shaft that are arranged side by side in a longitudinal direction; a main bearing that pivotally supports the main shaft; and an eccentric bearing that pivotally supports the eccentric shaft. A shaft part that is at least one of the main shaft and the eccentric shaft, or a bearing part that is at least one of the main bearing and the eccentric bearing, is provided with a tapered portion, the tapered portion being formed on at least one of one end side and another end side of the shaft part or the bearing part in an axial direction of the bearing part such that a diameter of the shaft part or the bearing part changes from an outer side toward a central side in the longitudinal direction of the crankshaft, the tapered portion allowing the shaft part and the bearing part to come into line contact with each other in a state where an axis of the shaft part is tilted relative to an axis of the bearing part. A ratio C / D, which is a ratio of a clearance C between the shaft part and the bearing part to a diameter D of the shaft part, is set to a value within a range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}. A taper depth d_B, which corresponds to a distance between one end and another end of the tapered portion in the axial direction of the bearing part, the distance being in a direction perpendicular to the axis of the bearing part, is set to a value not less than 2.0×10^{-3} mm. In a combination of the shaft part and the bearing part corresponding thereto, a ratio G / D, which is a ratio of a maximum gap G to the diameter D of the shaft part, is set to a value not greater than 4.0×10^{-3}, the maximum gap G being a sum of the clearance C and a total value of the taper depth d_B of the tapered portion.

[0010] According to the above configuration, the ratio C / D, the taper depth d_B, and the ratio G / D are set to the values within the aforementioned ranges. With such settings, the distance between the shaft part and the bearing part can be suitably set in relation to the diameter D of the shaft part, and also, the tapered portion having a favorable sloped surface can be formed. Consequently, local metal contact between the shaft part and the bearing part can be prevented, and the formation of an oil film between a sliding portion of the shaft part and a sliding portion of the bearing part can be facilitated. This makes it possible to provide the refrigerant compressor, which is a low-input and highly efficient refrigerant compressor with excellent long-term durability.

[0011] A refrigeration apparatus according to another aspect of the present invention includes a refrigerant circuit including: the above refrigerant compressor; a radiator that radiates heat of a refrigerant; a decompressor that decompresses the refrigerant; and a heat absorber that absorbs the heat of the refrigerant. In the refrigerant circuit, the refrigerant compressor, the radiator, the decompressor, and the heat absorber are connected by piping in an annular manner.

[0012] Provided by the above configuration is the refrigeration apparatus which can, by including the above-described refrigerant compressor, reduce electric power consumption, i.e., realize energy saving, and have improved long-term reliability.

Advantageous Effects of Invention

[0013] The present invention can provide a refrigerant compressor that makes it possible to achieve improvement in efficiency while preventing reduction in durability and reliability by preventing wear of sliding portions, and provide a refrigeration apparatus in which the refrigerant compressor is used.

Brief Description of Drawings

[0014]

FIG. 1 is a schematic sectional view of a reciprocating refrigerant compressor according to Embodiment 1.
FIG. 2 is an enlarged sectional view of an E region of the refrigerant compressor of FIG. 1.
FIG. 3 is a sectional view of main components of the refrigerant compressor of FIG. 1.
FIG. 4A is a characteristic diagram showing an input ratio between the refrigerant compressor of Example, which is shown in FIG. 1, and a refrigerant compressor of Conventional Example.
FIG. 4B is a characteristic diagram showing a COP ratio between the refrigerant compressor of Example, which is shown in FIG. 1, and the refrigerant compressor of Conventional Example.
FIG. 5 is a functional diagram illustrating a compressive load applied to the refrigerant compressor of FIG. 1.
FIG. 6 illustrates contact states each between a main shaft of FIG. 1 and a main bearing when the main shaft is tilted inside the main bearing; FIG. 6 shows, for each contact state, a correlation between the contact state and relational expressions that hold true in the contact state.

FIG. 7 is a graph showing setting ranges of Examples and Comparative Examples.
FIG. 8 is a schematic sectional view of a rotating (rotary) refrigerant compressor according to Embodiment 2.
FIG. 9 is an enlarged sectional view of a B region of the refrigerant compressor of FIG. 8.
FIG. 10 is a sectional view of the refrigerant compressor of FIG. 8, the sectional view being taken along line A-A' of FIG. 8 as viewed in the direction of the arrows of line A-A'.
FIG. 11 is a schematic diagram of a refrigeration apparatus according to Embodiment 3.
FIG. 12 is a schematic sectional view of a conventional refrigerant compressor.

Description of Embodiments

[0015] Hereinafter, embodiments are described with reference to the drawings.

(Embodiment 1)

[Refrigerant Compressor]

[0016] FIG. 1 is a schematic sectional view of a reciprocating refrigerant compressor 100 according to Embodiment 1. As shown in FIG. 1, the refrigerant compressor 100 includes a sealed container 101, an electric element 106, a compression element 107, and an oil-feeding pump 120. The inside of the sealed container 101 is filled with refrigerant gas (e.g., R600a). Refrigerating machine oil 103 (e.g., mineral oil) is stored in the bottom of the sealed container 101).

[0017] The electric element 106 is accommodated in the sealed container 101, and is driven by electric power supplied from the outside. The electric element 106 includes a stator 104 and a rotor 105. The compression element 107 is accommodated in the sealed container 101, and is covered with the refrigerating machine oil 103. The compression element 107 is driven by the electric element 106 to compress the refrigerant gas supplied from the outside. The compression element 107 includes a crankshaft 108, a cylinder block 112, a piston pin 115, a coupling member 117, a piston 132, a valve plate 139, and a cylinder head 140.

[0018] As one example, the crankshaft 108 is made of cast iron. The crankshaft 108 is disposed in a manner to extend vertically. The crankshaft 108 includes a main shaft 109 and an eccentric shaft 110, which are arranged side by side in the longitudinal direction. The rotor 105 is fixed to the main shaft 109 by press-fitting. As one example, the eccentric shaft 110 is disposed above the main shaft 109. The eccentric shaft 110 is disposed eccentrically with respect to the main shaft 109.

[0019] As described below, in the refrigerant compressor 100, the main shaft 109 is pivotally supported by a main bearing 111, and the eccentric shaft 110 is pivotally supported by an eccentric bearing 119. The lower side of the crankshaft 108 is provided with the oil-feeding pump 120, and is fed with the refrigerating machine oil 103.

[0020] As one example, the cylinder block 112 is made of cast iron. A substantially cylindrical cylinder bore 113 is formed inside the cylinder block 112. The cylinder bore 113 extends horizontally, and one end thereof is sealed by the valve plate 139. The cylinder block 112 includes the main bearing 111, which pivotally supports the main shaft 109.

[0021] The piston 132 is inserted in the cylinder bore 113 in a reciprocable manner. An interior space of the cylinder bore 113 between the piston 132 and the valve plate 139 serves as a compression chamber 134. A piston pin hole 116 is formed in the piston 132. The piston pin 115 is locked to the piston pin hole 116 in a non-rotatable manner. The piston pin 115 has a substantially cylindrical shape, and is disposed parallel to the eccentric shaft 110.

[0022] The eccentric shaft 110 and the piston 132 are coupled together by the coupling member 117. The coupling member 117 is an aluminum casting product, and includes the eccentric bearing 119. The coupling member 117 couples the eccentric shaft 110 and the piston 132 via the piston pin 115.

[0023] The cylinder head 140 is disposed on the opposite side of the valve plate 139 from the cylinder bore 113. The cylinder head 140 forms a high-pressure chamber (not shown), and is fixed to the valve plate 139.

[0024] A suction tube (not shown) is fixed to the sealed container 101. The suction tube is connected to the low-pressure side (not shown) of the refrigeration cycle of the refrigerant compressor 100, such that the suction tube leads the refrigerant gas into the sealed container 101. A suction muffler 142 is held between the valve plate 139 and the cylinder head 140.

[0025] At the time of driving the refrigerant compressor 100, electric power supplied from the outside, such as electric power from a commercial power source, is supplied to the electric element 106 via an external inverter drive circuit (not shown). Accordingly, the electric element 106 is inverter-driven at a plurality of operating frequencies.

[0026] The crankshaft 108 is rotated by the rotor 105 of the electric element 106. As a result, the eccentric shaft 110 makes eccentric motion. The coupling member 117 causes, via the piston pin 115, the piston 132 to make reciprocating motion in the cylinder bore 113. As a result, the refrigerant gas that has been led into the sealed container 101 through the suction tube is sucked from the suction muffler 142 into the compression chamber 134, and is compressed in the compression chamber 134.

[0027] In accordance with the rotation of the crankshaft 108, the refrigerating machine oil 103 is fed to sliding portions from the oil-feeding pump 120 to lubricate the sliding portions. Also, the refrigerating machine oil 103 seals between the piston 132 and the cylinder bore 113. Hereinafter, a shaft part and a bearing part of the refrigerant compressor 100 are illustratively described in detail.

[Shaft Part and Bearing Part]

[0028] The refrigerant compressor 100 includes a shaft part and a bearing part. The shaft part is at least one of the main shaft 109 and the eccentric shaft 110. The bearing part is at least one of the main bearing 111 and the eccentric bearing 119. Either the shaft part or the bearing part is provided with a tapered portion. The tapered portion is formed on at least one of one end side and the other end side of the shaft part or the bearing part in the axial direction of the bearing part such that the diameter of the shaft part or the bearing part changes from the outer side toward the central side in the longitudinal direction of the crankshaft 108. The tapered portion allows the shaft part and the bearing part to come into line contact with each other in a state where the axis of the shaft part is tilted relative to the axis of the bearing part.

[0029] In the refrigerant compressor 100, a ratio C / D, which is the ratio of a clearance C between the shaft part and the bearing part to the diameter D of the shaft part, is set to a value within the range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}. A taper depth d_B, which corresponds to a distance between one end and the other end of the tapered portion in the axial direction of the bearing part, the distance being in a direction perpendicular to the axis of the bearing part, is set to a value not less than 2.0×10^{-3} mm.

[0030] In the refrigerant compressor 100, the shaft part or the bearing part is provided with a pair of the tapered portions that are formed on both sides of the shaft part or the bearing part in the axial direction of the bearing part. In a case where the shaft part is provided with the tapered portions, the external diameter of each tapered portion changes from its one end toward the other end in the axial direction of the shaft part. In a case where the bearing part is provided with the tapered portions, the internal diameter of each tapered portion changes from its one end toward the other end in the axial direction of the bearing part.

[0031] In the refrigerant compressor 100, the following elements satisfy Math. 1 and Math. 2 indicated below: a taper depth d_{BU}, which is the taper depth d_B of the tapered portion on the one end side of the shaft part or the bearing part in the axial direction of the bearing part; a taper width W_{BU}, which is the width, in the axial direction of the bearing part, of the tapered portion on the one end side of the shaft part or the bearing part in the axial direction of the bearing part; a taper depth d_{BL}, which is the taper depth d_B of the tapered portion on the other end side of the shaft part or the bearing part in the axial direction of the bearing part; a taper width W_{BL}, which is the width, in the axial direction of the bearing part, of the tapered portion on the other end side of the shaft part or the bearing part in the axial direction of the bearing part; a bearing length B of the bearing part; and the clearance C.

[Math. 1]

$$d_{BU} / W_{BU} \leq (C + d_{BU} + d_{BL}) / B$$

[Math. 2]

$$d_{BL} / W_{BL} \leq (C + d_{BU} + d_{BL}) / B$$

[0032] Here, $(C + d_{BU} + d_{BL})$ corresponds to a maximum gap G, which is the sum of the clearance C, the taper depth d_{BU}, and the taper depth d_{BL}. In other words, in a combination of the shaft part and the bearing part corresponding thereto, the maximum gap G is the sum of the clearance C and a total value of the taper depths d_B of the tapered portions. Hereinafter, $(C + d_{BU} + d_{BL})$ is also referred to as the maximum gap G.

[0033] Next, the refrigerant compressor 100 with the above-described configuration is illustratively described in detail. FIG. 2 is an enlarged sectional view of an E region of the refrigerant compressor 100 of FIG. 1. FIG. 3 is a sectional view of main components of the refrigerant compressor 100 of FIG. 1. As shown in FIGS. 1 to 3, the main shaft 109 extends vertically.

[0034] In the refrigerant compressor 100, of the shaft part or the bearing part, an opposite-to-taper surface that faces the opposite surface of the tapered portion on the one end side in the axial direction of the bearing part is provided with a first sliding surface formed thereon. Also, of the shaft part or the bearing part, an opposite-to-taper surface that faces the opposite surface of the tapered portion on the other end side in the axial direction of the bearing part is provided with a second sliding surface formed thereon.

[0035] In the refrigerant compressor 100, at least one of the shaft part and the bearing part is provided with a pair of the tapered portions that are formed on both sides of the at least one of the shaft part and the bearing part in the axial direction of the bearing part, and at least one of the shaft part and the bearing part includes a small-diameter portion that has a less diameter than the maximum diameter of the tapered portions.

[0036] As one example, in the refrigerant compressor 100 of the present embodiment, the main shaft 109 includes a first sliding surface 151, a small-diameter portion 152, and a second sliding surface 153. The first sliding surface 151 is disposed on the upper portion of the main shaft 109. The second sliding surface 153 is disposed on the lower portion of the main shaft 109. The small-diameter portion 152 is disposed between the first sliding surface 151 and the second sliding surface 153.

[0037] The small-diameter portion 152 has a less diameter than that of the first sliding surface 151. A diameter D_{LO} of a portion of the main shaft 109, the portion being provided with the second sliding surface 153, is equal to a diameter Duo of a portion of the main shaft 109, the portion being provided with the first sliding surface 151 (see FIG. 5).

[0038] The main bearing 111, which pivotally supports the main shaft 109, is disposed such that the axis of the main bearing 111 extends vertically. The upper end of the inner peripheral surface of the main bearing 111 is provided with a tapered portion 170U. The lower end of the inner peripheral surface of the main bearing 111 is provided with a tapered portion 170L. That is, in the present embodiment, the bearing part is provided with a pair of tapered portions. The internal diameter of the main bearing 111, except the tapered portions 170U and 170L, is constant.

[0039] Each tapered portion 170U or 170L has a linear or continuously curved surface when seen in a direction perpendicular to the axis of the tapered portion 170U or 170L. FIG. 2 shows a configuration in which the tapered portion 170U has a linear surface between an inner one end 171 and an outer other end 172 in the axial direction of the main bearing 111. The tapered portion 170L is configured in the same manner.

[0040] Each tapered portion 170U or 170L is formed over the entire circumferential direction of the inner peripheral surface of the main bearing 111. The taper depth d_B (d_{BU} or d_{BL}), which corresponds to a distance between the one end 171 and the other end 172 of the tapered portion 170U or 170L in the axial direction of the main bearing 111, the distance being in a direction perpendicular to the axis of the main bearing 111, is set to a value in units of μm.

[0041] The method of forming each of the tapered portions 170U and 170L is not particularly limited. Each of the tapered portions 170U and 170L of the present embodiment is formed by using a prototype tool. The prototype tool includes a radial needle bearing and a rotating shaft. The radial needle bearing has an internal diameter of 12 mm, an external diameter of 16 mm, and a roller diameter of 2 mm. The rotating shaft is provided with a minute slope. A bearing that is to be formed into the main bearing 111 is prepared. The prototype tool is, while being rotated, press-fitted into the bearing, and thereby an end portion of the bearing is deformed. In this manner, each of the tapered portions 170U and 170L is formed.

[0042] Assuming that the tapered portions are absent, the clearance C herein corresponds to a difference between the internal diameter of the bearing part and the external diameter of a portion of the shaft part, the portion facing the inner peripheral surface of the bearing part. There may be a case where the portion of the shaft part, the portion facing the inner peripheral surface of the bearing part, has an external diameter that varies at a plurality of positions. In this case, assuming that the tapered portions are absent, the clearance C corresponds to a difference between the internal diameter of the bearing part and a maximum external diameter of the portion of the shaft part, the portion facing the inner peripheral surface of the bearing part.

[0043] Specifically, in a case where the diameters D_{LO} and Duo of the shaft part are equal to each other as described above, assuming that the tapered portions are absent, the clearance C corresponds to a difference between the internal diameter of the bearing part and the external diameter of portions of the shaft part, the portions being provided with the respective sliding surfaces 151 and 153. In other words, in the present embodiment, the clearance C is a difference between the internal diameter D_I of a portion of the main bearing 111, the portion including neither the tapered portion 170U nor the tapered portion 170L, and the diameter D_{LO} or D_{UO} of a portion of the main shaft 109, the portion being provided with the first or second sliding surface 151 or 153.

[0044] There may be a case where the diameters D_{LO} and D_{UO} of the shaft part are different from each other. In this case, assuming that the tapered portions are absent, the clearance C may be a difference between the internal diameter of the main bearing 111 and a greater one of the diameters D_{LO} and D_{UO} of the main shaft 109.

[0045] As shown in FIGS. 2 and 3, in the present embodiment, as one example, in a sectional view of an end portion of the main bearing 111 in a plane including the axis 111c of the main bearing 111, the taper width W_{BU} of the tapered portion 170U in a direction parallel to the axis 111c of the main bearing 111 (in other words, the taper width W_{BU}, which is the width, in the axial direction of the main bearing 111, of the tapered portion 170U on the one end side in the axial direction of the main bearing 111) is set to 10 mm, and the taper depth d_{BU} is set to 4.0×10^{-3} mm.

[0046] Also, in the sectional view of the end portion of the main bearing 111 in the plane including the axis 111c of the main bearing 111, the taper width W_{BL} of the tapered portion 170L in the direction parallel to the axis 111c of the main bearing 111 (in other words, the taper width W_{BL}, which is the width, in the axial direction of the main bearing 111, of the tapered portion 170L on the other end side in the axial direction of the main bearing 111) is set to 10 mm, and the taper depth d_{BL} is set to 4.0×10^{-3} mm.

[0047] Further, the bearing length B of the main bearing 111 is set to 43.5 mm. The internal diameter D_I of a portion of the main bearing 111, the portion including neither the tapered portion 170U nor the tapered portion 170L, is set to 16.026 mm. A diameter Do, which is the diameter of a portion of the main shaft 109, the portion being provided with the first sliding surface 151, or the diameter of a portion of the main shaft 109, the portion being provided with the second sliding surface 153, is set to 16.010 mm. The clearance C between the main shaft 109 and the main bearing 111 is set to 1.6×10^{-2} mm.

[0048] As a result, d_{BU} / w_{BU} and d_{BL} / w_{BL} are each set to 4.0×10^{-4}. Also, $(C + d_{BU} + d_{BL}) / B$ is set to 5.5×10^{-4}. That is, d_{BU} / w_{BU} and d_{BL} / w_{BL} are each less than $(C + d_{BU} + d_{BL}) / B$. Also, a ratio C / D_O, which is the ratio of the clearance C to the diameter D_O of the main shaft 109, is set to 1.0×10^{-3}.

[0049] In the refrigerant compressor 100, in a combination of the shaft part and the bearing part corresponding thereto, a ratio G / D, which is the ratio of the maximum gap G to the diameter D of the shaft part (in the above example, the diameter Do of the main shaft 109), is set to a value not greater than 4.0×10^{-3}. The maximum gap G is the sum of the clearance C and a total value of the taper depths d_B of the tapered portions (in this example, a total value $d_{BU} + d_{BL}$ of the taper depths of the two respective tapered portions 170U and 170L in the combination of the main shaft 109 and the main bearing 111) (i.e., the maximum gap $G = C + d_{BU} + d_{BL}$).

[0050] As described above, the ratio C / D, the taper depths d_B (d_{BU}, d_{BL}), and the ratio G / D are set to the values within the aforementioned ranges. With such settings, the distance between the shaft part and the bearing part can be suitably set in relation to the diameter D of the shaft part, and also, the tapered portions 170U and 170L each having a favorable sloped surface can be formed. Consequently, local metal contact between the shaft part and the bearing part can be prevented, and the formation of an oil film between a sliding portion of the shaft part and a sliding portion of the bearing part can be facilitated. This makes it possible to provide the refrigerant compressor 100, which is a low-input and highly efficient refrigerant compressor with excellent long-term durability.

[0051] In the refrigerant compressor 100, the bearing length B of the bearing part and the clearance C satisfy the relational expressions Math. 1 and Math. 2. Accordingly, the degree of the slope of each of the tapered portions 170U and 170L is adjusted to be suitably small. Therefore, during the refrigerant compressor 200 being driven, when run-out of the shaft part occurs, the surface of the tapered portion 170U or 170 and the opposite surface of the shaft part are allowed to extend along with each other in a more aligned manner (see FIG. 6). Consequently, the formation of an oil film between the surface of the tapered portion 170U or 170 and the opposite surface of the shaft part can be further facilitated.

[0052] Further, in the refrigerant compressor 100, as one example, the first sliding surface 151 faces the surface of the tapered portion 170U; a sliding width L_1 of the first sliding surface 151 is less than the taper width W_{BU} of the tapered portion 170U; the second sliding surface 153 faces the surface of the tapered portion 170L; and a sliding width L_2 of the second sliding surface 153 is less than the taper width W_{BL} of the tapered portion 170L. Consequently, the viscous resistance between the shaft part and the bearing part is reduced effectively.

[0053] Still further, in the refrigerant compressor 100, the ratio G / D is set to a value not greater than 4.0×10^{-3}. With such a setting, the ratio of the maximum gap G to the diameter D of the shaft part can be suitably adjusted, which makes it possible to prevent a situation where tilting of the crankshaft 108 in the bearing part becomes excessively steep, causing increase in edge contact (the edge contact will be described below). This consequently makes it possible to prevent, for example, a situation where the edge contact causes wear at the distal end of the piston 132, and the amount of leakage of the refrigerant from the worn portion increases, causing deterioration in refrigeration capacity.

[0054] Still further, the shaft part of the refrigerant compressor 100 includes a coating formed on its surface portion that slides on the bearing part. The hardness of the coating is higher than or equal to the hardness of the surface of the bearing part facing the coating. In the present embodiment, at least one of (in this example, both) the main shaft 109 and the eccentric shaft 110 includes the coating.

[0055] The coating is not limited to a specific type of coating. The coating may be, for example, an oxide coating. Examples of the oxide coating include an iron oxide coating. For example, compared to a phosphatic coating, an iron oxide coating is chemically very stable, and has higher hardness. As a result of the oxide coating being formed, problems such as the generation of wear debris and the adhesion of wear debris to the coating can be prevented effectively. This makes it possible to effectively avoid increase in the amount of wear of the oxide coating, and thereby high wear resistance can be imparted to the coating.

[0056] The coating is required to be harder than the material that the coating is facing, as with the oxide coating. Assume that the base material of a portion of the shaft part, the portion being provided with the coating formed thereon, is a ferrous material. In this case, the coating may be formed not only by general quenching, but also by impregnating the surface layer of the shaft part with, for example, carbon or nitrogen. Alternatively, the coating may be formed by a steam oxidization process, or by an oxidization process in which a material is immersed in an aqueous solution such as sodium hydroxide.

[0057] The coating is not limited to a compound layer that is formed by, for example, the oxidization process mentioned above, carburizing, nitriding, or a different oxidization process, but may be, for example, a strength reinforced layer whose base material strength is increased by suppressing dislocation slip by any of, for example, the following: cold working; work hardening; solid solution strengthening; precipitation strengthening; dispersion strengthening; and grain refining. The coating may be a treatment layer formed by a coating technique that is any of, for example, the following: plating; thermal spraying; PVD; and CVD.

[Validation Test]

[0058] The refrigerant compressor 100 of Embodiment 1 was produced as Example. Another refrigerant compressor was produced as Conventional Example. The refrigerant compressor of Conventional Example is the same as the refrigerant compressor 100, except that the refrigerant compressor of Conventional Example is not provided with the tapered portions 170U and 170L. The performance of each of these refrigerant compressors was evaluated when each compressor was inverter-driven to perform low-speed operation (at an operating frequency of 17Hz).

[0059] FIG. 4A is a characteristic diagram showing an input ratio between the refrigerant compressor of Example, which is shown in FIG. 1, and the refrigerant compressor of Conventional Example. FIG. 4B is a characteristic diagram showing a coefficient of performance (COP) ratio between the refrigerant compressor of Example, which is shown in FIG. 1, and the refrigerant compressor of Conventional Example.

[0060] The coefficient of performance is a coefficient used as an index indicating the energy consumption efficiency of refrigerator-freezer equipment or the like. The coefficient of performance is a value that is obtained by dividing a refrigeration capacity (W) by an applied input (W). FIG. 4A shows the ratio (input ratio) when the applied input value in Conventional Example is assumed as 100. FIG. 4B shows the ratio (COP ratio) when the COP value in Conventional Example is assumed as 100.

[0061] From the results shown in FIGS. 4A and 4B, it has been validated that Example exhibits a lower input and a higher COP than the comparative Conventional Example for the reason that the refrigerant compressor of Example is additionally provided with the tapered portions 170U and 170L.

[0062] FIG. 5 is a functional diagram illustrating a compressive load applied to the refrigerant compressor 100 of FIG. 1. FIG. 5 schematically shows the compressive load applied to the refrigerant compressor 100. An analysis of the validation test results on Example and Conventional Example is given below with reference to FIG. 5.

[0063] In the case of a reciprocating refrigerant compressor such as the refrigerant compressor 100, generally speaking, in the compression chamber 134 formed between the cylinder bore 113 and the piston 132, the internal pressure of the sealed container 101 is lower compared to a compressive load P that occurs in the cylinder axial direction of the cylinder bore 113. While the compressive load P is being applied to the eccentric shaft 110, the main shaft 109 is supported at one end by the single main bearing 111. Accordingly, during the refrigerant compressor being driven, run-out of the crankshaft 108, in which the crankshaft 108 rotates in a state where the crankshaft 108 is tilted inside the main bearing 111, occurs due to the influence of the compressive load P as described in a literature of Ito, et al. (Proceedings of The Japan Society of Mechanical Engineers Annual Meeting Vol. 5-1 (2005) p. 143).

[0064] As a result, component force PI of the compressive load P is applied to a portion of the main shaft 109, the portion corresponding to the upper end portion of the main bearing 111, and component force P2 of the compressive load P is applied to a portion of the main shaft 109, the portion corresponding to the lower end portion of the main bearing 111. Consequently, so-called edge contact occurs. In a conventional refrigerant compressor, there is a case where when the main shaft 109 is tilted inside the main bearing 111, local contact between the main shaft 109 and the main bearing 111 occurs, which causes increase in surface pressure. When the speed of the operation decreases, the thickness of the oil film formed between the main shaft 109 and the main bearing 111 becomes thinner, which may even result in breakage of the oil film. Consequently, solid contact between the main shaft 109 and the main bearing 111 occurs, which causes increase in sliding loss.

[0065] On the other hand, in the present embodiment (Example), since the main bearing 111 is provided with the tapered portions 170U and 170L, even if the main shaft 109 is tilted inside the main bearing 111, the surface of the main shaft 109 and the opposite surface of the main bearing 111 are arranged such that, when seen in a direction perpendicular to the axis of the main bearing 111, these surfaces extend along with each other. Accordingly, local metal contact between the main shaft 109 and the main bearing 111 is prevented.

[0066] Further, in the present embodiment (Example), the total value of the clearance C, the taper depth d_{BU}, and the taper depth d_{BL}, i.e., the maximum gap G (= C + d_{BU} + d_{BL}), is set to be relatively large. It is considered that, owing to such setting, the viscous resistance of the refrigerating machine oil 103 is reduced, and sliding loss is reduced significantly, which effectively lowers the input to the refrigerant compressor.

[0067] From the above results, it is understood that, by providing the bearing part of the refrigerant compressor with the tapered portions, local metal contact between the bearing part and the shaft part can be prevented, thereby achieving improvement in durability, and also, the performance of the refrigerant compressor can be improved.

[Evaluation Test]

[0068] Next, after the above validation test results were obtained, a performance evaluation test and a reliability evaluation test were performed on the refrigerant compressor, and thereby numerical value ranges with which to achieve improvement in the performance of the refrigerant compressor were clarified. In the performance evaluation test, the clearance C between the main shaft and the main bearing, the bearing length B, the diameter Do of the main shaft, the ratio C / Do of the clearance C to the diameter Do, the taper depths d_{BU} and d_{BL} of the tapered portions 170U and 170L, and the taper widths W_{BU} and W_{BL} were used as parameters. In the performance evaluation test, the refrigerant compressor was inverter-driven to perform low-speed operation (at an operating frequency of 17 Hz).

[0069] In the reliability evaluation test, the refrigerant compressor was operated for 160 hours in a high-temperature and high-load intermittent operation mode. Thereafter, the refrigerant compressor was disassembled, and wear of sliding components (such as the crankshaft and piston) was measured, based on which evaluations were made.

[0070] The description hereinafter refers to a graph (shown in FIG. 7). Plotted in the graph is a relationship among the taper width W_{BU} and the taper depth d_{BU} of the tapered portion 170U, the taper width W_{BL} and the taper depth d_{BL} of the tapered portion 170L, the bearing length B of the main bearing 111, and the clearance C between the main shaft 109 and the main bearing 111. In the graph, a range that satisfies the aforementioned relational expressions Math. 1 and Math. 2 is referred to as an area A1, and a range that satisfies relational expressions Math. 3 and Math. 4 shown below is referred to as an area A2.

[Math. 3]

$$d_{BU} / W_{BU} > (C + d_{BU} + d_{BL}) / B$$

[Math. 4]

$$d_{BL} / W_{BL} > (C + d_{BU} + d_{BL}) / B$$

[0071] In each of these tests, the clearance C between the main shaft 109 and the main bearing 111 was set to 1.6×10^{-2} mm, and the bearing length B of the main bearing 111 was set to 43.5 mm. FIG. 6 illustrates contact states each between the main shaft 109 of FIG. 1 and the main bearing 111 when the main shaft 109 is tilted inside the main bearing 111. FIG. 6 shows, for each contact state, a correlation between the contact state and relational expressions that hold true in the contact state. FIG. 7 is a graph showing setting ranges of Examples 1 to 2 and Comparative Examples 1 to 2. Table 1 below shows evaluations on Examples 1 to 2 and Comparative Examples 1 to 2 in the performance evaluation test and the reliability evaluation test.

[Table 1]

		Surface Treatment on Crankshaft			
		Coating softer than the opposite side		Coating harder than the opposite side	
		Performance	Reliability	Performance	Reliability
Comp. Ex. 1	Area A1	C	C	C	B
	Area A2	C	C	C	B
Ex. 1	Area A1	A	B	A	A
Ex. 2	Area A2	B	B	B	A
Comp. Ex. 2	Area A1	C	C	C	C
	Area A2	C	C	C	C

[0072] In the graph of FIG. 7, the lower horizontal axis represents each of the taper depths d_{BU} and d_{BL}, and the upper horizontal axis represents the ratio of the maximum gap G to the diameter D of the shaft part, i.e., $(C + d_{BU} + d_{BL}) / D$. In FIG. 7, the vertical axis represents each of the taper widths W_{BU} and W_{BL}. FIG. 7 shows a solid line representing positions that satisfy relational expressions Math. 5 and Math. 6 shown below.

[Math. 5]

$$d_{BU} / w_{BU} = (C + d_{BU} + d_{BL}) / B$$

[Math. 6]

$$d_{BL} / w_{BL} = (C + d_{BU} + d_{BL}) / B$$

[0073] In the tests of FIG. 7, Example 1 was the refrigerant compressor 100 with such settings that the relational expressions Math. 1 and Math. 2 are satisfied in an area in which each of the taper depths d_{BU} and d_{BL} is not less than 2.0×10^{-3} mm and the ratio of the maximum gap G to the diameter D of the shaft part, i.e., $(C + d_{BU} + d_{BL}) / D$, is not greater than 4.0×10^{-3}. On the other hand, Example 2 was a refrigerant compressor with such settings that the relational expressions Math. 3 and Math. 4 are satisfied in an area in which each of the taper depths d_{BU} and d_{BL} is not less than 2.0×10^{-3} mm and the ratio of the maximum gap G to the diameter D of the shaft part, i.e., $(C + d_{BU} + d_{BL}) / D$, is not greater than 4.0×10^{-3}.

[0074] Comparative Example 1 was a refrigerant compressor with each of the taper depths d_{BU} and d_{BL} set to a value less than 2.0×10^{-3} mm. Comparative Example 2 was a refrigerant compressor with the ratio of the maximum gap G to the diameter D of the shaft part, i.e., $(C + d_{BU} + d_{BL}) / D$, set to a value greater than 4.0×10^{-3}.

[0075] In each of Examples 1 to 2 and Comparative Examples 1 to 2, a coating was formed on a surface portion of the shaft part, the surface portion sliding on the bearing part. Formed as the coating was a manganese phosphate coating whose hardness is lower than the hardness of the material of the opposite main bearing, or an iron oxide coating whose hardness is higher than the hardness of the material of the opposite main bearing.

[0076] The evaluations shown in Table 1 were made with reference to the performance of the refrigerant compressor of Conventional Example provided with no tapered portions and to the wear result of the refrigerant compressor of Conventional Example in the reliability test. In Table 1, "A" indicates significant improvement in characteristics compared to Conventional Example. Specifically, "A" indicates an evaluation that the compressor performance has improved and that the wear between the shaft part and the bearing part has been reduced to the greatest degree. Indicated by "B" is an evaluation next to "A", and "B" indicates an evaluation that slight improvement in characteristics has been made compared to the refrigerant compressor of Conventional Example. Indicated by "C" is an evaluation next to "B", and "C" indicates an evaluation that no improvement in characteristics has been made compared to the refrigerant compressor of Conventional Example.

[0077] It is understood from FIG. 7 and Table 1 that both Examples 1 and 2 exhibit improvement in performance and reliability compared to Comparative Examples 1 and 2. Example 1 exhibits higher performance and reliability than Example 2. It has been confirmed that particularly when the coating of the shaft part is formed as a coating whose hardness is higher than the hardness of the material of the opposite main bearing, excellent compressor performance is obtained and the wear between the shaft part and the bearing part is reduced significantly, realizing further improvement in reliability.

[0078] On the other hand, Comparative Example 1 with the taper depths d_{BU} and d_{BL} set to a value less than 2.0×10^{-3} mm exhibits no improvement in performance compared to Conventional Example, regardless of the areas A1 and A2. The reason for this is considered that, in Comparative Example 1, for example, the taper depths of the tapered portions are too shallow, and therefore, advantages owing to shape differences from the tapered portions of Examples 1 and 2 cannot be obtained.

[0079] Also, Comparative Example 2 with the ratio $(C + d_{BU} + d_{BL}) / D$ set to a value greater than 4.0×10^{-3} exhibits no improvement in performance compared to Conventional Example, regardless of the areas A1 and A2. The reason for this is considered that, in Comparative Example 2, for example, the tilting of the crankshaft 108 inside the bearing part was excessively steep, and edge contact occurred. Specifically, in Comparative Example 2, it is considered that the occurrence of edge contact caused wear at the distal end of the piston 132, and the amount of leakage of the refrigerant from the worn portion increased, causing deterioration in refrigeration capacity. It is considered that, for these reasons, no improvement in performance was observed in Comparative Example 2.

[0080] Also in the separately performed compressor reliability test, significant wear on the distal end portion of the piston was found in Comparative Example 2, which appeared to have been caused by edge contact. This backs up the above analysis.

[0081] In the above tests, the clearance C between the main shaft 109 and the main bearing 111 was set to 1.6×10^{-2} mm, and the bearing length B of the main bearing 111 was set to 43.5 mm, and the test results obtained under these conditions are indicated herein. It should be noted that it has been confirmed that the same advantageous effects are obtained also in a case where the ratio C / D is set to a value within the range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}.

[0082] The diameter D_O of the main shaft 109 can be suitably set. For example, the diameter D_O can be set to a value within the range of not less than 10 mm and not greater than 28 mm. For example, desirably, in accordance with the shaft part diameter D thus set, the clearance C, the taper depths d_{BU} and d_{BL}, and the taper widths W_{BU} and W_{BL} are set such that the ratio C / D and the ratio $(C + d_{BU} + d_{BL}) / D$ fall within suitable value ranges, respectively.

[0083] In the refrigerant compressor 100 of the present embodiment, the inner peripheral surface of the main bearing 111 is provided with the tapered portions 170U and 170L. However, as an alternative, the outer peripheral surface of the main shaft 109 may be provided with the tapered portions. Also in this case, the same advantageous effects are obtained. As another alternative, the inner peripheral surface of the eccentric bearing 119 may be provided with the tapered portions. Yet another alternative, the outer peripheral surface of the eccentric shaft 110 may be provided with the tapered portions. In these cases, in a combination of the eccentric shaft 110 and the eccentric bearing 119, the ratio C / D, the taper depths, and the ratio G / D are set in the same manner as in the above-described combination of the main shaft 109 and the main bearing 111. These configurations can also contribute to improvement in the performance and reliability of the refrigerant compressor as with the present embodiment.

[0084] The present embodiment has described an advantageous effect that the performance of the refrigerant compressor 100 is improved in a case where the refrigerant compressor 100 is driven to perform low-speed operation (e.g., at an operating frequency of 17 Hz). However, the same advantageous effect is obtained not only when the refrigerant compressor 100 performs low-speed operation, but also when the refrigerant compressor 100 operates at a commercial rotation speed or when the refrigerant compressor 100 performs high-speed operation at an even higher rotation speed.

[0085] The refrigerant compressor is not limited to a reciprocating compressor, but may be a different type of compressor, such as a rotary compressor or a scroll compressor. That is, in the case of a different type of refrigerant compressor, such as a rotary refrigerant compressor or a scroll refrigerant compressor, the same performance-improving and reliability-improving effects are obtained when the tapered portion is applied to a sliding portion (a journal bearing sliding portion) that is constituted by the outer peripheral surface of a shaft and the inner peripheral surface of a bearing. Hereinafter, other embodiments are described focusing on differences from Embodiment 1.

(Embodiment 2)

[0086] FIG. 8 is a schematic sectional view of a rotating (rotary) refrigerant compressor 200 according to Embodiment 2. FIG. 9 is an enlarged sectional view of a B region of the refrigerant compressor 200 of FIG. 8. FIG. 9 is an enlarged sectional view of the B region (positioned at the lower side of a main bearing 209) surrounded by a dashed circle in FIG. 8. FIG. 10 is a sectional view of the refrigerant compressor 200 of FIG. 8, the sectional view being taken along line A-A' of FIG. 8 as viewed in the direction of the arrows of line A-A'.

[0087] As shown in FIGS. 8 to 10, the refrigerant compressor 200 includes a sealed container 201, an electric element 202, and a compression element 203. Refrigerating machine oil 220 is stored in the bottom of the sealed container 101. The electric element 202 and the compression element 203 are accommodated in the sealed container 201. The electric element 202 includes a stator 202a and a rotor 202b. The compression element 203 includes a crankshaft 208, the main bearing 209, an auxiliary bearing 211, a cylinder 210, and a roller 213.

[0088] The crankshaft 208 extends vertically, and includes a main shaft 206 and an eccentric shaft 212. The eccentric shaft 212 is disposed on a non-end portion of the main shaft 206. The main shaft 206 is, above the eccentric shaft 212, pivotally supported by the main bearing 209, and below the eccentric shaft 212, pivotally supported by the auxiliary bearing 211. The rotor 202b of the electric element 202 is fixed to the main shaft 206. The outer periphery of the rotor 202b is surrounded by the stator 202a.

[0089] The eccentric shaft 212 is disposed inside the cylinder 210, which extends vertically. The roller 213 is formed in a cylindrical shape, and is disposed such that the axis thereof extends vertically. Inside the cylinder 210, the main shaft 206 and the eccentric shaft 212 are inserted in the roller 213. The eccentric shaft 212 is supported by the inner peripheral surface of the cylinder 210 via the roller 213. In the present embodiment, the roller 213 corresponds to the eccentric bearing of the eccentric shaft 212. During the refrigerant compressor 200 being driven, the roller 213 makes planetary motion about the axis of the main shaft 206 of the crankshaft 208.

[0090] The cylinder 210 is provided with a through groove 222 extending horizontally. A shaft-shaped vane 214 is inserted in the through groove 222. One end (distal end) of the vane 214 in the longitudinal direction is pressed against a peripheral surface 231 of the roller 213 by a spring 215 and back pressure (delivery pressure). Accordingly, the space between the cylinder 210 and the roller 213 is divided into a suction chamber 216 and a compression chamber 217. The suction chamber 216 sucks refrigerant gas from the outside. The compression chamber 217 compresses the refrigerant gas.

[0091] The cylinder 210 is further provided with a suction hole 205. One end of a suction tube 204 is inserted in the suction hole 205. The refrigerant compressor 200 is connected to an accumulator (not shown) via the suction tube 204. The inner peripheral surface of the cylinder 210 is provided with a delivery notch 219.

[0092] During the refrigerant compressor 200 being driven, the electric element 202 causes the crankshaft 208 to rotate about the axis of the main shaft 206, and the roller 213 makes planetary motion (left rotation in FIG. 10). Consequently, the refrigerant gas is sucked into the suction chamber 216 from the outside through the suction tube 204 and the suction hole 205. As a result of the internal pressure of the compression chamber 217 increasing, the refrigerant gas is compressed, and the compressed refrigerant gas is, after passing through the delivery notch 219, delivered into the sealed container 201 through an unshown delivery hole.

[0093] The vane 214, which partitions off the suction chamber 216 and the compression chamber 217 from each other, has its one end in the longitudinal direction pressed against the peripheral surface 231 of the roller 213 by the spring 215 and back pressure. As a result, the vane 214 moves while sliding at a contact point with the peripheral surface 231 of the roller 213. Due to such motion of the vane 214, the crankshaft 208 receives pressure from a direction perpendicular to the axis of the main shaft 206, and thereby deflects. Consequently, the crankshaft 208 rotates in such a manner that run-out occurs in each clearance between the main bearing 209 and the auxiliary bearing 211.

[0094] Due to the run-out, the crankshaft 208 may make edge contact with at least one of the following: the upper end of the main bearing 209 (in FIG. 8, an end portion on the electric element 202 side); the lower end of the main bearing 209 (in FIG. 8, an end portion on the roller 213 side); the upper end of the auxiliary bearing 211 (in FIG. 8, an end portion on the roller 213 side); and the lower end of the auxiliary bearing 211 (in FIG. 8, an end portion on an oil feeder 221 side, the oil feeder 221 being provided at the lower end of the crankshaft 208). Due to the edge contact, there are risks that the sliding surface may get damaged and adhesive wear may occur. Adhesive wear is a phenomenon in which the sliding surface gets cut and worn by minute wear debris.

[0095] In view of the above, in the refrigerant compressor 200, a tapered portion 270U is provided at the upper end of the main bearing 209, which pivotally supports the crankshaft 208, and a tapered portion 270L is provided at the lower end of the main bearing 209. Also, a tapered portion 280U is provided at the upper end of the auxiliary bearing 211, and a tapered portion 280L is provided at the lower end of the auxiliary bearing 211. The tapered portions 270U and 280U correspond to the tapered portion 170U, and the tapered portions 270L and 280L correspond to the tapered portion 170L. It should be noted that FIG. 9 shows only the tapered portion 270L among these tapered portions.

[0096] Each of the tapered portions 270U and 270L is formed over the entire circumferential direction of the inner peripheral surface of the main bearing 209. The taper depth d_B (d_{BU} or d_{BL}), which corresponds to a distance between one end 271 and the other end 272 of the tapered portion 270U or 270L in the axial direction of the main bearing 209, the distance being in a direction perpendicular to the axis of the main bearing 209, is set to a value in units of μm.

[0097] As shown in FIGS. 8 and 9, the ratio C / D of the clearance C between the crankshaft 208 (the main shaft 206) and the bearing part (the main bearing 209) to the diameter D of the crankshaft 208 (the main shaft 206) is set to a value within the range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}. The ratio G / D in a combination of the shaft part and the bearing part corresponding thereto (in this example, a combination of the main shaft 206 and the main bearing 209) is set to a value not greater than 4.0×10^{-3}.

[0098] Although not illustrated, the ratio C / D of the clearance C between the crankshaft 208 (the main shaft 206) and the bearing part (the auxiliary bearing 211) to the diameter D of the crankshaft 208 (the main shaft 206) is also set to a value within the range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3}.

[0099] Further, at least one of the taper depth d_{BU} (not shown) of each of the tapered portions 270U and 280U and the taper depth d_{BL} of each of the tapered portions 270L and 280L is (in this example, both the taper depth d_{BU} and the taper depth d_{BL} are) set to a value within the range of not less than 2.0×10^{-3} mm. The crankshaft 208 includes a coating formed on its surface portions that slide on the main bearing 209 and the auxiliary bearing 211. This coating is the same as the coating described in Embodiment 1.

[0100] By adopting these settings, even if run-out of the crankshaft 208 occurs and the aforementioned edge contact occurs, the surface of the main shaft 206 and the opposite surface of the main bearing 209 are arranged such that, when seen in a direction perpendicular to the axis of the crankshaft 208, these surfaces extend along with each other, and also, the surface of the main shaft 206 and the opposite surface of the auxiliary bearing 211 are arranged such that, when seen in the direction perpendicular to the axis of the crankshaft 208, these surfaces extend along with each other. Accordingly, local metal contact is prevented between the main shaft 206 and the main bearing 209, and between the main shaft 206 and the auxiliary bearing 211. Therefore, the refrigerant compressor 200 has favorable frictional wear characteristics, high performance, and high reliability.

[0101] The tapered portion 270L shown in FIG. 9 is, when seen in a direction perpendicular to the axis thereof, formed to have a curved shape that has a continuously curved surface. However, as an alternative, the tapered portion 270L may be formed to have a linear surface. In a case where a plurality of tapered portions are provided, the tapered portions may have different shapes from each other. Although the refrigerant compressor 200 described above includes the four tapered portions 270U, 270L, 280U, and 280L, the refrigerant compressor 200 is only required to include at least one of these tapered portions.

[0102] An object on which the above-described coating is formed is not limited to the crankshaft 208. The coating may be provided on a sliding portion of any of the components (e.g., mechanical parts, devices, and units such as a pump and a motor) of a refrigerant compressor or a refrigeration apparatus in which the refrigerant compressor is used. Next, the configuration of a refrigeration apparatus in which the refrigerant compressor 100 or 200 is used is illustratively described.

(Embodiment 3)

[0103] FIG. 11 is a schematic diagram of a refrigeration apparatus 300 according to Embodiment 3. Hereinafter, a fundamental configuration of the refrigeration apparatus 300 is described briefly. As shown in FIG. 11, the refrigeration apparatus 300 includes a body 301, a dividing wall 307, and a refrigerant circuit 309.

[0104] The body 301 includes a thermally insulated box and a door. The box is provided with an opening that communicates with the inside of the box. The opening of the box is opened and closed by the door. The body 301 includes a storage space 303 and a machinery room 305. The storage space 303 is a space in which a product is stored. In the machinery room 305, the refrigerant circuit 309 is disposed, which cools the inside of the storage space 303. The storage space 303 and the machinery room 305 are divided by the dividing wall 307. An air feeder (not shown) is disposed in the storage space 303. FIG. 11 is a partially cutaway view of the box, showing the inside of the body 301.

[0105] The refrigerant circuit 309 includes: the refrigerant compressor 100 or 200; a radiator 313; a decompressor 315; and a heat absorber 317. The refrigerant compressor 100 or 200, the radiator 313, the decompressor 315, and the heat absorber 317 are connected by piping in an annular manner.

[0106] The radiator 313 radiates the heat of the refrigerant. The decompressor 315 decompresses the refrigerant. The heat absorber 317 absorbs the heat of the refrigerant. The heat absorber 317 is disposed in the storage space 303, and generates cooling heat. As indicated by arrows in FIG. 11, the air feeder causes the cooling heat generated by the heat absorber 317 to circulate inside the storage space 303. In this manner, the air in the storage space 303 is stirred, and the inside of the storage space 303 is cooled.

[0107] In the refrigeration apparatus 300 configured as above, high wear resistance between the shaft part and the bearing part is obtained in the refrigerant compressor 100 or 200. Also, since the formation of the oil film between the shaft part and the bearing part is facilitated, local metal contact between the shaft part and the bearing part can be prevented, and thereby high reliability and high compressor performance are obtained. Accordingly, by including the refrigerant compressor 100 or 200, the refrigeration apparatus 300 can reduce electric power consumption, i.e., realize energy saving, and can have improved long-term reliability.

[0108] The present invention is not limited to the above-described embodiments. Various modifications, additions, or deletions can be made to the configurations without departing from the scope of the present invention. The above-described embodiments may be combined with each other in any manner. For example, part of a configuration in one embodiment may be applied to another embodiment. The scope of the present invention is defined by the appended claims, and all changes that fall within metes and bounds of the claims, or equivalence of such metes and bounds thereof are therefore intended to be embraced by the claims.

Industrial Applicability

[0109] As described above, the present invention has an excellent advantageous effect of being able to provide a refrigerant compressor that makes it possible to achieve improvement in efficiency while preventing reduction in durability and reliability by preventing wear of sliding portions, and to provide a refrigeration apparatus in which the refrigerant compressor is used. Therefore, the present invention is useful when widely applied to refrigerant compressors and refrigeration apparatuses using the same, because such applications make it possible to exert the above advantageous effect meaningfully.

Reference Signs List

[0110]

100, 200	refrigerant compressor
101, 201	sealed container
103, 220	refrigerating machine oil
106, 202	electric element
107, 203	compression element
108, 208	crankshaft
109, 206	main shaft (shaft part)
110, 212	eccentric shaft (shaft part)
111, 209	main bearing (bearing part)

111c	axis of the main bearing
119	eccentric bearing (bearing part)
151	first sliding surface
152	small-diameter portion
153	second sliding surface
170U, 170L, 270U, 270L, 280U, 280L	tapered portion
211	auxiliary bearing (bearing part)
300	refrigeration apparatus
309	refrigerant circuit
313	radiator
315	decompressor
317	heat absorber
B	bearing width
C	clearance
D	diameter of the shaft part
D_O	diameter of the main shaft
D_I	diameter (internal diameter) of the main bearing
d_B, d_{BU}, d_{BL}	taper depth
L_1	sliding width of the first sliding surface
L_2	sliding width of the second sliding surface
w_{BU}, w_{BL}	taper width

Claims

1. A refrigerant compressor comprising:

a sealed container in which refrigerating machine oil is stored;
an electric element accommodated in the sealed container and driven by electric power supplied from outside; and
a compression element accommodated in the sealed container and covered with the refrigerating machine oil, the compression element being driven by the electric element to compress refrigerant gas supplied from outside, wherein
the compression element includes:

a crankshaft including a main shaft and an eccentric shaft that are arranged side by side in a longitudinal direction;
a main bearing that pivotally supports the main shaft; and
an eccentric bearing that pivotally supports the eccentric shaft,

a shaft part that is at least one of the main shaft and the eccentric shaft, or a bearing part that is at least one of the main bearing and the eccentric bearing, is provided with a tapered portion, the tapered portion being formed on at least one of one end side and another end side of the shaft part or the bearing part in an axial direction of the bearing part such that a diameter of the shaft part or the bearing part changes from an outer side toward a central side in the longitudinal direction of the crankshaft, the tapered portion allowing the shaft part and the bearing part to come into line contact with each other in a state where an axis of the shaft part is tilted relative to an axis of the bearing part,
a ratio C / D, which is a ratio of a clearance C between the shaft part and the bearing part to a diameter D of the shaft part, is set to a value within a range of not less than 4.0×10^{-4} and not greater than 3.0×10^{-3},
a taper depth d_B, which corresponds to a distance between one end and another end of the tapered portion in the axial direction of the bearing part, the distance being in a direction perpendicular to the axis of the bearing part, is set to a value not less than 2.0×10^{-3} mm, and
in a combination of the shaft part and the bearing part corresponding thereto, a ratio G / D, which is a ratio of a maximum gap G to the diameter D of the shaft part, is set to a value not greater than 4.0×10^{-3}, the maximum gap G being a sum of the clearance C and a total value of the taper depth d_B of the tapered portion.

2. The refrigerant compressor according to claim 1, wherein

the shaft part or the bearing part is provided with a pair of the tapered portions that are formed on both sides of the shaft part or the bearing part in the axial direction of the bearing part, and
the following elements satisfy Math. 1 and Math. 2:

a taper depth d_{BU}, which is the taper depth d_B of the tapered portion on the one end side of the shaft part or the bearing part in the axial direction of the bearing part;
a taper width W_{BU}, which is a width, in the axial direction of the bearing part, of the tapered portion on the one end side of the shaft part or the bearing part in the axial direction of the bearing part;
a taper depth d_{BL}, which is the taper depth d_B of the tapered portion on the other end side of the shaft part or the bearing part in the axial direction of the bearing part;
a taper width W_{BL}, which is a width, in the axial direction of the bearing part, of the tapered portion on the other end side of the shaft part or the bearing part in the axial direction of the bearing part;
a bearing length B of the bearing part; and
the clearance C, where
Math. 1 is $d_{BU} / W_{BU} \leq (C + d_{BU} + d_{BL}) / B$,
Math. 2 is $d_{BL} / W_{BL} \leq (C + d_{BU} + d_{BL}) / B$.

3. The refrigerant compressor according to claim 2, wherein

of the shaft part or the bearing part, an opposite-to-taper surface that faces an opposite surface of the tapered portion on the one end side in the axial direction of the bearing part is provided with a first sliding surface formed thereon,
of the shaft part or the bearing part, an opposite-to-taper surface that faces an opposite surface of the tapered portion on the other end side in the axial direction of the bearing part is provided with a second sliding surface formed thereon, and
a sliding width L_1 of the first sliding surface facing the opposite surface of the tapered portion on the one end side is less than the taper width W_{BU}, and a sliding width L_2 of the second sliding surface facing the opposite surface of the tapered portion on the other end side is less than the taper width W_{BL}, when seen in a direction perpendicular to the axis of the opposite shaft part or bearing part.

4. The refrigerant compressor according to any one of claims 1 to 3, wherein

at least one of the shaft part and the bearing part is provided with a pair of the tapered portions that are formed on both sides of the at least one of the shaft part and the bearing part in the axial direction of the bearing part, and
at least one of the shaft part and the bearing part includes a small-diameter portion that has a less diameter than a maximum diameter of the tapered portions.

5. The refrigerant compressor according to any one of claims 1 to 4, wherein
the tapered portion has a linear or continuously curved surface when seen in a direction perpendicular to an axis of the tapered portion.

6. The refrigerant compressor according to any one of claims 1 to 5, wherein

the shaft part includes a coating formed on its surface portion that slides on the bearing part, and
a hardness of the coating is higher than or equal to a hardness of a surface of the bearing part, the surface facing the coating.

7. The refrigerant compressor according to any one of claims 1 to 6, wherein
the electric element is inverter-driven at a plurality of operating frequencies.

8. A refrigeration apparatus comprising a refrigerant circuit including:

the refrigerant compressor according to any one of claims 1 to 7;
a radiator that radiates heat of a refrigerant;
a decompressor that decompresses the refrigerant; and
a heat absorber that absorbs the heat of the refrigerant, wherein
in the refrigerant circuit, the refrigerant compressor, the radiator, the decompressor, and the heat absorber are connected by piping in an annular manner.

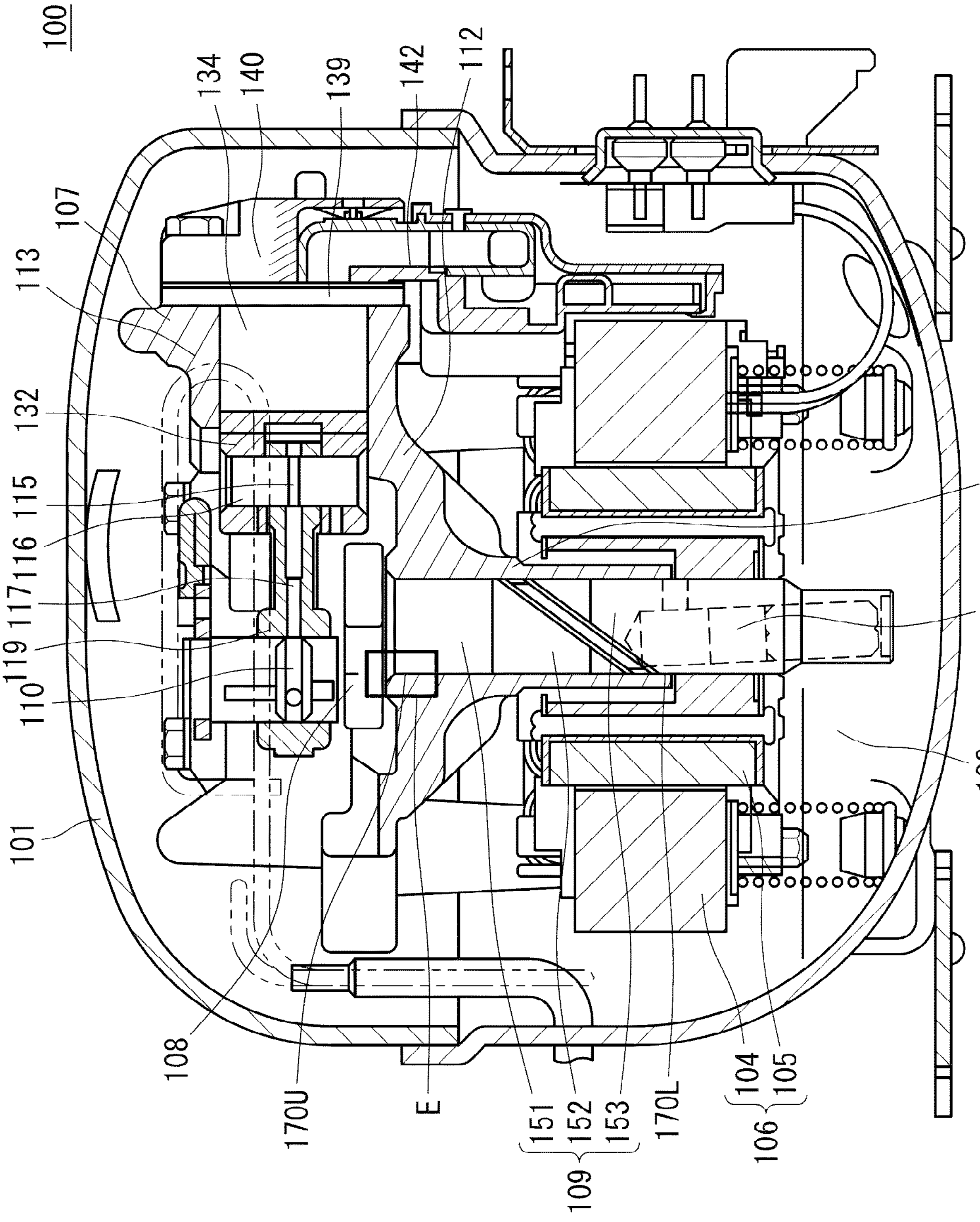

100
134
140
139
142
112
107
113
132
115
116
117
119
110
101
111
120
103
108
170U
E
151
152
153
170L
104
105
106
109

FIG.1

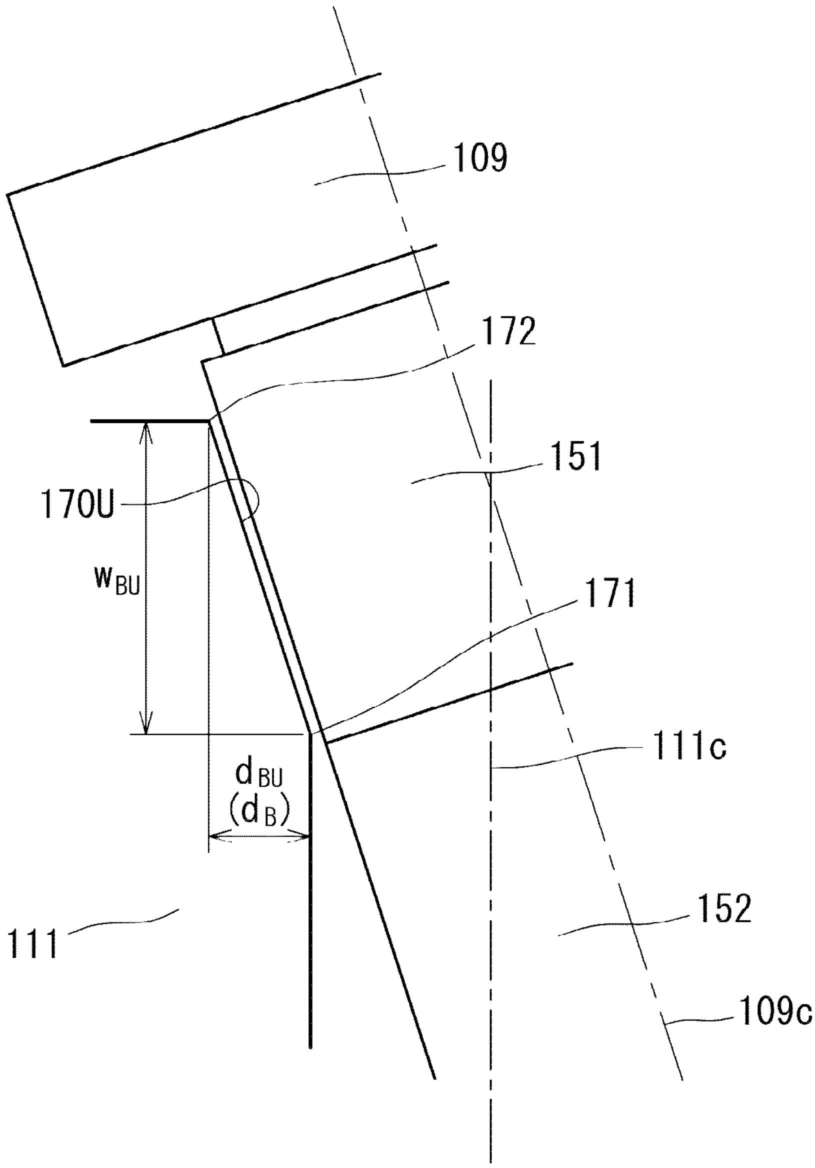


FIG.2

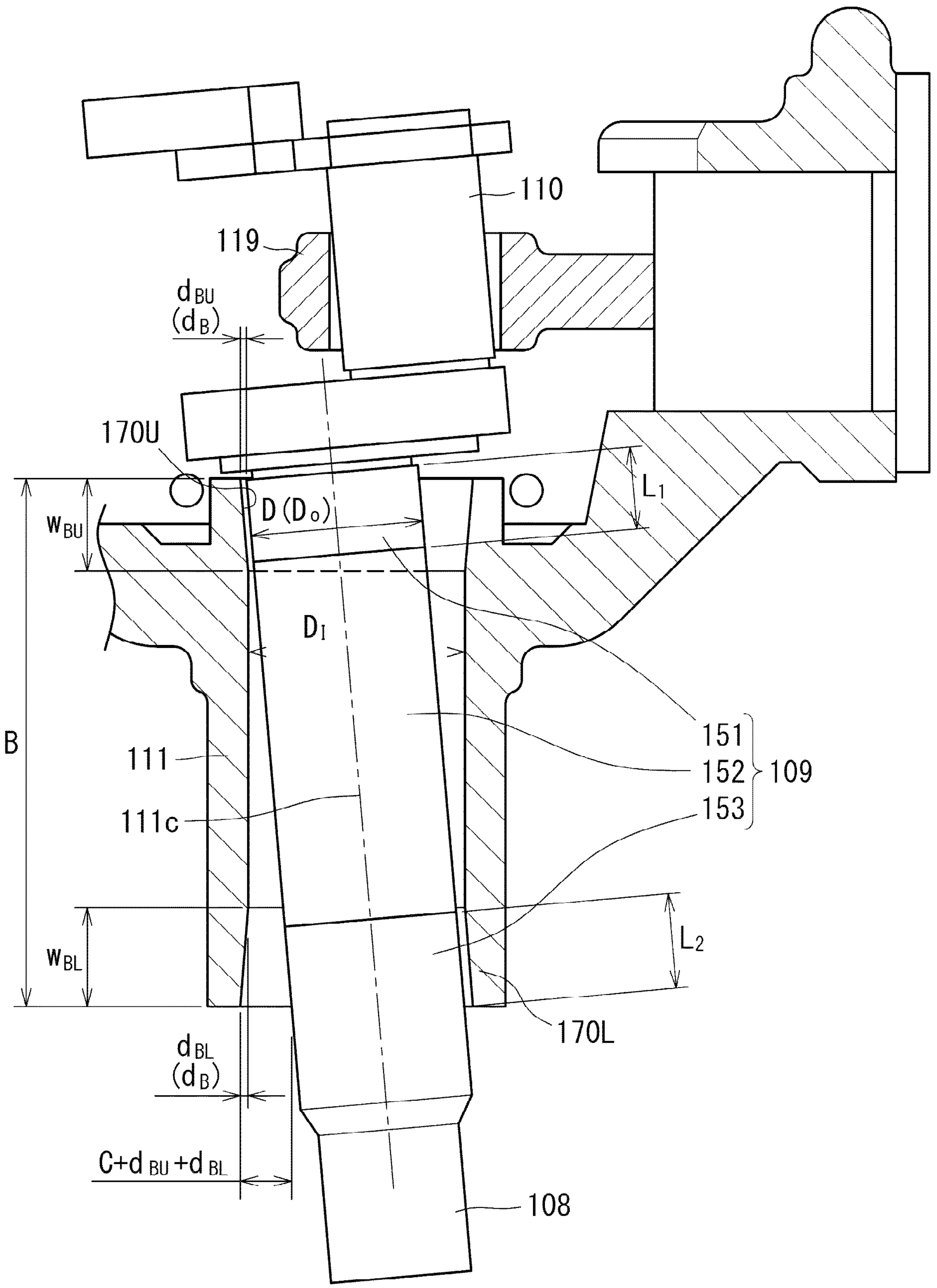

110
119
d_{BU}
(d_B)
170U
L_1
w_{BU}
$D(D_o)$
D_I
B
111
111c
151
152
153
109
w_{BL}
L_2
170L
d_{BL}
(d_B)
$C+d_{BU}+d_{BL}$
108

FIG.3

FIG.4A

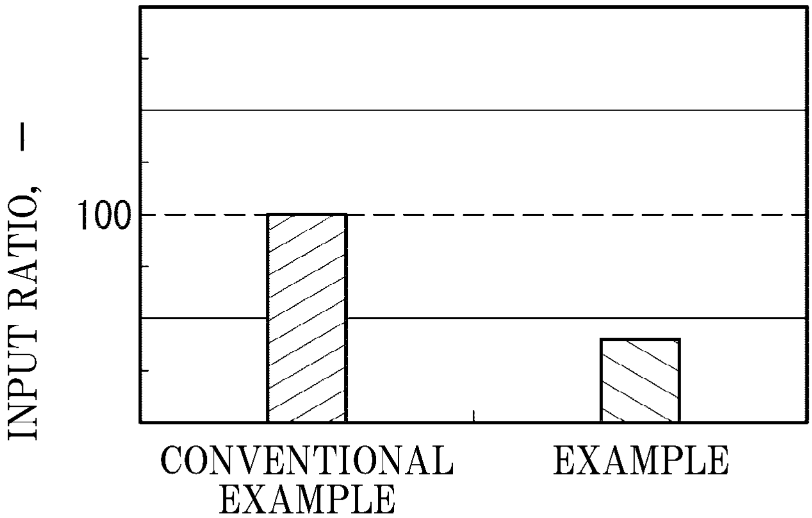
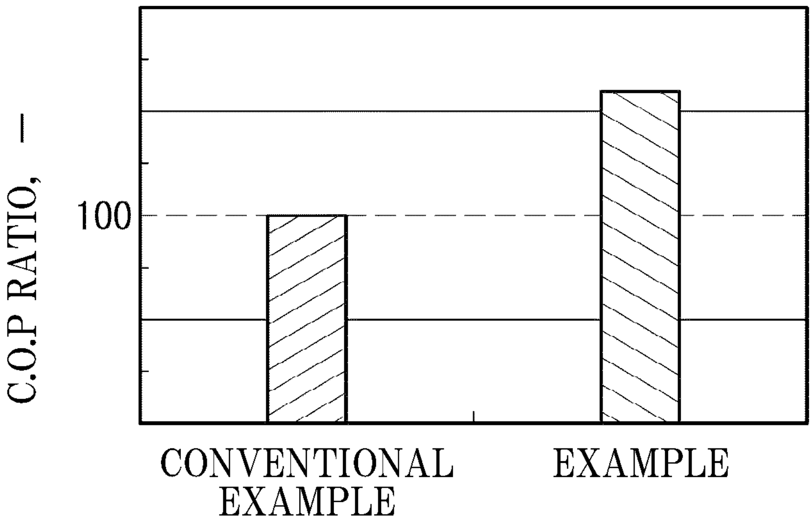

INPUT RATIO, -
100
CONVENTIONAL EXAMPLE
EXAMPLE

FIG.4B


C.O.P RATIO, -
100
CONVENTIONAL EXAMPLE
EXAMPLE

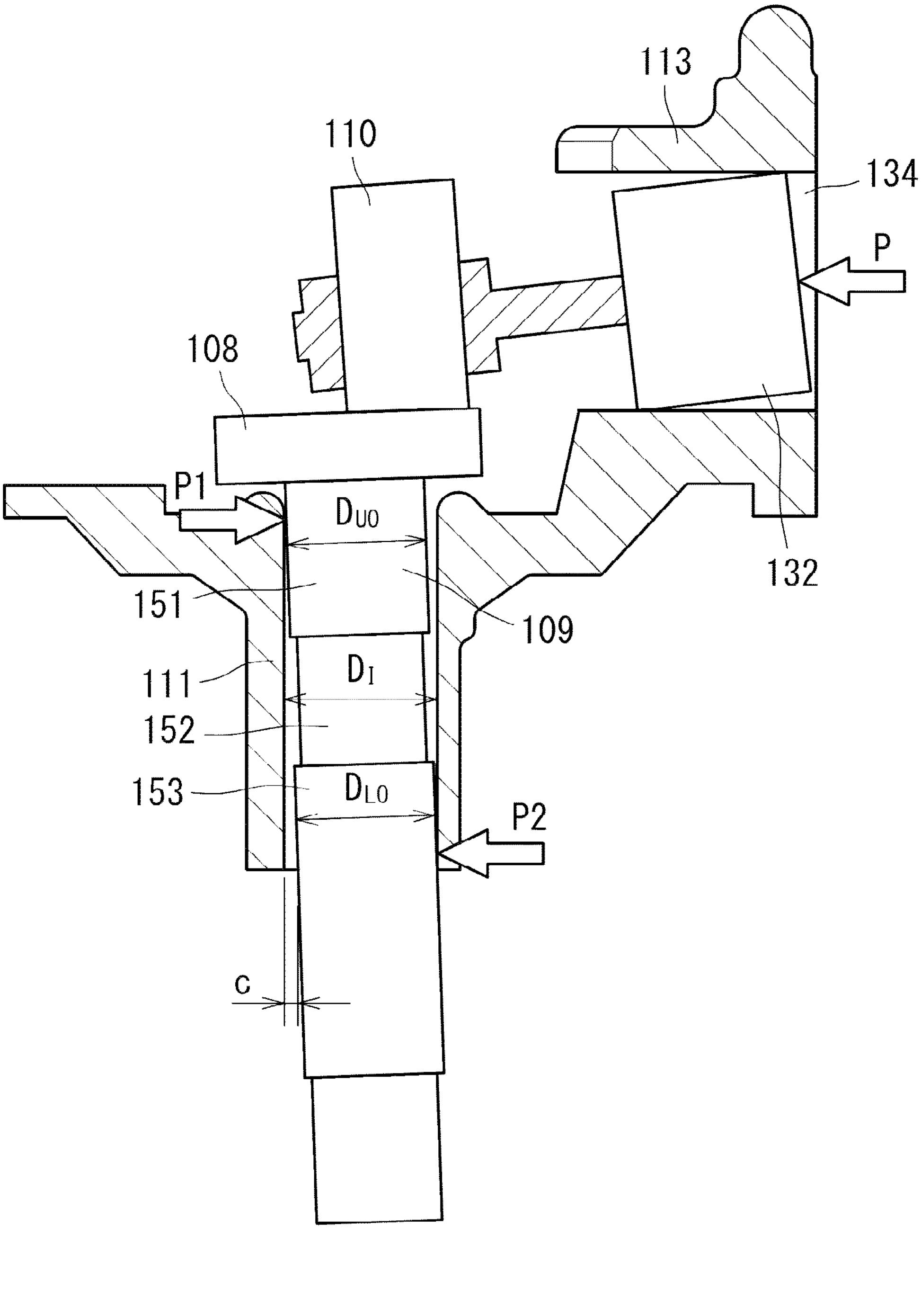


FIG.5

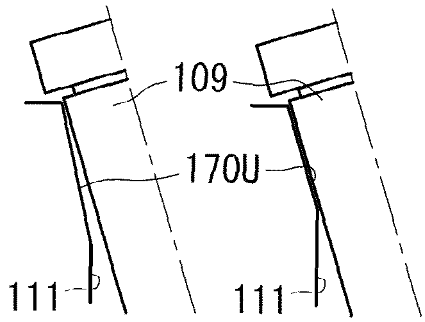
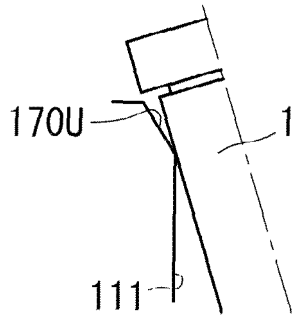
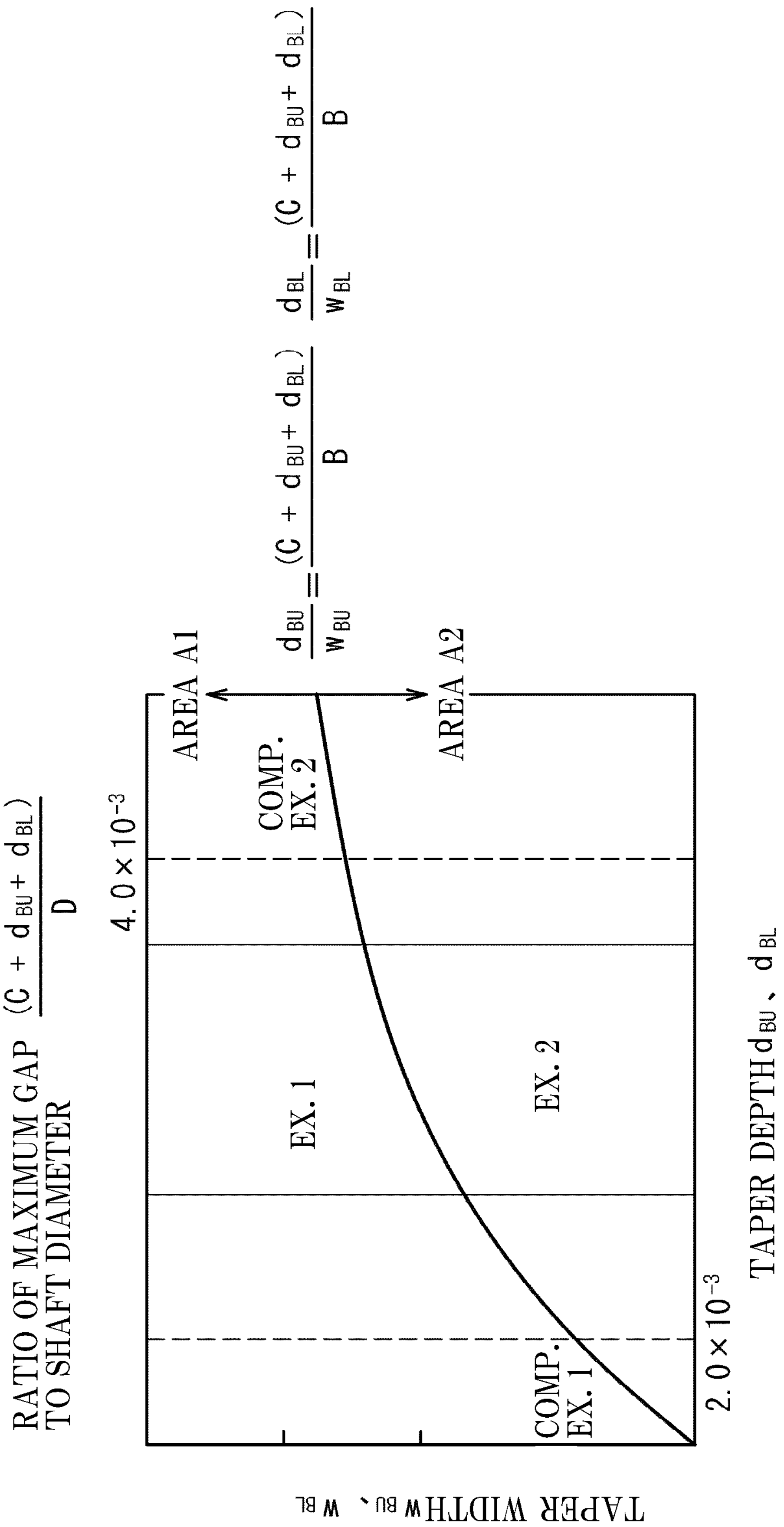
	
$\frac{d_{BU}}{w_{BU}} \leqq \frac{(C + d_{BU} + d_{BL})}{B}$ $\frac{d_{BL}}{w_{BL}} \leqq \frac{(C + d_{BU} + d_{BL})}{B}$ MATH. 1, MATH. 2	$\frac{d_{BU}}{w_{BU}} > \frac{(C + d_{BU} + d_{BL})}{B}$ $\frac{d_{BL}}{w_{BL}} > \frac{(C + d_{BU} + d_{BL})}{B}$ MATH. 3, MATH. 4
AREA A1	AREA A2

FIG.6



RATIO OF MAXIMUM GAP TO SHAFT DIAMETER $\frac{(C + d_{BU} + d_{BL})}{D}$
4.0×10^{-3}
2.0×10^{-3}
TAPER DEPTH d_{BU}、d_{BL}
TAPER WIDTH w_{BU}、w_{BL}
AREA A1
AREA A2
COMP. EX. 1
EX. 1
EX. 2
COMP. EX. 2
$\frac{d_{BU}}{w_{BU}} = \frac{(C + d_{BU} + d_{BL})}{B}$
$\frac{d_{BL}}{w_{BL}} = \frac{(C + d_{BU} + d_{BL})}{B}$

FIG.7

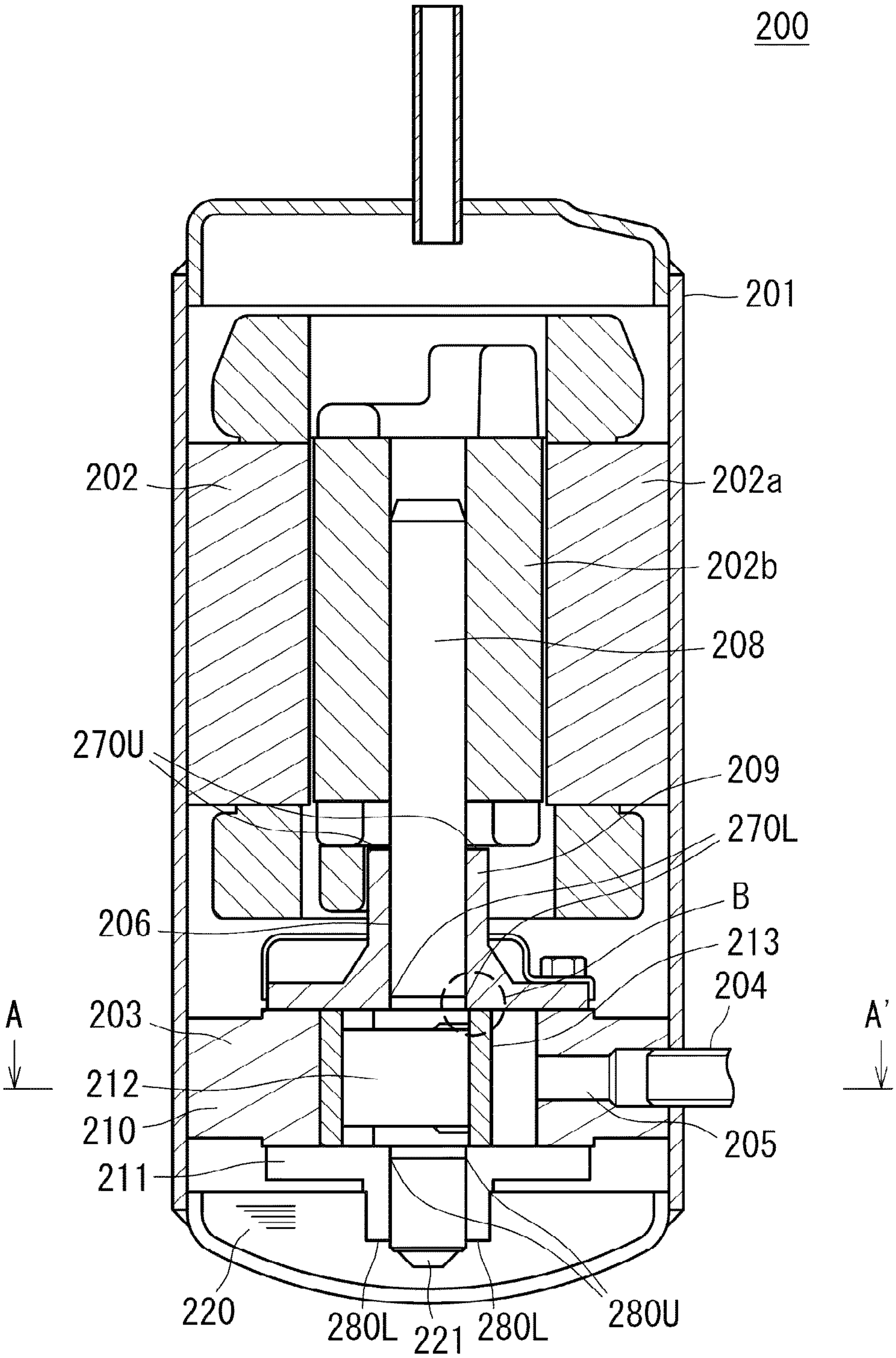

200
201
202
202a
202b
208
270U
209
270L
206
B
213
204
A
A'
203
212
210
205
211
220
280L
221
280L
280U

FIG.8

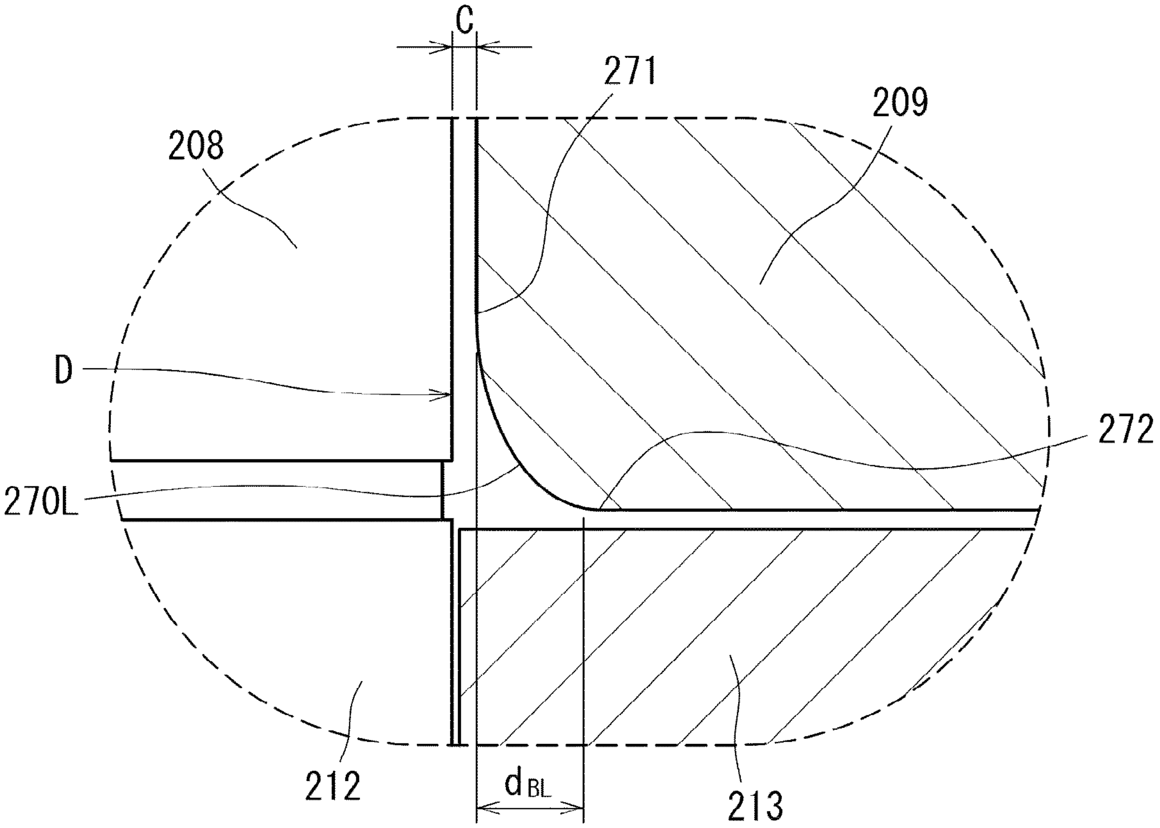


FIG.9

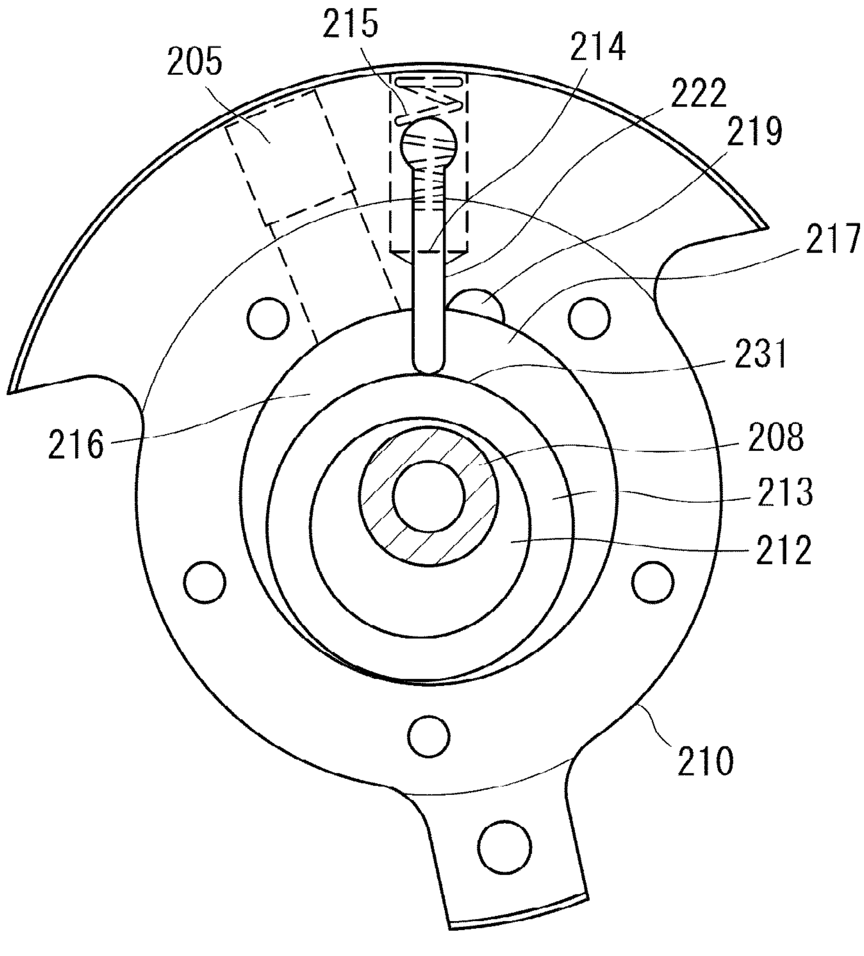


FIG.10

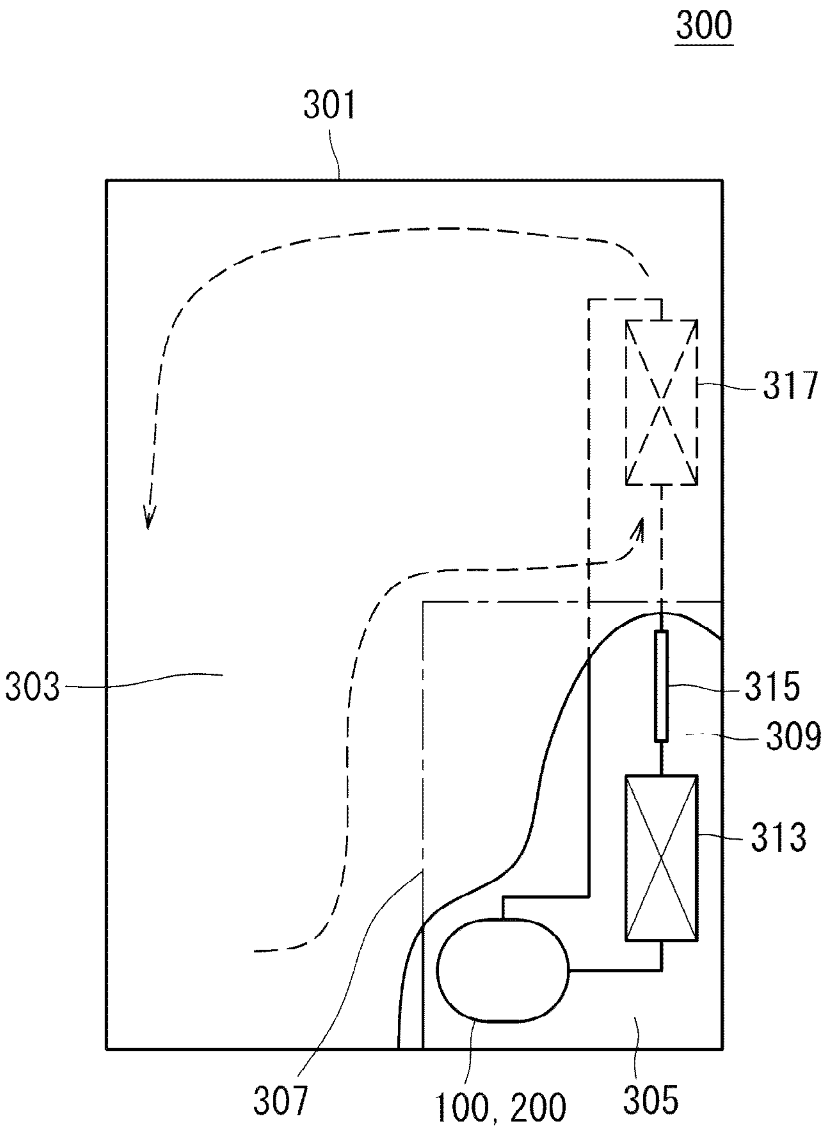


FIG.11

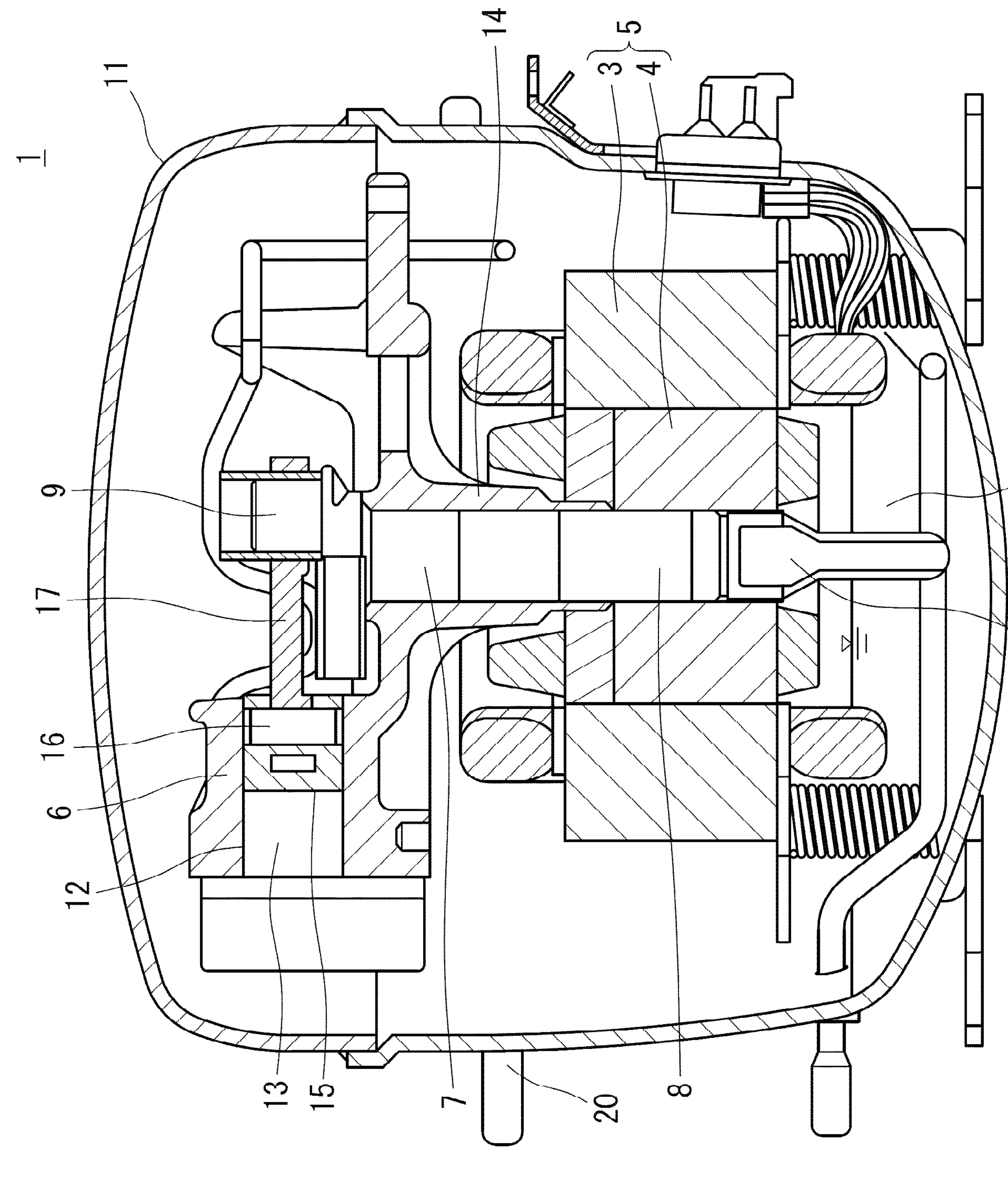


FIG.12

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2019/043312

A. CLASSIFICATION OF SUBJECT MATTER
Int.Cl. F04B39/00(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
Int.Cl. F04B39/00

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Published examined utility model applications of Japan	1922-1996
Published unexamined utility model applications of Japan	1971-2019
Registered utility model specifications of Japan	1996-2019
Published registered utility model applications of Japan	1994-2019

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	JP 2016-205134 A (HITACHI APPLIANCES, INC.) 08 December 2016, paragraphs [0018]-[0067], fig. 1-7 (Family: none)	1-5, 7-8 2-3, 6
X Y	WO 2018/092853 A1 (PANASONIC IP MANAGEMENT CO., LTD.) 24 May 2018, paragraphs [0028]-[0192], fig. 1-15 & EP 3543530 A1, paragraphs [0028]-[0193], fig. 1-15	1, 4-8 2-3, 6
X Y	JP 7-4355 A (HITACHI, LTD.) 10 January 1995, paragraphs [0015]-[0045], fig. 1-5 & CN 1090376 A	1, 4-5, 7-8 2-3, 6

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
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"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
16 December 2019 (16.12.2019)	24 December 2019 (24.12.2019)

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3, Kasumigaseki, Chiyoda-ku, Tokyo 100-8915, Japan	Telephone No.

Form PCT/ISA/210 (second sheet) (January 2015)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP H07238885 A **[0006]**

Non-patent literature cited in the description

- **ITO et al.** *Proceedings of The Japan Society of Mechanical Engineers Annual Meeting,* 2005, vol. 5-1, 143 **[0063]**