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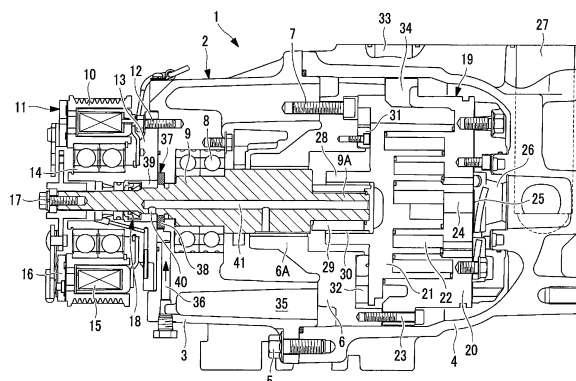
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(54) **OPEN-TYPE COMPRESSOR**

(57) Provided is an open-type compressor with which it is possible to prevent a reduction in oil supply amount caused by an increase in flow-path pressure loss particularly in a high rotational speed range, and to improve reliability in lubrication performance. Disclosed is an open-type compressor (1) comprising: a drive shaft (9) having one end protruding outside a housing (2); an oil supply pump (37) provided in an outer circumferential section of the drive shaft (9); a pump chamber (39) that is formed around the drive shaft (9), and into which oil pumped up by the oil supply pump (37) is discharged; an axial-direction oil supply passage (41) that is pierced in-

side the drive shaft (9) along the direction of the axis L thereof, and that supplies the oil from the pump chamber (39) to a sliding part; and a radial-direction oil supply passage (40) that guides the oil in the pump chamber (39) to the axial-direction oil supply passage (41). The axial-direction oil supply passage (41) is provided in a position that is decentered by a predetermined dimension Δh with respect to the axis L of the drive shaft (9). The radial-direction oil supply passage (40) is provided on a side toward which the axial-direction oil supply passage (41) is decentered.

FIG. 1



Description

Technical Field

[0001] The present invention relates to an open-type compressor which is provided on an outer circumference of a drive shaft and can supply oil which is pumped up by an oil supply pump driven by the drive shaft to a sliding part through an oil supply passage which is pierced inside the drive shaft along an axial direction.

Background Art

[0002] In a horizontally laid open-type compressor in which one end of a drive shaft rotatably supported inside a housing via a bearing protrudes outside the housing and which is driven by driving power from the outside, in a case where a forcible oil-supply type compressor which pumps a lubricant in an oil sump by an oil supply pump and supplies the oil to a sliding part such as a bearing so as to lubricate the sliding part is adopted, as described in Patent Document 1, the oil supply pump which is driven by the drive shaft is provided on the outer circumference of the drive shaft, and oil pumped up by the oil supply pump is supplied to the sliding part through an oil supply passage which is pierced inside the drive shaft along the axial direction.

[0003] In a case of a closed-type compressor, in both a vertically laid closed-type compressor and a horizontally laid closed-type compressor, in general, a centrifugal-type oil supply pump, a displacement-type oil supply pump, or the like is provided on the end of a drive shaft, a lubricant which fills a sealed vessel is pumped up by the oil supply pump, and the oil is supplied to a sliding part through an oil supply passage which is pierced inside the drive shaft along the axial direction. Regarding forcible oil-supply type compressors, Patent Document 2 discloses a forcible oil-supply type compressor in which the oil supply passage pierced inside the drive shaft along the axial direction is provided so as to be offset with respect to the axis by a predetermined dimension and oil supply performance is improved using a centrifugal force.

Citation List

Patent Literature

[0004]

[PTL 1] Japanese Unexamined Patent Application Publication No. 2005-282446

[PTL 2] Japanese Unexamined Patent Application Publication No. 8-219063

Summary of Invention

Technical Problem

[0005] As described above, in the case where a forcible oil-supply type compressor is adopted, when the compressor is a closed-type compressor, external driving power is not required, and the oil supply passage can be pierced in the axial direction in a state where one end of the drive shaft is an open end. Accordingly, it is possible to increase a rotating speed or a centrifugal force of the oil supply pump in proportion to the rotating speed of the drive shaft without it being easily influenced by the centrifugal force. Therefore, since an oil supply amount increases and oil supply performance can be improved, significant problems do not occur in the closed-type compressor. However, in a case of an open-type compressor, external driving power is required, and it is difficult to set one end of the drive shaft to an open end. Accordingly, it is necessary to temporarily discharge oil pumped up by the oil supply pump to a pump chamber formed around the drive shaft, and to supply the oil to an oil supply passage which is pierced in the axial direction through an oil supply passage which is provided from the pump chamber to the drive shaft in a radial direction.

[0006] Therefore, it is not possible to prevent pressure loss due to a centrifugal force from occurring in an inlet portion of the oil supply passage which is structurally pierced in the radial direction. Since the flow-path pressure loss is likely to increase as the rotating speed of the drive shaft increases, in a case where an axial-direction oil supply passage is provided on the axis of the drive shaft (decentering is not present), as shown in Fig. 3, the oil supply amount decreases particularly in a high rotational speed range.

[0007] The present invention is made in consideration of the above-described circumstances, and an object thereof is to provide an open-type compressor in which a decrease in the oil supply amount due to an increase of the flow-path pressure loss can be prevented particularly in the high rotational speed range, and reliability with respect to lubrication performance can be improved.

Solution to Problem

[0008] According to a first aspect of the present invention, there is provided an open-type compressor, including: a drive shaft which is rotatably supported inside a housing and includes one end which protrudes outside the housing; an oil supply pump which is provided on an outer circumferential section of the drive shaft and is driven by rotation of the drive shaft; a pump chamber which is formed around the drive shaft and into which oil pumped up by the oil supply pump is discharged; an axial-direction oil supply passage which is pierced inside the drive shaft along the direction of an axis of the axial-direction oil supply passage, and through which oil from the pump chamber is supplied to a sliding part; and a

radial-direction oil supply passage which is provided in the drive shaft and through which oil in the pump chamber is introduced to the axial-direction oil supply passage, in which the axial-direction oil supply passage is provided at a position which is decentered by a predetermined dimension with respect to the axis of the drive shaft, and the radial-direction oil supply passage is provided on a side toward which the axial-direction oil supply passage is decentered.

[0009] According to the first aspect, the axial-direction oil supply passage which is pierced inside the drive shaft is provided at the position decentered by a predetermined dimension with respect to the axis of the drive shaft, and the radial-direction oil supply passage through which oil in the pump chamber pumped up by the oil supply pump is introduced to the axial-direction oil supply passage is provided in the direction in which the axial-direction oil supply passage is decentered. Accordingly, the passage length of the radial-direction oil supply passage decreases as the axial-direction oil supply passage is decentered and the radial-direction oil supply passage is provided on the decentered side, and it is possible to decrease flow-path pressure loss due to a centrifugal force generated in an inlet portion of the radial-direction oil supply passage. That is, the passage length of the radial-direction oil supply passage becomes the radius of the drive shaft if the axial-direction oil supply passage is not decentered. However, in the case where the radial-direction oil supply passage is provided on the side toward which the axial-direction oil supply passage is decentered, the passage length can be shorter than the radius of the drive shaft, and it is possible to decrease the flow-path pressure loss according to the decrease in the passage length. In addition, the axial-direction oil supply passage is decentered by the predetermined dimension, and it is possible to increase oil supply performance with respect to the sliding part using centrifugal pump effects acting on the oil in the axial-direction oil supply passage. Therefore, according to synergistic effects between the decrease in the flow-path pressure loss in the radial-direction oil supply passage and the improvement in oil supply performance in the axial-direction oil supply passage, it is possible to prevent the oil supply amount from decreasing particularly in a high rotational speed range, and it is possible to increase reliability with respect to lubrication performance.

[0010] In the open-type compressor of the first aspect of the present invention, the radial-direction oil supply passage may be provided at a position at which a passage length is shortest on the axis in the decentered direction.

[0011] According to the first aspect of the present invention, since the radial-direction oil supply passage is provided at the position at which the passage length is shortest on the axis in the decentered direction, the passage length of the radial-direction oil supply passage can be decreased by the degree corresponding to the decentered dimension of the axial-direction oil supply passage,

and the passage length becomes the shortest length. Therefore, it is possible to minimize the flow-path pressure loss due to the centrifugal force generated in the inlet portion of the radial-direction oil supply passage.

Accordingly, it is possible to improve the oil supply amount in the high rotational speed range and to further improve the oil supply performance.

[0012] In the above-described open-type compressors of the first aspect of the present invention, a passage diameter of the axial-direction oil supply passage may be greater than a passage diameter of the radial-direction oil supply passage.

[0013] According to the first aspect, since the passage diameter of the axial-direction oil supply passage is greater than the passage diameter of the radial-direction oil supply passage, it is possible to easily obtain centrifugal pump effects due to decentering and to decrease the flow-path pressure loss in the passage. In addition, by processing a connection between the radial-direction oil supply passage and the axial-direction oil supply passage such that these communicate with each other such that steps, burrs, and the like do not occur, it is possible to prevent flow-path pressure loss from occurring in the connection portion between the radial-direction oil supply passage and the axial-direction oil supply passage, and it is possible to prevent the oil supply amount from decreasing in the high rotational speed range and to improve the oil supply performance.

Advantageous Effects of Invention

[0014] According to the present invention, the passage length of the radial-direction oil supply passage decreases as the axial-direction oil supply passage is decentered and the radial-direction oil supply passage is provided on the decentered side, and it is possible to decrease flow-path pressure loss due to a centrifugal force generated in the inlet portion of the radial-direction oil supply passage. In addition, since the axial-direction oil supply passage is decentered by the predetermined dimension, and it is possible to increase oil supply performance with respect to the sliding part using centrifugal pump effects acting on the oil in the axial-direction oil supply passage, according to synergistic effects between the decrease in the flow-path pressure loss in the radial-direction oil supply passage and the improvement of oil supply performance in the axial-direction oil supply passage, it is possible to prevent the oil supply amount from decreasing particularly in a high rotational speed range, and it is possible to increase reliability with respect to lubrication performance.

Brief Description of Drawings

[0015]

Fig. 1 is a longitudinal sectional view of an open-type compressor according to an embodiment of the

present invention.

Fig. 2A is a sectional view of a drive shaft in the open-type compressor, and Fig. 2B is a right-side view of the drive shaft.

Fig. 3 is a sectional view taken along line a-a of Fig. 2A.

Fig. 4 is a sectional view corresponding to Fig. 3 of a modification example of a radial-direction oil supply passage provided in the drive shaft.

Fig. 5 is a graph showing oil supply characteristics in the open-type compressor.

Description of Embodiments

[0016] Hereinafter, an embodiment of the present invention will be described with reference to Figs. 1 to 5.

[0017] Fig. 1 is a longitudinal sectional view of an open-type compressor according to the embodiment of the present invention, Fig. 2A is a sectional view of a drive shaft, Fig. 2B is a right-side view of the drive shaft, and Figs. 3 and 4 are sectional views taken along line a-a of Fig. 2A.

[0018] The open-type compressor 1 includes a tubular housing 2 in which a bottomed front housing 3 and a bottomed rear housing 4 are integrally connected to each other by a bolt 5.

[0019] A bearing member 6 is fixed and installed to the opening end side in the housing 2 on the front housing 3 side by a bolt 7, and a drive shaft 9 is rotatably supported by a radial bearing portion 6A of the bearing member 6 and a rolling bearing 8 which is installed in the front housing 3. One end of the drive shaft 9 penetrates the front housing 3 and protrudes outside the front housing 3, and driving power from an external drive source such as an engine is input to the protrusion portion of the drive shaft 9 via a pulley 10 and an electromagnetic clutch 11.

[0020] The pulley 10 is rotatably supported to an outer circumference of a flange member 13, which is fixed and installed to the front end surface of the front housing 3 by a bolt 12, via a rolling bearing 14, and a coil assembly 15 of the electromagnetic clutch 11 is incorporated into the pulley 10. In addition, an armature assembly 16 of the electromagnetic clutch 11 is assembled to the external protrusion end of the drive shaft 9 by a bolt 17 via a boss portion so as to face the pulley 10. In addition, a mechanical seal 18 for hermetically sealing the penetration portion of the drive shaft 9 is installed on the inner circumference of the flange member 13.

[0021] A compression mechanism 19 is incorporated into the rear housing 4 side of the housing 2. Here, the compression mechanism 19 is a scroll compression mechanism 19 which includes a pair of fixed scroll 20 and orbiting scroll 21. In the scroll compression mechanism 19, the pair of fixed scroll 20 and orbiting scroll 21 engage with each other so as to be deviated by a phase of 180°, multiple compression chambers 22 are formed between both scrolls 20 and 21, and the scroll compression mechanism 19 is known.

[0022] The fixed scroll 20 is fastened and fixed to the bearing member 6 by a bolt 23, and a discharge cavity 26 is formed between the back face of the end plate of the fixed scroll and the inner surface of the rear housing 4. A discharge port 24 which discharges compressed gas into the discharge cavity 26 and a discharge valve 25 which opens and closes the discharge port 24 are provided on the end plate of the fixed scroll 20. In addition, a discharge port 27 through which compressed gas which has been discharged into the discharge cavity 26 is discharged to the outside is open to the rear housing 4, and a discharge pipe configuring a refrigerating cycle can be connected to the rear housing 4.

[0023] The orbiting scroll 21 has a boss portion 28 on the back face of the end plate of the orbiting scroll 21, a crank pin 9A provided on the inner end side of the drive shaft 9 is connected to the boss portion 28 via a drive bush 29 and a turning bearing 30, and the orbiting scroll 21 is pivotally driven via the crank pin 9A by the rotation of the drive shaft 9. In addition, the back face of the end plate of the orbiting scroll 21 is supported by a thrust bearing 31 provided in the bearing member 6, the rotation of the orbiting scroll 21 is prevented by a known rotation prevention mechanism 32 including an oldham link, a pin ring, or the like which is interposed between the back plate of the end plate and the bearing member 6, and the orbiting scroll 21 is driven so as to be pivotally revolved with respect to the fixed scroll 20.

[0024] An intake port 33 which is connected to an intake pipe on the refrigerating cycle side is provided on the outer circumference on the front end side of the rear housing 4, and the compressed gas entering from the intake port 33 into an intake cavity 34 is suctioned to the compression chamber 22 of the scroll compression mechanism 19 so as to be compressed. In the present embodiment, the compression mechanism 19 is a so-called stepped scroll compression mechanism 19 in which a step portion which changes a lap height is provided in spiral directions of the fixed scroll 20 and the orbiting scroll 21, the outer circumferential side lap height is higher than the inner circumferential side lap height, gas can be compressed not only in the circumferential direction but also in the axial direction, and a three-dimensional compression can be realized. However, the present invention is not limited to this.

[0025] Meanwhile, the inside of the front housing 3 is filled with a lubricant having a required amount, the lower space inside the front housing 3 becomes an oil sump 35, and oil is collected in the oil sump 35. The oil in the oil sump 35 is suctioned to an oil supply pump 37 via an intake passage 36.

[0026] The oil supply pump 37 is a known a rotary-type displacement pump in which a decentered portion 9B (Fig. 2) is formed on the outer circumferential section of the drive shaft 9 penetrating the front end surface of the front housing 3, and a rotor 38 which is rotated so as to be decentered in the cylinder formed between the front end surface of the front housing 3 and the end surface

of the flange member 13 is fitted to the decentered portion 9B.

[0027] The oil which is pumped up from the oil sump 35 by the oil supply pump 37 is discharged into a pump chamber 39 which is formed between the decentered portion 9B around the drive shaft 9 and the mechanical seal 18. The oil pumped to the pump chamber 39 is supplied to a sliding part such as the radial bearing portion 6A, the drive bush 29, the turning bearing 30, or the thrust bearing 31, a sliding part such as the mechanical seal 18, or the like through a radial-direction oil supply passage 40 and an axial-direction oil supply passage 41 which are provided inside the drive shaft 9.

[0028] As shown in Figs. 2 and 3, the axial-direction oil supply passage 41 which is provided along the axis L of the drive shaft 9 is inside the drive shaft 9 at a position decentered to an axis L of the drive shaft 9 by a predetermined dimension (decentered dimension) Δh . In addition, the radial-direction oil supply passage 40 through which the oil inside the pump chamber 39 is introduced to the axial-direction oil supply passage 41 is provided in the decentered direction of the axial-direction oil supply passage 41. In the present embodiment, the radial-direction oil supply passage 40 is provided at the position at which a passage length h_1 thereof is shortest on the axis in the decentered direction. Accordingly, compared to a passage length h with which the axial-direction oil supply passage 41 is provided on the axis L of the drive shaft 9, the passage length h_1 of the radial-direction oil supply passage 40 can be shortened by Δh ($\Delta h = h - h_1$).

[0029] However, the present invention is not limited to the above-described case where the radial-direction oil supply passage 40 is provided on the axis of the decentered direction and the passage length h_1 is shortest. That is, as shown in Fig. 4, a radial-direction oil supply passage 40A may be provided in a direction which has a predetermined angle with respect to the axis in the decentered direction. According to this configuration, a passage length h_2 of the radial-direction oil supply passage 40A can be shorter than the passage length h with which the axial-direction oil supply passage 41 is provided on the axis L of the drive shaft 9, and in this case, the passage h_2 satisfies that $h_1 < h_2 < h$. That is, the radial-direction oil supply passage through which the oil in the pump chamber 39 is introduced to the axial-direction oil supply passage 41 is not limited to be positioned on the axis in the decentered direction, the radial-direction oil supply passage is provided on the decentered side of the axial-direction oil supply passage 41, and the passage length of the radial-direction oil supply passage can be shorter than that when the axial-direction oil supply passage 41 is provided on the axis L of the drive shaft 9.

[0030] In addition, as described above, according to the case where the axial-direction oil supply passage 41 and the radial-direction oil supply passages 40 and 40A are provided, the axial-direction oil supply passage 41 is a blind hole which is pierced along the axis L from one end of the crank pin 9A. Meanwhile, the radial-direction

oil supply passages 40 and 40A are hole which are provided to be radially perpendicular in the vicinity of the tip portion of the blind hole, and it is necessary to prevent steps, burrs, or the like generating flow-path pressure loss from occurring in the intersection portion (connection portion) between the axial-direction oil supply passage 41 and the radial-direction oil supply passages 40 and 40A. Accordingly, by causing a passage diameter d_1 of the axial-direction oil supply passage 41 to be greater than a passage diameter d_2 of each of the radial-direction oil supply passages 40 and 40A ($d_1 > d_2$), it is possible to easily obtain centrifugal pump effect due to decentering in the axial-direction oil supply passage 41 and to decrease the flow-path pressure loss in the passage, and both passages 40 and 40A, and 41 can be processed to communicate with each other such that steps, burrs, and the like do not occur in the connection portion between both passages 40 and 40A, and 41.

[0031] According to the above-described configuration, the following effects of the present embodiment are exerted.

[0032] In the open-type compressor 1, if the electromagnetic clutch 11 is turned on, the driving power input from the external drive source via the pulley 10 is transmitted to the drive shaft 9, and the drive shaft 9 is rotationally driven. Accordingly, the orbiting scroll 21 of the scroll compression mechanism 19 is driven so as to be pivotally revolved around the fixed scroll 20, low-pressure gas suctioned from the intake port 33 into the intake cavity 34 is suctioned into the compression chamber 22 and is compressed so as to be high-pressure gas, and the high-pressure gas is discharged from the discharge port 24 into the discharge cavity 26 and discharged from the discharge port 27 to the refrigerating cycle.

[0033] During this, a lubricant in the oil sump 35 is suctioned via the intake passage 36 by the oil supply pump 37 driven by the rotation of the drive shaft 9, and is pumped to the pump chamber 39. The oil pumped into the pump chamber 39 lubricates the sliding part of the mechanical seal 18, is introduced into the axial-direction oil supply passage 41 via the radial-direction oil supply passages 40 and 40A, and is supplied to the sliding part such as the radial bearing portion 6A, the drive bush 29, the turning bearing 30, or the thrust bearing 31 through the axial-direction oil supply passage 41 so as to lubricate the sliding part. The oil which has lubricated the sliding part is collected in the oil sump 35 which is the bottom portion of the housing 2 and is recirculated.

[0034] Here, in the present embodiment, the axial-direction oil supply passage 41 is provided at the position which is decentered to the axis L of the drive shaft 9 by the predetermined dimension Δh , and the radial-direction oil supply passages 40 and 40A through which the oil in the pump chamber 39 is introduced to the axial-direction oil supply passage 41 is introduced are provided on the decentered side of the axial-direction oil supply passage 41. Accordingly, the passage lengths h_1 and h_2 can be shorter than those in the case where the axial-direction

oil supply passage 41 is provided on the axis L of the drive shaft 9 ($h_1 < h_2 < h$).

[0035] Accordingly, the passage lengths h_1 and h_2 of the radial-direction oil supply passages 40 and 40A can be shortened by the dimension Δh by which the axial-direction oil supply passage 41 is decentered, and it is possible to decrease the flow-path pressure loss due to the centrifugal force generated in the inlet portions of the radial-direction oil supply passages 40 and 40A. In addition, since the axial-direction oil supply passage 41 is decentered by the predetermined dimension Δh , it is possible to increase oil supply performance with respect to the sliding part using centrifugal pump effects acting on the oil in the axial-direction oil supply passage 41.

[0036] Fig. 5 is a graph showing oil supply characteristics in the case where the forcible oil-supply type compressor is adopted, in which a horizontal axis indicates a rotating speed (rpm) of the drive shaft 9 and a vertical axis indicates an oil supply amount (cm^3/min). With respect to a theoretical value indicated by a solid line, since the passage length h of the radial-direction oil supply passage is lengthened in the case where the axial-direction oil supply passage 41 is provided on the axis L, the oil supply amount with respect to the theoretical value decreases in a high rotational speed range as shown by plots. Like the present embodiment, in the case the axial-direction oil supply passage 41 is provided so as to be decentered to the axis L, the radial-direction oil supply passages 40 and 40A are provided on the decentered side, and the passage lengths h_1 and h_2 are shortened, the oil supply amount is improved in the high rotational speed range as shown by plots $\blacktriangle$ and can approach the theoretical value.

[0037] Therefore, in the present embodiment, according to synergistic effects between the decrease in the flow-path pressure loss in the radial-direction oil supply passages 40 and 40A and the improvement of oil supply performance in the axial-direction oil supply passage 41, it is possible to prevent the oil supply amount from decreasing in the high rotational speed range, and it is possible to increase reliability with respect to lubrication performance.

[0038] Particularly, the radial-direction oil supply passage 40 is provided at the position at which the passage length h_1 is shortest on the axis in the decentered direction. Accordingly, the passage length h_1 of the radial-direction oil supply passage 40 can be decreased by the degree corresponding to the decentered dimension Δh of the axial-direction oil supply passage 41, and the passage length h_1 becomes the shortest length. Therefore, it is possible to minimize the flow-path pressure loss due to the centrifugal force generated in the inlet portion of the radial-direction oil supply passage 40. Accordingly, it is possible to improve the oil supply amount in the high rotational speed range and to further improve the oil supply performance.

[0039] In addition, in the radial-direction oil supply passages 40 and 40A and the axial-direction oil supply pas-

sage 41, since the passage diameter d_1 of the axial-direction oil supply passage 41 is greater than the passage diameter d_2 of each of the radial-direction oil supply passages 40 and 40A ($d_1 > d_2$), it is possible to easily obtain centrifugal pump effect due to decentering in the axial-direction oil supply passage 41 and to decrease the flow-path pressure loss in the passage. In addition, by processing a connection between the radial-direction oil supply passages 40 and 40A and the axial-direction oil supply passage 41 so as to communicate with each other such that steps, burrs, and the like do not occur, it is possible to prevent the flow-path pressure loss from occurring in the connection portion between the radial-direction oil supply passages 40 and 40A and the axial-direction oil supply passage 41, and it is possible to prevent the oil supply amount from decreasing in the high rotational speed range due to synergistic effects and to improve the oil supply performance.

[0040] Particularly, in a case where the open-type compressor 1 according to the present embodiment is applied to an open-type scroll compressor 1 which adopts the forcible oil-supply type compressor which is operated at a high speed of 3600 rpm or more, it is possible to improve lubrication performance.

[0041] In addition, the present invention is not limited to the invention according to the present embodiment, and may be appropriately modified. For example, in the above-described embodiment, as an example of the open-type compressor 1, the example in which the scroll type compressor is applied is described. However, other-type compressors, for example, a rotary open-type compressor, a swash plate open-type compressor, a reciprocating open-type compressor, or the like may be similarly applied.

[0042] In addition, in the above-described embodiment, the example in which the rotary displacement-type pump is applied as the oil supply pump 37 is described. However, the present invention is not limited to this, and other type oil supply pumps such as a screw type pump may be applied.

Reference Signs List

[0043]

- 1: open-type compressor
- 2: housing
- 9: drive shaft
- 37: oil supply pump
- 39: pump chamber
- 40, 40A: radial-direction oil supply passage
- 41: axial-direction oil supply passage
- L: axis of drive shaft
- Δh : decentered dimension
- h_1, h_2 : passage length of radial-direction oil supply passage
- d_1 : passage diameter of axial-direction oil supply passage

d2: passage diameter of radial-direction oil supply passage

Claims

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1. An open-type compressor, comprising:

a drive shaft which is rotatably supported inside
a housing and includes one end which protrudes
outside the housing; 10
an oil supply pump which is provided on an outer
circumferential section of the drive shaft and is
driven by rotation of the drive shaft;
a pump chamber which is formed around the 15
drive shaft and from which oil pumped up by the
oil supply pump is discharged;
an axial-direction oil supply passage which is
pierced inside the drive shaft along the direction
of an axis of the axial-direction oil supply pas- 20
sage, and through which oil from the pump
chamber is supplied to a sliding part; and
a radial-direction oil supply passage which is
provided in the drive shaft and through which oil 25
in the pump chamber is introduced to the axial-
direction oil supply passage,
wherein the axial-direction oil supply passage is
provided at a position which is decentered by a
predetermined dimension with respect to the ax- 30
is of the drive shaft, and
wherein the radial-direction oil supply passage
is provided on a side toward which the axial-
direction oil supply passage is decentered.

2. The open-type compressor according to claim 1, 35 wherein the radial-direction oil supply passage is provided at a position at which a passage length is shortest on the axis in the decentered direction.

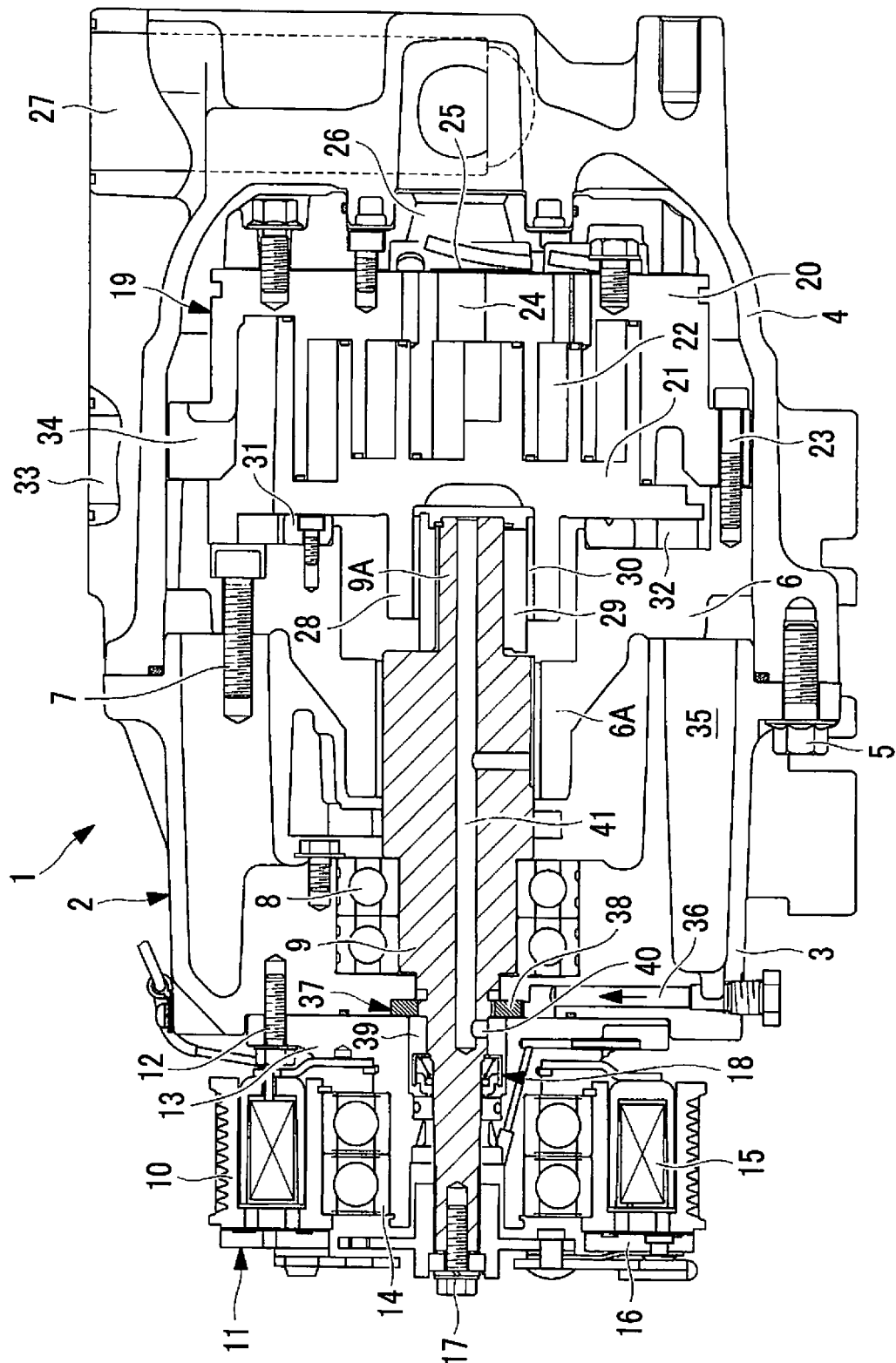
3. The open-type compressor according to claim 1, 40 wherein a passage diameter of the axial-direction oil supply passage is greater than a passage diameter of the radial-direction oil supply passage.

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FIG. 1



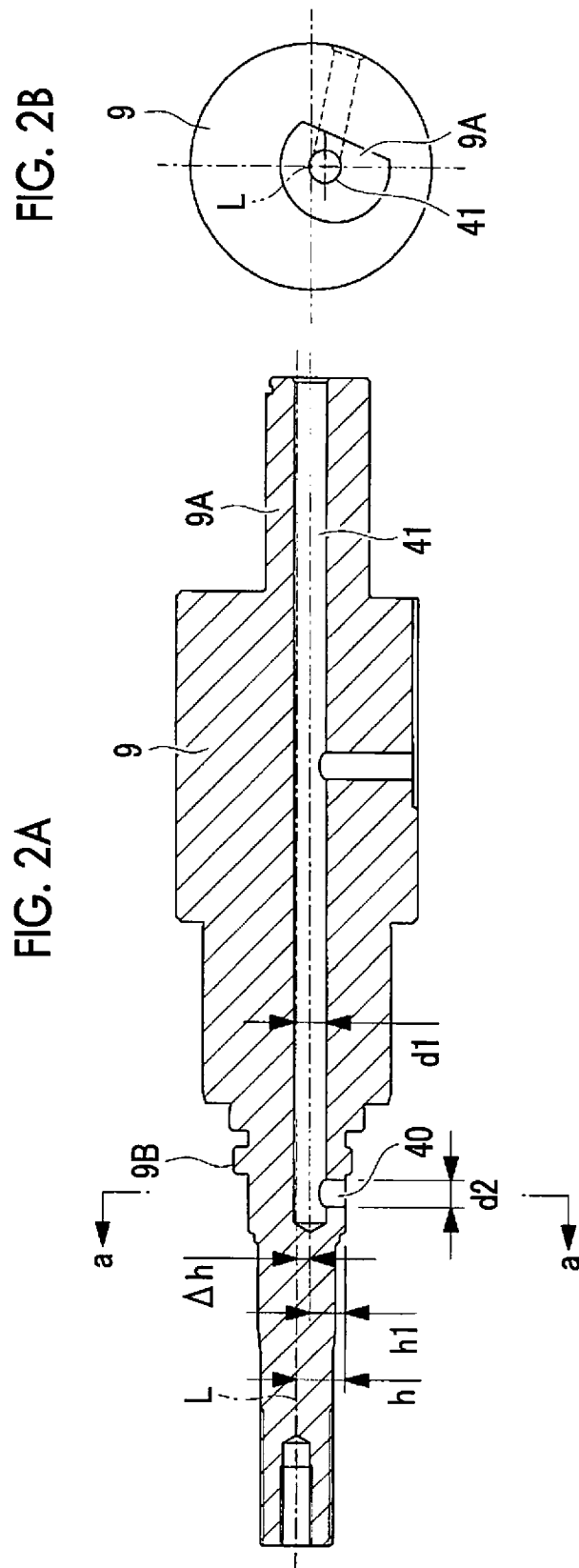


FIG. 3

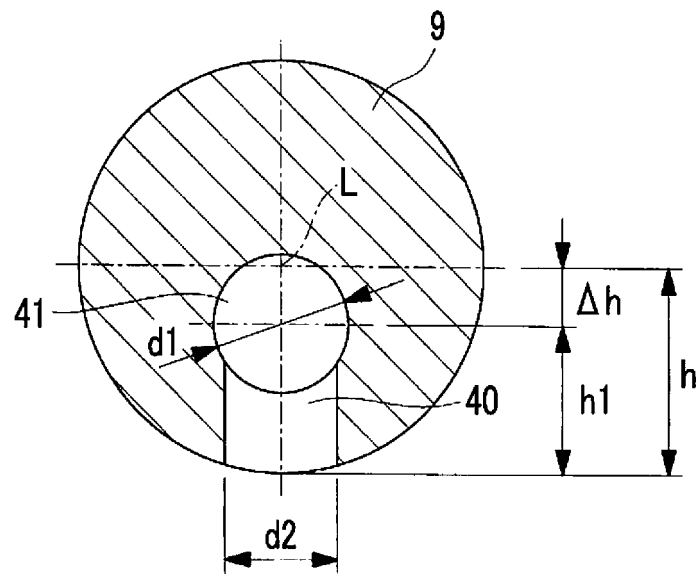


FIG. 4

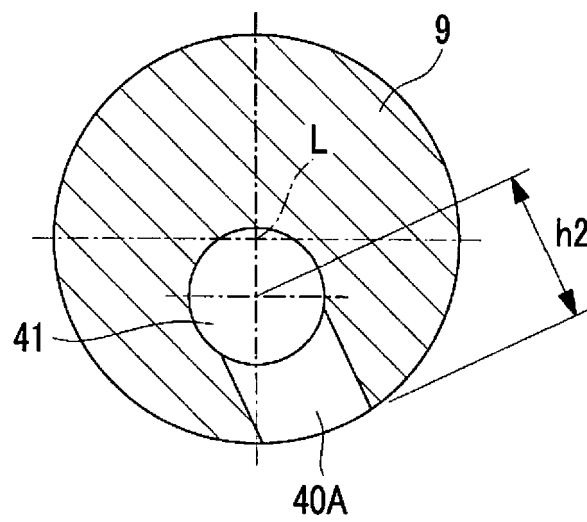
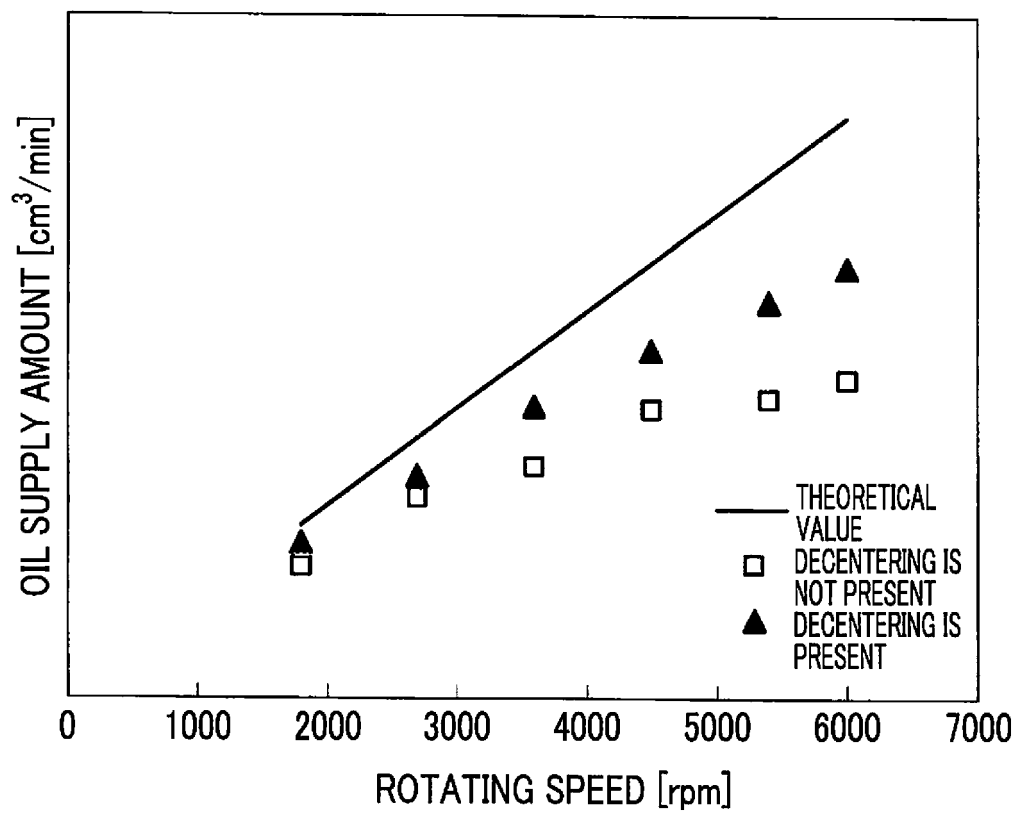


FIG. 5



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2015/072069

A. CLASSIFICATION OF SUBJECT MATTER

F04B39/02(2006.01) i, F04C29/02(2006.01) i

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F04B39/02, F04C29/02

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

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2015

Kokai Jitsuyo Shinan Koho 1971-2015 Toroku Jitsuyo Shinan Koho 1994-2015

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.
Y	JP 2005-282446 A (Mitsubishi Heavy Industries, Ltd.), 13 October 2005 (13.10.2005), fig. 1; claims (Family: none)	1-3
Y	JP 5-149277 A (Mitsubishi Heavy Industries, Ltd.), 15 June 1993 (15.06.1993), fig. 3; paragraph [0007] (Family: none)	1-3
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☐ Further documents are listed in the continuation of Box C.☐ See patent family annex.

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