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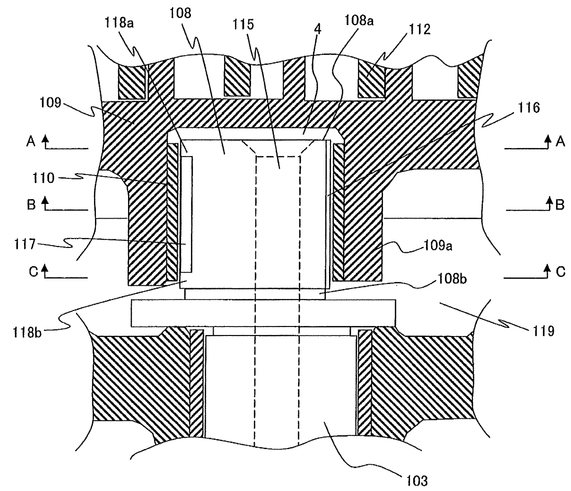
(54) SCROLL COMPRESSOR

(57) A scroll compressor includes: a fixed scroll 112; an orbiting scroll 109; a crankshaft 103 having an eccentric section 108 at an end portion; an oil supply hole 115 penetrating an inside of the crankshaft in an axial direction and having an opening portion on an end surface of the eccentric section: an orbiting sliding bearing 110 provided at the orbiting scroll and sliding by engaging with the eccentric section; and an oil supply passage 116 provided at an outer circumference of the eccentric section. The scroll compressor is structured so as to lubricate a gap between the eccentric section and the orbiting sliding bearing by lubricating oil supplied through the oil supply hole. At the outer circumference of the eccentric section, an axial loss reduction groove 117 is provided separately from the oil supply passage, and seal sections 118a, 118b are provided on at least one of the end surface side and a base end side of the eccentric section of the loss reduction groove.

With this configuration, shear resistance of an oil film by the lubricating oil, which exists between an outer circumferential surface of the eccentric section of the crankshaft and an inner circumferential surface of the orbiting sliding bearing, can be reduced, and a bearing loss during fluid lubrication can be reduced.

FIG. 3



EP 2 700 818 A1

Description

Technical Field

[0001] The present invention relates to a scroll compressor used for a refrigeration and air conditioning system, and more particularly to a scroll compressor in which an orbiting scroll includes an orbiting sliding bearing which slides by engaging with an eccentric section of a crankshaft.

Background Art

[0002] A scroll compressor is a compressor which carries out compression of a gas, such as a refrigerant, by causing two scroll members having a volute teeth-like configuration to orbit relatively. In general, the scroll compressor is structured in such a manner that one movable orbiting scroll carries out orbital motion around a fixed scroll restrained with screws, by welding, or the like. The orbiting scroll is provided with an orbiting sliding bearing which slides by engaging with an eccentric section of a crankshaft. A mechanism in which, while the eccentric section of the crankshaft and the orbiting sliding bearing slide through lubricating oil, whirling rotary motion of the crankshaft is conveyed to the orbiting scroll so as to cause the orbiting scroll to orbit has been widely employed.

[0003] In recent years, in order to reduce an amount of energy consumption of the scroll compressor and to reduce a load of an electric motor, reduction of a bearing loss caused by sliding of the shaft and the bearing has become an issue.

[0004] A conventional art configured to reduce this bearing loss is described in JP 2003-239876 A (PTL 1). In PTL 1, "A floating ring member is held at an insertion groove of a hub formed at a bottom of an orbiting scroll so as to freely rotate and idle, and a slide bush fixed to an eccentric section of a rotary shaft is inserted into a center of the floating ring member, thereby constituting a friction loss reducing device of a scroll compressor."

[0005] Generally, it is known that, in a sliding section, such as a bearing, in which two surfaces slide through lubricating oil, a friction loss increases with the increase in a sliding speed. The scroll compressor described in PTL 1 has a structure in which the rotatable slide bush is inserted into a space between the eccentric section of the rotary shaft and the hub (orbiting boss section) filled with the lubricating oil. With this configuration, sliding caused between the rotary shaft and the hub can be divided into sliding between an outer circumference of the rotary shaft and an inner circumference of the slide bush and sliding between an outer circumference of the slide bush and an inner circumference of the hub. Moreover, a relative sliding speed at each sliding section is reduced, and in particular, a friction loss of a bearing during high speed operation is reduced.

[0006] Another conventional art is described in JP 11-159484 A (PTL 2). In PTL 2, "A D-cut is formed on an outer circumferential surface of an eccentric pin section within a range of 30° or more and 120° or less in a direction opposite to a rotation direction of a crankshaft from an eccentricity direction of the eccentric pin section relative to the crankshaft." Further, PTL 2 describes the D-cut as "an oil supply notch provided in a comparatively large area of a gap between an eccentric shaft section and a bearing section."

[0007] By providing the oil supply notch (D-cut) in the comparatively large area of the gap between the eccentric shaft section and the bearing section, stable oil supply is promoted even during start-up, low speed operation, and transient operation. In this way, absence of an oil film is prevented and fluid lubrication is secured. As a result, a direct contact between an outer circumference of the crankshaft and an inner circumference of the bearing is prevented, and the increase in friction loss can be prevented.

Citation List

Patent Literature

[0008]

PTL 1: JP 2003-239876 A
PTL 2: JP 11-159484 A

Summary of Invention

Technical Problem

[0009] However, in the scroll compressor described in PLT 1, there are new problems such that the structure becomes complicated because of the increase in number of parts and that high management accuracy is required in dimensions of a bearing gap because of the increase in sliding places. Additionally, there are problems such that an oil film is difficult to be formed due to the decrease in sliding speed during low speed operation and that a direct contact with the slide bush easily occurs.

[0010] Further, the scroll compressor described in PTL 2 is effective in preventing the oil film absence by promoting the inflow of the lubricating oil from an outside to the bearing gap, and preventing the direct contact between the shaft and the bearing. However, since the notch is provided in an area where the gap between the eccentric shaft section and the bearing section is large, reduction of shear resistance of the fluid lubricated oil film cannot be expected or a reduction effect thereof is small. Therefore, once the fluid lubricated oil film is formed and a condition without a direct contact portion is formed, the further loss reduction is not made or limited.

[0011] An object of the present invention is to reduce a bearing loss during fluid lubrication by reducing shear resistance of an oil film by lubricating oil, which exists between an outer circumferential surface of an eccentric section of the crankshaft and an inner circumferential surface of the orbiting sliding bearing.

Solution to Problem

[0012] To achieve the above object, the present invention provides a scroll compressor, including: a fixed scroll; an orbiting scroll engaging with the fixed scroll; a crankshaft having an eccentric section at an end portion for causing the orbiting scroll to carry out orbital motion; an oil supply hole penetrating an inside of the crankshaft in an axial direction and having an opening portion on an end surface of the eccentric section: an orbiting sliding bearing provided at the orbiting scroll and sliding by engaging with the eccentric section of the crankshaft; and an oil supply passage provided at an outer circumference of the eccentric section of the crankshaft so as to communicate with an end surface side and a base end side of the eccentric section, the scroll compressor is structured so as to lubricate a gap between the eccentric section and the orbiting sliding bearing by lubricating oil supplied through the oil supply hole, further including: an axial loss reduction groove provided at the outer circumference of the eccentric section of the crankshaft separately from the oil supply passage; and a seal section which is provided on at least one of the end surface side and the base end side of the eccentric section of the loss reduction groove.

Advantageous Effects of Invention

[0013] According to the present invention, there are effects such that the shear resistance of the oil film by the lubricating oil, which exists between the outer circumferential surface of the eccentric section of the crankshaft and the inner circumferential surface of the orbiting sliding bearing, can be reduced and that the bearing loss during fluid lubrication can be reduced.

Brief Description of Drawings

[0014]

[FIG. 1] FIG. 1 is a longitudinal cross-sectional view illustrating a scroll compressor according to Embodiment 1 of the present invention.
[FIG. 2] FIG. 2 is an enlarged perspective view of a vicinity of an eccentric section illustrated in FIG. 1.
[FIG. 3] FIG.. 3 is an enlarged cross-sectional view of the vicinity of the eccentric section illustrated in FIG. 1.
[FIG. 4] FIG. 4 is a cross-sectional view taken along the line A-A in FIG 3.
[FIG. 5] FIG. 5 is a cross-sectional view taken along the line B-B in FIG 3.
[FIG. 6] FIG. 6 is a cross-sectional view taken along the line C-C in FIG. 3.
[FIG. 7] FIG. 7 is a diagram explaining a shaft rotation direction, an angular position, and a bearing load direction in the cross section taken along the line B-B in FIG. 3.
[FIG. 8] FIG. 8 is a diagram explaining a start position of a loss reduction groove in the present invention; (a) is a graph explaining a relationship between the start position of the loss reduction groove and a relative bearing loss; and (b) is a graph explaining a relationship between the start position of the loss reduction groove and a relative minimum oil film thickness.
[FIG. 9] FIG. 9 is a graph explaining a relationship between a depth of the loss reduction groove and a relative bearing loss in the present invention.
[FIG. 10] FIG. 10 is a graph explaining a relationship between a circumferential angular width of the loss reduction groove and a relative bearing loss in the present invention.
[FIG. 11] FIG. 11 is a graph explaining a relationship between a rotation speed of a shaft and a relative bearing loss in the present invention.
[FIG. 12] FIG. 12 is an enlarged perspective view of a vicinity of an eccentric section of a loss reduction groove provided at an outer circumference of a crankshaft according to another example.
[FIG. 13] FIG. 13 is an enlarged perspective view of a vicinity of an eccentric section of a loss reduction groove provided at an outer circumference of a crankshaft according to still another example.
[FIG. 14] FIG. 14 is an enlarged perspective view of a vicinity of an eccentric section of a loss reduction groove provided at an outer circumference of a crankshaft according to yet still another example.

Description of Embodiments

[0015] A scroll compressor according to a concrete embodiment of the present invention is described below using the drawings. In each drawing, the portions having the same reference numerals designate the same or corresponding portions.

Embodiment 1

[0016] FIG. 1 is a longitudinal cross-sectional view illustrating a scroll compressor according to Embodiment 1 of the present invention.

[0017] A scroll compressor 1 illustrated in FIG. 1 is a closed scroll compressor used for refrigeration and air conditioning, such as an air conditioning system, e.g., an air conditioner, and a refrigeration system. A fixed scroll 112 and an orbiting scroll 109, which engages with this fixed scroll 112 and carries out orbital motion, are provided at an upper portion of an inside of a closed container 2. Further, an electric motor 102 is provided inside the closed container 2, and a crankshaft 103 is connected to this electric motor 102. This crankshaft 103 is rotatably supported by a main bearing 105 provided at a frame

104, which is fixedly provided inside the closed container 2, and an auxiliary bearing 107 provided at a lower frame 106.

[0018] An eccentric section 108 is provided at the upper portion of the crankshaft 103. This eccentric section 108 slides by engaging with an orbiting sliding bearing 110 provided on a lower surface of an end plate of the orbiting scroll 109, and whirling rotary motion (eccentric motion) of the eccentric section 108 is conveyed to the orbiting scroll 109. Rotation of this orbiting scroll 109 is limited by an Oldham ring 111, and the orbiting scroll 109 carries out orbital motion relative to the fixed scroll 112. In this way, low-pressure refrigerant gas is sucked in through a suction port 113, is compressed, and thereafter, is discharged to outside via a discharge port 114.

[0019] It should be noted that an oil supply hole 115, which penetrates from a lower end of the crankshaft 103 to an end surface (upper end surface) side of the eccentric section 108, is provided inside the crankshaft 103. Lubricating oil 3 stored at a lower portion of the closed container is pushed up through the oil supply hole 115 by a pressure difference or by a pump separately mounted to a lower end portion of the crankshaft, and is supplied to sliding sections, such as the respective bearing sections (the main bearing 105, the auxiliary bearing 107, and the orbiting sliding bearing 110). In the present embodiment, the inside of the closed container 2 is at discharge pressure, and an intermediate chamber (a back pressure chamber) 119, which is on a back surface of the end plate of the orbiting scroll 109, is at intermediate pressure between discharge pressure and suction pressure. For this reason, it is structured that the lubricating oil stored at the lower portion of the closed container is supplied to the respective bearing sections through the oil supply hole 115 by the pressure difference between the discharge pressure and the intermediate pressure.

[0020] FIG. 2 is an enlarged perspective view of a vicinity of the eccentric section 108 illustrated in FIG. 1. The oil supply hole 115 opens on the upper end surface of the eccentric section 108. The eccentric section 108 is provided with an axial oil supply passage 116 in such a manner that an upper end (end surface) 108a side communicates with a lower end (base end) 108b side. Further, besides the oil supply passage 116, a loss reduction groove 117 engraved on an outer circumferential surface of the eccentric section 108 is formed in the axial direction. Seal sections 118a, 118b, which are not engraved on the outer circumferential surface of the eccentric section 108, are respectively provided on the upper end 108a side and the lower end 108b side of this loss reduction groove 117.

[0021] FIG. 3 is an enlarged cross-sectional view of the vicinity of the eccentric section 108 illustrated in FIG. 1. The orbiting sliding bearing 110 is provided inside an orbiting boss section 109a of the orbiting scroll 109. The eccentric section 108 of the crankshaft 103 is inserted into this orbiting sliding bearing 110 and engaged therewith, thereby sliding this eccentric section 108 and the orbiting sliding bearing 110. A space surrounded by the upper end (end surface) 108a of the eccentric section 108 and the orbiting scroll 109 communicates with the oil supply hole 115. Since the space is located upstream of the oil supply passage, the pressure of the lubricating oil supplied thereto is substantially discharge pressure.

[0022] In contrast to this, the lower end (base end) 108b side of the eccentric section 108 communicates with the intermediate chamber 119 having pressure lower than that of the upper end 108 side. The lubricating oil supplied through the oil supply hole 115 fills a space (orbiting boss section inner space) 4 surrounded by the orbiting scroll 109, the eccentric section 108, and the orbiting sliding bearing 110, and thereafter, is discharged to the lower intermediate chamber 119 through the oil supply passage 116 and the like.

[0023] The oil supply passage 116 is formed by an engraved groove engraved into the outer circumference of the eccentric section 108 or a notch. This oil supply passage 116 enlarges a gap between the eccentric section 108 and the orbiting sliding bearing 110, and communicates with both the upper end 108a side and the lower end 108b side of the eccentric section 108 across the orbiting sliding bearing 110 in the axial direction. Accordingly, flow path resistance of the lubricating oil in the orbiting boss section inner space 4 flowing to the intermediate chamber 119 through the oil supply passage 116 is smaller than flow path resistance of the lubricating oil flowing in a portion of the outer circumferential surface of the eccentric section 108 other than the oil supply passage 116.

[0024] On the other hand, the loss reduction groove 117 is also formed by an engraved groove engraved into the outer circumference of the eccentric section 108 or a notch. A seal section 118 having the same diameter as the outer circumferential surface of the eccentric section 108 is formed on the upper end 108a side and the lower end 108b side of this loss reduction groove 117, and an axial length of the loss reduction groove 117 is shorter than that of the orbiting sliding bearing 110. Accordingly, the loss reduction groove 117 is structured in such a manner that the loss reduction groove 117 does not cross the orbiting sliding bearing 110 in the axial direction and does not simultaneously open on the upper end 108a side and the lower end 108b side of the eccentric section 108.

[0025] Further, at the portions of the seal sections 118a, 118b, the gap between the outer circumference of the eccentric section 108 and the inner circumference of the orbiting sliding bearing 110 is the same as that of a portion other than the oil supply passage 116 at the outer circumference of the eccentric section 108. In other words, flow path resistance from the upper end 108a to the lower end 108b of the eccentric section 108 through the loss reduction groove 117 is substantially the same as the flow path resistance flowing in the portion of the outer circumferential surface of the eccentric section 108 other than the oil supply passage 116. As a result, the lubricating oil supplied to the orbiting boss section inner space 4 through the oil supply hole 115 preferentially passes through the oil supply passage 116 and easily flows to the intermediate chamber 119 side. Moreover, in the entire portion of the orbiting sliding bearing 110, the flow path resistance of the lubricating oil flowing in the axial direction is substantially the same as that of a case where the loss reduction groove 117 is not provided. Accordingly, even if the loss reduction groove 117 is provided, the increase in an amount of oil supply is prevented.

[0026] FIG. 4 is a cross-sectional view taken along the line A-A in FIG 3, FIG. 5 is a cross-sectional view taken along the line B-B in FIG 3, and FIG. 6 is a cross-sectional view taken along the line C-C in FIG. 3. Since a diameter of the outer circumferential surface of the eccentric section 108 is smaller than a diameter of the inner circumferential surface of the orbiting sliding bearing 110, there is a gap between the eccentric section 108 and the orbiting sliding bearing 110, and this gap is filled with the lubricating oil.

[0027] As illustrated in FIG. 4, the oil supply passage 116 opens at a portion of the upper end 108a of the eccentric section 108, and a gap between this oil supply passage 116 and the orbiting sliding bearing 110 is particularly wider than a gap between a portion of the eccentric section 108, at which the oil supply passage 116 is not provided, and the orbiting sliding bearing 110.

[0028] Further, in an axial intermediate region of the eccentric section 108, as illustrated in FIG. 5, a gap between the oil supply passage 116 of the eccentric section 108 and the orbiting sliding bearing 110 and a gap between the loss reduction groove 117 of the eccentric section 108 and the orbiting sliding bearing 110 are particularly wider than a gap between the other portion of the eccentric section 108 and the orbiting sliding bearing 110.

[0029] Furthermore, in a vicinity of an axial lower end (base end) of the eccentric section 108, as illustrated in FIG. 6, the loss reduction groove 117 does not exist, and a gap between the oil supply passage 116 of the eccentric section 108 and the orbiting sliding bearing 110 is particularly wider than a gap between the portion of the eccentric section 108, at which the oil supply passage 116 is not provided, and the orbiting sliding bearing 110.

[0030] FIG. 7 is a diagram explaining a shaft rotation direction, an angular position, and a bearing load direction in the cross section taken along the line B-B in FIG. 3. FIG. 7 illustrates positions of the oil supply passage 116, the loss reduction groove 117, a rotation direction 120 of the crankshaft 103, and a bearing load direction 121 in which the orbiting sliding bearing 110 is pressed against the eccentric section 108. A description is given in more detail. First, angular positions of various regions are described using a coordinate system, in which a center of the eccentric section 108 is set as a reference and an opposite side of an eccentricity direction of the eccentric section 108 is set to 0°.

[0031] As illustrated in the shaft rotation direction 120, when the crankshaft 103 carries out rotation motion in a clockwise direction in the drawing, a bearing load is generated in the bearing load direction 121 as resultant force of reaction force, in which the orbiting scroll 109 compresses gas, and centrifugal force, in which the orbiting scroll is swung around in the eccentricity direction. At this time, the gap between the eccentric section 108 and the orbiting sliding bearing 110 is not uniform in the circumferential direction and biased, and is smallest at a minimum gap portion 122, which is shifted from the bearing load direction 121 in a direction opposite to the rotation direction.

[0032] Evaluation of a bearing loss caused by oil film shear were made on a case where the loss reduction groove 117 is provided at the outer circumference of the shaft, and this shaft is slid against a cylindrical sliding bearing through the lubricating oil. Then, effects of reducing the bearing loss by the loss reduction groove were verified. The results of verification are illustrated in FIGS. 8 to 11. Further, a position, a depth, and a width of the loss reduction groove 117, which can effectively reduce the bearing loss, were considered from these results. It should be noted that in this verification, verification is carried out on the assumption that the scroll compressor is used for an air conditioner and a shaft diameter of the eccentric section is 14 to 18 mm.

[0033] Description is given below in detail using FIGS. 8 to 11.

[0034] FIG. 8 is a diagram explaining a start position of the loss reduction groove 117 in the present invention; (a) is a graph explaining a relationship between the start position of the loss reduction groove 117 and a relative bearing loss; and (b) is a graph explaining a relationship between the start position of the loss reduction groove and a relative minimum oil film thickness.

[0035] FIG. 8(a) illustrates the bearing loss caused by oil film shear in a case where the loss reduction groove 117 is formed in an engraved groove by engraving the outer circumference of the shaft and is slid against the cylindrical sliding bearing through the lubricating oil. Evaluations were made by variously changing a circumferential start position of the loss reduction groove 117 formed at the outer circumference of the shaft. The circumferential start position of the loss reduction groove 117 is expressed on the abscissa, and where a bearing loss of a case in which a shaft without the loss reduction groove 117 was used is set to 100%, the relative bearing loss relative to this case is expressed on the ordinate. Further, the loss reduction groove 117 was an engraved groove with a depth of 0.1 mm and with an angular range (angular width) of 30 degrees in the circumferential direction.

[0036] As a result of verification, as illustrated in FIG. 8(a), the relative bearing loss shows a tendency to decrease particularly when a start angle of the loss reduction groove is within a range of 140 degrees to 210 degrees. Accordingly, the start position of the loss reduction groove 117 is preferably set within the range of 140° to

210°, and this range can reduce the bearing loss by at least 2% or more. Moreover, when the start position is set within the range of 145° to 180°, the largest reduction effect is obtained. It should be noted that, if the loss reduction groove 117 is provided in such a manner that at least a part thereof exists within the range of 140° to 210°, the bearing loss reduction effect can be reduced more than conventional examples.

[0037] FIG. 8(b) is the graph explaining the relationship between the start position of the loss reduction groove and the relative minimum oil film thickness. The circumferential start position of the loss reduction groove 117 is expressed on the abscissa, and where a minimum oil film thickness of a case in which a shaft without the loss reduction groove 117 was used is set to 100%, the relative minimum oil film thickness relative to this case is expressed on the ordinate. As illustrated in this drawing, when the start position of the loss reduction groove 117 is 140 degrees or less, the start position of the loss reduction groove 117 is in the vicinity of an angle having the minimum oil film thickness. Accordingly, when the bearing reduction groove 117 is provided within an angular range of less than 140 degrees, bearing behavior becomes unstable, and wear due to the contact between the shaft and the bearing is easily developed. Therefore, it is preferable that the start position of the loss reduction groove 117 be determined within the angular range of at least 140 degrees or more. It should be noted that, even if a part of the loss reduction groove 117 is located at a portion of 210 degrees or more, the minimum oil film thickness is sufficiently large. Accordingly, if the start position of the loss reduction groove 117 is within a range of 140° to 210°, an end position thereof may be at a position of 210 degrees or more.

[0038] FIG. 9 is a graph explaining a relationship between a depth of the loss reduction groove 117 and a relative bearing loss in the present invention. FIG 9 illustrates a bearing loss caused by oil film shear in a case where the loss reduction groove 117 is formed in an engraved groove by engraving the outer circumference of the shaft and is slid against the cylindrical sliding bearing through the lubricating oil. Evaluations were made by variously changing a depth of the loss reduction groove 117 formed at the outer circumference of the shaft (radial depth from an unprocessed outer circumferential circle of the eccentric section). The depth (engraved depth) of the loss reduction groove 117 is expressed on the abscissa, and where a bearing loss of a case in which a shaft without the loss reduction groove 117 was used is set to 100%, the relative bearing loss relative to this case is expressed on the ordinate. Further, the loss reduction groove 117 was formed within an angular range (angular width) of 30 degrees in the circumferential direction, and a start angle of this loss reduction groove was set to 150 degrees.

[0039] As a result of verification, as illustrated in FIG. 9, the relative bearing loss can be reduced by at least 2% or more by having the engraved depth of the loss reduction groove of 0.002 mm or more. Further, if the depth of the loss reduction groove is 0.01 mm or more, the bearing loss reduction effect of at least 5% or more is obtained. If the depth of the loss reduction groove is 0.05 mm or more, the largest bearing loss reduction effect is obtained. It should be noted that, if the depth of the loss reduction groove 117 is too large, stiffness of the shaft is decreased. Accordingly, it is preferable that the depth of the loss reduction groove 117 be set to at most 20% or less of a shaft diameter (shaft diameter of the eccentric section of the crankshaft). Therefore, in general, it is preferable that the depth of the loss reduction groove be approximately 0.05 to 0.5 mm.

[0040] FIG. 10 is a graph explaining a relationship between a circumferential angular width of the loss reduction groove and a relative bearing loss in the present invention. FIG 10 illustrates a bearing loss caused by oil film shear in a case where the loss reduction groove 117 is formed in an engraved groove by engraving the outer circumference of the shaft and is slid against the cylindrical sliding bearing through the lubricating oil. Evaluations were made by variously changing a circumferential angular width of the loss reduction groove 117 formed at the outer circumference of the shaft. The circumferential angular width of the loss reduction groove 117 is expressed on the abscissa, and where a bearing loss of a case in which a shaft without the loss reduction groove 117 was used is set to 100%, the relative bearing loss relative to this case is expressed on the ordinate. Further, a depth of the loss reduction groove 117 was set to 0.1 mm, and a start angle of the groove was set to 150 degrees.

[0041] As a result of verification, as illustrated in FIG. 10, regarding the relative bearing loss, by having the circumferential angular width of the loss reduction groove 117 of 10 degrees or more, i.e., in this example, the angular width of 10 degrees or more in the circumferential direction from the position of the start angle of 150 degrees, the relative bearing loss can be reduced by at least 2% or more. Further, if the circumferential angular width of the loss reduction groove is gradually increased by 60 degrees from the start angle of 150°, the relative bearing loss is reduced in response to the increase in the circumferential angular width. Even if the angular width is larger than 60 degrees, the relative bearing loss shows little tendency to decrease. Accordingly, considering processability and the like of the bearing loss groove 117, the angular width is preferably set within a range of 20° to 60°.

[0042] According to the above-description, when the loss reduction groove 117 is formed at the outer circumference of the eccentric section 108 of the crankshaft 103, the bearing loss can be reduced by the engraved grooved or the notch, by providing the portion with the engraved depth of 0.002 mm or more and at the angle within the range of 140 degrees to 210 degrees in the coordinate system illustrated in FIG. 7. In particular, when the start position of the loss reduction groove 117 is set to the position in the vicinity of the angle of 150 degrees in the coordinate system illustrated in FIG. 7, and the angular width of the groove is set to 20° to 60°, and the depth of the groove is 0.01 mm, it was verified that a large bearing loss reduction effect was obtained.

[0043] FIG. 11 is a graph explaining a relationship between a rotation speed of a shaft and a relative bearing loss in the present invention. In other words, FIG 10 illustrates a bearing loss caused by oil film shear in a case where the shaft provided with the loss reduction groove 117 at the outer circumference and the shaft without the loss reduction groove are used and slid against the cylindrical bearing through the lubricating oil. Evaluations were made by variously changing a rotation speed of the shaft. In FIG. 11, the rotation speed is expressed on the abscissa, and where a bearing loss of a case in which the shaft without the loss reduction groove was used and rotated at a rate of 6000 rev/min is set to 100%, the relative bearing loss relative to this case is expressed on the ordinate. Further, the loss reduction groove 117 was an engraved groove with a depth of 0.1 mm. The loss reduction groove 117 was formed at an angular range (angular width) of 30 degrees in the circumferential direction, and a start angle of this loss reduction groove was set to 150 degrees.

[0044] As a result of verification, as illustrated in FIG. 11, by providing the loss reduction groove 117, reduction of the bearing loss was able to be verified in each number of revolutions. Further, it is found that, though the bearing loss increases with the increase in the rotation speed, the bearing loss reduction effect of the present invention, at which the loss reduction groove was provided, relatively increases with the increase in the rotation speed, compared to that of the conventional one, at which the loss reduction groove is not provided.

[0045] FIGS. 12 to 14 are respectively enlarged perspective views of vicinities of eccentric sections of loss reduction grooves 117 provided at outer circumferences of crankshafts according to other examples.

[0046] As is apparent from FIG. 9, it is preferable that the loss reduction groove 117 be an engraved groove engraved at a depth of 0.05 mm or more from the unprocessed outer circumferential surface of the shaft. However, as illustrated in FIG. 12, the loss reduction groove 117 may be a notch shape. The bearing loss reduction effect which is substantially the same as that of the engraved groove can be obtained.

[0047] In the case of such notch shape, a depth of the shaft from the unprocessed outer circumferential surface thereof to the shaft center direction continuously increases or decreases from 0 mm depending on a circumferential angular position. Accordingly, in order to secure a depth of 0.05 mm or more, in which the bearing loss reduction effect becomes particularly large, the loss reduction groove 117 needs to have a larger circumferential angular width. However, the case of forming the loss reduction groove 117 as the notch shape illustrated in FIG. 12 has a greater effect of reducing a manufacturing cost than the case of forming the loss reduction groove 117 as the engraved groove illustrated in FIG. 2.

[0048] Further, in the example of FIG. 12, a seal section 118a is provided on an end surface 108a side of the eccentric section of the loss reduction groove 117, and a seal section 118b is provided on a base end 108b side thereof, respectively. However, this seal section 118 may be provided at at least one of the seal sections 118a and 118b. The example illustrated in FIG. 13 has a structure in which a seal section 118 is provided only on a base end 108b side (intermediate chamber side) of the loss reduction groove 117. In this case, lubricating oil flowed from an oil supply hole 115, which opens at an upper end of an eccentric section 108, is easily flown into the loss reduction groove 117, and an amount of oil supply slightly increases. However, in the case of the example in FIG. 13, even if the circumferential width of the loss reduction groove 117 is the same, an area of the loss reduction groove 117 can be increased. Accordingly, there is an advantage of improving a bearing loss reduction effect. Further, if the seal section 118 is provided adjacent to the intermediate chamber side (base end side), the extreme increase in the amount of oil supply can be prevented.

[0049] Further, as illustrated in FIG. 14, a plurality of loss reduction grooves 117 may be formed in the circumferential direction of the eccentric section 108. In other words, if a start position of the loss reduction groove 117 is set to a position of 140 degrees in a direction opposite to a rotation direction of a crankshaft from a direction opposite to an eccentricity direction of the eccentric section 108, and an end position thereof is set to a position of 210 degrees, a circumferential angular width of the loss reduction groove 117 is 70 degrees. In a case of the groove with a large angular width as such, as illustrated in FIG. 14, a portion without a notch is provided, on the way, on an outer circumferential surface of the eccentric shaft with a width of, for example, approximately 20 degrees in the circumferential direction. The loss reduction groove 117 is separately provided at the angular position in such a manner that a loss reduction groove 117a is within a range of 140° to 165° and a loss reduction groove 117b is within a range of 185° to 210°. A portion without the notch is left within a range of 165° to 185°, which is an intermediate portion of the loss reduction groove 117.

[0050] It is difficult to form an oil film pressure supporting a bearing load at the portion of the loss reduction groove 117. However, as illustrated in FIG. 14, the plurality of loss reduction grooves is separately provided in the circumferential direction and, therebetween, the outer circumferential surface of the eccentric shaft is left within the certain circumferential angular range without forming the engraved groove or the notch. By so doing, an ability of producing a certain amount of oil film pressure can be secured at the portion left of the outer circumferential surface of this eccentric shaft. With this configuration, if an earthquake or a big vibration occurs and an unexpected dynamic load is applied to a portion of the loss reduction groove, a collision between an eccentric shaft 108 and an orbiting sliding bearing 110 can be prevented, and occurrence of galling, seizure, or the like can be prevented.

[0051] It should be noted that, in the example illustrated in FIG. 14, the circumferential start positions of the both loss reduction grooves 117a, 117b, which are formed more than one, are preferably provided at the positions of 140° to 210° around the center of the eccentric section in the direction opposite to the rotation direction of the crankshaft from the direction opposite to the eccentricity direction of the eccentric section. However, by providing the circumferential start position of at least one of the plurality of loss reduction grooves at the positions of 140° to 210°, the bearing loss reduction effect can be obtained.

[0052] According to the embodiment described above, in the structure in which the outer circumferential surface of the eccentric section of the crankshaft and the inner circumferential surface of the orbiting sliding bearing slide through the lubricating oil, the shear resistance of the oil film by the lubricating oil, which exists between the outer circumferential surface of the eccentric section and the inner circumferential surface of the orbiting sliding bearing, can be reduced. Accordingly, the bearing loss during the fluid lubrication can be reduced. This effect is further described in detail.

[0053] Generally, a shear stress τ of a thin fluid lubricating oil film is known to have an increasing relationship with a variation gradient dU/dh, which relates to a shaft rotation direction flow rate U of the lubricating oil relative to an oil film thickness h direction, and a lubricating oil viscosity η. According to the present embodiment, the gap of a portion has a very small role of carrying out a load support by the oil film pressure and the gap between the outer circumference of the eccentric shaft and the inner circumference of the orbiting sliding bearing is relatively small. The gap can be enlarged by providing the loss reduction groove formed by the engraved groove or the notch at the outer circumference of the eccentric shaft. As a result, the oil film thickness h by the lubricating oil filling the portion of the loss reduction groove can be increased. Thus, the variation gradient dU/dh is reduced and the shear stress of the oil film can be reduced. Accordingly, the shear resistance of the oil film, which is an integral value of the shear stress, is reduced, and the bearing loss can be reduced.

[0054] Further, since the seal section serving as the flow path resistance of the crankshaft in the axial direction is provided on at least the end surface (upper end) side and the base end (lower end) side of the eccentric section of the loss reduction groove, the loss reduction groove does not communicate simultaneously with the end surface side and the base end side of the eccentric section across the orbiting sliding bearing in the axial direction. On the other hand, the oil supply passage has the structure of communicating with the end surface side and the base end side of the eccentric section across the orbiting sliding bearing in the axial direction. As a result, the axial flow path resistance is smallest in the oil supply passage, and the lubricating oil supplied through the oil supply hole is preferentially flowed to the oil supply passage. Accordingly, an oil supply condition of the lubricating oil at the eccentric section is maintained substantially the same as that of the conventional one without the loss reduction groove. Even if the loss reduction groove is provided as in the present embodiment, an increase in the amount of oil or deterioration of the oil supply condition can be prevented.

[0055] In the present embodiment, the start position of the loss reduction groove is provided within the angular range of 140° to 210° around the center of the eccentric shaft in the direction opposite to the rotation direction of the crankshaft from the direction opposite to the eccentricity direction of the eccentric section. As described above, this is because when the start position of the loss reduction groove is 140 degrees or less, generation of the oil film pressure supporting the load is prevented, and the effect of reducing the bearing loss is lost. Also, since the oil film pressure is decreased, the oil film absence can occur. Consequently, the start position of the loss reduction groove is set to 140 degrees or more. Moreover, even if the start position of the loss reduction groove is 210 degrees or more, the region is originally a portion in which the gap between the shaft and the bearing is comparatively large during the operation of the scroll compressor. Accordingly, it is difficult to obtain the effect of reducing the variation gradient dU/dh, of reducing the oil film shear stress, and of reducing the bearing loss. Therefore, the start position of the loss reduction groove is set to 210 degrees or less.

[0056] Further, in the present embodiment, the loss reduction groove is formed in such a manner that a portion with the radial direction depth of 0.002 mm or more (preferably, 0.01 to 0.05 mm or more) from the unprocessed outer circumferential circle of the eccentric section exists within the range of 140° to 210° (preferably, 145° to 180°) in the direction opposite to the rotation direction of the crankshaft from the direction opposite to the eccentricity direction of the eccentric section. Therefore, the stable bearing loss reduction effect having a few variations caused by dimension errors or the like can be obtained.

Reference Signs List

[0057]

1	scroll compressor
2	closed container
3	lubricating oil
4	orbiting boss section inner space
102	electric motor
103	crankshaft
104	frame
105	main bearing
106	lower frame
107	auxiliary bearing

108	eccentric section
108a	end surface (upper end)
108b	base end (lower end)
109	orbiting scroll
110	orbiting sliding bearing
111	Oldham ring
112	fixed scroll
113	suction port
114	discharge port
115	oil supply hole
116	oil supply passage
117, 117a, 117b	loss reduction groove
118, 118a, 118b	seal section
119	intermediate chamber (backpressure chamber)
120	shaft rotation direction
121	bearing load direction
122	minimum gap portion

Claims

1. A scroll compressor, comprising:

a fixed scroll;
an orbiting scroll engaging with the fixed scroll;
a crankshaft having an eccentric section at an end portion for causing the orbiting scroll to carry out orbital motion;
an oil supply hole penetrating an inside of the crankshaft in an axial direction and having an opening portion on an end surface of the eccentric section:

an orbiting sliding bearing provided at the orbiting scroll and sliding by engaging with the eccentric section of the crankshaft; and
an oil supply passage provided at an outer circumference of the eccentric section of the crankshaft so as to communicate with an end surface side and a base end side of the eccentric section,
the scroll compressor is structured so as to lubricate a gap between the eccentric section and the orbiting sliding bearing by lubricating oil supplied through the oil supply hole, further comprising:

an axial loss reduction groove provided at the outer circumference of the eccentric section of the crankshaft separately from the oil supply passage; and
a seal section which is provided on at least one of the end surface side and the base end side of the eccentric section of the loss reduction groove.

2. The scroll compressor according to claim 1, wherein a start position of the loss reduction groove is provided at a position of 140° to 210° around a center of the eccentric section in a direction opposite to a rotation direction of the crankshaft from a direction opposite to an eccentricity direction of the eccentric section.

3. The scroll compressor according to claim 2, wherein the start position of the loss reduction groove is provided at a position of 145° to 180° around the center of the eccentric section in the direction opposite to the rotation direction of the crankshaft from the direction opposite to the eccentricity direction of the eccentric section.

4. The scroll compressor according to claim 2, wherein the oil supply passage and the loss reduction groove are respectively formed at the outer circumference of the eccentric section by an engraved groove or a notch, and the loss reduction groove is formed so as to have a portion with a radial direction depth of 0.002 mm or more from an unprocessed outer circumferential circle of the eccentric section.

5. The scroll compressor according to claim 4, wherein the depth of the loss reduction groove is set to 0.01 mm or more, and to 20% or less of a shaft diameter of the eccentric section.

6. The scroll compressor according to claim 5, wherein the depth of the loss reduction groove is set to 0.05 to 0.5 mm.

7. The scroll compressor according to claim 1, wherein a plurality of the loss reduction grooves is formed in a circumferential direction of the eccentric section.

8. The scroll compressor according to claim 7, wherein at least one of plurality of the loss reduction grooves formed is provided at a position of 140° to 210° around a center of the eccentric section in a direction opposite to a rotation direction of the crankshaft from a direction opposite to an eccentricity direction of the eccentric section.

9. The scroll compressor according to claim 1, wherein the seal section is provided on at least the base end side of the eccentric section of the loss reduction groove.

FIG. 1

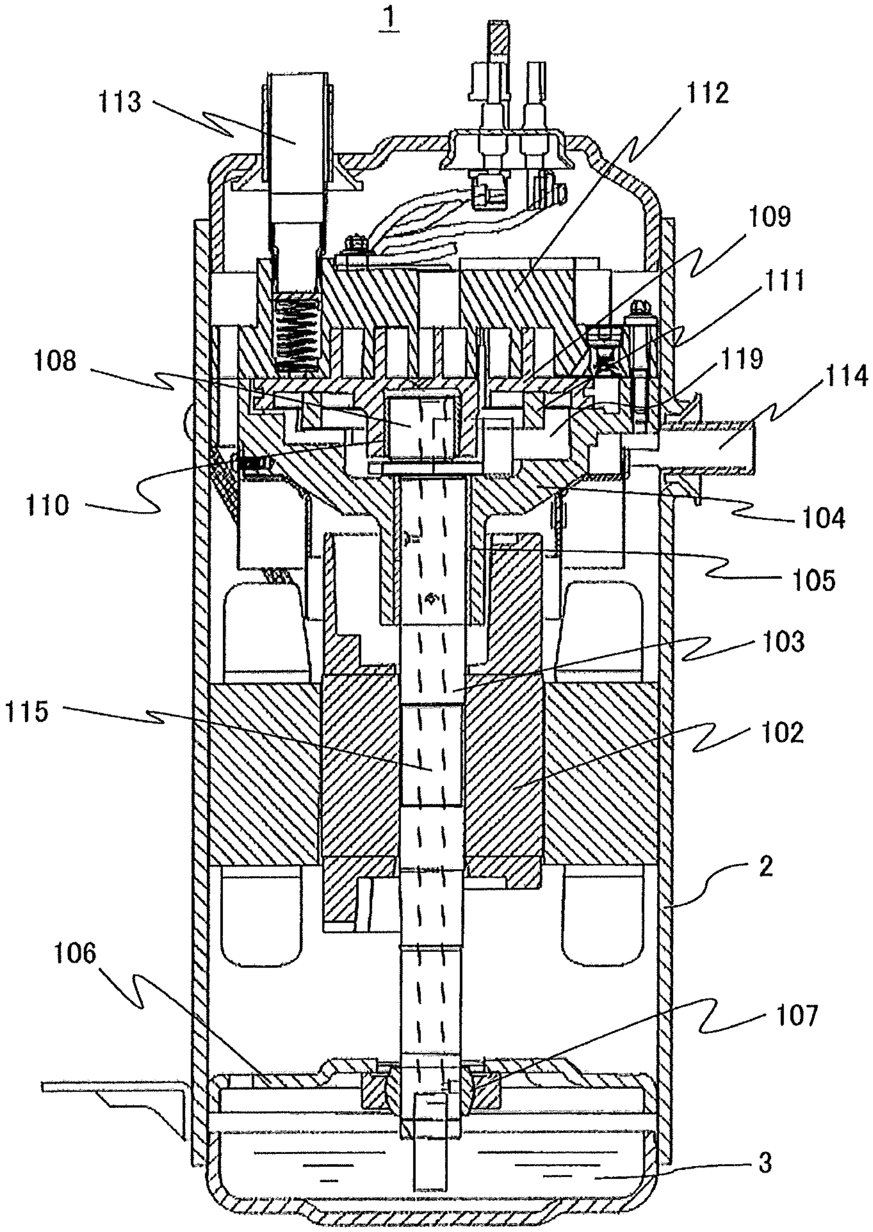


FIG. 2

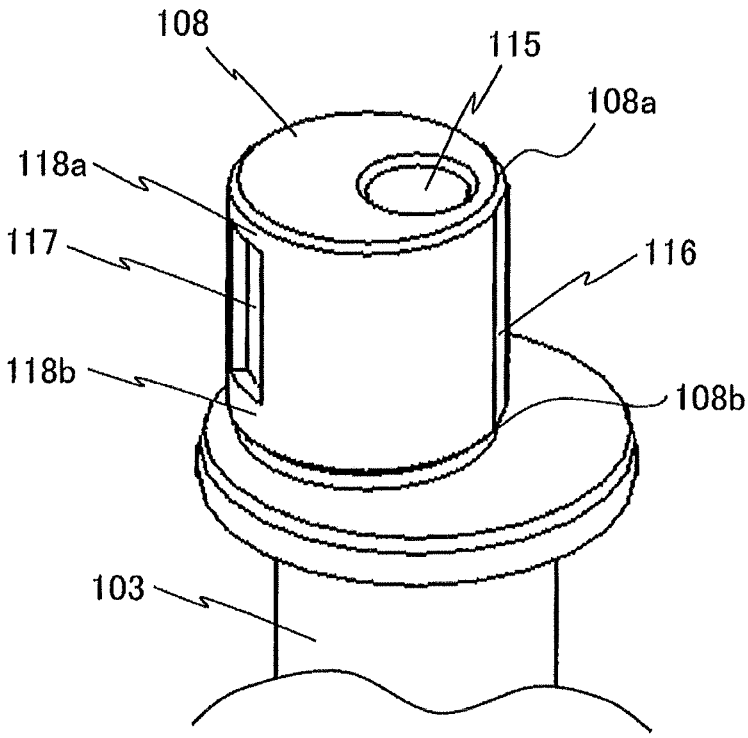


FIG. 3

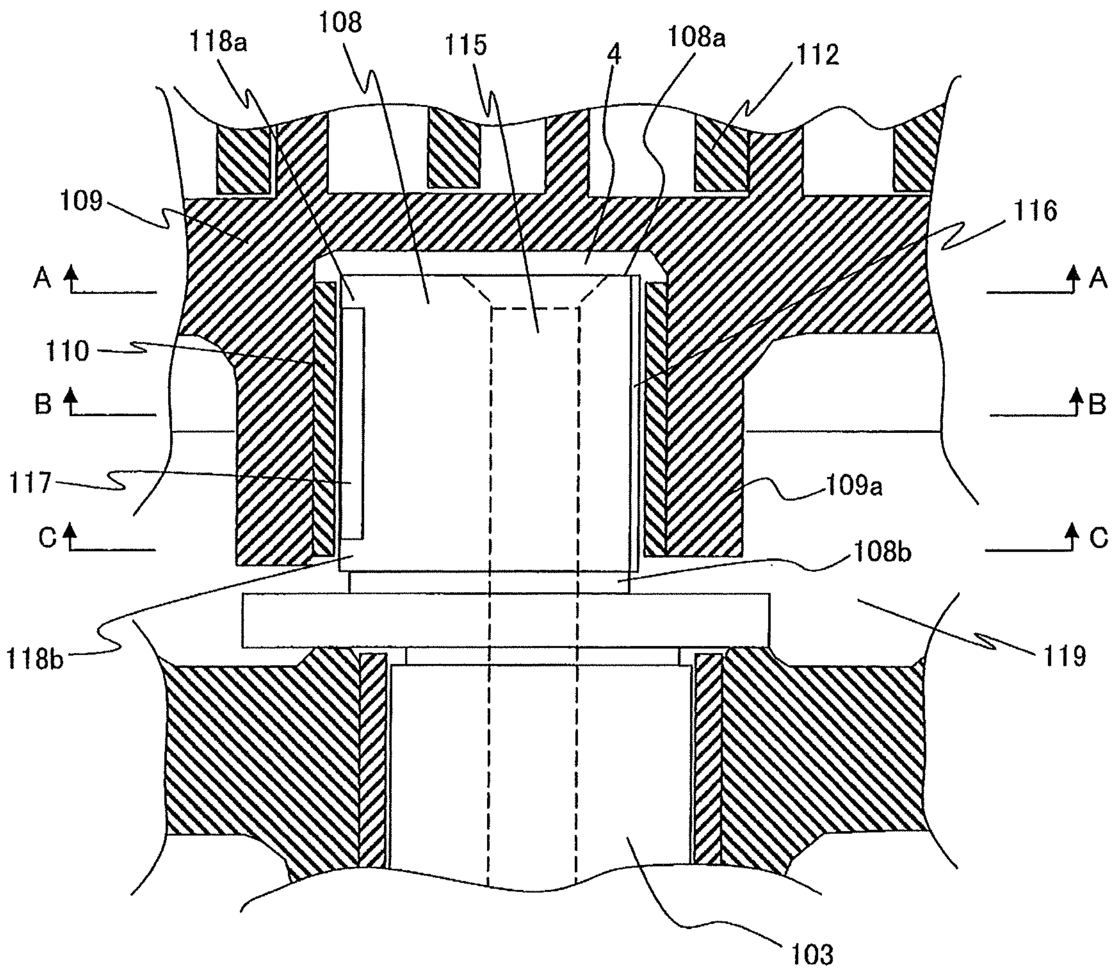

118a
108
115
4
108a
112
109
116
A
A
110
B
B
117
109a
C
C
108b
118b
119
103

FIG. 4

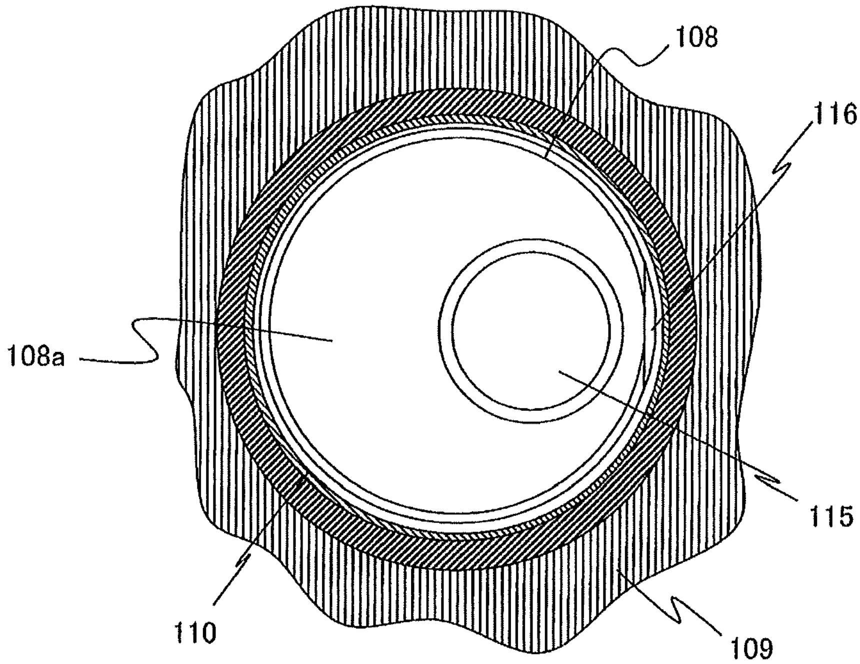


FIG. 5

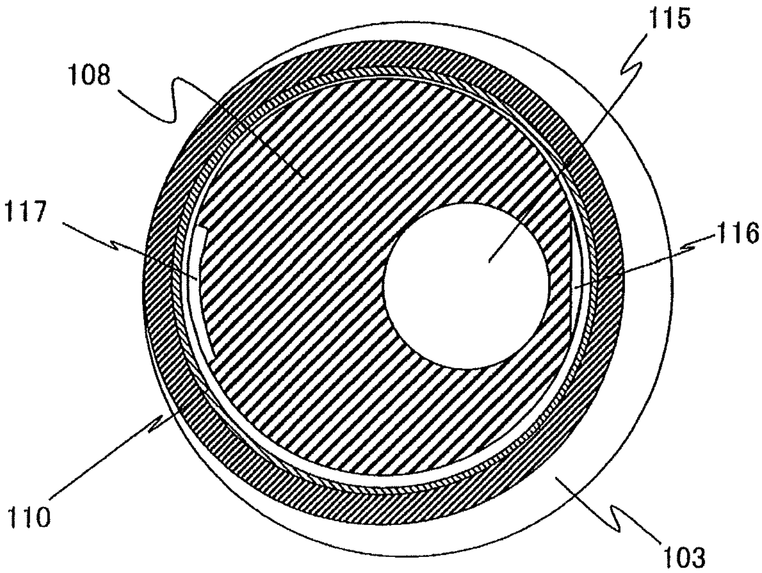


FIG. 6

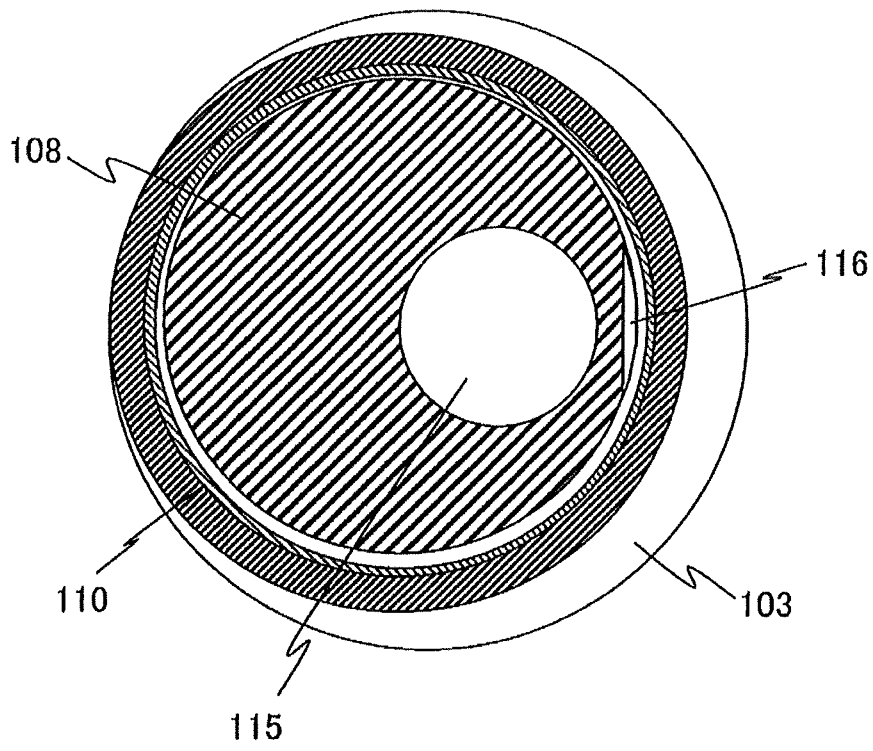


FIG. 7

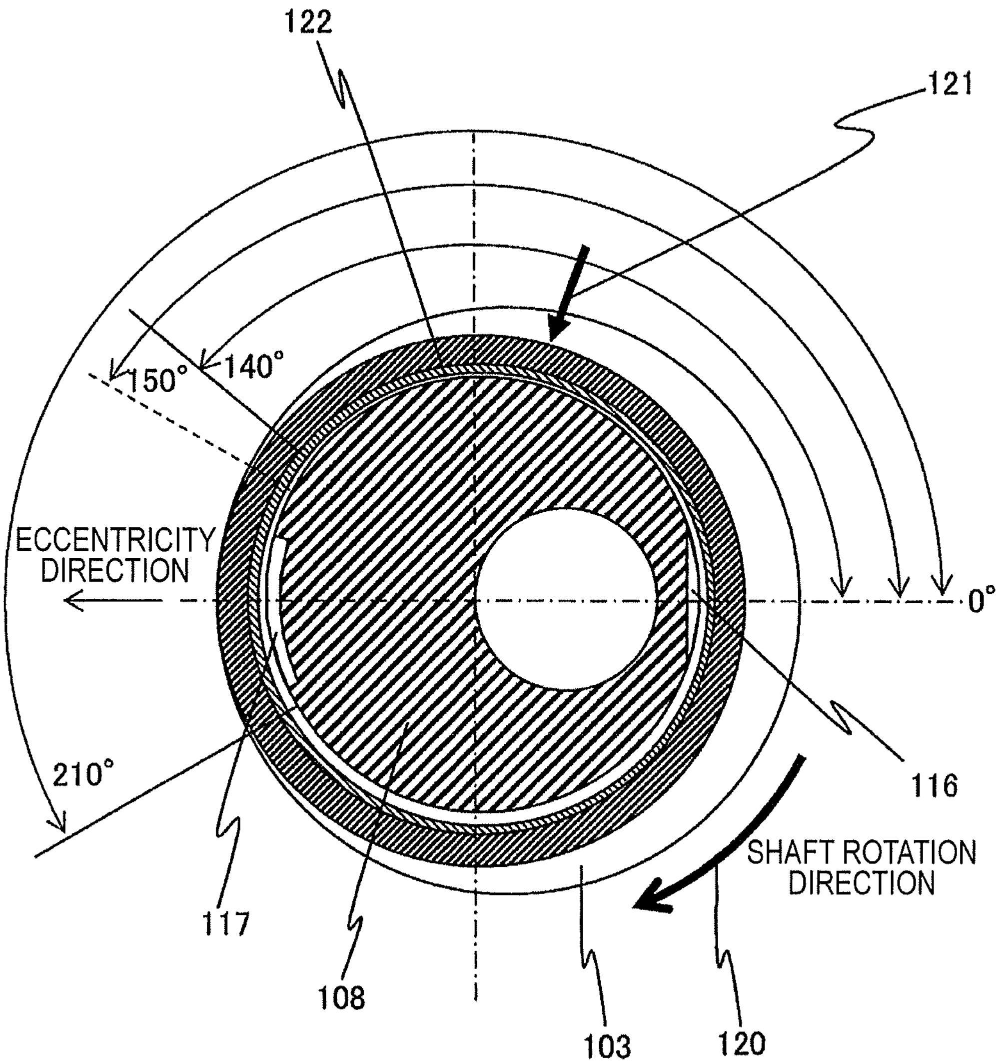
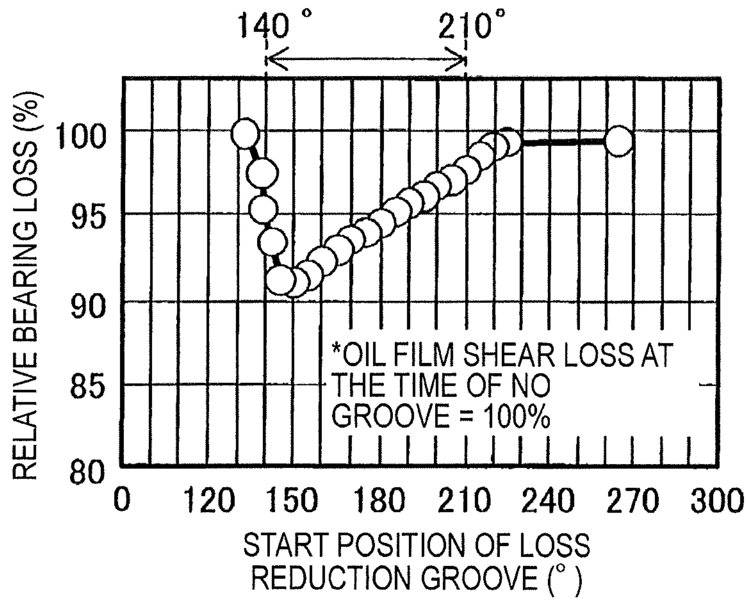

122
121
140°
150°
ECCENTRICITY
DIRECTION
0°
210°
116
SHAFT ROTATION
DIRECTION
117
108
103
120

FIG. 8

(a)


140°
210°
RELATIVE BEARING LOSS (%)
100
95
90
85
80
*OIL FILM SHEAR LOSS AT THE TIME OF NO GROOVE = 100%
0
120
150
180
210
240
270
300
START POSITION OF LOSS REDUCTION GROOVE (°)

(b)

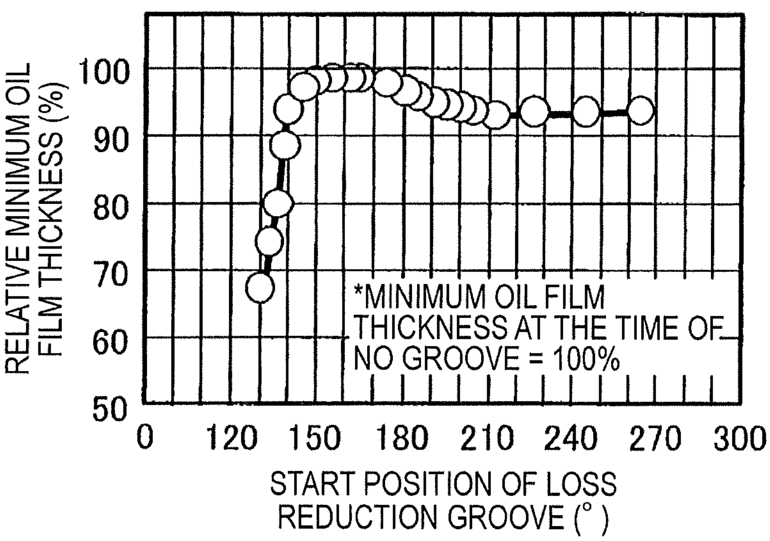

RELATIVE MINIMUM OIL FILM THICKNESS (%)
100
90
80
70
60
50
*MINIMUM OIL FILM THICKNESS AT THE TIME OF NO GROOVE = 100%
0
120
150
180
210
240
270
300
START POSITION OF LOSS REDUCTION GROOVE (°)

FIG. 9

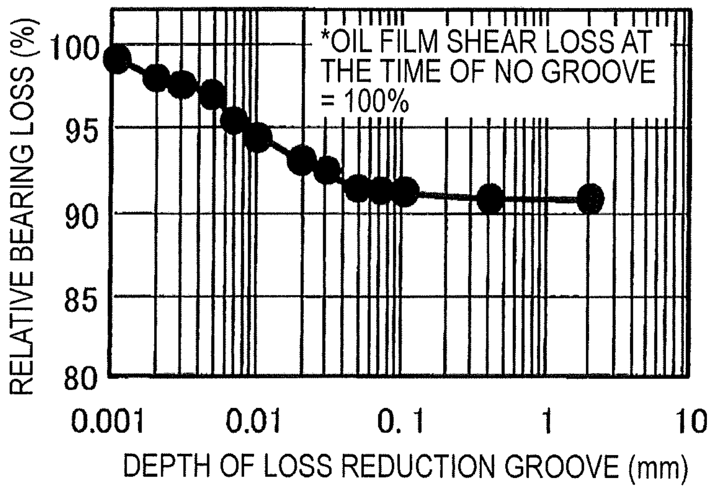


FIG. 10

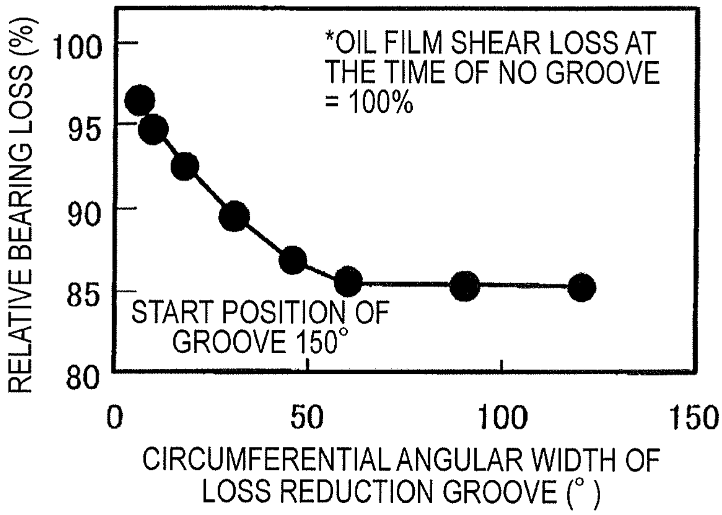


FIG. 11

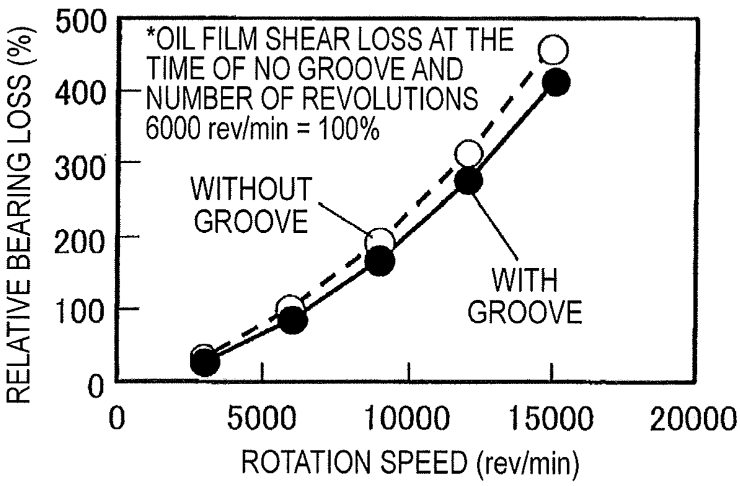


FIG. 12

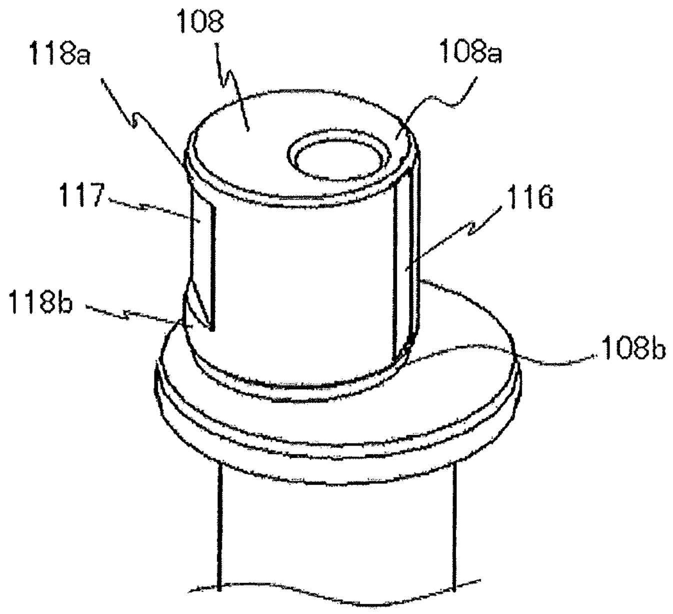


FIG. 13

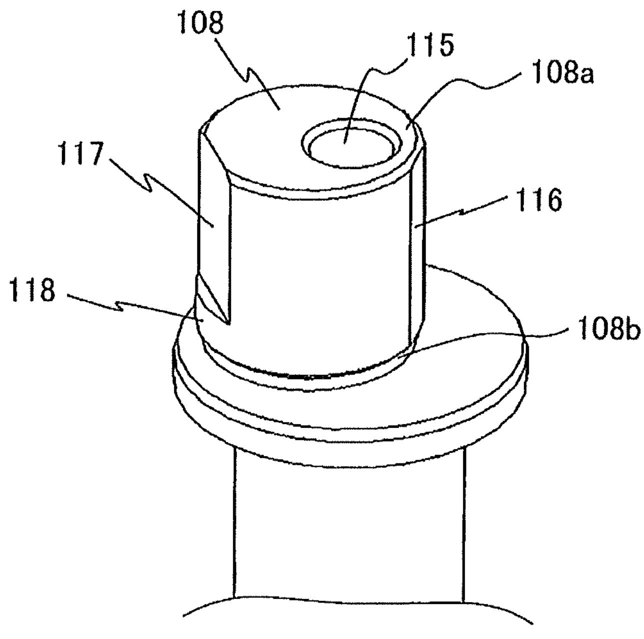
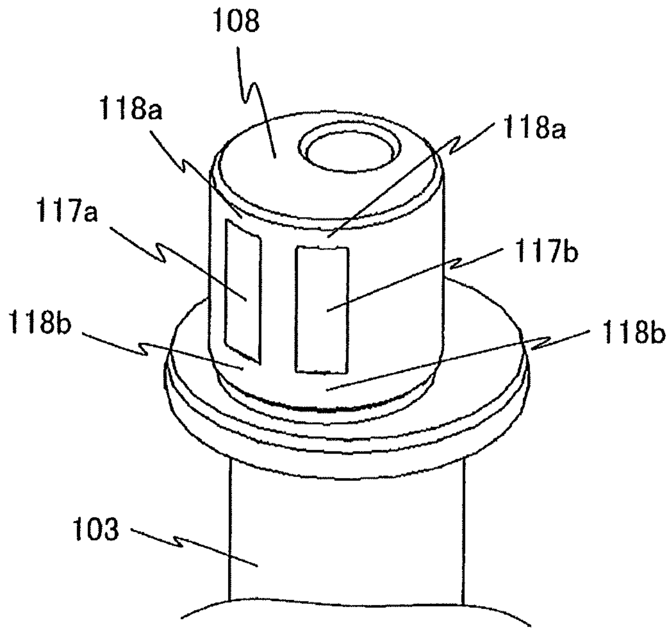


FIG. 14



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2011/059938

A. CLASSIFICATION OF SUBJECT MATTER
F04C18/02(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)
F04C18/02

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2011
Kokai Jitsuyo Shinan Koho	1971-2011	Toroku Jitsuyo Shinan Koho	1994-2011

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.
A	JP 4-54296 A (Hitachi, Ltd.), 21 February 1992 (21.02.1992), entire text; all drawings (Family: none)	1-9
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 39667/1987(Laid-open No. 146222/1988) (Daikin Industries, Ltd.), 27 September 1988 (27.09.1988), entire text; all drawings (Family: none)	1-9

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

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
04 July, 2011 (04.07.11)	12 July, 2011 (12.07.11)

Name and mailing address of the ISA/	Authorized officer
Japanese Patent Office	
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 2009)

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2011/059938

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 64972/1990(Laid-open No. 24684/1992) (Mitsui Seiki Kogyo Co., Ltd.), 27 February 1992 (27.02.1992), entire text; all drawings (Family: none)	1-9
A	JP 6-173954 A (Hitachi, Ltd.), 21 June 1994 (21.06.1994), entire text; all drawings (Family: none)	1-9
A	JP 11-159484 A (Daikin Industries, Ltd.), 15 June 1999 (15.06.1999), entire text; all drawings (Family: none)	1-9

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2003239876 A **[0004] [0008]**
- JP 11159484 A **[0006] [0008]**