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(54) **SCROLL COMPRESSOR**

SPIRALVERDICHTER

COMPRESSEUR À VOLUTES

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Description

Technical Field

[0001] The present invention relates to a scroll-type compressor which is used in a refrigerating cycle of an air conditioning device for a vehicle, and more particularly to a scroll-type compressor having the constitution where an eccentric shaft which is provided to an end portion of a drive shaft is pivotally supported on a boss portion of a revolving scroll member by way of a bushing and a radial bearing.

Background Art

[0002] A scroll-type compressor includes: a fixed scroll member which has an end plate and a spiral wall formed on the end plate in an erected manner; and a revolving scroll member which is arranged so as to face the fixed scroll member in an opposed manner and has an end plate and a spiral wall formed on the end plate in an erected manner. By combining the respective spiral walls of pair of scroll members with each other and by making the revolving scroll member perform the rocking rotation (revolving motion) in a state where the rotation of the revolving scroll member is restricted, a compression chamber formed between the spiral walls of both scroll members is moved toward the center while a volume of the compression chamber is gradually decreased thus compressing a working fluid.

[0003] In such a scroll-type compressor, the revolving scroll member is configured such that a boss portion is formed on a back surface of the end plate and an eccentric shaft which is provided to one end of a drive shaft is pivotally supported on the boss portion by way of a radial bearing so that the revolving scroll member is supported in a state where the revolving scroll member performs the rocking rotation (revolving motion) about an axis of the drive shaft.

[0004] Here, when a clearance exists between side surfaces of the spiral walls of both scroll members, a high pressure gas leaks to an outer peripheral portion side (upstream side) from the compression chamber on a center side (downstream side) and hence, compression efficiency is deteriorated. Accordingly, there has been adopted a mechanism where a bushing is disposed between the eccentric shaft of the drive shaft and the radial bearing (bearing portion) mounted on the boss portion of the revolving scroll member as an intermediate member, and a revolution radial amount of the revolving scroll member is made variable such that by making use of a component force of a compression reaction force which acts on the revolving scroll member, the revolving scroll member is biased in the radial direction of the fixed scroll member so as to bring a side surface of the spiral wall of the revolving scroll member and a side surface of the spiral wall of the fixed scroll member into contact with each other without being separated from each other (patent literatures 1 to 3).

[0005] Among the scroll-type compressors disclosed in patent literatures 1 to 3, the scroll-type compressor disclosed in patent literature 1 is, as shown in Fig. 6, configured such that a bushing 103 is fitted in a boss portion 102 of a revolving scroll member 101, and the bushing 103 is of a so-called slide-type bushing which is non-rotatable relative to an eccentric shaft 105 provided to one end of a drive shaft 104 and is slightly movable in the radial direction. A curved surface portion 106 is formed on the eccentric shaft 105 so that even when the deformation by deflection is generated in the drive shaft 104 due to a compressive force or a centrifugal force of the revolving scroll member 101, the occurrence of a non-uniform contact on a bearing portion is prevented. This publication also discloses the constitution where a balance weight 107 is mounted on the drive shaft 104 at a position axially displaced from the bushing 103.

[0006] The scroll-type compressor disclosed in patent literature 2 is, as shown in Fig. 7, configured such that an eccentric shaft 112 provided to one end of a drive shaft 111 is pivotally supported on a boss portion 117 of a revolving scroll member 115 by way of a bushing 113 and a radial bearing 114, wherein a balance weight 116 is supported on the bushing 113 by being press-fitted on an outer peripheral surface of the bushing 113 so that the balance weight 116 is rotatable together with the bushing 113. The balance weight 116 is arranged so as to project over the outer side of the boss portion 117 of the revolving scroll member 115 in a radial direction such that a point of action where a centrifugal force acts on the balance weight 116 is positioned as close as possible to a point of action where a centrifugal force acts on the revolving scroll member 115.

[0007] The scroll-type compressor disclosed in patent literature 3 is, as shown in Fig. 8, configured such that a bushing 122 with which a balance weight 121 is integrally formed is assembled to an eccentric shaft 124 provided to a distal end of a drive shaft 123, and the bushing 122 is pivotally supported on a boss portion 126 formed on a back surface of a revolving scroll member 125 by way of a radial bearing 127 thus making the revolving scroll member 125 perform the revolving motion. While the balance weight 121 is arranged so as to project over an outer side of the boss portion 126 of the revolving scroll member 125 in the same manner as patent literature 2, the balance weight 121 is also arranged such that the balance weight 121 further projects toward a side opposite to the revolving scroll member 125 (motor side). In this manner, the scroll-type compressor disclosed in patent literature 3 is designed such that the projecting of the balance weight 121 in the radial direction is suppressed thus preventing the revolving scroll member 125 from becoming large-sized.

Citation List

Patent Literature

[0008]

PTL 1: JP-A-8-42467
PTL 2: JP-A-8-42477
PTL 3: JP-A-2010-196630

[0009] The European patent application EP0921316 discloses a further example of a scroll compressor of the background art.

Summary of Invention

Technical Problem

[0010] In the constitution disclosed in patent literature 1 (Fig. 6), the portion on which the balance weight 107 is mounted is not the bushing 103 but the drive shaft 104. Accordingly, centrifugal forces are cancelled each other and balanced in the compressor as a whole. However, the balance weight 107 does not act so as to cancel a centrifugal force of the revolving scroll member 101 which acts so as to push out the revolving scroll member 101 radially outward. Accordingly, a contact load between a spiral wall 101a of the revolving scroll member 101 and a spiral wall 108a of the fixed scroll member 108 is increased thus giving rise to drawbacks such as the increase in the rolling resistance between the spiral walls 101a, 108a at the high speed rotation or the loss of reliability of the spiral walls.

[0011] In the constitution disclosed in patent literature 2 (Fig. 7), the balance weight 116 is formed so as to project over the outside of the boss portion 117 and hence, a rotation prevention mechanism 118 is disposed radially outside the balance weight 116 so as to prevent the interference with the balance weight 116. Accordingly, the revolving scroll member 115 becomes large-sized thus hampering the miniaturization of the compressor. Further, a weight of the revolving scroll member 115 is also increased and hence, there also arises a drawback that it is necessary to make the balance weight 116 large-sized to take a balance with the revolving scroll member 115.

[0012] In the constitution disclosed in patent literature 3 (Fig. 8), the balance weight 121 is formed such that the balance weight 121 projects also toward the side opposite to the revolving scroll member 125 (motor side) and hence, a centrifugal force of the balance weight 121 acts on an end portion of the bushing 122 which projects from the radial bearing 127 whereby an axis of the bushing 122 tends to be inclined with respect to an axis of the eccentric shaft 124. Accordingly, to decrease such inclination of the bushing 122, conventionally, the scroll-type compressor is designed such that a clearance between the radial bearing 127 and an outer peripheral surface of the bushing 122 is set as small as possible so as to make the radial bearing 127 support a force by which the bushing 122 is inclined. However, at the high speed rotation, a local contact of an inner ring of the bearing exceeds a yield strength thus giving rise to a drawback that flaking occurs.

[0013] Particularly, in the constitution of a scroll-type compressor where a needle bearing is used as a radial bearing, under a conventional design concept, as shown in Fig. 9A, the scroll-type compressor is designed such that a clearance between an inner peripheral surface of a radial bearing 24 and an outer peripheral surface of a bushing 23 (a clearance between a roller 24a and an outer peripheral surface of the bushing 23 in the case of the needle bearing) C is made as small as possible (such that the clearance C becomes smaller than a clearance A between the bushing 23 and an eccentric shaft 17). However, a balance weight 32 is mounted on an end portion of the bushing 23 so that the center of gravity of the balance weight 32 is displaced from the center of the eccentric shaft 17 provided to an end portion of a drive shaft 12 and hence, as shown in Fig. 9B, an axis of the bushing 23 is inclined with respect to the axis of the eccentric shaft 17. In such a state, a centrifugal force of the balance weight 32 acts on a revolving scroll member 22 by way of the bushing 23 and the radial bearing 24. Since the clearance C is set smaller than the clearance A, to support the bushing 23 which is inclined due to a centrifugal force F1 of the balance weight 32, loads (F2, F3) are received by two portions of an inner surface of the radial bearing 24 (that is, two portions consisting of a portion of the inner surface of the radial bearing 24 in the vicinity of an end portion of the bushing 23 where the balance weight 32 is mounted [a portion of the bushing 23 which is away from a portion of the bushing on which a centrifugal force of the balance weight 32 acts 23 by L1 in the axial direction] and an end portion of the bushing 23 which is away from the balance weight 32 or a portion of the bushing 23 in the vicinity of such a portion) [a portion which is away from a portion of the bushing 23 on which a centrifugal force of the balance weight 32 acts by L2 in the axial direction: L2>L1]). Accordingly, a load F2 is increased by an amount corresponding to the generation of F3 and hence, a local load is remarkably increased at the portion which receives the load F2 whereby flaking is liable to occur.

[0014] The present invention has been made in view of the above, and it is a primary object of the present invention to provide a scroll-type compressor which can, even at the high speed rotation, prevent the occurrence of flaking at a contact portion of a radial bearing which is in contact with a bushing while suppressing the increase in rolling resistance of spiral walls of a pair of scroll members. It is another obj ect of the present invention to realize the miniaturization of the compressor. Solution to Problem

[0015] To achieve the above-mentioned objects, a scroll-type compressor according to the present invention is directed to a scroll-type compressor which includes: a fixed scroll member whose movement in the radial direction with respect to a housing is restricted, the fixed scroll member having an end plate and a spiral wall which is formed on the end plate in an erected manner; a revolving scroll member which is arranged so as to face the fixed scroll member in an opposed manner, the revolving scroll member having an end plate and a spiral wall which is formed on the end plate in an erected manner; a drive shaft which transmits rotational power; an eccentric shaft which is provided to an end portion of the drive shaft at a position offset from an axis of the drive shaft; a radial bearing which is fitted into a boss portion formed on a back surface of the revolving scroll member; a bushing which has an eccentric hole into which the eccentric shaft is inserted, the bushing being fitted on the eccentric shaft by way of the eccentric hole, and the bushing being relatively rotatably supported on the radial bearing; and a balance weight which is mounted on one end portion of the bushing and forms an integral body with the bushing, characterized in that the bushing is configured such that, when the bushing is inclined, an outer peripheral surface of the bushing is brought into contact with the radial bearing only on an end portion side on which the balance weight is mounted.

[0016] Accordingly, when a point at which a centrifugal force of the balance weight acts is a portion displaced from the radial bearing in the axial direction (an end portion of the bushing projecting from the radial bearing), an axis of the bushing tends to be inclined with respect to an axis of the eccentric shaft due to the centrifugal force of the balance weight. However, when the bushing is inclined, an outer peripheral surface of the bushing is brought into contact with the radial bearing only on an end portion side on which the balance weight is mounted and hence, it is possible to prevent the occurrence of a state where a local load at such a contact portion is remarkably increased as an end portion of the bushing away from the balance weight is brought into contact with the radial bearing. Accordingly, an offset load generated in the radial bearing can be decreased and hence, a yield strength of the radial bearing is relatively enhanced whereby the occurrence of flaking which occurs at a contact portion of the radial bearing which is in contact with the outer peripheral surface of the bushing can be suppressed.

[0017] Here, with respect to the specific constitution which brings the bushing into contact with the radial bearing only on an end portion side on which the balance weight is mounted when the bushing is inclined, assuming a clearance between the eccentric shaft and an inner peripheral surface of the eccentric hole formed in the bushing as A, a length of fitting between the eccentric shaft and the inner peripheral surface of the eccentric hole formed in the bushing as B, a clearance between the radial bearing and an outer peripheral surface of the bushing as C, and a length of fitting between the radial bearing and the outer peripheral surface of the bushing as D, the relationship $A/B < C/D$ may preferably be established.

[0018] Due to such a constitution, even when the point at which a centrifugal force of the balance weight acts is a position displaced from the radial bearing in the axial direction, an offset load (a local load at a contact portion between the bushing and the radial bearing) caused by inclining of the bushing can be suppressed to a value which falls within an allowable range. Accordingly, when a rotation prevention mechanism is arranged outside the boss portion of the revolving scroll member in the radial direction, it may be possible to prevent the interference of the balance weight with the rotation prevention mechanism by forming the balance weight in such a manner that the balance weight positively projects in the direction away from the revolving scroll member.

[0019] Due to such a constitution, even when the rotation prevention mechanism is provided outside and in the vicinity of the boss portion of the revolving scroll member, it is possible to mount the balance weight while preventing the interference between the balance weight and the rotation prevention mechanism and hence, a profile of the rotation prevention mechanism can be made small leading to the decrease of an outer diameter of the revolving scroll member.

[0020] A needle bearing may preferably be used as the radial bearing. Although the needle bearing has a compact shape and is light-weighted, a contact face pressure becomes excessively large when a shaft is brought into contact with the needle bearing in a state that the shaft is inclined with respect to an axis of the bearing and hence, the use of the needle bearing is supposed not to be desirable for supporting an axial load or the inclining of the shaft. However, a local load between the radial bearing and the bushing can be decreased as described above and hence, a yield strength of the radial bearing can be relatively enhanced whereby the needle bearing can sufficiently cope with the supporting of an axial load or the inclining of the shaft as the radial bearing. With the use of the needle bearing, a weight of the bearing can be reduced, and a size of the radial bearing in the radial direction can be also made compact so that the balance weight can be also made light-weighted.

Advantageous Effects of Invention

[0021] As has been described above, according to the present invention, in pivotally supporting the eccentric shaft provided to the end portion of the drive shaft on the radial bearing which is mounted in the boss portion of the revolving scroll member by way of the bushing on which the balance weight is integrally formed, when the bushing is inclined, an outer peripheral surface of the bushing is brought into contact with the radial bearing only on the end portion side of the bushing on which the balance weight is mounted. Accordingly, even when the bushing is inclined, there is no possibility that a local load is remarkably increased at a contact portion between the outer peripheral surface of the bushing and the radial bearing and hence, it is possible to prevent the occurrence of flaking at the contact portion between the radial bearing and the bushing. Accordingly, even at the high speed rotation, it is possible to prevent the occurrence of flaking at the contact portion between the radial bearing and the bushing while suppressing the increase in rolling resistance between the spiral walls of a pair of the scroll members.

[0022] Further, by adopting the above-mentioned constitution, even when the balance weight is formed such that the balance weight projects in the direction away from the revolving scroll member, a local load at the contact portion between the outer peripheral surface of the bushing and the radial bearing can be suppressed to a value which falls within an allowable range. Accordingly, by forming the balance weight such that the balance weight projects in the direction away from the revolving scroll member, even when a rotation prevention mechanism is arranged outside and in the vicinity of the boss portion, it may be possible to prevent the interference of the balance weight with the rotation prevention mechanism whereby a size of the rotation prevention mechanism and an outer diameter of the revolving scroll member can be made small.

[0023] Since a local load at the contact portion between the bushing and the radial bearing can be decreased and hence, the needle bearing can be sufficiently used as the radial bearing. With the use of the needle bearing, the revolving scroll member in which the needle bearing is mounted becomes light-weighted, and a weight of the balance weight which is provided for canceling a centrifugal force of the revolving scroll member can be also decreased.

Brief Description of Drawings

[0024]

[Fig. 1] Fig. 1 is a cross-sectional view showing an example of the whole constitution of a scroll-type compressor according to the present invention.

[Fig. 2] Fig. 2A is a cross-sectional view showing a state where a bushing is fitted on an eccentric shaft which is provided to an end portion of a drive shaft, and the bushing is supported on a radial bearing which is mounted on a boss portion of a revolving scroll member, and Fig. 2B is an exploded perspective view of the constitution shown in Fig. 2A.

[Fig. 3] Fig. 3A to Fig. 3C are views showing the bushing, wherein Fig. 3A is a view of the bushing as viewed from one side in the axial direction, Fig. 3B is a cross-sectional side view of the bushing, and Fig. 3C is a view of the bushing as viewed from the other side in the axial direction.

[Fig. 4] Fig. 4A to Fig. 4C are views showing a balance weight, wherein Fig. 4A is a view of the balance weight as viewed from one side in the axial direction, Fig. 4B is a cross-sectional side view of the balance weight, and Fig. 4C is a view of the balance weight as viewed from the other side in the axial direction.

[Fig. 5] Fig. 5A and Fig. 5B are enlarged views showing a state where the eccentric shaft which is provided to the end portion of the drive shaft is pivotally supported on the radial bearing which is mounted on the boss portion of the revolving scroll member by way of the bushing having one end portion on which a balance weight is mounted, wherein Fig. 5A is a view showing a state where the drive shaft is not rotated, and Fig. 5B is a view showing a state where the drive shaft is rotated so that the bushing is inclined.

[Fig. 6] Fig. 6 is a cross-sectional view showing a conventional scroll-type compressor.

[Fig. 7] Fig. 7 is a cross-sectional view showing another conventional scroll-type compressor.

[Fig. 8] Fig. 8 is a cross-sectional view showing still another conventional scroll-type compressor.

[Fig. 9] Fig. 9A and Fig. 9B are enlarged views showing a conventional state where an eccentric shaft which is provided to an end portion of a drive shaft is pivotally supported on a radial bearing which is mounted on a boss portion of a revolving scroll member by way of a bushing having one end on which a balance weight is mounted, wherein Fig. 9A is a view showing a state where the drive shaft is not rotated, and Fig. 9B is a view showing a state where the drive shaft is rotated and the bushing is inclined.

Description of Embodiments

[0025] Hereinafter, a scroll-type compressor according to the present invention is explained in conjunction with drawings.

[0026] In Fig. 1, a scroll-type compressor 1 is an electrically-operated compressor suitable for a refrigerating cycle which uses a refrigerant as a working fluid, wherein, in the inside of a housing 2 which is made of an aluminum alloy, a compression mechanism 3 is arranged on a left side in the drawing, and an electrically-operated motor 4 for driving the compression mechanism 3 is arranged on a right side in the drawing. In Fig. 1, a front side of the compressor 1 is arranged on a right side in the drawing, and a rear side of the compressor is arranged on a left side in the drawing.

[0027] The housing 2 includes: a compression mechanism housing member 2a which houses the compression mechanism 3 therein, an electrically-operated motor housing member 2b which houses the electrically-operated motor 4 for driving the compression mechanism 3 therein; and an inverter housing member 2c which houses an inverter device not shown in the drawing for controlling driving of the electrically-operated motor 4 therein. These housing members are positioned by positioning pins 7 and are fastened to each other in the axial direction using fastening bolts 8.

[0028] In the electrically-operated motor housing member 2b, a partition wall 10 which is integrally formed with a shaft support portion 9a is provided on a side which faces the compression mechanism housing member 2a. Also in the inverter housing member 2c, a partition wall 11 which is integrally formed with a shaft support portion 9b is provided on a side which faces the electrically-operated motor housing member 2b. A drive shaft 12 is rotatably supported on the shaft support portions 9a, 9b of the partition walls 10, 11 by way of bearings 13, 14. Due to the formation of the partition walls 10, 11 which are formed on the electrically-operated motor housing member 2b and the inverter housing member 2c respectively, the inside of the housing 2 is partitioned into: a compression mechanism housing portion 15a which houses the compression mechanism 3 therein; an electrically-operated motor housing portion 15b which houses the electrically-operated motor 4 therein; and an inverter housing portion 15c which houses the inverter device therein in order from a rear side of the housing 2.

[0029] In this embodiment, the inverter housing portion 15c is defined by fixing a lid body 16 to the inverter housing member 2c using bolts or the like not shown in the drawing.

[0030] The compression mechanism 3 is a scroll-type compression mechanism which includes a fixed scroll member 21, and a revolving scroll member 22 which is arranged so as to face the fixed scroll member 21 in an opposed manner. The fixed scroll member 21 is configured such that, while the movement of the fixed scroll member 21 in the axial direction with respect to the housing 2 is allowed, the movement of the fixed scroll member 21 in the radial direction relative to the housing 2 is restricted by a positioning pin 28. The fixed scroll member 21 is constituted of: a circular disc-shaped end plate 21a; a cylindrical outer peripheral wall 21b which is formed on and along an outer periphery of the end plate 21a over the whole circumference in an erected manner toward a front side; and a spiral wall 21c having a spiral shape which is formed on the end plate 21a in an erected manner inside the outer peripheral wall 21b such that the spiral wall 21c extends toward a front side.

[0031] The revolving scroll member 22 is constituted of: a circular disc-shaped end plate 22a; and a spiral wall 22c having a spiral shape which is formed on the end plate 22a in an erected manner such that the spiral wall 22c extends toward a rear side. An eccentric shaft 17 which is provided to a rear end portion of the drive shaft 12 and is disposed eccentrically with respect to an axis of the drive shaft 12 is supported on a boss portion 22b which is formed on a back surface of the end plate 22a in an erected manner by way of a bushing 23 and a radial bearing 24 so that the eccentric shaft 17 is capable of performing the revolving motion about the axis of the drive shaft 12.

[0032] The spiral wall 21c of the fixed scroll member 21 and the spiral wall 22c of the revolving scroll member 22 are meshed with each other so that a compression chamber 25 is defined in a space surrounded by the end plate 21a of the fixed scroll member 21, the spiral wall 21c of the fixed scroll member 21, the end plate 22a of the revolving scroll member 22, and the spiral wall 22c of the revolving scroll member 22.

[0033] An annular thrust race 26 having a thin plate shape is sandwiched between the outer peripheral wall 21b of the fixed scroll member 21 and the partition wall 10, and the fixed scroll member 21 and the partition wall 10 are made to abut to each other by way of the thrust race 26.

[0034] The thrust race 26 is formed using a material having excellent wear resistance, and a center opening through which the boss portion 22b of the revolving scroll member 22 and an Oldham ring 27 described later pass is formed in a center portion of the thrust race 26. Further, the respective positions of the fixed scroll member 21, the thrust race 26, and the electrically-operated motor housing member 2b in the radial direction are restricted by the positioning pin 28 which is inserted into a pin insertion hole formed in the thrust race 26.

[0035] The shaft support portion 9a which is integrally formed with the partition wall 10 of the electrically-operated motor housing member 2b has a through hole at the center thereof, and a diameter of an inner surface of the shaft support portion 9a is increased toward the thrust race 26 in a stepwise manner. In the shaft support portion 9a, in order from a front side thereof remotest from the thrust race 26, a bearing housing portion 31 in which the bearing 13 is housed, a weight housing portion 33 in which a balance weight 32 which is integrally formed with the bushing 23 or is fitted on the bushing in a non-rotatable manner relative to the bushing 23 and is rotatable along with the rotation of the drive shaft 12 as an integral body with the busing 23 (in this embodiment, the balance weight 32 being formed as a body separate from the bushing 23, and being fitted on the bushing 23 in a non-rotatable manner relative to the bushing 23) is housed, and an Oldham housing portion 34 which is formed continuously with the weight housing portion 33 and in which the Oldham ring 27 which prevents the rotation between the revolving scroll member 22 and the Oldham housing portion 34 is housed are formed.

[0036] Accordingly, although the revolving scroll member 22 generates a rotational force due to the rotation of the drive shaft 12, the revolving scroll member 22 performs the revolving motion about the axis of the drive shaft 12 while the rotation of the revolving scroll member 22 is restricted by the Oldham ring 27.

[0037] A suction chamber 35 which sucks a refrigerant introduced from a suction port 40 described later through an intake passage 45 is formed between the outer peripheral wall 21b of the above-mentioned fixed scroll member 21 and an outermost peripheral portion of the spiral wall 22c of the revolving scroll member 22. On a back side of the fixed scroll member 21 in the inside of the housing, a discharge chamber 37 into which a refrigerant gas which is compressed in the compression chamber 25 is discharged through a discharge hole 36 which is formed on the approximately center of the fixed scroll member 21 is formed between a rear end wall of the compression mechanism housing member 2a and the fixed scroll member 21. A refrigerant gas discharged into the discharge chamber 37 is pressure-fed to an external refrigerant circuit through the discharge port 38.

[0038] On the other hand, in an electrically-operated motor housing portion 15b which is formed in the inside of the housing 2 in front of the partition wall 10, a stator 41 and a rotor 42 which constitute the electrically-operated motor 4 are arranged. The stator 41 is constituted of a core 43 having a cylindrical shape and a coil 44 which is wound around the core 43, and is fixed to an inner surface of the housing 2 (the electrically-operated motor housing member 2b). The rotor 42 made of magnet which is housed in the inside of the stator 41 in a rotatable manner is fixedly mounted on the drive shaft 12. The rotor 42 is rotated by a rotational magnetic force generated by the stator 41 so that the drive shaft 12 is rotated. The electrically-operated motor 4 which is formed of a brushless DC motor is constituted of the stator 41 and the rotor 42.

[0039] The suction port 40 through which a refrigerant gas is sucked into the electrically-operated motor housing portion 15b is formed in a side surface of the housing 2 (electrically-operated motor housing member 2b). The suction passage 45 which introduces a refrigerant flown into the electrically-operated motor housing portion 15b from the suction port 40 into the suction chamber 35 is formed through a gap formed between the stator 41 and the housing 2 (electrically-operated motor housing member 2b), a hole formed in the partition wall 10, and a gap formed between the fixed scroll member 21 and the housing 2.

[0040] The inverter device which is housed in the inverter housing member 2c is electrically connected with the stator 41 via a terminal (airtight terminal) 60 which is mounted in a through hole 61 formed in the partition wall 11, and electricity is supplied to the electrically-operated motor 4 from the inverter device.

[0041] Accordingly, when the rotor 42 is rotated with the supply of electricity to the electrically-operated motor 4 so that the drive shaft 12 is rotated, in the compression mechanism 3, the revolving scroll member 22 is rotated around an axis of the eccentric shaft 17 and hence, the revolving scroll member 22 revolves around the axis of the fixed scroll member 21. Since the rotation of the revolving scroll member 22 is prevented by the rotation prevention mechanism which is constituted of the Oldham ring 27, only the revolving motion of the revolving scroll member 22 is allowed.

[0042] Due to the revolving motion of the revolving scroll member 22, the compression chamber 25 is moved toward the center from outer peripheral sides of the spiral walls 21c, 22c of both the scroll members while a volume of the compression chamber 25 is gradually decreased and hence, a refrigerant gas sucked into the compression chamber 25 from the suction chamber 35 is compressed, and the compressed refrigerant gas is discharged into the discharge chamber 37 through the discharge hole 36 which is formed in the end plate 21a of the fixed scroll member 21. Then, the refrigerant gas is fed out to the external refrigerant circuit through the discharge port 38.

[0043] In the above-mentioned constitution, as also shown in Fig. 2 and Fig. 3, the bushing 23 has a columnar shape. An eccentric hole 23a which extends in the axial direction and allows the insertion of the eccentric shaft 17 therein is formed in the bushing 23 at a position offset from the axis of the bushing 23 in the radial direction. A recessed portion 23b whose diameter is set larger than a diameter of the eccentric hole 23a is formed in a revolving scroll member -side end portion of the bushing 23. A weight fitting margin 23c which has a small outer diameter and on which the balance weight 32 is fitted is formed on the periphery of an electrically-operated-motor-side end portion of the bushing 23.

[0044] As also shown in Fig. 4, the balance weight 32 is constituted of a fitting portion 32a which is formed in an annular shape, and a fan-shaped weight body 32b which is integrally formed on the periphery of the fitting portion 32a over a predetermined angular range. The fitting portion 32a is fitted on the outer periphery of the weight fitting margin 23c of the bushing 23 by press-fitting, for example, so that the balance weight 32 is rotatable with the bushing 23.

[0045] The balance weight is formed such that the balance weight projects in the direction that the balance weight approaches the revolving scroll member and also in the direction away from the revolving scroll member. Due to such a constitution, it is possible to ensure a required mass while preventing the balance weight from interfering with the Oldham ring by decreasing a size of the balance weight in the radial direction.

[0046] The eccentric shaft 17 is a shaft having a circular columnar shape. An annular groove 17a is formed on the eccentric shaft 17 at a position in the vicinity of one end of the eccentric shaft 17. An end portion of the eccentric shaft 17 on a side opposite to a side where the annular groove 17a is formed is press-fitted into and fixed to a fitting hole 12a formed in an end surface of the drive shaft 12 which faces the revolving scroll member 22 in an opposed manner. An annular-groove-side end portion of the eccentric shaft 17 is inserted into the eccentric hole 23a of the bushing 23 in a relatively rotatable manner, and is projected into the recessed portion 23b. A snap ring 29 is fitted in the annular groove 17a at the portion of the eccentric shaft 17 projected into the recessed portion 12b. The bushing 23 is mounted on the eccentric shaft 17 in the above-mentioned manner. Accordingly, the bushing 23 is mounted on the eccentric shaft 17 in a relatively rotatable manner with respect to the eccentric shaft 17 while the movement of the bushing 23 in the axial direction is restricted.

[0047] The radial bearing 24 which is fitted into the boss portion 22b of the revolving scroll member 22 is constituted of a needle bearing where a large number of needle-shaped rollers 24a are arranged equidistantly in the circumferential direction, and the bushing 23 can be fitted into the radial bearing 24 in a relatively rotatable manner with a predetermined clearance between an outer peripheral surface of the bushing 23 and the radial bearing 24.

[0048] In this embodiment, the radial bearing 24 is configured such that the number of rollers is set to 14, a length of a roller portion (a length of fitting between the roller and the outer peripheral surface of the bushing 23 in the axial direction) is set to 10mm, and a diameter of the roller is set to 2.5mm. Further, a length of fitting between the bushing and the eccentric shaft in the axial direction is set to 15mm, and a diameter of the eccentric shaft is set to 6mm.

[0049] According to the conventional general designing method, a clearance between a radial bearing and a member which is supported on the radial bearing (bushing in this embodiment) is usually set as small as possible by taking into account tolerance of the member such that the radial bearing and the member are brought into contact with each other in a state where they are arranged as parallel as possible. That is, although the radial bearing 24 formed of the needle bearing is light-weighted and has a compact shape, the radial bearing 24 is not suitable for supporting an axial load or an inclination of the shaft. Accordingly, it is a common knowledge in designing that the inclination of the bushing is suppressed by controlling a clearance between the radial bearing and the member which is inserted into the radial bearing to a small value by matching-machining so that an offset load between the bushing and the needles is decreased.

[0050] However, the balance weight 32 is mounted on a portion of one end portion of the bushing 23 projecting from the radial bearing 24 in the axial direction and hence, the axis of the bushing 23 tends to be inclined with respect to the axis of the eccentric shaft 17 (drive shaft 12) due to a centrifugal force of the balance weight 32. Accordingly, to compensate for this inclination of the bushing 23 by making the clearance between the outer peripheral surface of the bushing 23 and the radial bearing 24 small, as described previously, the bushing 23 is brought into non-uniform contact with the radial bearing 24 at two different portions in the axial direction whereby an offset load is remarkably increased at a contact portion of an end portion side of the bushing 23 where the balance weight 23 is mounted thus giving rise to a drawback that flaking occurs at the contact portion.

[0051] According to the present invention, to decrease a local load (offset load) on the end portion side of the bushing where the balance weight is mounted, as shown in Fig. 5A, a clearance A between the outer peripheral surface of the eccentric shaft 17 which is inserted into the eccentric hole 23a and the inner peripheral surface of the eccentric hole 23a is set smaller than a clearance C between the outer peripheral surface of the bushing 23 and the inner peripheral surface of the radial bearing 24.

[0052] To be more specific, assuming the clearance between the outer peripheral surface of the eccentric shaft 17 and the inner peripheral surface of the eccentric hole 23a formed in the bushing 23 as A, a length of fitting between the outer peripheral surface of the eccentric shaft 17 and the inner peripheral surface of the eccentric hole 23a as B, a clearance between the outer peripheral surface of the bushing 23 and the inner peripheral surface of the radial bearing 24 as C, and a length of fitting between the outer peripheral surface of the bushing 23 and the inner peripheral surface of the radial bearing 24 as D, A is set to 6 to 22μm, C is set to 24 to 48μm, and these values are also set so as to satisfy the relationship of $A/B < C/D$.

[0053] In such a constitution, the center of gravity of the balance weight 32 which is mounted on the end portion of the bushing 23 projecting from the radial bearing 24 is displaced from the center of the eccentric shaft 17 (a centrifugal force of the balance weight acts on the end portion of the bushing 23 projecting from the radial bearing 24). Accordingly, as shown in Fig. 5B, the axis of the bushing 23 is inclined with respect to the axis of the eccentric shaft 17. In such a state, the outer peripheral surface of the bushing 23 is brought into non-uniform contact with the radial bearing 24, a centrifugal force of the balance weight 32 acts on the revolving scroll member 22 by way of the bushing 23 and the radial bearing 24 so that the centrifugal force of the balance weight 32 and a centrifugal force of the revolving scroll member 22 cancel each other. In this case, the clearance A between the outer peripheral surface of the eccentric shaft 17 which is inserted into the eccentric hole 23a and the inner peripheral surface of the eccentric hole 23a is set smaller than the clearance C between the outer peripheral surface of the bushing 23 and the inner peripheral surface of the radial bearing 24 and hence, the inclination of the bushing 23 is restricted by the eccentric shaft 17 (the bushing 23 being mainly supported on the eccentric shaft 17), and the bushing 23 is brought into contact with the radial bearing 24 only at one portion (the outer peripheral surface of the bushing 23 being brought into contact with only the end portion side of the radial bearing 24 on which the balance weight 32 is mounted but not being brought into contact with the end portion side of the radial bearing remote from the balance weight 32). Accordingly, the previously-mentioned load F3 (shown in Fig. 9) is not generated and hence, there exists no possibility that a load F2 which acts on a portion of the end portion side of the bushing 23 which is brought into contact with the radial bearing 24 on which the balance weight 32 is mounted is remarkably increased whereby the load F2 becomes substantially equal to a load F1.

[0054] As a matter of course, in an actual operation, due to a compressive reaction force generated along with the compression of a refrigerant gas in the inside of the compression chamber 25 defined between the revolving scroll member 22 and the fixed scroll member 21, a force which acts in the direction from a viewer's side to a depth side with respect to a paper plane on which the drawing is described is applied and hence, the bushing 23 and the radial bearing 24 are brought into contact with each other at a point where a resultant of these forces acts on. Accordingly, in the conventional configuration, forces F3 and F2 which act in the directions opposite to each other are generated at two portions at a front side and a rear side of the bushing 23 respectively and hence, a contact portion of the bushing 23 is twisted three-dimensionally. However, according to the present invention, a load F3 does not act on the bushing 23 and hence, such a twisted contact can be suppressed.

[0055] Further, in the above-mentioned constitution, the inclination of the bushing 23 is supported by the outer peripheral surface of the eccentric shaft 17 and the fitting portion of the eccentric hole 23a which is formed in the bushing 23 and hence, local loads (α1, α2) are generated at contact portions between the eccentric shaft 17 and the eccentric hole 23a. However, the bushing 23 is not rotatable relative to the eccentric shaft 17 and hence, sliding and rolling are not generated at the contact portion whereby there exists no possibility that flaking or wear is generated.

[0056] As described previously, in the general designing method, the clearance C is controlled by applying forming (so-called matching forming) to the outer peripheral surface of the bushing 23 in conformity with a size of the inner peripheral surface of the radial bearing 24. In the above-mentioned constitution, however, it is no more necessary to control the clearance C such that the clearance C becomes small and hence, matching forming can be omitted. The clearance A may be set to 6 to 12μm by applying matching forming to the control of clearance A (by forming the outer peripheral surface of the eccentric shaft 17 in accordance with a size of the inner peripheral surface of the eccentric hole 23a). Due to such a constitution, the inclination of the bushing which is supported with the clearance A can be further decreased without increasing man-hours for forming compared to the conventional technique.

[0057] Accordingly, an offset load which acts on the radial bearing 24 (roller 24a) can be decreased. Even when the bushing 23 is brought into local contact with the roller 24a of the radial bearing 24 due to a non-uniform contact, the generation of flaking can be suppressed and hence, a yield strength of the radial bearing 24 can be relatively enhanced (resulting in that large-sizing of the radial bearing 24 for ensuring a yield strength of the radial bearing 24 becomes unnecessary).

[0058] In the above-mentioned constitution, the balance weight 32 is formed such that the balance weight 32 projects also in the direction away from the revolving scroll member 22 and hence, a length of the balance weight in the radial direction can be suppressed and, further, the balance weight 32 can be mounted such that the balance weight 32 does not interfere with the rotation prevention mechanism (Oldham ring 27). Accordingly, the increase in diameter of the Oldham ring 27 can be prevented so that an outer diameter of the revolving scroll member can be made small.

[0059] In the above-mentioned constitution, the needle bearing is used as the radial bearing 24 and hence, a weight of the bearing per se can be decreased. Further, a size of the radial bearing in the radial direction can be also made compact and hence, the balance weight can be made light-weighted.

Reference Signs List

[0060]

1:	scroll-type compressor
2:	housing
21:	fixed scroll member
21a:	end plate
21c:	spiral wall
22:	revolving scroll member
22a:	end plate
22b:	boss portion
22c:	spiral wall
23:	bushing
23a:	eccentric hole
24:	radial bearing
32:	balance weight

Claims

1. Scroll-type compressor comprising:

a fixed scroll member (21) whose movement in the radial direction with respect to a housing (2) is restricted, the fixed scroll member (21) having an end plate (21a) and a spiral wall (21c) which is formed on the end plate in an erected manner;
a revolving scroll member (22) which is arranged so as to face the fixed scroll member in an opposed manner, the revolving scroll member having an end plate (22a) and a spiral wall (22b) which is formed on the end plate in an erected manner;
a drive shaft (12) which transmits rotational power;
an eccentric shaft (17) which is provided to an end portion of the drive shaft at a position offset from an axis of the drive shaft;
a radial bearing (24) which is fitted into a boss portion (22b) formed on a back surface of the revolving scroll member; a bushing (23) which has an eccentric hole (23a) into which the eccentric shaft (17) is inserted, the bushing (23) being fitted on the eccentric shaft by way of the eccentric hole, and the bushing being rotatably supported on the radial bearing; and
a balance weight (32) which is mounted on one

end portion of the bushing (23) and forms an integral body with the bushing,

characterized in that

the bushing (23) is configured such that, when the bushing is inclined, an outer peripheral surface of the bushing is brought into contact with the radial bearing (24) only on an end portion side on which the balance weight (32) is mounted.

2. Scroll-type compressor according to claim 1, wherein assuming a clearance between the eccentric shaft and an inner peripheral surface of the eccentric hole formed in the bushing as A, a length of fitting between the eccentric shaft and the inner peripheral surface of the eccentric hole formed in the bushing as B, a clearance between the radial bearing and an outer peripheral surface of the bushing as C, and a length of fitting between the radial bearing and the outer peripheral surface of the bushing as D, the relationship A/B < C/D is established.

3. Scroll-type compressor according to claim 1 or claim 2, wherein a rotation prevention mechanism (27) is arranged outside the boss portion (22b) of the revolving scroll member in the radial direction, and the interference of the balance weight with the rotation prevention mechanism is prevented by forming the balance weight in such a manner that the balance weight projects in the direction away from the revolving scroll member.

4. Scroll-type compressor according to any one of claims 1 to 3, wherein the radial bearing is a needle bearing.

Patentansprüche

1. Scrollkompressor, umfassend:

ein feststehendes Spiralelement (21), dessen Bewegung in der radialen Richtung bezüglich eines Gehäuses (2) eingeschränkt ist, wobei das feststehende Spiralelement (21) eine Endplatte (21a) und eine Spiralwand (21c) hat, die aufrecht an der Endplatte ausgebildet ist,
ein sich drehendes Spiralelement (22), das so angeordnet ist, dass es dem feststehenden Spiralelement gegenüberliegt, wobei das sich drehende Spiralelement eine Endplatte (22a) und eine Spiralwand (22b) hat, die aufrecht an der Endplatte ausgebildet ist,
eine Antriebswelle (12), die Drehenergie überträgt,
eine exzentrische Welle (17), die an einem Endabschnitt der Antriebswelle an einer von einer Achse der Antriebswelle versetzten Position vorgesehen ist,
ein radiales Lager (24), das in einem an einer Rückfläche des sich drehenden Spiralelements ausgebildeten Buckelabschnitt (22b) angeordnet ist,
eine Buchse (23), die ein exzentrisches Loch (23a) hat, in das die exzentrische Welle (17) eingeführt ist, wobei die Buchse (23) über das exzentrische Loch an der exzentrischen Welle angeordnet ist und am radialen Lager drehgestützt ist, und
ein Gegengewicht (32), das an einem Endabschnitt der Buchse (23) montiert ist und einen integralen Körper mit der Buchse bildet,
dadurch gekennzeichnet, dass
die Buchse (23) dazu konfiguriert ist, dass eine äußere Umfangsfläche der Buchse nur an einer Endabschnittseite, an der das Gegengewicht (32) montiert ist, mit dem radialen Lager (24) in Kontakt gebracht wird, wenn die Buchse geneigt ist.

2. Scrollkompressor nach Anspruch 1, wobei, wenn ein Abstand zwischen der exzentrischen Welle und einer inneren Umfangsfläche des in der Buchse ausgebildeten exzentrischen Lochs A ist, eine Einbaulänge zwischen der exzentrischen Welle und der inneren Umfangsfläche des in der Buchse ausgebildeten exzentrischen Lochs B ist, ein Abstand zwischen dem radialen Lager und einer äußeren Umfangsfläche der Buchse C ist und eine Einbaulänge zwischen dem radialen Lager und der äußeren Umfangsfläche der Buchse D ist, das Verhältnis A/B < C/D entsteht.

3. Scrollkompressor nach Anspruch 1 oder Anspruch 2, wobei ein Drehverhinderungsmechanismus (27) außerhalb des Buckelabschnitts (22b) des sich drehenden Spiralelements in der radialen Richtung angeordnet ist und die Störung des Drehverhinderungsmechanismus durch das Gegengewicht verhindert wird, indem das Gegengewicht derart ausgebildet ist, dass das Gegengewicht in der vom sich drehenden Spiralelement weg gehenden Richtung vorragt.

4. Scrollkompressor nach einem der Ansprüche 1 bis 3, wobei das radiale Lager ein Nadellager ist.

Revendications

1. Compresseur du type à volutes, comprenant :

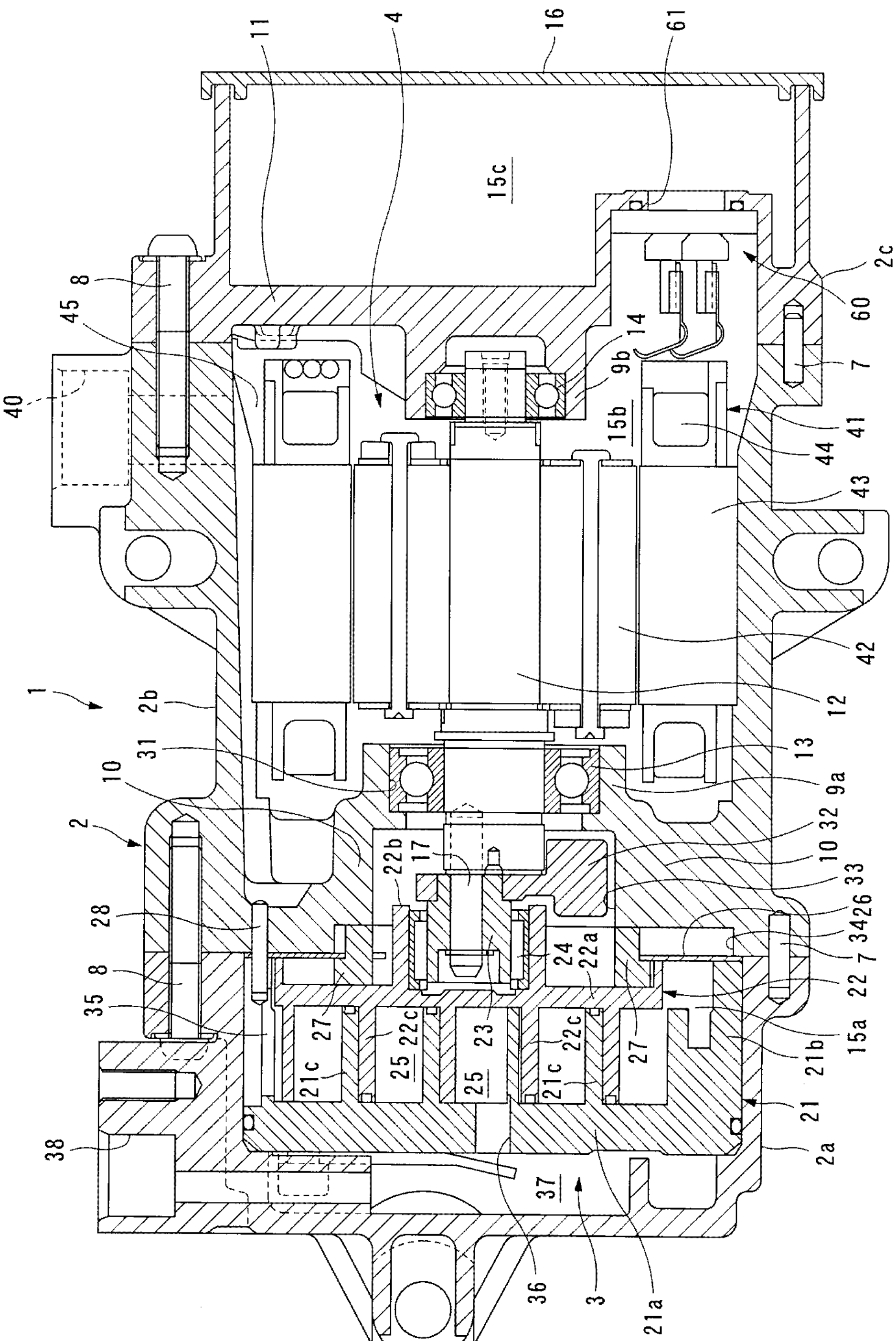
un élément de volute fixe (21) dont le déplacement dans la direction radiale par rapport à un

boîtier (2) est limité, l'élément de volute fixe (21) présentant une plaque d'extrémité (21a) et une paroi hélicoïdale (21c) qui est formée sur la plaque d'extrémité d'une manière érigée ;
un élément de volute rotatif (22) qui est agencé de manière à faire face à l'élément de volute fixe d'une manière opposée, l'élément de volute rotatif présentant une plaque d'extrémité (22a) et une paroi hélicoïdale (22b) qui est formée sur la plaque d'extrémité d'une manière érigée ;
un arbre d'entraînement (12) qui transmet une puissance de rotation ;
un arbre excentrique (17) qui est prévu sur une partie d'extrémité de l'arbre d'entraînement à une position décalée d'un axe de l'arbre d'entraînement ;
un palier radial (24) qui est agencé dans une partie de bosse (22b) formée sur une surface arrière de l'élément de volute rotatif ;
une douille (23) comportant un trou excentrique (23a) dans lequel l'arbre excentrique (17) est inséré, la douille (23) étant agencée sur l'arbre excentrique au moyen du trou excentrique, et la douille étant supportée de façon rotative sur le palier radial ; et
un poids d'équilibrage (32) qui est monté sur une partie d'extrémité de la douille (23) et qui forme un corps intégral avec la douille,
caractérisé en ce que la douille (23) est configurée de telle sorte que, lorsque la douille est inclinée, une surface périphérique extérieure de la douille soit amenée en contact avec le palier radial (24) seulement sur un côté de la partie d'extrémité sur lequel le poids d'équilibrage (32) est monté.

2. Compresseur du type à volutes selon la revendication 1, dans lequel, en supposant la présence d'un jeu entre l'arbre excentrique et une surface périphérique intérieure du trou excentrique formé dans la douille comme A, une longueur d'agencement entre l'arbre excentrique et la surface périphérique intérieure du trou excentrique formé dans la douille comme B, un jeu entre le palier radial et une surface périphérique extérieure de la douille comme C, et une longueur d'agencement entre le palier radial et la surface périphérique extérieure de la douille comme D, la relation $A/B < C/D$ est établie.

3. Compresseur du type à volutes selon la revendication 1 ou la revendication 2, dans lequel un mécanisme de prévention de rotation (27) est agencé à l'extérieur de la partie de bosse (22b) de l'élément de volute rotatif dans la direction radiale, et l'interférence du poids d'équilibrage avec le mécanisme de prévention de rotation est empêchée en formant le poids d'équilibrage de telle sorte que le poids d'équilibrage fasse saillie dans la direction opposée à l'élément de volute rotatif.

4. Compresseur du type à volutes selon l'une quelconque des revendications 1 à 3, dans lequel le palier radial est un roulement à aiguilles.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2b
2c
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16
17
21
21a
21b
21c
22
22a
22b
22c
23
24
25
26
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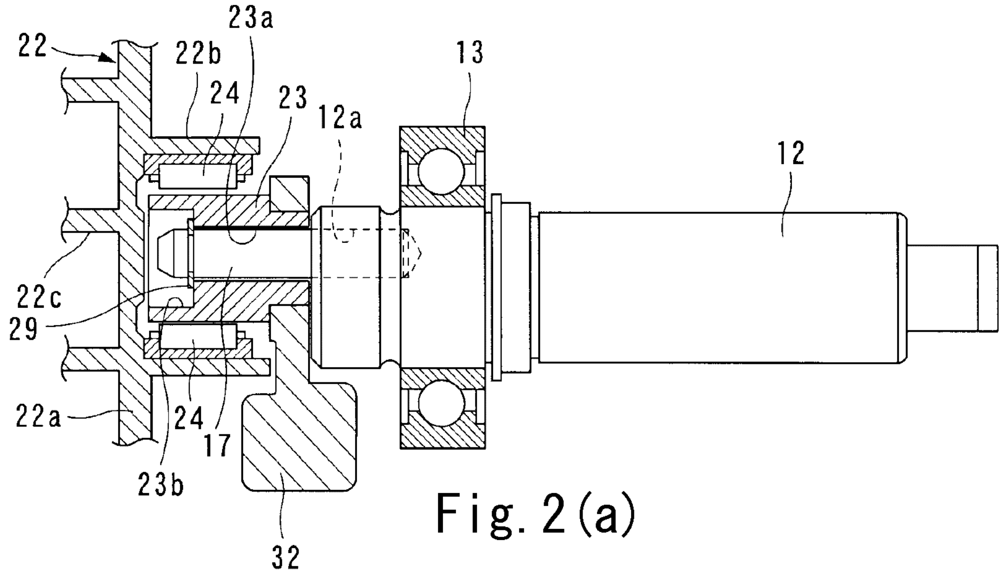
Fig. 1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23b
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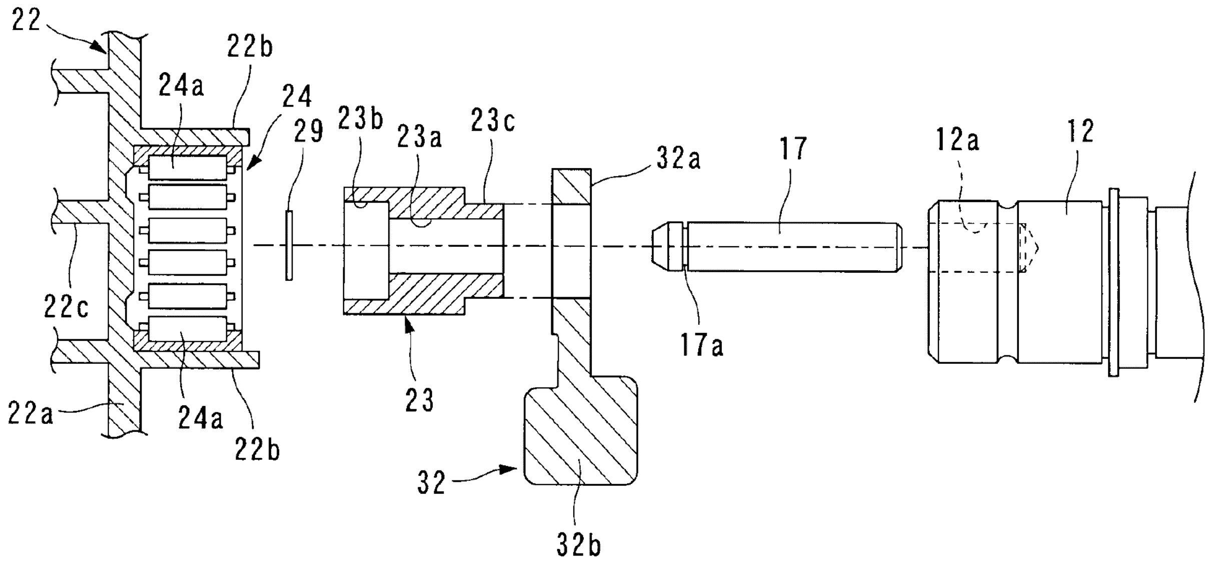
Fig. 2(a)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32
32b

Fig. 2(b)

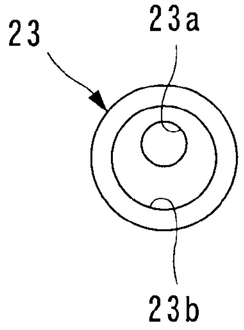


Fig. 3A

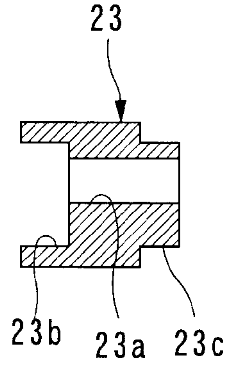


Fig. 3B

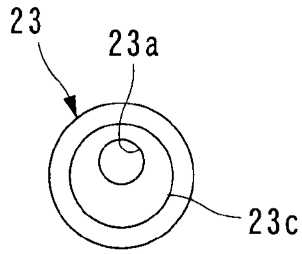


Fig. 3C

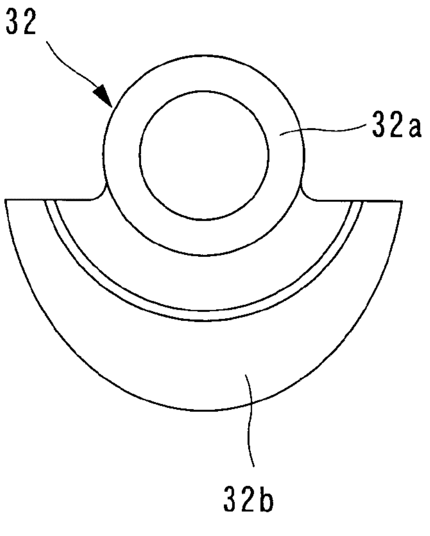


Fig. 4A

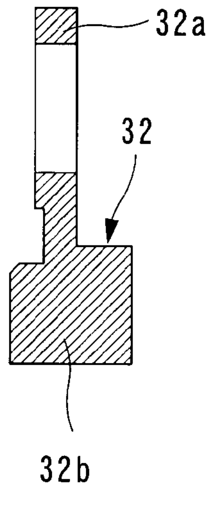


Fig. 4B

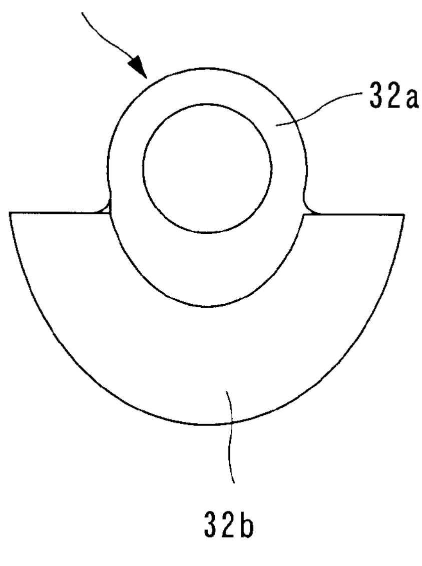
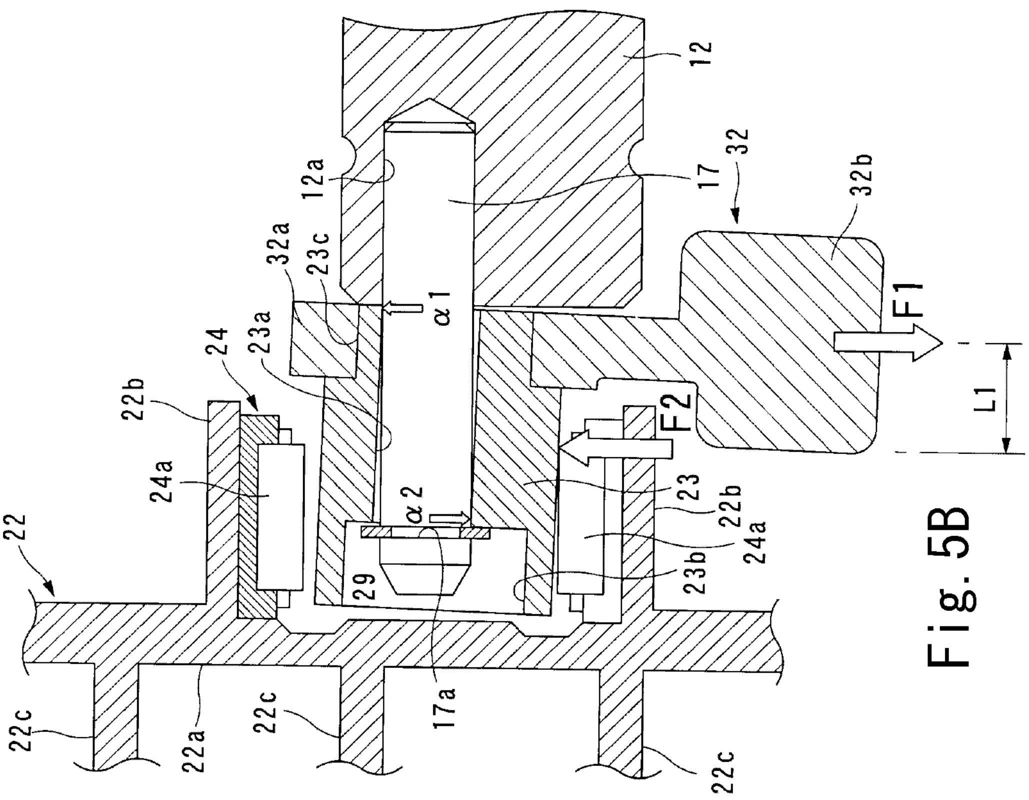


Fig. 4C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32b
32a
23c
α1
23a
24
22b
F1
F2
L1
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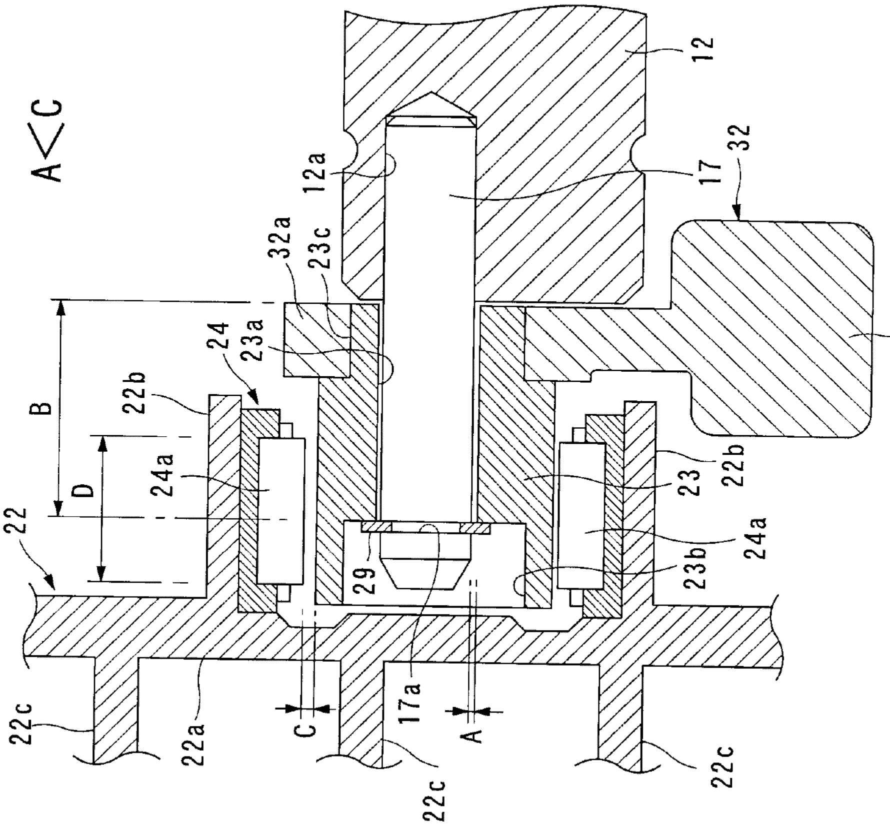
22b
22
α2
24a
29
23b
17a
22c
22a
22c
22c

Fig. 5B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32a
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B
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C
A
22c
22c

Fig. 5A

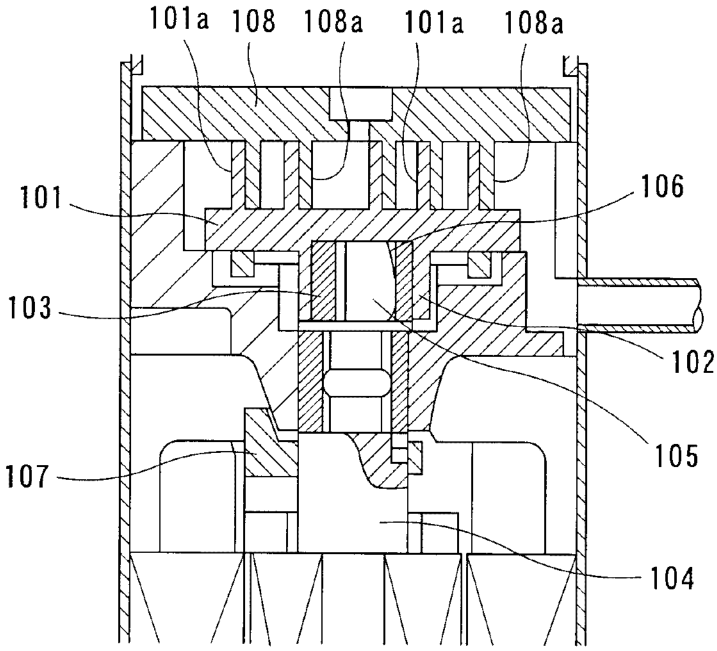
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Fig. 6

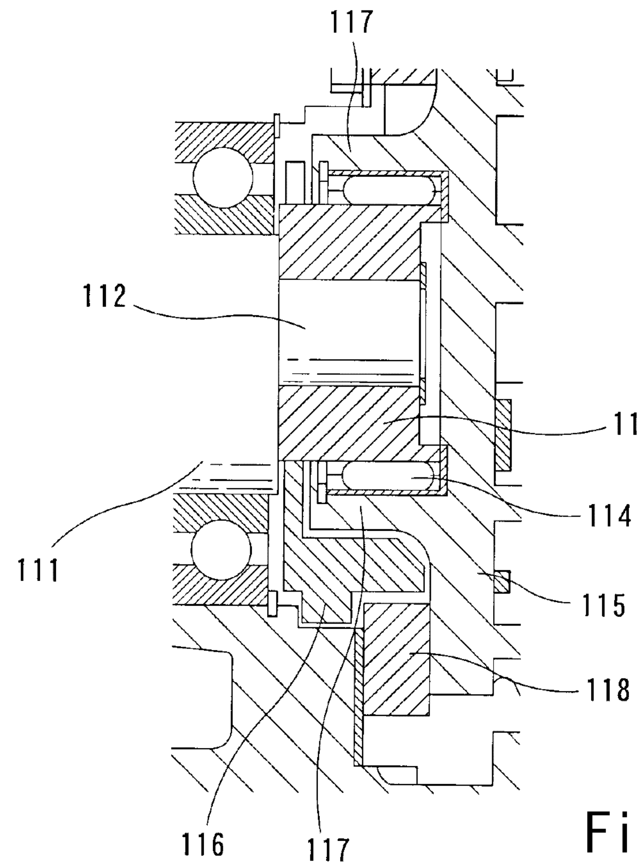
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Fig. 7

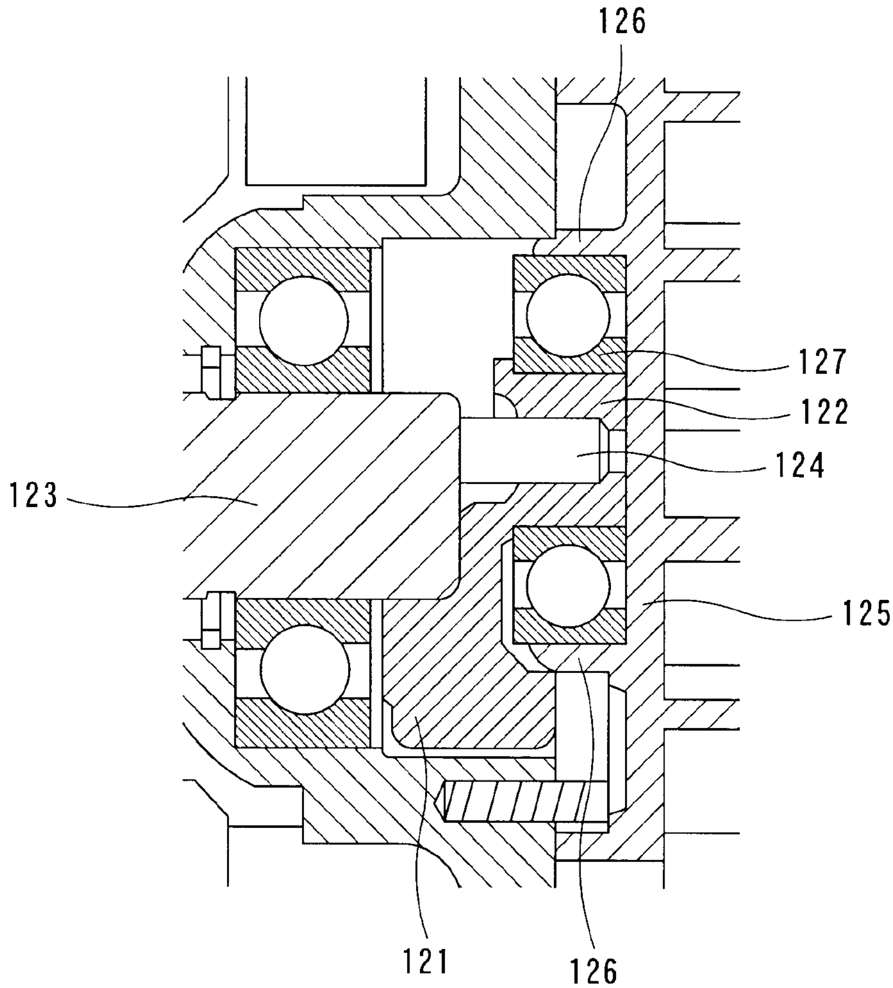
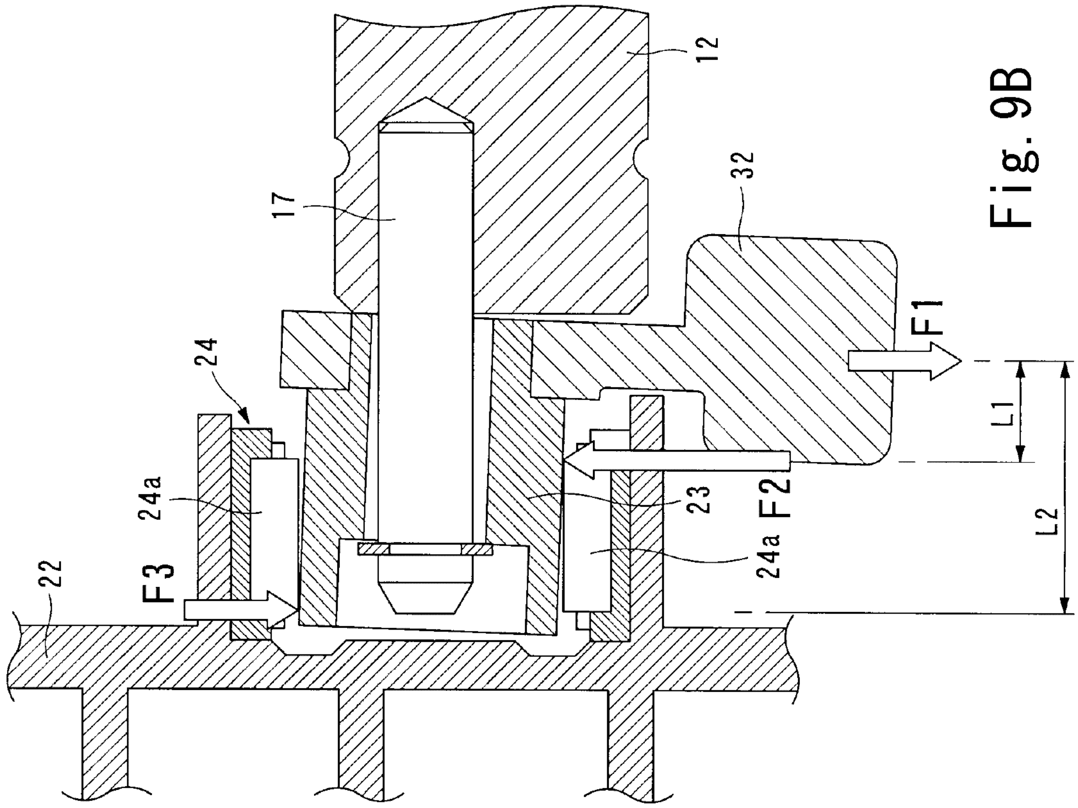


Fig. 8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32
24
24a
23
24a
22
F1
F2
F3
L1
L2

Fig. 9B

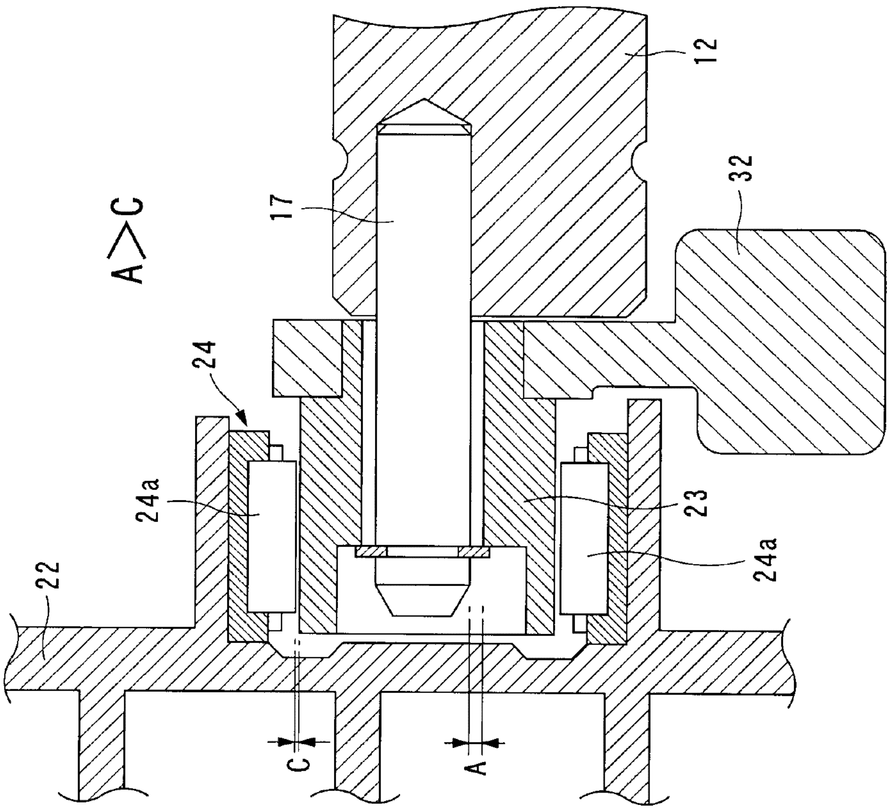
A>C
12
17
32
24
24a
23
24a
22
C
A

Fig. 9A

REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

- JP 8042467 A **[0008]**
- JP 8042477 A **[0008]**
- JP 2010196630 A **[0008]**
- EP 0921316 A **[0009]**