(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 2 256 344 A1

(12) **EUROPEAN PATENT APPLICATION**
published in accordance with Art. 153(4) EPC

(43) Date of publication:
01.12.2010 Bulletin 2010/48

(21) Application number: **09823255.6**

(22) Date of filing: **19.10.2009**

(51) Int Cl.:
F04B 39/12 (2006.01) ***F04B 39/00*** (2006.01)

(86) International application number:
PCT/JP2009/005449

(87) International publication number:
WO 2010/050141 (06.05.2010 Gazette 2010/18)

(84) Designated Contracting States:
AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO SE SI SK SM TR
Designated Extension States:
AL BA RS

(30) Priority: **29.10.2008 JP 2008277972**
29.10.2008 JP 2008277973

(71) Applicant: **Panasonic Corporation**
Kadoma-shi
Osaka 571-8501 (JP)

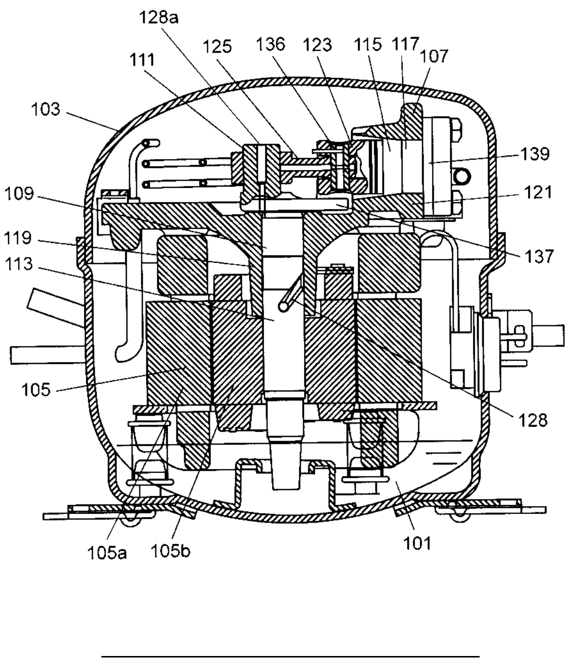
(72) Inventors:
- **YAGI, Akio**
Osaka-shi, Osaka 540-6207 (JP)
- **MORITA, Ichiro**
Osaka-shi, Osaka 540-6207 (JP)

(74) Representative: **Grünecker, Kinkeldey, Stockmair & Schwanhäusser**
Anwaltssozietät
Leopoldstrasse 4
80802 München (DE)

(54) SEALED COMPRESSOR

(57) A hermetic compressor includes a cylindrical hole (117) forming a compression space (115) and having a tapered portion (127) so as to increase its inner diameter from the top dead center to the bottom dead center of a piston (123). The piston (123) is reversed in its inclination direction with respect to the axial center of the cylindrical hole (117) in the initial stage of the compression stroke. This reduces the contact between the piston (123) and the cylindrical hole (117) at the time of the reversal, as compared with the case in which the piston (123) is reversed in its inclination direction in the middle or later stage of the compression stroke. As a result, the hermetic compressor is reliable and has low noise level.

FIG. 1



EP 2 256 344 A1

Description

TECHNICAL FIELD

[0001] The present invention relates to a hermetic compressor used in a refrigeration cycle of a fridge-freezer or the like.

BACKGROUND ART

[0002] One conventional hermetic compressor having a reciprocating compression mechanism is disclosed on Patent Literature 1. The hermetic compressor of Patent Literature 1 includes a cylinder, a piston, and a connecting rod. The cylinder has a cylindrical compression space on the inner-diameter side. The piston is cylindrical on the outer-diameter side and reciprocates in the cylinder. The connecting rod connects the piston to an eccentric shaft portion of a shaft via a piston pin. The compression mechanism is driven by the rotation of a rotor of a motor with the shaft fixed to the axial center of the rotor.

[0003] In such hermetic compressors, it is generally necessary to provide a gap for allowing the inner-diameter side of the cylinder and the outer-diameter side of the reciprocation piston to slide with each other. When the gap is large, the high-temperature, high-pressure refrigerant gas compressed in the compression space leaks out, having low compression efficiency. When the gap is small, on the other hand, the sliding loss is large, also having low compression efficiency.

[0004] To avoid this problem, the hermetic compressor of Patent Literature 1 has a cylinder which is tapered so as to increase its inner diameter from the top dead center to the bottom dead center of the piston.

[0005] This conventional hermetic compressor will be described as follows with reference to drawings. Figs. 12A and 12B are longitudinal sectional views of a compression section of the hermetic compressor disclosed in Patent Literature 1. Fig. 12A shows a state in which the piston is in the bottom dead center position, and Fig. 12B shows a state in which the piston is in the top dead center position.

[0006] In Figs. 12A and 12B, the compression section includes cylinder block 14 having cylindrical hole 16 and piston 23 reciprocable therein, and connecting rod 26 connected to piston 23 via piston pin 25. Connecting rod 26 reciprocates piston 23 between the bottom dead center shown in Fig. 12A and the top dead center shown in Fig. 12B by the eccentric movement of an eccentric shaft portion of a shaft (not shown).

[0007] The compression section also includes an unillustrated valve plate at an end of cylindrical hole 16 that is opposite (the right side in the drawing) to connecting rod 26. Piston 23, cylindrical hole 16, and the valve plate together form compression space 15.

[0008] Cylindrical hole 16 includes tapered portion 17 so as to increase its inner diameter from Dt to Db (> Dt) from the top dead center to the bottom dead center of piston 23. Piston 23 has a uniform outer diameter throughout its length.

[0009] With this structure, the pressure in compression space 15 does not increase very much while the outer surface of piston 23 is moving from the bottom dead center shown in Fig. 12A toward the top dead center along tapered portion 17 in the compression stroke to compress the refrigerant gas. Therefore, even when the gap is comparatively large, there is almost no leakage of the refrigerant gas, and piston 23 has low sliding resistance due to the sealing effect of lubricating oil.

[0010] As the compression stroke proceeds, the pressure of the refrigerant gas in compression space 15 gradually increases. When piston 23 comes close to the top dead center position shown in Fig. 12B, the pressure in compression space 15 reaches a predetermined discharge pressure, making compression space 15 susceptible to leakage of the refrigerant gas. However, the gap is small on the top dead center side, thereby obtaining the sealing effect of the lubricating oil, and hence, reducing the leakage of the refrigerant gas.

[0011] In the above described conventional structure, however, piston 23 comes into contact at edge 30 on the compression space 15 side with tapered portion 17 in the compression stroke. As a result, starting from edge 30, piston 23 is reversed in its inclination direction with respect to the axial center of cylindrical hole 16. Then, the region of the outer surface of piston 23 that did not slide with tapered portion 17 before the reversal comes into contact with tapered portion 17, possibly making the sliding too much or causing contact noise when the contact is severe at the time of the reversal.

[0012] Patent Literature 1: Japanese Patent Unexamined Publication No. 2002-89450

SUMMARY OF THE INVENTION

[0013] In view of the conventional problems, it is an object of the present invention to provide a hermetic compressor in which the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole in the initial stage of the compression stroke. This reduces the contact between the piston and the tapered portion at the time of the reversal and reduces the noise, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke.

[0014] According to the present invention, the hermetic compressor includes an airtight container having lubricating oil therein; an electric element; and a compression element, the compression element being driven by the electric element, and the electric element and the compression element being housed in the airtight container. The compression element includes a shaft, a cylinder block, a piston, and a connection mechanism. The shaft has a main shaft portion and an eccentric shaft portion, the main shaft portion being rotatable by the electric element, and the eccentric shaft portion moving in unison with the main shaft portion. The cylinder block has a cylindrical hole and a bearing, the cylindrical hole forming a compression space, and the bearing supporting the main shaft portion. The piston is reciprocable in the cylindrical hole. The connection mechanism connects the eccentric shaft portion and the piston. The cylindrical hole includes a tapered portion so as to increase an inner diameter thereof from the top dead center to the bottom dead center of the piston, and the piston is reversed in the inclination direction thereof with respect to the axial center of the cylindrical hole in the initial stage of the compression stroke.

[0015] This structure reduces the sliding resistance, that is, the sliding loss between the piston and the cylindrical hole. This structure also reduces the load when the region of the outer surface of the piston, which did not slide with the tapered portion at the time of the reversal, comes into contact with the tapered portion. The load reduction can be achieved because in the initial stage of the compression stroke, the end face on the compression space side of the piston is subjected to only a small compressive load. This reduces the contact between the piston and the tapered portion, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke. This results in a reduction in the contact when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole, thereby achieving noise reduction.

BRIEF DESCRIPTION OF THE DRAWINGS

[0016]

Fig. 1 is a longitudinal sectional view of a hermetic compressor according to a first exemplary embodiment of the present invention.

Fig. 2 is a longitudinal sectional view of an essential part of a compression section of the hermetic compressor according to the first exemplary embodiment.

Fig. 3 is a longitudinal sectional view of the essential part of the compression section, including its design dimensions, of the hermetic compressor according to the first exemplary embodiment.

Fig. 4 is a cross sectional view of the essential part of the compression section, including its design dimensions, of the hermetic compressor according to the first exemplary embodiment.

Fig. 5A is a schematic diagram showing a behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 5B is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 6A is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 6B is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 7A is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 7B is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 8A is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 8B is a schematic diagram showing another behavior of piston 123 in the compression stroke of the hermetic compressor according to the first exemplary embodiment.

Fig. 9 is a characteristic diagram showing the relationship between rotation angle and noise obtained from an example of the design dimensions of the hermetic compressor according to the first exemplary embodiment.

Fig. 10 is a longitudinal sectional view of an essential part of a compression section, including its design dimensions, of a hermetic compressor according to a second exemplary embodiment of the present invention.

Fig. 11 is a cross sectional view of the essential part of the compression section, including its design dimensions, of the hermetic compressor according to the second exemplary embodiment.

Fig. 12A is a longitudinal sectional view of a compression section of a conventional hermetic compressor.

Fig. 12B is another longitudinal sectional view of the compression section of the conventional hermetic compressor.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

[0017] Exemplary embodiments of the hermetic compressor of the present invention will be described as follows. Note that the present invention is not limited to these exemplary embodiments.

FIRST EXEMPLARY EMBODIMENT

[0018] Fig. 1 is a longitudinal sectional view of a hermetic compressor according to a first exemplary embodiment of the present invention. Fig. 2 is a longitudinal sectional view of an essential part of a compression section of the hermetic compressor. Fig. 3 is a longitudinal sectional view of the essential part of the compression section including its design dimensions. Fig. 4 is a cross sectional view of the essential part of the compression section including its design dimensions.

[0019] In Figs. 1 to 4, the hermetic compressor includes airtight container 103 having electric element 105 and compression element 107 driven by electric element 105. Electric element 105 includes stator 105a and rotor 105b. Airtight container 103 has lubricating oil 101 at its bottom. Compression element 107 includes shaft 113 having main shaft portion 109 and eccentric shaft portion 111, which is eccentric thereto to move in unison therewith. Main shaft portion 109 is fixed to the axial center of rotor 105b.

[0020] Compression element 107 also includes bearing 119, which forms a cantilever bearing by supporting the end on the eccentric shaft portion 111 side of main shaft portion 109 of shaft 113.

[0021] Compression element 107 also includes balance weight 137 between main shaft portion 109 and eccentric shaft portion 111. Balance weight 137 is eccentric in the direction opposite to the eccentric direction of eccentric shaft portion 111 so as to strike a rotational balance with the eccentric weight attached to main shaft portion 109. The eccentric weight is the load of eccentric shaft portion 111 or the pressure load of the refrigerant gas in compression space 115 which acts on eccentric shaft portion 111.

[0022] Compression element 107 also includes cylinder block 121 having cylindrical hole 117 with a substantially cylindrical shape and bearing 119, which are arranged in fixed positions relative to each other. Cylindrical hole 117 has piston 123 reciprocable therein.

[0023] Compression element 107 also includes connecting rod 125 as a connection mechanism, whose one end is connected with eccentric shaft portion 111, and whose other end is connected with piston 123 via piston pin 136. Shaft 113 is provided on its inside and outer surface with oil supply passage 128. Oil supply passage 128 is communicated at one end (upper end) thereof with oil supply hole 128a formed in eccentric shaft portion 111. The end of main shaft portion 109 on the side opposite to eccentric shaft portion 111, that is, the bottom end of main shaft portion 109 is extended so that oil supply passage 128 can reach a predetermined depth of lubricating oil 101.

[0024] Compression element 107 also includes valve plate 139 at an end of cylindrical hole 117. Cylindrical hole 117 is formed in cylinder block 121 in such a manner to form compression space 115 together with piston 123 and valve plate 139. As shown in Fig. 3, cylindrical hole 117 includes tapered portion 127 so as to increase its inner diameter from D1 to D3 (> D1) from the top dead center to the bottom dead center of piston 123. Cylindrical hole 117 also includes straight portion 129 in the position corresponding to the end on the compression space 115 side of piston 123 when piston 123 is in the top dead center position. Straight portion 129 has a uniform inner diameter in an axial length L1. Piston 123 has a uniform outer diameter D2 throughout its length.

[0025] As shown in Fig. 3, cylindrical hole 117 of cylinder block 121 is formed in such a manner that when piston 123 is in the bottom dead center position, the side opposite to the compression space 115 side of piston 123 is exposed in airtight container 103.

[0026] Piston 123 is provided on its outer surface 133 on the compression space 115 side with concave oil supply groove 131 which is substantially annular (including completely annular). Cylindrical hole 117 has notch 120 on its peripheral wall so that oil supply groove 131 is exposed at least partly from cylindrical hole 117 and is communicated with airtight container 103 when piston 123 is in the bottom dead center position.

[0027] Piston 123 has the outer diameter D2, and eccentric shaft portion 111 has an eccentricity "e" with respect to main shaft portion 109. The center of piston pin 136 and the compression-space-side end face 134 of piston 123 have a distance (hereinafter, main sliding surface length) L2 therebetween. The center of piston pin 136 corresponds to the connection center between connecting rod 125 and piston 123. Main shaft portion 109, which has a rotation angle of 0 (zero) degrees when the piston 123 is in the top dead center position, has a rotation angle "θ". The axial center of compression space 115 and tapered portion 127 form an angle "α" therebetween.

[0028] The inner diameter D1 of cylindrical hole 117, the outer diameter D2 of piston 123, the length L1 of straight portion 129, the main sliding surface length L2, the eccentricity "e", and the rotation angle "θ" are design dimensions to find the coordinates of the tip position of piston 123 in cylindrical hole 117 when the behaviors of piston 123 in cylindrical hole 117 are simulated.

[0029] The angle "α" formed by tapered portion 127 when the above-mentioned design dimensions are selected is set in the range obtained by multiplying a coefficient in the range of 0.4 to 2.0 by a value "γ". The value "γ" (hereinafter, dimension value) is obtained by dividing the dimensional numerical value 3/2 of the difference (D1 - D2) between the inner diameter D1 of cylindrical hole 117 and the outer diameter D2 of piston 123 by the coordinate position {L1 - L2 + 2e(1 - cosθ)} of the tip on the top-dead-center side of piston 123 when the top dead center position is zero.

[0030] The dimensional numerical value 3/2 is a value derived from the above-mentioned design dimensions (values) to find the coordinates of the tip position of piston 123 in cylindrical hole 117.

[0031] In other words, according to the present exemplary embodiment, the angle "α" defines the dimension value "γ" expressed by Mathematical Formula 1 based on the above-mentioned design dimensions: the inner diameter D1 of cylindrical hole 117, the outer diameter D2 of piston 123, the length L1 of straight portion 129, the main sliding surface length L2, the eccentricity "e", and the rotation angle "θ". The angle "α" is defined by Mathematical Formula 2 which is based on the dimension value "γ".

[0032] In this case, the rotation angle "θ" of main shaft portion 109 is in the range of π to 4 π/3 (rad) in the initial stage of the compression stroke.

$$\gamma = \{3(D1 - D2)/2\}/\{L1 - L2 + 2e(1 - \cos\theta)\} \quad \text{Mathematical Formula 1}$$

$$0.4\gamma \leq \tan(\alpha) \leq 2.0\gamma \text{ where } \alpha > 0 \quad \text{Mathematical Formula 2}$$

[0033] The coefficients of the dimension value "γ" (0.4 and 2.0 in the present exemplary embodiment) are values properly determined in view of the machining tolerance of tapered portion 127 and the like, and can be set according to the material of cylinder block 121.

[0034] The following is a description of the operation of the hermetic compressor structured as above. First, sealing effect and sliding resistance in tapered portion 127 and straight portion 129 of cylindrical hole 117 will be described as follows.

[0035] Rotor 105b of electric element 10 rotates shaft 113, and the rotational motion of eccentric shaft portion 111 is transmitted to piston 123 via connecting rod 125, allowing piston 123 to reciprocate in cylindrical hole 117. The reciprocation of piston 123 allows refrigerant gas to be suctioned into compression space 115 from an unillustrated cooling system, to be compressed, and to be discharged into the cooling system.

[0036] By the rotation of shaft 113, oil supply passage 128 performs a pumping action at its bottom end so that lubricating oil 101 at the bottom of airtight container 103 is pumped through oil supply passage 128 and reaches oil supply hole 128a. Lubricating oil 101 reached oil supply hole 128a is sprinkled all around airtight container 103 horizontally from the upper end of shaft 113 so as to be supplied to piston pin 136, piston 123, and other components, thereby lubricating them.

[0037] The pressure in compression space 115 does not increase very much while piston 123 is moving from the bottom dead center shown in Fig. 3 toward the top dead center in the compression stroke to compress the refrigerant gas. Therefore, even when the gap is comparatively large between outer surface 133 of piston 123 and tapered portion 127, there is almost no leakage of the refrigerant gas, and piston 23 has low sliding resistance due to the sealing effect of lubricating oil 101.

[0038] As the compression stroke proceeds, the pressure of the refrigerant gas in compression space 115 gradually increases. When piston 123 comes close to the top dead center position, the pressure in compression space 115 suddenly increases. However, the gap is small between outer surface 133 of piston 123 and tapered portion 127 on the top-dead-center side, thereby reducing the leakage of the refrigerant gas. In this case, straight portion 129 reduces the leakage of the refrigerant gas that has increased to reach the predetermined discharge pressure, as compared with the case in which straight portion 129 is tapered.

[0039] When piston 123 is in the bottom dead center position, the connecting rod 125 side of piston 123 is exposed from cylinder block 121. This allows lubricating oil 101 sprinkled from the upper end of shaft 113 to be sufficiently supplied to outer surface 133 of piston 123 and to be held there.

[0040] When piston 123 is in the bottom dead center position, concave substantially annular oil supply groove 131 formed on outer surface 133 on the compression space 115 side of piston 123 is also exposed at least partly from cylindrical hole 117 via notch 120. This allows lubricating oil 101 sprinkled from the upper end of shaft 113 to be sufficiently supplied to oil supply groove 131 and to be held therein.

[0041] As a result, a sufficient amount of lubricating oil 101 can be supplied to the gap between the inner surface of cylindrical hole 117 of cylinder block 121 and outer surface 133 of piston 123 in the compression stroke.

[0042] Substantially annular oil supply groove 131 can move to the position facing straight portion 129 of cylindrical hole 117, making it easy to carry lubricating oil 101 to straight portion 129 in which the sliding resistance becomes the largest.

[0043] As a result, a larger amount of lubricating oil 101 is supplied to the sliding portion between cylinder block 121 and piston 123 and is adequately held there. Furthermore, the sliding resistance when piston 123 comes close to the top dead center position can be reduced, thereby achieving efficiency improvement.

[0044] Behaviors of piston 123 in the compression stroke will be described as follows with reference to Figs. 5A, 5B to 8A, 8B, which are schematic diagrams showing the behaviors of piston 123 in the present exemplary embodiment.

[0045] Figs. 5A, 5B to 8A, 8B are schematic diagrams showing the behaviors of piston 123 in the compression stroke. Figs. 5A to 8A are schematic diagrams showing a side surface of compression space 115. Figs. 5B to 8B are schematic diagrams showing a side surface of shaft 113. Figs. 5A, 5B to 7A, 7B show the initial stage of the compression stroke, and Figs. 8A and 8B show the latter stage of the compression stroke. Fig. 9 is a characteristic diagram showing the relationship between rotation angle and noise obtained from an example of the design dimensions of the hermetic compressor according to the present exemplary embodiment.

[0046] In the hermetic compressor of the present exemplary embodiment, bearing 119 forms a cantilever bearing which supports the end on the eccentric shaft portion 111 side of main shaft portion 109 of shaft 113. Therefore, shaft 113 is inclined in the clearance between main shaft portion 109 and bearing 119. It is known that shaft 113 has intricate behaviors, changing its direction and inclination angle according to operating and other conditions.

[0047] This is because shaft 113 is affected by various forces such as the pressure load in compression space 115 or the inertia force of piston 123 and connecting rod 125. Therefore, the schematic diagrams of Figs. 5B to 8B showing inclinations of shaft 113 are inferentially drawn by the applicant.

[0048] First, the initial stage of the compression stroke will be described. It is not known how shaft 113 is inclined in the initial stage of the compression stroke. As mentioned above, shaft 113 has intricate inclination behaviors, and hence, piston 123 is considered to have intricate behaviors.

[0049] In the initial stage of the compression stroke, piston 123 is near the bottom dead center and in the region of tapered portion 127 in cylindrical hole 117. In this case, piston 123 is easily inclined only by a small force, so that, under normal conditions, piston 123 is considered to slide along somewhere on the inner wall surface of tapered portion 127.

[0050] The following description is based on the case in which piston 123 is inclined nearly in the same manner as shaft 113 and is slid along an upper region of tapered portion 127 in cylindrical hole 117.

[0051] Assume that outer surface 133a, which is an upper region of outer surface 133 of piston 123, moves toward compression space 115 while sliding with the upper region of tapered portion 127 in cylindrical hole 117. Then, as shown in Figs. 6A and 6B, edge 135 on the outer surface 133b side of outer surface 133 of piston 123 that does not slide with tapered portion 127 comes into contact with the region of tapered portion 127 that faces outer surface 133b.

[0052] The inventors have obtained experimental results suggesting that as shown in Figs. 7A and 7B, piston 123 is reversed in its inclination direction with respect to the axial center of cylindrical hole 117, and consequently, outer surface 133b that did not slide with tapered portion 127 before that slides with tapered portion 127.

[0053] The inventors have another supposition as follows. Edge 135 on the outer surface 133b side of piston 123 that does not slide with tapered portion 127 comes into contact with tapered portion 127. At this moment, shaft 113 is inclined largely toward the side opposite to the compression space 115 side, making piston 123 reversed in its inclination direction with respect to the axial center of cylindrical hole 117.

[0054] At any rate, when the compression stroke proceeds to the middle or later stage and the refrigerant gas pressure in compression space 115 is increased, the compressive load of the refrigerant gas applied to eccentric shaft portion 111 of shaft 113 is supported only by main shaft portion 109 of the cantilever bearing. As a result, as shown in Figs. 8A and 8B, shaft 113 is inclined in the clearance between main shaft portion 109 and bearing 119. Although it keeps changing its direction, shaft 113 is inclined on the side opposite to the compression space 115 side.

[0055] As a result, the inclination of piston 123 is adjusted in such a manner that its axial center substantially coincides with the axial center of straight portion 129 in cylindrical hole 117, and then piston 123 further moves toward the compression space 115 side. Thus, compression is performed in such a manner as to reduce the leakage of the refrigerant gas that has increased to reach the predetermined discharge pressure, as compared with the case in which straight portion 129 is tapered.

[0056] In the above-described case, in the initial stage of the compression stroke, piston 123 is inclined nearly in the same manner as shaft 113 and is slid along the upper region of tapered portion 127 in cylindrical hole 117. Even when piston 123 and shaft 113 are inclined differently, however, at least piston 123 is considered to be inclined along some region of tapered portion 127. As a result, it is likely that piston 123 is reversed in its inclination direction, and consequently, the region of outer surface 133 that did not slide with tapered portion 127 before that slides with tapered portion 127.

[0057] Those are the inference-based description of the behaviors of piston 123. The inventors have performed experiments changing the design dimensions of tapered portion 127, while watching the behaviors of piston 123 shown in Figs. 5A, 5B to 8A, 8B. The experimental results indicate that noise is smaller when tapered portion 127 is designed by making a timing range suggestive of edge 135 of piston 123 coming into contact with tapered portion 127 as the initial stage of the compression stroke than by making the timing range as the middle or later stage of the compression stroke. The timing range is hereinafter referred to as a rotation angle θ1.

[0058] The reason is considered as follows. In the middle or later stage of the compression stroke in which compression space 115 has a high gas pressure and a large compressive load, shaft 113 or piston 123 is reversed in its inclination direction at high speed. This causes outer surface 133 of piston 123 to come into contact and to collide with tapered portion 127 severely.

[0059] From these results and inferences, piston 123 should be designed to be reversed in its inclination direction with respect to the axial center of cylindrical hole 117 in the initial stage of the compression stroke. This reduces the contact between piston 123 and cylindrical hole 117 at the time of the reversal, as compared with the case in which piston 123 is reversed in its inclination direction in the middle or later stage of the compression stroke. As a result, noise reduction can be achieved.

[0060] In order to allow piston 123 to be reversed in its inclination direction with respect to the axial center of cylindrical hole 117 in the initial stage of the compression stroke, tapered portion 127 and compression element 107 can be designed to meet the following requirements. When outer surface 133a of piston 123 is moved toward compression space 115 along tapered portion 127, edge 135 of outer surface 133b of piston 123 that does not slide with tapered portion 127 comes into contact with the region of tapered portion 127 with which outer surface 133 does not slide.

[0061] It is also possible that piston 123 is reversed in its inclination direction without edge 135 of piston 123 coming into contact with tapered portion 127. Even in such a case, noise reduction is expected to be achieved in the initial stage of the compression stroke.

[0062] Therefore, according to the present exemplary embodiment, as part of the design to make edge 135 of piston 123 come into contact with tapered portion 127 in the initial stage of the compression stroke, straight portion 129 is formed in cylindrical hole 117 as follows. Straight portion 129 is formed in a region which is adjacent to tapered portion 127 and corresponds to the upper end on the compression space 115 side of piston 123. Straight portion 129 is uniform in the inner diameter direction.

[0063] As described above, straight portion 129 designed as above can reduce the leakage of the refrigerant gas that has increased to reach the predetermined discharge pressure, as compared with the case in which straight portion 129 is tapered.

[0064] More specifically, edge 135 of piston 123 comes into contact with tapered portion 127 at the timing at which the difference between the outer diameter D2 of piston 123 and the minimum inner diameter of compression space 115 (the inner diameter D1 of straight portion 129 in the present exemplary embodiment) is reduced. This means that edge 135 of piston 123 comes into geometric contact with the region of tapered portion 127 that is in the vicinity of straight portion 129.

[0065] Thus, providing straight portion 129 advances the timing at which edge 135 of piston 123 comes into contact with tapered portion 127 as early as the initial stage of the compression stroke.

[0066] Increasing the axial length of straight portion 129 can more advance the timing at which edge 135 of piston 123 comes into contact with tapered portion 127. This reduces, however, the axial length of tapered portion 127 by just that much, decreasing the effect of reducing the sliding resistance in tapered portion 127.

[0067] Therefore, it is necessary to balance two opposing actions. One action is to reduce the refrigerant gas leakage in compression space 115 by providing straight portion 129, and also to make the timing at which edge 135 of piston 123 comes into contact with tapered portion 127 in the initial stage of the compression stroke. The other action is to ensure the axial length of tapered portion 127 by reducing the axial length of straight portion 129, thereby reducing the sliding resistance in tapered portion 127.

[0068] The inventors have examined the angle "α" formed by the axial center of compression space 115 and tapered portion 127, and other design dimensions of compression element 107, while watching the timing at which edge 135 of piston 123 comes into contact with tapered portion 127 in the initial stage of the compression stroke.

[0069] As a result, it has turned out that the angle "α" of tapered portion 127 and the design dimensions of compression element 107 can be determined so that the dimension value "γ" and the angle "α" of tapered portion 127 satisfy Mathematical Formula 2. As described above, the dimension value "γ" is expressed by the Mathematical Formula 1 when the design dimensions of compression element 107 are used as parameters, and the rotation angle "θ" of main shaft portion 109 is set in the range of π to 4 π/3 (rad) in the initial stage of the compression stroke.

[0070] Properly designing the design values such as the axial length of straight portion 129 and the angle "α" of tapered portion 127 in the range of the design dimensions can provide a hermetic compressor having higher performance.

[0071] Experimental results of an example of the design dimensions are shown in Fig. 9. In Fig. 9, solid line 91 represents a noise level in the case of using the design dimensions of the present invention. Dotted line 92 represents a noise level in the case of using conventional design dimensions. Solid line 93 represents the range of rotation angle θ1 in the case of using the design dimensions of the present invention. Dotted line 94 represents the range of rotation angle θ1 in the case of using the conventional design dimensions. The experimental results have been obtained by measuring noise values under the following conditions: the inner diameter D1 of cylindrical hole 117 is about 22.01 mm, the outer diameter D2 of piston 123 is about 22 mm (D1 > D2), the main sliding surface length L2 is about 13 mm, the eccentricity "e" is 10 mm, and the length L1 of straight portion 129, which is one of the design dimensions, is about 4 mm, about 8 mm, or about 10 mm (the rotation angle "θ" is about 190 degrees, about 210 degrees, or about 225 degrees). The angle "α" obtained in this experiment is in the range of 0.03 to 0.05 degrees. It goes without saying that this range includes some tolerance.

[0072] These results indicate that an improvement in noise characteristics can be expected under the following conditions. The design dimensions of cylindrical hole 117, piston 123, and other components are set, and the timing at which edge 135 of piston 123 comes into contact with tapered portion 127 is set to between about 180 degrees (the initial stage of the compression stroke) at which compression is started and about 240 degrees in the middle stage of the compression stroke.

[0073] In other words, as shown in Fig. 9, the design dimensions in the conventional design are determined to be used over a wide range including the middle stage of the compression stroke, and therefore, the design dimensions include those having high noise levels. In the present exemplary embodiment, on the other hand, the dimension value "γ" is defined by Mathematical Formula 1 above, and the timing at which edge 135 of piston 123 comes into contact with tapered portion 127 is set in the range of π to 4 π/3 (rad). As a result, design can be implemented in a reasonable manner to improve noise characteristics, thereby simplifying the design.

[0074] According to the compressor designed as defined in Mathematical Formulas 1 and 2, when piston 123 is reversed in its inclination direction with respect to the axial center of cylindrical hole 117, outer surface 133b that did not slide with tapered portion 127 before that slides with tapered portion 127. In this case, even when the axial length of outer surface 133 of piston 123 that comes into contact with tapered portion 127 is short, outer surface 133 can be supplied with a sufficient amount of lubricating oil 101 sprinkled all around airtight container 103 horizontally from the upper end of shaft 113.

[0075] As a result, lubricating oil 101 sufficiently supplied to outer surface 133 of piston 123 reduces the contact between outer surface 133 of piston 123 and tapered portion 127, thereby achieving efficiency improvement and noise reduction.

[0076] In addition, piston 123 is provided on its outer surface with concave oil supply groove 131, and cylindrical hole 117 has notch 120 on its peripheral wall so that oil supply groove 131 can be communicated with airtight container 103 near the bottom dead center of piston 123.

[0077] With this structure, lubricating oil 101 sprinkled all around airtight container 103 from the upper end of oil supply hole 128a formed at eccentric shaft portion 111 of shaft 113 can be held in oil supply groove 131 and sufficiently supplied to tapered portion 127 and straight portion 129 of cylindrical hole 117. As a result, the sealing effect of lubricating oil 101 is achieved, which prevents the refrigerant gas leakage. In addition, lubricating oil 101 sufficiently supplied to outer surface 133 of piston 123 reduces the contact between outer surface 133 of piston 123 and tapered portion 127, thereby achieving efficiency improvement and noise reduction.

[0078] According to the present exemplary embodiment, eccentric shaft portion 111 and piston 123 are connected using connecting rod 125 as a connection mechanism, but the same effect can be obtained by using a connection mechanism having a movable portion such as a ball joint.

SECOND EXEMPLARY EMBODIMENT

[0079] A present exemplary embodiment has the same structure as the first exemplary embodiment except that bearing 119 and compression space 115 are arranged differently. Therefore, the following description of the present exemplary embodiment will be mainly focused on the difference.

[0080] Fig. 10 is a longitudinal sectional view of an essential part of a compression section, including its design dimensions, of a hermetic compressor according to the present exemplary embodiment. Fig. 11 is a cross sectional view of the essential part of the compression section, including its design dimensions, of the hermetic compressor.

[0081] As shown in Figs. 10 and 11, in the present exemplary embodiment, bearing 119 and compression space 115 are arranged in such a manner that third center line 142 and second center line 143 cross each other. Third center line 142 is parallel with first center line 141 representing the axial center of bearing 119. Second center line 143 represents the axial center of compression space 115. In Fig. 11, each of first and third center lines 141 and 142 is illustrated as a dot because Fig. 11 is a cross sectional view.

[0082] In the present exemplary embodiment, second center line 143 and offset line 144, which passes through first center line 141 and is parallel with second center line 143 have a distance (hereinafter, offset distance) "s" therebetween. Thus, bearing 119 is arranged offset with respect to compression space 115. The first exemplary embodiment does not have this offset.

[0083] In the present exemplary embodiment shown in Fig. 10, shaft 113 rotates clockwise when viewed from above in Fig. 1. Therefore, the offset arrangement of bearing 119 and compression space 115 plays a role in reducing the sliding loss between cylinder block 121 and piston 123. The offset distance "s" is one of the design dimensions in the present exemplary embodiment, additional to the design dimensions of the first exemplary embodiment. More specifically, the offset distance "s" is designed in the range of 1 to 4 mm, and is designed to be 2 mm in a hermetic compressor for a refrigerator.

[0084] In the present exemplary embodiment, too, the angle "α" formed by the axial center of compression space 115 and tapered portion 127 is defined by Mathematical Formula 2 described in the first exemplary embodiment.

[0085] More specifically, the angle "α" is set based on the following design dimensions: the inner diameter D1 of cylindrical hole 117, the outer diameter D2 of piston 123, the length L1 of straight portion 129, the main sliding surface length L2, the eccentricity "e", and the rotation angle "θ" of main shaft portion 109, which are defined in the first exemplary embodiment, and also offset distance "s".

[0086] Further more specifically, the angle "α" is set in the range obtained by multiplying a coefficient in the range of 0.4 to 2.0 by the dimension value "γ". The dimension value "γ" is obtained by dividing the dimensional numerical value 3/2 of the difference (D1 - D2) between the inner diameter D1 of cylindrical hole 117 and the outer diameter D2 of piston 123 by the coordinate position {L1 - L2 + 2A} of the tip on the top-dead-center side of piston 123 when the top dead center position of piston 123 is 0 (zero).

[0087] The "A" is an assignment expression used to simplify the calculation formula because the offset arrangement of bearing 119 and compression space 115 makes it necessary to correct the coordinate position of the tip of the piston.

[0088] More specifically, as shown in Mathematical Formula 4, the offset distance "s" is taken into consideration in addition to the eccentricity "e".

[0089] The dimensional numerical value 3/2 is a value derived from the design dimensions (values) to find the coordinates of the tip position of piston 123 in cylindrical hole 117 in the same manner as in the first exemplary embodiment.

[0090] In other words, according to the present exemplary embodiment, because bearing 119 is arranged offset with respect to compression space 115, the angle "α" is defined by Mathematical Formula 2 shown in the first exemplary embodiment, which is based on the dimension value "γ" expressed by Mathematical Formula 3.

$$\gamma = \{3(D1 - D2)/2\}/\{L1 - L2 + 2A\} \qquad \text{Mathematical Formula 3}$$

$$A = \sqrt{\{(e^2(1 - \cos\theta)^2 - s^2\}} \qquad \text{Mathematical Formula 4}$$

[0091] As described above, according to the present exemplary embodiment, bearing 119 is arranged offset with respect to compression space 115. This makes it possible to reduce the sliding loss between cylinder block 121 and piston 123 in addition to the effect of the first exemplary embodiment.

[0092] As described hereinbefore, the hermetic compressor of the present invention includes an airtight container having lubricating oil therein; an electric element; and a compression element, the compression element being driven by the electric element, and the electric element and the compression element being housed in the airtight container. The compression element includes a shaft, a cylinder block, a piston, and a connection mechanism. The shaft has a main shaft portion and an eccentric shaft portion, the main shaft portion being rotatable by the electric element, and the eccentric shaft portion moving in unison with the main shaft portion. The cylinder block has a cylindrical hole and a bearing, the cylindrical hole forming a compression space, and the bearing supporting the main shaft portion. The piston is reciprocable in the cylindrical hole. The connection mechanism connects the eccentric shaft portion and the piston. The cylindrical hole includes a tapered portion so as to increase an inner diameter thereof from the top dead center to the bottom dead center of the piston, and the piston is reversed in the inclination direction thereof with respect to the axial center of the cylindrical hole in the initial stage of the compression stroke.

[0093] This structure reduces the sliding resistance, that is, the sliding loss between the piston and the cylindrical hole. This structure also reduces the load when the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. The load reduction can be achieved because in the initial stage of the compression stroke, the end face on the compression space side of the piston is subjected to only a small compressive load. This reduces the contact between the piston and the tapered portion at the time of the reversal, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke. This results in a reduction in the contact when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole, thereby achieving sliding loss reduction, and hence, achieving efficiency improvement and noise reduction.

[0094] According to the present invention, the piston is revered in its inclination direction with respect to the axial center of the cylindrical hole at the moment when the edge on the compression space side of the piston comes into contact with the tapered portion.

[0095] With this structure, when the edge on the compression space side of the piston comes into contact with the tapered portion, this contact is likely to cause the piston to be reversed in its inclination direction with respect to the axial center of the cylindrical hole. Even in such a case, however, it is possible to reduce the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole, and hence, to achieve efficiency improvement and noise reduction.

[0096] According to the present invention, the cylindrical hole includes the straight portion designed as follows. The straight portion has a uniform inner diameter in the axial direction, and is formed in the region which is adjacent to the tapered portion and corresponds to the upper end on the compression space side of the piston when the piston is near the top dead center position.

[0097] This structure advances the timing at which the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole as early as the initial stage of the compression stroke, much earlier than the middle or later stage. In the initial stage of the compression stroke, the end face on the compression space of the piston is subjected to only a small compressive load. Therefore, this structure also reduces the load when the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. This reduces the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole, thereby achieving efficiency improvement and noise reduction. When the outer surface of piston 123 is moving toward the top dead center in the compression stroke, there is almost no leakage of refrigerant gas, and piston has low sliding resistance. As the compression stroke proceeds, and the piston comes close to the top dead center position, the refrigerant gas leakage due to an increased pressure to compress the refrigerant gas can be reduced as compared with the case in which the tapered portion is formed along the entire length of the cylindrical hole, thereby obtaining high freezing capacity.

[0098] According to the present invention, assume that the axial length of the straight portion is L1, the minimum inner diameter of the compression space is D1, the outer diameter of the piston is D2, the eccentricity of the eccentric shaft portion with respect to the main shaft portion is "e", the distance from the connection center of the connection mechanism and the piston to the compression-space-side end face of the piston is L2, the rotation angle of the main shaft portion whose rotation angle is 0 (zero) degrees when the piston is in the top dead center position is "θ", and the angle formed by the axial center of the compression space and the tapered portion is "α". In this case, the angle "α" defines the dimension value "γ" expressed by Mathematical Formula 1 based on the design dimensions: the inner diameter D1 of the cylindrical hole, the outer diameter D2 of the piston, the length L1 of the straight portion, the main sliding surface length L2, the eccentricity "e", and the rotation angle "θ". The angle "α" is defined by Mathematical Formula 2 which is based on the dimension value "γ".

[0099] With this structure, the design dimensions of the hermetic compressor that are involved in the behaviors of the piston can be determined in such a manner as to reduce the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole. As a result, this reduces the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke.

[0100] As an example of a specific design, the angle "α" formed by the axial center of the compression space and the tapered portion can be determined by setting the rotation angle "θ" of the main shaft portion with respect to which the piston is reversed in its inclination direction, and also by setting the following design values: the inner diameter D1 of the cylindrical hole, the outer diameter D2 of the piston, the length L1 of the straight portion, the main sliding surface length L2, and the eccentricity "e".

[0101] According to the present invention, when the piston is in the bottom dead center position, at least the bottom end of the piston is exposed from the cylindrical hole, and the rotation angle "θ" of the main shaft portion is in the range of π to 4 $\pi/3$ (rad).

[0102] With this structure, the bottom end of the piston is exposed from the cylindrical hole when the piston is returned to the bottom dead center position. This allows a large amount of lubricating oil to be supplied and held, thereby reducing the sliding loss between the piston and the cylindrical hole, and hence, achieving efficiency improvement. Furthermore, when the piston is reversed in its inclination direction, the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. In this case, even when the axial length of the outer surface of the piston that comes into contact with the tapered portion is short, a sufficient amount of lubricating oil is supplied. As a result, the lubricating oil can reduce the contact between the outer surface of the piston and the tapered portion, thereby achieving efficiency improvement and noise reduction.

[0103] According to the present invention, the piston is provided on its outer surface with the concave oil supply groove, which is communicated with the airtight container near the bottom dead center of the piston.

[0104] With this structure, a sufficient amount of lubricating oil is supplied to the cylindrical hole so as to achieve the sealing effect of the lubricating oil, thereby reducing the refrigerant gas leakage. Furthermore, the lubricating oil lubricates the sliding portion, thereby achieving a hermetic compressor having high freezing capacity and reliability. In addition, when the piston is reversed in its inclination direction, the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. In this case, even when the axial length of the outer surface of the piston that comes into contact with the tapered portion is short, the outer surface is supplied with a sufficient amount of lubricating oil. As s result, the lubricating oil reduces the contact between the outer surface of the piston and the tapered portion, and ensures the sealing between the outer surface of the piston and the tapered portion, thereby achieving efficiency improvement and noise reduction.

[0105] According to the present invention, the bearing and the compression space are arranged in such a manner that the third center line and the second center line cross each other. The third center line is parallel with the first center line representing the axial center of the bearing. The second center line represents the axial center of the compression space.

[0106] This structure reduces the sliding resistance, that is, the sliding loss between the piston and the cylindrical hole. This structure also reduces the load when the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. The load reduction can be achieved because in the initial stage of the compression stroke, the end face on the compression space of the piston is subjected to only a small compressive load. This reduces the contact between the piston and the tapered portion at the time of the reversal, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke. In other words, this reduces the contact when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole, thereby achieving efficiency improvement and noise reduction. Furthermore, the offset arrangement of the bearing and the compression space reduces the sliding loss between the cylinder block and the piston.

[0107] According to the present invention, assume that the axial length of the straight portion is L1, the minimum inner diameter of the compression space is D1, the outer diameter of the piston is D2, the eccentricity of the eccentric shaft portion with respect to the main shaft portion is "e", the distance from the connection center of the connection mechanism and the piston to the compression-space-side end face of the piston is L2, the rotation angle of the main shaft portion whose rotation angle is 0 (zero) degrees when the piston is in the top dead center position is "θ", the offset distance (the distance between the first and third center lines) is "s", and the angle formed by the axial center of the compression space and the tapered portion is "α". In this case, the angle "α" is defined by Mathematical Formula 2 which is based on the dimension value "γ" expressed by Mathematical Formula 3 based on the design dimensions: the inner diameter D1 of the cylindrical hole, the outer diameter D2 of the piston, the length L1 of the straight portion, the main sliding surface length L2, the eccentricity "e", the rotation angle "θ", and the offset distance "s".

[0108] With this structure, even when the bearing and the compression space are arranged offset with respect to each other, the design dimensions of the hermetic compressor that are involved in the behaviors of the piston can be determined in such a manner as to reduce the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction with respect to the axial center of the cylindrical hole. As a result, it becomes possible to specifically design a hermetic compressor which reduces the contact between the outer surface of the piston and the tapered portion when the piston is reversed in its inclination direction, as compared with the case in which the piston is reversed in its inclination direction in the middle or later stage of the compression stroke. As an example of a specific design, the angle "α" formed by the axial center of the compression space and the tapered portion can be determined by setting the rotation angle "θ" of the main shaft portion with respect to which the piston is reversed in its inclination direction, and also by setting the following design values: the inner diameter D1 of the cylindrical hole, the outer diameter D2 of the piston, the length L1 of the straight portion, the main sliding surface length (the distance from the center of the piston pin to the compression-space-side end face of the piston) L2, the eccentricity "e", and the offset distance "s".

[0109] According to the present invention, when the piston is in the bottom dead center position, at least the bottom end of the piston is exposed from the cylindrical hole, and the rotation angle "θ" of the main shaft portion is in the range of π to 4 $\pi/3$ (rad).

[0110] With this structure, even when the bearing and the compression space are arranged offset with respect to each other, the bottom end of the piston is exposed from the cylindrical hole when the piston is returned to the bottom dead center position. This allows a large amount of lubricating oil to be supplied and held, thereby reducing the sliding loss between the piston and the cylindrical hole, and hence, achieving efficiency improvement. Furthermore, when the piston is reversed in its inclination direction, the region of the outer surface of the piston that did not slide with the tapered portion before the reversal comes into contact with the tapered portion. In this case, even when the axial length of the outer surface of the piston that comes into the tapered portion is short, a sufficient amount of lubricating oil is supplied. As a result, the lubricating oil can reduce the contact between the outer surface of the piston and the tapered portion, thereby achieving efficiency improvement and noise reduction.

INDUSTRIAL APPLICABILITY

[0111] Thus, the hermetic compressor of the present invention, which has low sliding loss of the piston, low input, high efficiency, and low collision impact with low noise, is applicable to any device having a refrigeration cycle such as domestic refrigerators, dehumidifiers, showcases, and vending machines.

Reference marks in the drawings

[0112]

101	lubricating oil
103	airtight container
105	electric element
105a	stator
105b	rotor
107	compression element
109	main shaft portion
111	eccentric shaft portion
113	shaft
115	compression space
117	cylindrical hole
119	bearing
120	notch
121	cylinder block
123	piston
125	connecting rod
127	tapered portion
128	oil supply passage
128a	oil supply hole
129	straight portion
131	oil supply groove
133, 133a, 133b	outer surface
134	compression-space-side end face
135	edge

136	piston pin
137	balance weight
139	valve plate
141	first center line
142	third center line
143	second center line
144	offset line

Claims

1. A hermetic compressor comprising:

an airtight container having lubricating oil therein;
an electric element; and
a compression element, the compression element being driven by the electric element, and the electric element and the compression element being housed in the airtight container;
the compression element including:

a shaft having a main shaft portion and an eccentric shaft portion, the main shaft portion being rotatable by the electric element, and the eccentric shaft portion moving in unison with the main shaft portion;
a cylinder block having a cylindrical hole and a bearing, the cylindrical hole forming a compression space, and the bearing supporting the main shaft portion;
a piston reciprocable in the cylindrical hole; and
a connection mechanism for connecting the eccentric shaft portion and the piston, wherein
the cylindrical hole includes a tapered portion so as to increase an inner diameter thereof from a top dead center to a bottom dead center of the piston, and
the piston is reversed in an inclination direction thereof with respect to an axial center of the cylindrical hole in an initial stage of a compression stroke.

2. The hermetic compressor of claim 1, wherein
the piston is reversed in the inclination direction thereof with respect to the axial center of the cylindrical hole at a moment when the piston comes into contact with the tapered portion at an edge on a compression space side thereof.

3. The hermetic compressor of claim 1, wherein
the cylindrical hole includes a straight portion having a uniform inner diameter in an axial direction, the straight portion being formed in a region adjacent to the tapered portion and corresponding to an upper end on the compression space side of the piston when the piston is near the top dead center position.

4. The hermetic compressor of claim 3, wherein when the straight portion has an axial length L1,
the compression space has a minimum inner diameter D1,
the piston has an outer diameter D2,
the eccentric shaft portion has an eccentricity "e" with respect to the main shaft portion,
a connection center of the connection mechanism and the piston, and an end face on the compression space side of the piston have a distance L2 therebetween,
the main shaft portion whose rotation angle is 0 (zero) degrees when the piston is in the top dead center position has a rotation angle "θ", and
the axial center of the compression space and the tapered portion form an angle "α" therebetween,
the angle "α" and a dimension value "γ" satisfy Mathematical Formula 2, the dimension value "γ" being expressed by Mathematical Formula 1 based on the D1, the D2, the L1, the L2, the "e", and the "θ";

$$\gamma = \{3(D1 - D2)/2\}/\{L1 - L2 + 2e(1 - \cos\theta)\} \qquad \text{Mathematical Formula 1}$$

$$0.4\gamma \leq \tan(\alpha) \leq 2.0\gamma \qquad \text{where } \alpha > 0 \qquad \text{Mathematical Formula 2}$$

5. The hermetic compressor of claim 4, wherein
when the piston is in the bottom dead center position, at least a bottom end of the piston is exposed from the cylindrical hole, and
the rotation angle "θ" of the main shaft portion is in a range of π to 4 π/3 (rad).

6. The hermetic compressor of claim 1, wherein
the piston is provided on an outer surface thereof with a concave oil supply groove, the oil supply groove being communicated with the airtight container near the bottom dead center of the piston.

7. The hermetic compressor of claim 3, wherein
the bearing and the compression space are arranged in such a manner that a third center line and a second center line cross each other, the third center line being parallel to a first center line representing an axial center of the bearing, and the second center line representing an axial center of the compression space.

8. The hermetic compressor of claim 7, wherein
when the straight portion has an axial length L1, the compression space has a minimum inner diameter D1, the piston has an outer diameter D2, the eccentric shaft portion has eccentricity "e" with respect to the main shaft portion, a connection center of the connection mechanism and the piston and an end face on the compression space side of the piston have a distance L2 therebetween, the main shaft portion whose rotation angle is zero degrees when the piston is in the top dead center position has a rotation angle "θ", the first center line and the third center line have a distance "s" therebetween, and the axial center of the compression space and the tapered portion form an angle "α" therebetween, the angle "α" and a dimension value "γ" satisfy Mathematical Formula 2, the dimension value "γ" being expressed by Mathematical Formula 3 based on the D1, the D2, the L1, the L2, the "e", the "θ", and the "s";

$$\gamma = \{3(D1 - D2)/2\}/\{L1 - L2 + 2A\} \qquad \text{Mathematical Formula 3,}$$

wherein

$$A = \sqrt{\{(e^2(1 - \cos\theta)^2 - s^2\}} \qquad \text{Mathematical Formula 4}$$

9. The hermetic compressor of claim 8, wherein
when the piston is in the bottom dead center position, at least a bottom end of the piston is exposed from the cylindrical hole, and
the rotation angle "θ" of the main shaft portion is in the range of
π to 4 π/3 (rad).

FIG. 1

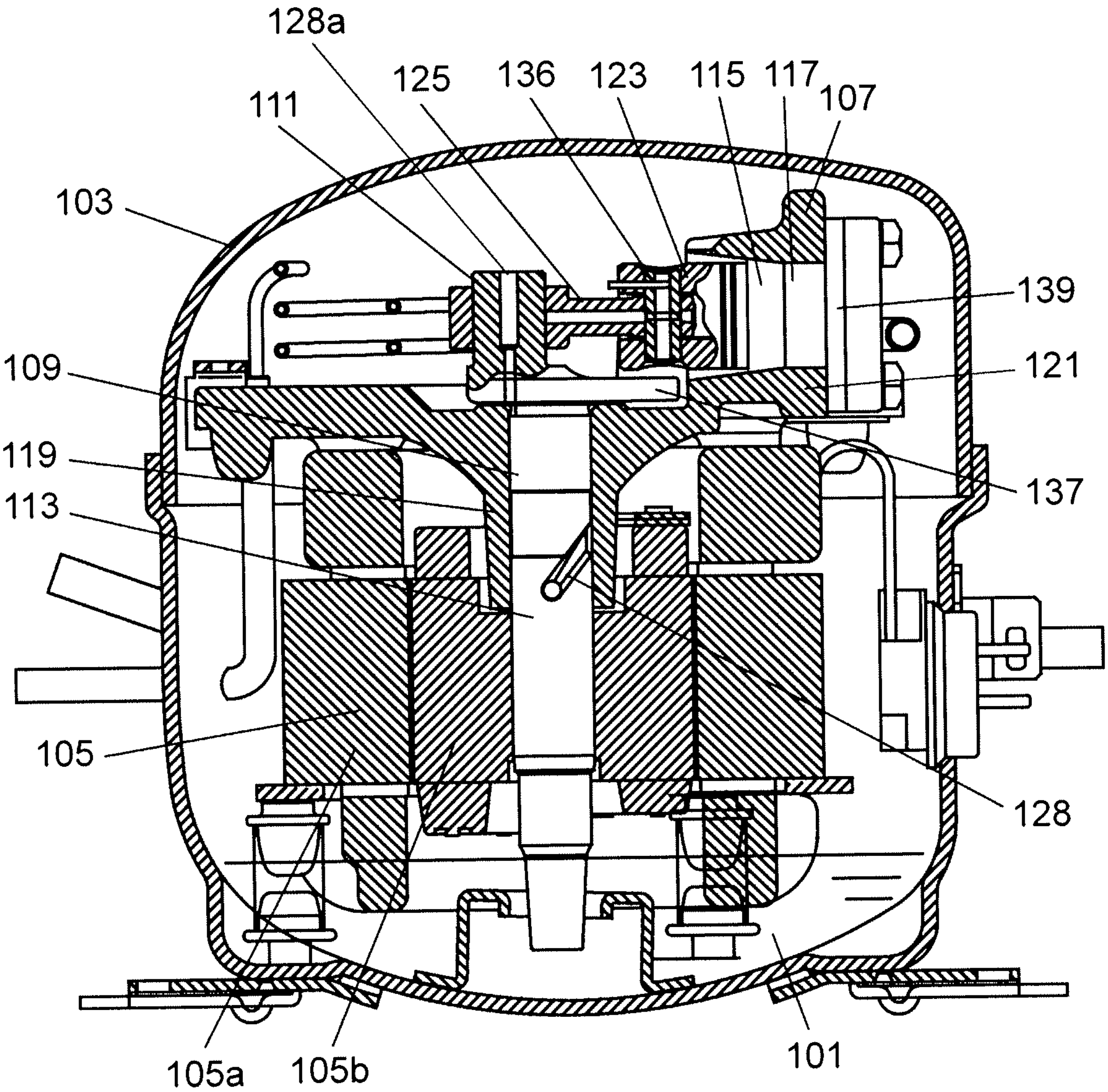

128a
125
136
123
115
117
107
111
103
139
109
121
119
137
113
105
128
105a
105b
101

FIG. 2

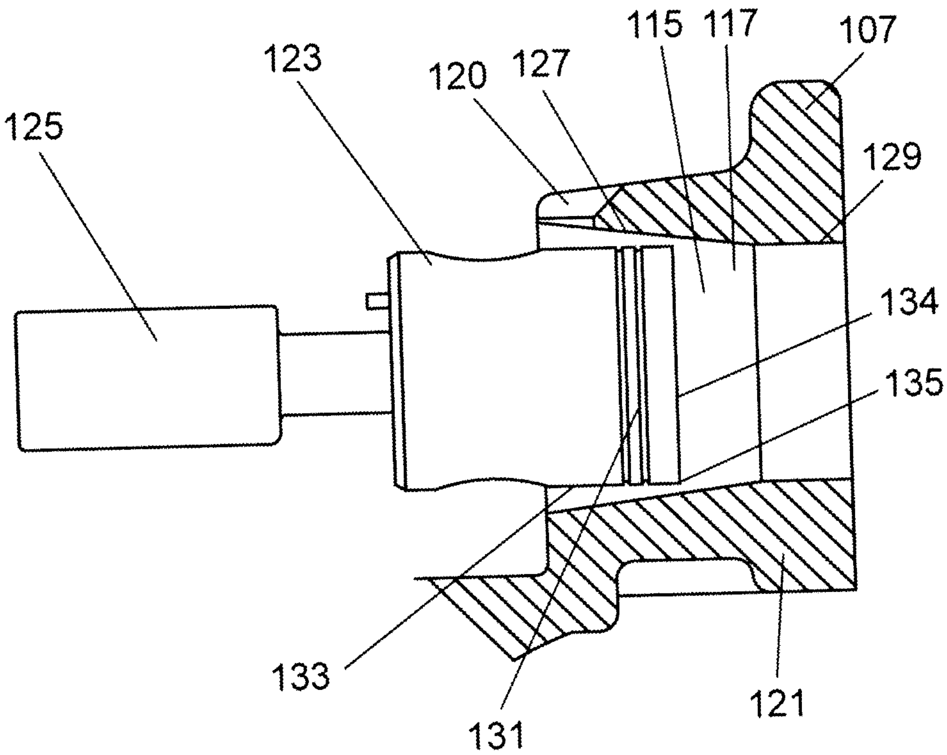


FIG. 3

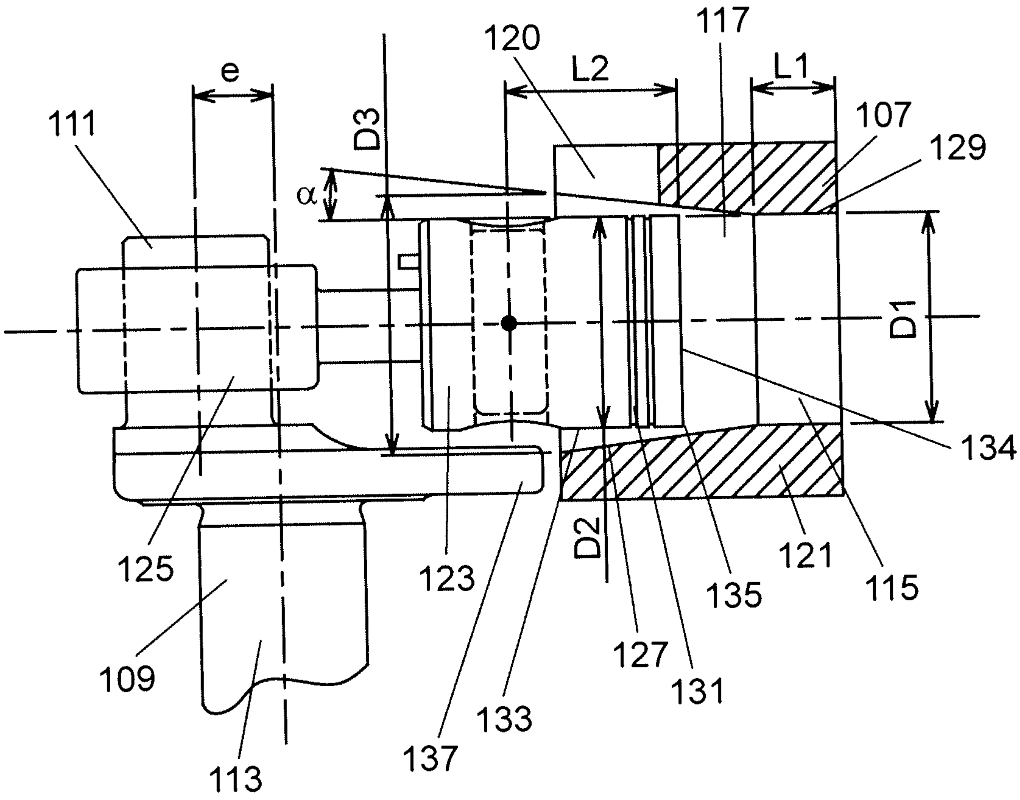

120
117
L2
L1
e
107
129
111
D3
α
D1
134
125
123
D2
135
121
115
127
131
109
137
133
113

FIG. 4

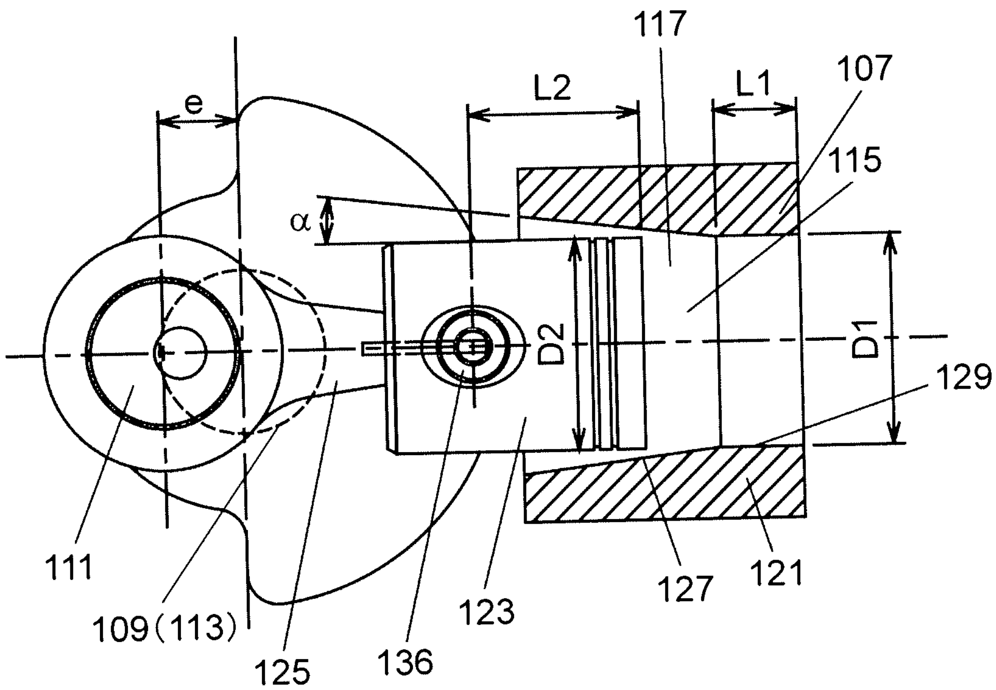

117
107
L2
L1
115
e
α
D2
D1
129
111
127
121
123
109(113)
125
136

FIG. 5A

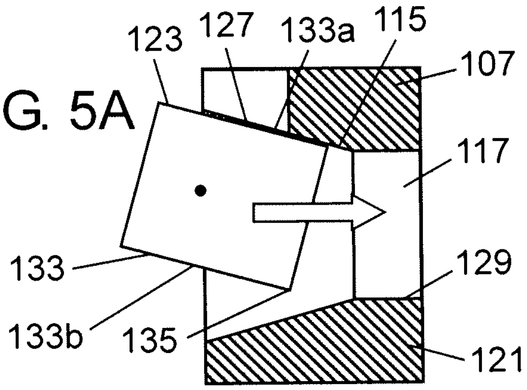


FIG. 5B

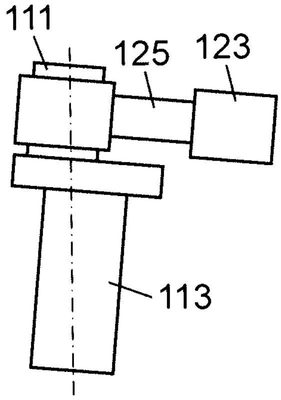


FIG. 6A

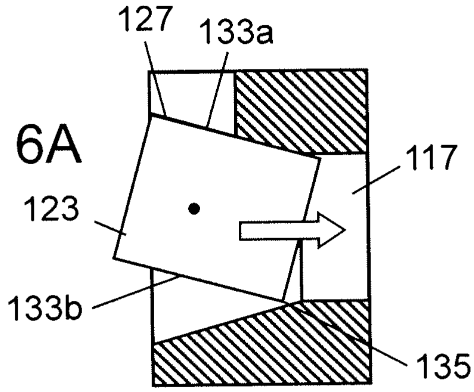


FIG. 6B

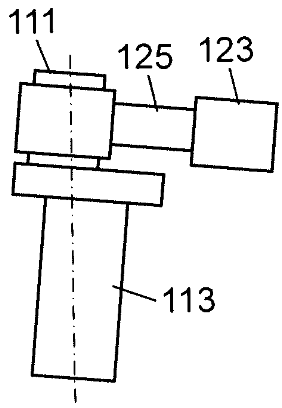


FIG. 7A

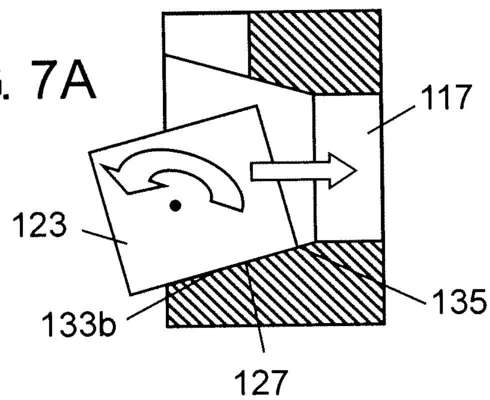


FIG. 7B

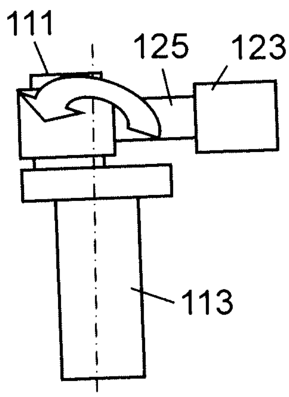


FIG. 8A

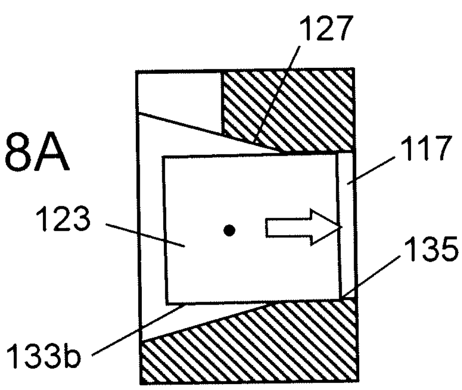


FIG. 8B

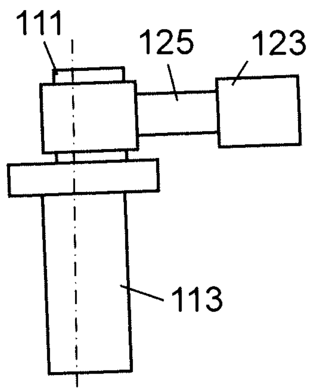


FIG. 9

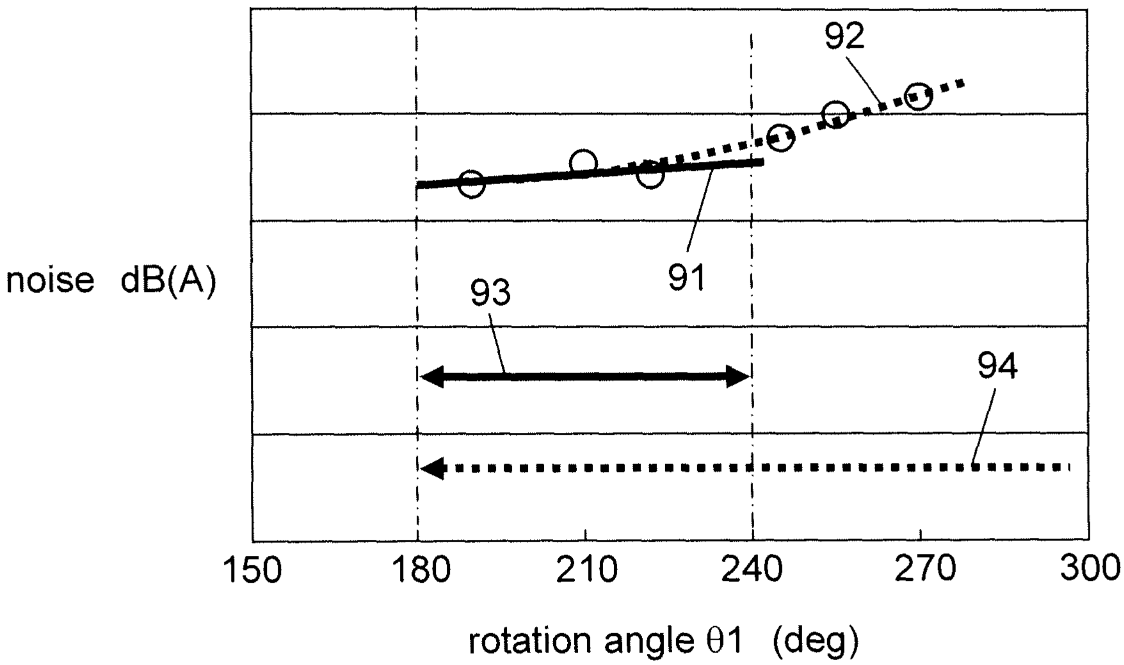

92
91
93
94
noise dB(A)
150
180
210
240
270
300
rotation angle θ1 (deg)

FIG. 10

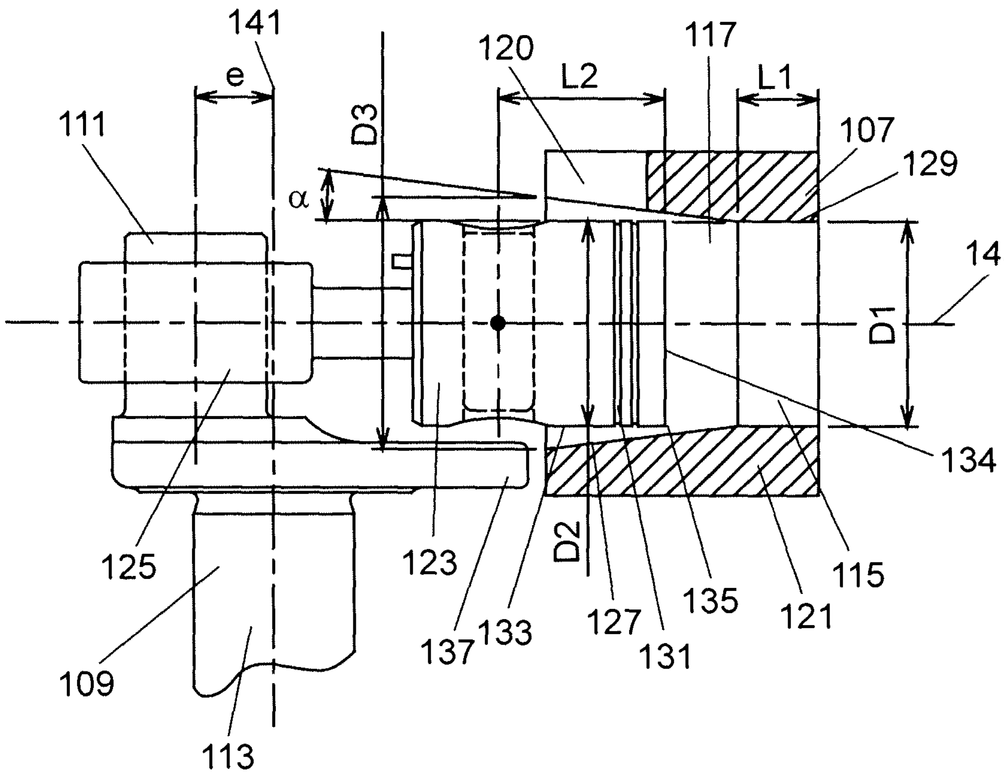

141
120
117
e
L2
L1
111
D3
107
129
α
143
D1
134
125
123
D2
115
137
133
127
131
135
121
109
113

FIG. 11

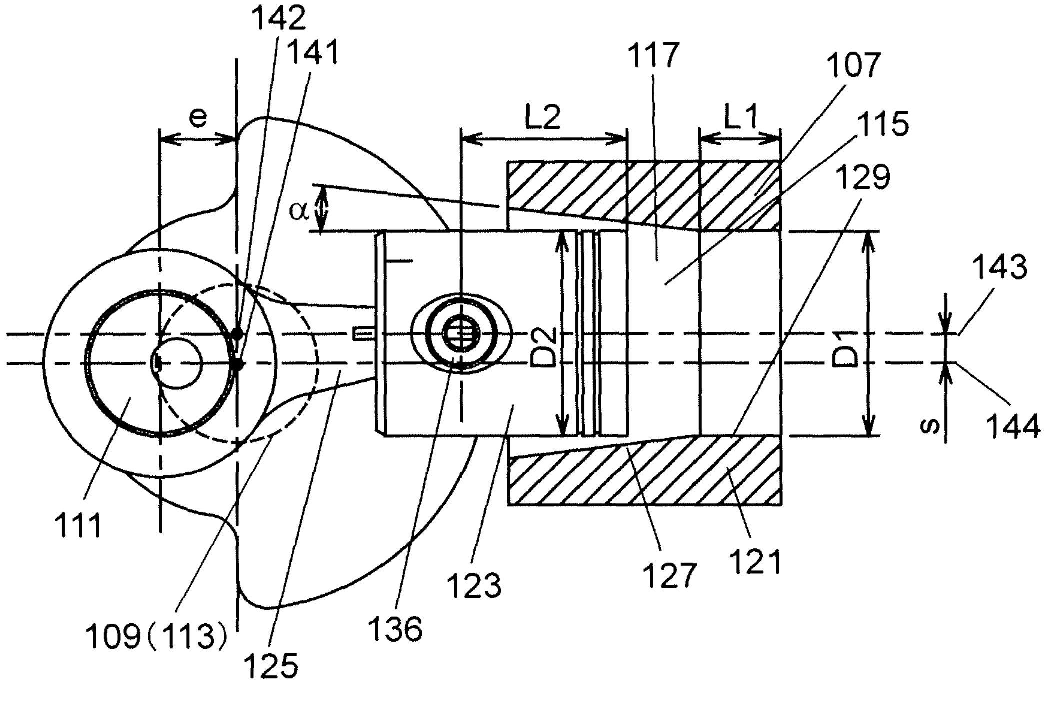

142
141
117
107
e
L2
L1
115
129
α
143
D2
D1
s
144
111
127
121
123
109(113)
136
125

FIG. 12A

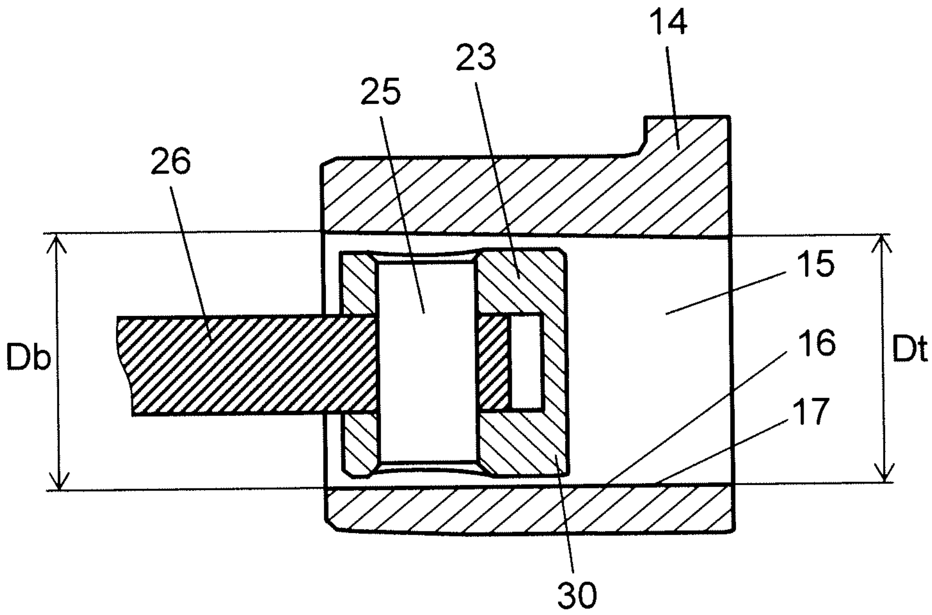
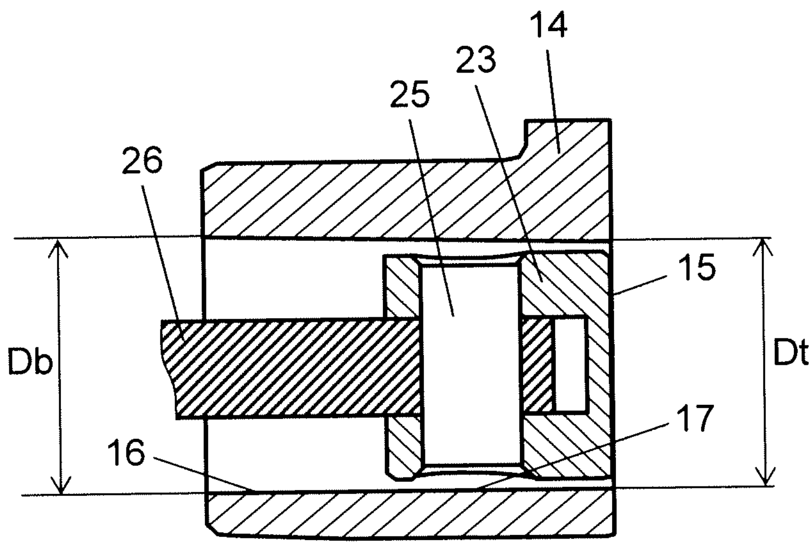


FIG. 12B



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2009/005449

A. CLASSIFICATION OF SUBJECT MATTER
F04B39/12(2006.01)i, *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)
F04B39/12, F04B39/00

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	1996-2009
Kokai Jitsuyo Shinan Koho	1971-2009	Toroku Jitsuyo Shinan Koho	1994-2009

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 A	JP 2002-89450 A (Sanyo Electric Co., Ltd.), 27 March 2002 (27.03.2002), claim 1; fig. 1 to 4 (Family: none)	1-3,7 6 4,5,8,9
Y	JP 2008-101532 A (Matsushita Electric Industrial Co., Ltd.), 01 May 2008 (01.05.2008), claim 1; fig. 1 (Family: none)	6

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

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"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

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"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 04 November, 2009 (04.11.09)	Date of mailing of the international search report 17 November, 2009 (17.11.09)
Name and mailing address of the ISA/ Japanese Patent Office Facsimile No.	Authorized officer Telephone No.

Form PCT/ISA/210 (second sheet) (April 2007)

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- JP 2002089450 A **[0012]**