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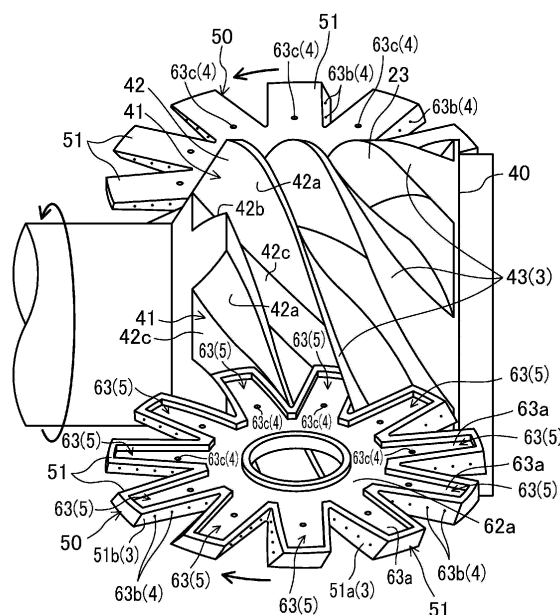
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(54) **SCREW COMPRESSOR**

(57) In at least one of a screw rotor (40) or a gate rotor (50) of a screw compressor (1), an oil supply passage (5) which is connected to an oil supply port (5)

opened at a sliding surface (3) of the rotor (40, 50) and supplies a lubricant to the sliding surface (3) is formed.

FIG.4



Description

TECHNICAL FIELD

[0001] The present invention relates to a screw compressor.

BACKGROUND ART

[0002] As a conventional compressor for compressing a fluid such as a refrigerant or air, a screw compressor having a first rotor comprised of a screw rotor provided with helical grooves, and second rotors which mesh with the first rotor and rotate together with the first rotor has been used (see Patent document 1 below).

[0003] Patent Document 1 discloses a single-screw compressor including a screw rotor as a first rotor which is rotatably housed in a cylindrical wall, and gate rotors as second rotors which are arranged outside the cylindrical wall. Some of gates of each gate rotor enter the internal space of the cylindrical wall through an opening formed therein to mesh with the screw rotor, so that the gate rotors rotate together with the screw rotor. The cylindrical wall, the screw rotor, and the gates meshing with the screw rotor define a compression chamber in the helical grooves. When the screw rotor is driven by an electric motor to rotate, the gates meshing with the screw rotor are pushed to rotate the two gate rotors. When the position of the gate changes in the helical groove, the capacity of the compression chamber decreases to compress the fluid.

[0004] In the conventional screw compressor described above, a lubricant is injected toward the screw rotor from an oil supply port formed at a predetermined position of the cylindrical wall to supply the lubricant between sliding surfaces of two members, such as the screw rotor and the gate, or the screw rotor and the cylindrical wall, thereby lubricating the sliding surfaces, or sealing a minute gap, if any, formed between the two members when they do not slide. This configuration keeps the sliding surfaces of the screw compressor from wearing or seizing, and blocks a high pressure fluid from leaking from the compression chamber defined by the cylindrical wall, the screw rotor, and the gate.

CITATION LIST

PATENT DOCUMENT

[0005] [Patent Document 1] Japanese Published Patent Application No. 2009-197794

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0006] When the lubricant is injected toward the screw rotor from the oil supply port formed at a predetermined

position of the cylindrical wall, just like in the screw compressor described above, the lubricant does not reach the sliding surfaces in some cases when the injection amount of the lubricant is small. Therefore, the screw compressor described above requires the injection of a large amount of lubricant in order to supply the lubricant to the sliding surfaces with reliability.

[0007] However, when a large amount of lubricant is injected into the helical grooves of the screw rotor, the lubricant can be reliably supplied to the sliding surfaces, but power required for transporting the lubricant increases. Further, when a large amount of lubricant is supplied into the helical grooves of the screw rotor, an excess of the lubricant blocks the screw rotor from rotating, which increases power required for the rotation of the screw rotor. When the screw compressor is improved in speed and reduced in size, such an increase in the required power has been a problem because the efficiency of the compressor remarkably decreases.

[0008] In view of the foregoing, it is therefore an object of the present invention to provide a configuration of a screw compressor in which the lubricant can be reliably supplied to the sliding surfaces while reducing the supply amount of the lubricant.

SOLUTION TO THE PROBLEM

[0009] A first aspect of the present disclosure is directed to a screw compressor comprising: a first rotor (40) provided with a helical groove (41); a second rotor (50) which meshes with the first rotor (40) and rotates together with the first rotor (40); a rotor casing (30) which covers at least an outer periphery of the first rotor (40), and defines a compression chamber (23) in the helical groove (41) together with the first rotor (40) and the second rotor (50), wherein a fluid is compressed in the compression chamber (23), and at least one of the first rotor (40) or the second rotor (50) is provided with an oil supply passage (5) which is connected to an oil supply port (4) opened at a sliding surface (3) of the rotor (40, 50) to supply a lubricant to the sliding surface (3).

[0010] In the first aspect of the present disclosure, the oil supply passage (5) is formed in at least one of the rotors (40, 50), i.e., the first rotor (40) and the second rotor (50) which mesh with each other and rotate together, and the oil supply passage (5) is connected to the oil supply port (4) opened at the sliding surface (3) of the rotor (40, 50) in which the oil supply passage (5) is formed. Thus, in the rotor (40, 50) provided with the oil supply passage (5), the lubricant in the oil supply passage (5) flows from the oil supply port (4) to the sliding surface (3) to lubricate the sliding surface (3), or seal a gap, if any, between the sliding surface (3) and its counterpart sliding surface.

[0011] Further, in the first aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in a rotor casing which does not rotate, the oil supply port (4)

is opened at the sliding surface (3) of the rotor (40, 50) that rotates, from which the lubricant is allowed to flow to the sliding surface (3). Therefore, the lubricant flowing from the oil supply port (4) is rapidly spread over the rotating rotor (40, 50), and is rapidly supplied to the sliding surface (3) other than the sliding surface (3) at which the oil supply port (4) is formed. Since the first rotor (40) and the second rotor (50) mesh with each other and rotate together, the lubricant supplied to one of the rotors (40, 50) provided with the oil supply passage (5) is rapidly spread to the other rotor (50, 40). Thus, the lubricant is quickly supplied to the sliding surface (3) of the other rotor (50, 40).

[0012] A second aspect of the present disclosure is an embodiment of the first aspect. In the second aspect, a switching mechanism (6) switches the oil supply passage (5) between a supply state in which the lubricant is supplied to the sliding surface (3) and a non-supply state in which no lubricant is supplied to the sliding surface (3).

[0013] In the second aspect of the present disclosure, the oil supply passage (5) can be switched between the supply state in which the lubricant is supplied from the oil supply passage (5) to the sliding surface (3), and the non-supply state in which no lubricant is supplied from the oil supply passage (5) to the sliding surface (3).

[0014] A third aspect of the present disclosure is an embodiment of the second aspect. In the third aspect, the switching mechanism (6) is configured to switch the oil supply passage (5) to the supply state by causing an oil supply source (94c, 95c) for supplying the lubricant to the oil supply passage (5) to communicate with the oil supply passage (5) when a rotational angle position of the rotor (40, 50) provided with the oil supply passage (5) is in a predetermined angular range, and to switch the oil supply passage (5) to the non-supply state by blocking the oil supply source (94c, 95c) from the oil supply passage (5) when the rotational angle position of the rotor (40, 50) is out of the predetermined angular range.

[0015] In the third aspect of the present disclosure, when the rotational angle position of the rotor (40, 50) provided with the oil supply passage (5) is in the predetermined angle range, the oil supply source (94c, 95c) communicates with the oil supply passage (5), and the oil supply passage (5) is switched to the supply state. When the rotational angle position of the rotor (40, 50) is out of the predetermined angle range, the oil supply source (94c, 95c) and the oil supply passage (5) are blocked from each other, and the oil supply passage (5) is switched to the non-supply state.

[0016] A fourth aspect of the present disclosure is an embodiment of any one of the first to third aspects. In the fourth aspect, the first rotor (40) is a screw rotor (40) rotatably housed in a cylindrical wall (30) constituting the rotor casing (30), the second rotor (50) is a gear-shaped gate rotor (50) having a plurality of flat gates (51) and arranged outside the cylindrical wall (30), some of the gates (51) entering a space inside the cylindrical wall (30) via an opening (39) formed in the cylindrical wall (30) and

meshing with the screw rotor (40) so that the gate rotor (50) rotates together with the screw rotor (40), the oil supply passage (5) is formed in at least one of the gates (51) of the gate rotor (50), and the oil supply port (4) is a lateral oil supply port (63b) opened at a side surface (51a, 51b) of the at least one gate (51), the side surface (51a, 51b) serving as the sliding surface (3) which slides on the screw rotor (40).

[0017] In the fourth aspect of the present disclosure, the screw compressor (1) is configured as a single-screw compressor (1), and the gate rotor (50) which meshes with the screw rotor (40) rotates as the screw rotor (40) rotates. As a result, the position of the gate (51) in the helical groove (41) of the screw rotor (40) changes, the capacity of the compression chamber (23) gradually decreases, and the fluid is compressed. At this time, the lubricant in the oil supply passage (5) formed in the gate (51) of the gate rotor (50) flows from the lateral oil supply port (63b) opened at the side surface (51a, 51b) of the gate (51) sliding on the screw rotor (40). Thus, the lubricant is supplied between the side surface (51a, 51b) of the gate (51) and the screw rotor (40), thereby lubricating these sliding surfaces (3), or sealing a gap, if any, between these sliding surfaces (3). The lubricant supplied between the side surface (51a, 51b) of the gate (51) and the screw rotor (40) adheres to the screw rotor (40), and is spread toward the outer periphery of the screw rotor (40) by the effect of a centrifugal force generated by the rotation of the screw rotor (40). Thus, the lubricant is supplied to a gap between the screw rotor (40) and the cylindrical wall (30) to seal the gap.

[0018] A fifth aspect of the present disclosure is an embodiment of the fourth aspect. In the fifth aspect, the lateral oil supply port (63b) is opened at least at one of side surfaces (51b), including the side surface (51a, 51b), on a rear side in a direction of rotation of the at least one gate (51).

[0019] When the screw rotor (40) rotates, the lateral face of the helical groove (41) of the screw rotor (40) pushes the gate (51) to rotate the gear-shaped gate rotor (50) meshing with the screw rotor (40). Specifically, the side surface (51b) on the rear side in the rotation direction of the gate (51) is the sliding surface which reliably slides on the screw rotor (40) and is pushed by the screw rotor (40), and therefore, is probably worn through the sliding movement.

[0020] In the fifth aspect of the present disclosure, the lubricant is directly supplied to the rear side surface (51b) of the gate (51) in the rotation direction from the oil supply passage (5). This makes it possible to reliably supply the lubricant to the gap between the rear side surface (51b) of the gate (51) in the rotation direction, which is probably worn through the sliding movement, and the lateral faces of the helical groove (41) of the screw rotor (40), thereby lubricating the sliding surfaces (3).

[0021] A sixth aspect of the present disclosure is an embodiment of the fourth or fifth aspect. In the sixth aspect, the oil supply passage (5) is connected to a front

oil supply port (63c) opened at a front surface (51c) of the at least one gate (51) facing the compression chamber (23).

[0022] The rotation of the gate rotor (50) causes the gate (51) to come in and out of the space inside the cylindrical wall (30) via the opening (39). In general, a gap is formed between the front surface (51c) of the gate (51) and the cylindrical wall (30), but the front surface (51c) of the gate (51) may slide on the cylindrical wall (30) when the gate rotor (50) thermally expands. If the gap is present between the front surface (51c) of the gate (51) and the cylindrical wall (30), the lubricant may possibly leak from the high pressure compression chamber (23) through the gap to a low-pressure space outside the cylindrical wall (30) in which the gate rotor (50) is disposed. Thus, the gap needs to be sealed.

[0023] In the sixth aspect of the present disclosure, the oil supply passage (5) is also connected to the front oil supply port (63c) opened at the front surface (51c) of the gate (51). Therefore, in the gate (51) of the gate rotor (50), the lubricant in the oil supply passage (5) is supplied not only to the side surface (51a, 51b) that slide on the screw rotor (40) but also to the front surface (51c) that faces the compression chamber (23). Thus, the lubricant is supplied between the front surface (51c) of the gate (51) and the cylindrical wall (30), thereby lubricating the front surface (51c) and the cylindrical wall (30), or sealing a gap, if any, formed between the front surface (51c) and the cylindrical wall (30).

[0024] A seventh aspect of the present disclosure is an embodiment of any one of the fourth to sixth aspects. In the seventh aspect, the lateral oil supply port (63b) includes at least one lateral oil supply port (63b) formed at a position closer to a base end of the at least one gate (51) than a center, of the at least one gate (51), in a radial direction of the gate rotor (50).

[0025] In the seventh aspect of the present disclosure, the lubricant in the oil supply passage (5) is supplied to a portion of the side surface (51a, 51b) of the gate (51) sliding on the screw rotor (40) closer to the base end than the center thereof in the radial direction. The lubricant supplied to the portion of the side surface (51a, 51b) of the gate (51) closer to the base end is spread toward the distal end of the gate (51) by the effect of the centrifugal force generated by the rotation of the gate rotor (50).

[0026] An eighth aspect of the present disclosure is an embodiment of any one of the fourth to seventh aspects. In the eighth aspect, the screw compressor (1) further comprises a support member (55) supporting the gate rotor (50) from a rear side opposite to the compression chamber (23), wherein an oil sump (62) to which the lubricant is supplied is formed between the support member (55) and a coupling portion (52) of the gate rotor (50) coupling base ends of the plurality of gates (51), and the oil supply passage (5) extends in a radial direction of the gate rotor (50) of the at least one gate (51), and has a base end connected to the oil sump (62).

[0027] In the eighth aspect of the present disclosure,

the oil supply passage (5) extends radially outward from the oil sump (62) closer to the base end than the gate (51). In this configuration, the gate rotor (50) rotates to generate the centrifugal force, which causes the lubricant to enter and flow radially outward through the oil supply passage (5) extending from the oil sump (62) along the gate (51), and flow from the lateral oil supply port (63b) to be supplied between the side surface (51a, 51b) of the gate (51) and the screw rotor (40).

[0028] A ninth aspect of the present disclosure is an embodiment of any one of the first to third aspects. In the ninth aspect, the oil supply passage (5) is formed in the first rotor (40), and the oil supply port (4) is an in-groove oil supply port (66d) opened at an inner surface (42) of the helical groove (41) serving as the sliding surface (3) of the first rotor (40) sliding on the second rotor (50).

[0029] In the ninth aspect of the present disclosure, the oil supply passage (5) is formed in the first rotor (40), and connected to the in-groove oil supply port (66d) opened at the inner surface (42) of the helical groove (41) of the first rotor (40). In the first rotor (40) configured in this manner, the lubricant in the oil supply passage (5) flows from the in-groove oil supply port (66d) to the inner surface (42) of the helical groove (41) which slides on the second rotor (50), thereby lubricating the inner surface (42) of the helical groove, or sealing a gap, if any, between the inner surface (42) and the second rotor (50) sliding on the inner surface (42). That is, in the ninth aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing to be indirectly supplied to the inner surface (42) of the helical groove of the first rotor (40), the lubricant is directly supplied to the inner surface (42) of the helical groove serving as the sliding surface (3) from the in-groove oil supply port (66d) opened at the inner surface (42) of the helical groove of the first rotor (40).

[0030] Further, in the ninth aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing that does not rotate, the oil supply port (4) is opened at the inner surface (42) of the helical groove of the first rotor (40) that rotates, from which the lubricant is allowed to flow to the inner surface (42). Therefore, the lubricant which has flowed from the in-groove oil supply port (66d) is rapidly spread over the rotating first rotor (40) by the effect of the centrifugal force, and thus, the lubricant is quickly supplied to the sliding surfaces (3) other than the inner surface (42). Further, the lubricant supplied to the inner surface (42) of the helical groove of the first rotor (40) adheres to the second rotor (50) which meshes with and rotates with the first rotor (40), and is rapidly spread over the second rotor (50) by the effect of the centrifugal force. Thus, the lubricant is quickly supplied to the sliding surface (3) of the second rotor (50).

[0031] A tenth aspect of the present disclosure is an embodiment of any one of the first to third aspects. In the tenth aspect, the oil supply passage (5) is formed in the

first rotor (40), and the oil supply port (4) is an outer peripheral oil supply port (66c) opened at an outer peripheral surface (43) of the first rotor (40) serving as the sliding surface (3) of the first rotor (40) sliding on the rotor casing (30).

[0032] The outer peripheral surface (43) of the first rotor (40) provided with the helical grooves (41) slides on the inner surface of the rotor casing (30) covering the outer periphery of the first rotor (40). Thus, lubrication is required to keep the outer peripheral surface (43) of the first rotor (40) and the inner surface of the rotor casing (30) from seizing. On the other hand, when a gap is formed between the outer peripheral surface of the first rotor (40) and the inner surface of the rotor casing (30), the gap needs to be sealed so that the high pressure fluid does not leak to the low pressure side.

[0033] In the tenth aspect of the present disclosure, the oil supply passage (5) is formed in the first rotor (40), and connected to the outer peripheral oil supply port (66c) opened at the outer peripheral surface (43) of the first rotor (40) that slides on the rotor casing (30). In the first rotor (40) configured in this manner, the lubricant in the oil supply passage (5) flows from the outer peripheral oil supply port (66c) to the outer peripheral surface (43) of the first rotor (40) that slides on the inner surface of the rotor casing (30), thereby lubricating the outer peripheral surface (43), or sealing a gap, if any, between the outer peripheral surface (43) and the inner surface of the rotor casing (30).

[0034] Further, in the tenth aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing that does not rotate, the oil supply port (4) is opened at the outer peripheral surface (43) of the first rotor (40) that rotates, from which the lubricant is allowed to flow to the outer peripheral surface (43). Therefore, the lubricant that has flowed from the outer peripheral oil supply port (66c) is rapidly spread over the rotating first rotor (40), and is quickly supplied to the sliding surfaces (3) other than the outer peripheral surface (43) at which the outer peripheral oil supply port (66c) is formed. Since the first rotor (40) and the second rotor (50) mesh with each other to rotate together, the lubricant supplied to the first rotor (40) is rapidly spread to the second rotor (50). Thus, the lubricant can be quickly supplied to the sliding surface (3) of the second rotor (50).

[0035] An eleventh aspect of the present disclosure is an embodiment of the ninth or tenth aspect. In the eleventh aspect, the first rotor (40) has an oil sump (44) to which the lubricant is supplied, the oil sump (44) being formed at a position closer to a rotation axis of the first rotor (40) than a bottom face (42c) of the helical groove (41), and the oil supply passage (5) extends from the oil sump (44) toward an outer periphery of the first rotor (40).

[0036] In the eleventh aspect of the present disclosure, the oil supply passage (5) extends from the oil sump (44) closer to the rotation axis than the bottom face (42c) of the helical groove (41) of the first rotor (40) toward the

outer periphery of the first rotor (40). In this configuration, the first rotor (40) rotates to generate the centrifugal force, which causes the lubricant to enter the oil supply passage (5) from the oil sump (44), flow toward the outer periphery of the first rotor (40), and flow from the oil supply port (4) to be supplied to the sliding surface (3) of the first rotor (40).

ADVANTAGES OF THE INVENTION

[0037] According to the first aspect of the present disclosure, the oil supply passage (5) is formed in at least one of the rotors (40, 50), i.e., the first rotor (40) and the second rotor (50) which mesh with each other and rotate together, and the oil supply passage (5) is connected to the oil supply port (4) opened at the sliding surface (3) of the rotor (40, 50) so that the lubricant is directly supplied from the oil supply port (4) to the sliding surface (3). This makes it possible to reliably supply the lubricant to the sliding surface (3) of the rotor (40, 50).

[0038] Further, according to the first aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing which does not rotate, the oil supply port (4) is opened at the sliding surface (3) of the rotor (40, 50) that rotates, from which the lubricant is allowed to flow to the sliding surface (3). Therefore, the lubricant that has flowed from the oil supply port (4) is rapidly spread over the rotating rotor (40, 50), and can be quickly supplied to the sliding surface (3) other than the sliding surface (3) at which the oil supply port (4) is formed. Since the first rotor (40) and the second rotor (50) mesh with each other and rotate together, the lubricant supplied to one of the rotors (40, 50) in which the oil supply passage (5) is formed is rapidly spread to the other rotor (50, 40). Thus, the lubricant can be quickly supplied to the sliding surface (3) of the other rotor (50, 40).

[0039] As described above, according to the first aspect of the present disclosure, the efficiency of the compressor is not lowered because it is unnecessary to increase the power for the transport of the lubricant and the power for the rotation of the first and second rotors (40, 50), unlike in the conventional configuration in which a large amount of lubricant is supplied. Supplying the lubricant in a small amount to at least one of the sliding surface (3) of the first rotor (40) or the sliding surface (3) of the second rotor (50) makes it possible to lubricate the sliding surface (3) of each of the first rotor (40) and the second rotor (50), or to seal the gap, if any, between the sliding surface (3) and its counterpart sliding surface. That is, according to the first aspect of the present disclosure, the sliding surfaces (3) of the first rotor (40) and the second rotor (50) can be kept from seizing, and the high pressure fluid can be blocked from leaking from the compression chamber, even if the supply amount of the lubricant is reduced. Therefore, according to the first aspect of the present disclosure, the supply amount of the

lubricant can be reduced without lowering the reliability of the screw compressor (1), which can improve the compressor efficiency.

[0040] According to the second aspect of the present disclosure, the oil supply passage (5) can be switched between the supply state in which the lubricant is supplied from the oil supply passage (5) to the sliding surface (3), and the non-supply state in which no lubricant is supplied from the oil supply passage (5) to the sliding surface (3). Thus, in a situation where the sliding surface (3) of the rotor (40, 50) provided with the oil supply port (4) is not configured to slide constantly, the oil supply passage can be switched to the non-supply state to stop the supply of the lubricant to the sliding surface (3) when the sliding surface (3) does not slide and requires no lubrication. Therefore, according to the second aspect of the present disclosure, the lubricant can be reliably supplied to the sliding surface (3) of the rotor (40, 50), while reducing the amount of the lubricant supplied.

[0041] In the third aspect of the present disclosure, when the rotational angle position of the rotor (40, 50) provided with the oil supply passage (5) is in the predetermined angle range, the oil supply source (94c, 95c) communicates with the oil supply passage (5), and the oil supply passage (5) is switched to the supply state. When the rotational angle position of the rotor (40, 50) is out of the predetermined angle range, the oil supply source (94c, 95c) and the oil supply passage (5) are blocked from each other, and the oil supply passage (5) is switched to the non-supply state. Such a simple configuration of the third aspect of the present disclosure makes it possible to automatically switch the oil supply passage (5) between the supply state and the non-supply state while the rotor (40, 50) provided with the oil supply passage (5) makes a single rotation.

[0042] According to the fourth aspect of the present disclosure, each of the gates (51) of the gate rotor (50) is provided with the oil supply passage (5) which directly supplies the lubricant to the side surface (51a, 51b) which slide on the screw rotor (51) and need to be lubricated and sealed by the lubricant. Thus, as compared to the conventional configuration in which the lubricant is injected into the helical groove (41) to be indirectly supplied to the sliding surfaces (3) of the gate rotor (50) and the screw rotor (40), the lubricant can be reliably supplied to the sliding surfaces (3) of the gate (51) and the screw rotor (40) in a smaller amount, thereby lubricating the sliding surfaces (3), or sealing a gap, if any, between the sliding surfaces (3). Moreover, the lubricant supplied in this manner to the sliding surfaces (3) of the screw rotor (40) and the gate (51) also adheres to the screw rotor (40), and is spread toward the outer periphery of the screw rotor (40) by the effect of the centrifugal force generated by the rotation of the screw rotor (40). Thus, the lubricant can also be supplied to a gap between the screw rotor (40) and the cylindrical wall (30) to seal the gap.

[0043] As described above, according to the fourth aspect of the present disclosure, the efficiency of the com-

pressor is not lowered because it is unnecessary to increase the power for the transport of the lubricant and the power for the rotation of the screw rotor (40), unlike in the conventional configuration in which a large amount of lubricant is supplied. Directly supplying the lubricant in a small amount to the sliding surfaces (3) of the gate (51) and the screw rotor (40) makes it possible to lubricate the gate (51) and the screw rotor (40), and the screw rotor (40) and the cylindrical wall (30), and to seal the gap between the gate (51) and the screw rotor (40), and the gap between screw rotor (40) and the cylindrical wall (30), if any. That is, according to the fourth aspect of the present disclosure, the gate rotor (50) and the screw rotor (40) can be kept from seizing, and the high pressure fluid can be blocked from leaking from the compression chamber, even if the supply amount of the lubricant is reduced. Therefore, according to the fourth aspect of the present disclosure, the supply amount of the lubricant can be reduced without lowering the reliability of the single-screw compressor (1), which can improve the compressor efficiency.

[0044] According to the fifth aspect of the present disclosure, the lateral oil supply port (63b) of the oil supply passage (5) is opened at least at the side surface (51b) of the gate (51) on the rear side in the direction of rotation of the gate (51). The rear side surface (51b) in the rotation direction of the gate (51) is the sliding surface (3) which reliably slides on the screw rotor (40) and is pressed by the screw rotor (40), and therefore, is probably worn through the sliding movement. However, the lateral oil supply port (63b) opened at the rear side surface (51b) causes the lubricant to be reliably supplied between the rear side surface (51b) and the lateral face of the helical groove (41). This can protect the gate (51) and the screw rotor (40) from the sliding wear.

[0045] According to the sixth aspect of the present disclosure, the oil supply passage (5) of the gate (51) is connected to not only the lateral oil supply port (63b) which is opened at the side surface (51a, 51b) that slide on the screw rotor (40) of the gate (51), but also the front oil supply port (63c) which is opened at the front surface (51c) of the gate (51). Thus, in the gate (51) of the gate rotor (50), the lubricant in the oil supply passage (5) can be supplied not only to the side surface (51a, 51b) that slide on the screw rotor (40), but also to the front surface (51c) that faces the compression chamber (23). As a result, the lubricant is supplied between the front surface (51c) of the gate (51) and the cylindrical wall (30) to lubricate the front surface (51c) and the cylindrical wall (30), or seal a gap, if any, between the front surface (51c) and the cylindrical wall (30). This can keep the seizing caused by the sliding movement of the gate (51), and can block the fluid from leaking from the high pressure compression chamber (23) through the gap between the front surface (51c) of the gate (51) and the cylindrical wall (30) to the low-pressure space outside the cylindrical wall (30) where the gate rotor (50) is disposed.

[0046] According to the seventh aspect of the present

disclosure, the lateral oil supply port (63b) opened at the side surface (51a, 51b) of the gate (51) which slides on the screw rotor (40) includes at least one lateral oil supply port (63b) formed at a position closer to the base end of the gate (51) than the center thereof in the radial direction of the gate (51). The at least one lateral oil supply port (63b) formed at the position closer to the base end of the gate (51) than the center thereof in the radial direction makes it possible to supply the lubricant to the base end of the side surface (51a, 51b) of the gate (51), and to easily spread the lubricant toward the distal end of the side surface (51a, 51b) of the gate (51) by utilizing the centrifugal force. This configuration can minimize the number of the lateral oil supply ports (63b), and can further reduce the supply amount of the lubricant.

[0047] According to the eighth aspect of the present disclosure, the oil sump (62) is formed between the support member (55) supporting the gate rotor (50) and the coupling portion (52) of the gate rotor (50) coupling the base ends of the gates (51), and an end of the oil supply passage (5) toward the base ends of the gates (51) is connected to the oil sump (62). That is, the oil supply passage (5) extends radially outward from the oil sump (62) along the corresponding gate (51). In this configuration, the gate rotor (50) rotates to generate the centrifugal force, which causes the lubricant in the oil sump (62) to enter and flow radially outward through the oil supply passage (5) in the gate (51), and flows from the lateral oil supply port (63b) to be supplied between the side surface (51a, 51b) of the gate (51) and the screw rotor (40). That is, this simple configuration can supply the lubricant between the side surface (51a, 51b) of the gate (51) and the screw rotor (40) by utilizing the centrifugal force generated by the rotation of the gate rotor (50).

[0048] According to the ninth aspect of the present disclosure, the oil supply passage (5) is formed in the first rotor (40), and connected to the in-groove oil supply port (66d) opened at the inner surface (42) of the helical groove (41) of the first rotor (40), so that the lubricant is directly supplied from the in-groove oil supply port (66d) to the inner surface (42) of the helical groove, which is the sliding surface (3) which slides on the second rotor (50). Thus, as compared to the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing to be indirectly supplied to the inner surface (42) of the helical groove of the first rotor (40), the lubricant can be reliably supplied in a smaller amount to the inner surface (42) of the helical groove of the first rotor (40). Further, the in-groove oil supply port (66d) is opened at the inner surface (42) of the helical groove of the first rotor (40) that rotates, from which the lubricant is allowed to flow to the inner surface (42). Thus, the lubricant that has flowed from the in-groove oil supply port (66d) is rapidly spread over the rotating first rotor (40), and the lubricant can also be quickly supplied to the sliding surface (3) other than the inner surface (42). The lubricant supplied to the inner surface (42) of the helical groove of the first rotor (40) also adheres to the second

rotor (50) which meshes with and rotates with the first rotor (40), and is rapidly spread over the second rotor (50) by the effect of the centrifugal force. Thus, the lubricant can be quickly supplied to the sliding surface (3) of the second rotor (50).

[0049] According to the tenth aspect of the present disclosure, the oil supply passage (5) is formed in the first rotor (40), and connected to the outer peripheral oil supply port (66c) formed at the outer peripheral surface (43) which slides on the rotor casing (30) of the first rotor (40), so that the lubricant is directly supplied from the outer peripheral oil supply port (66c) to the outer peripheral surface (43) which is the sliding surface (3). This makes it possible to reliably supply the lubricant to the outer peripheral surface (43) of the first rotor (40) which slides on the inner surface of the rotor casing (30).

[0050] Further, according to the tenth aspect of the present disclosure, unlike the conventional configuration in which the lubricant is injected from the oil supply port formed in the rotor casing that does not rotate, the oil supply port (4) is opened at the outer peripheral surface (43) of the first rotor (40) that rotates, from which the lubricant is allowed to flow to the outer peripheral surface (43). Therefore, the lubricant that has flowed from the outer peripheral oil supply port (66c) is rapidly spread over the rotating first rotor (40), and is quickly supplied to the sliding surface (3) other than the outer peripheral surface (43) of the first rotor (40) at which the outer peripheral oil supply port (66c) is formed. Since the first rotor (40) and the second rotor (50) mesh with each other and rotate together, the lubricant supplied to the first rotor (40) is rapidly spread to the second rotor (50). Thus, the lubricant can be quickly supplied to the sliding surface (3) of the second rotor (50).

[0051] According to the eleventh aspect of the present disclosure, the oil sump (44) is formed at a position closer to the rotation axis of the first rotor (40) than the bottom face (42c) of the helical groove (41), and a base end of the oil supply passage (5) is connected to the oil sump (44). That is, the oil supply passage (5) extends from the oil sump (44) in the first rotor (40) toward the outer periphery. In this configuration, the first rotor (40) rotates to generate the centrifugal force, which causes the lubricant to enter the oil supply passage (5) from the oil sump (44), flow toward the outer periphery of the first rotor (40), and flow from the oil supply port (4) to be supplied to the sliding surface (3) of the first rotor (40). That is, this simple configuration can supply the lubricant to the sliding surface (3) of the first rotor (40) by utilizing the centrifugal force generated by the rotation of the first rotor (40).

BRIEF DESCRIPTION OF THE DRAWINGS

[0052]

FIG. 1 is a diagram schematically showing a general configuration of a screw compressor according to a first embodiment.

FIG. 2 is a vertical sectional view illustrating the vicinity of a compression mechanism of the screw compressor.

FIG. 3 is a cross-sectional view illustrating the vicinity of the compression mechanism of the screw compressor.

FIG. 4 is a perspective view illustrating a screw rotor and gate rotors taken out of the screw compressor.

FIG. 5 is an enlarged view illustrating a right side portion of FIG. 3.

FIG. 6 is a perspective view illustrating a support member shown in FIG. 5.

FIG. 7 is a vertical sectional view schematically illustrating the gate rotor and the screw rotor meshing with each other in an enlarged scale.

FIG. 8 is a sectional view illustrating a gate of the gate rotor and an arm of the support member in a helical groove of the screw rotor.

FIG. 9 is an enlarged view of a left side portion of FIG. 3.

FIGS. 10A to 10C are plan views respectively illustrating how a compression mechanism of a single-screw compressor is operated in a suction phase, a compression phase, and a discharge phase.

FIG. 11 is a cross-sectional view corresponding to FIG. 5, illustrating a screw compressor according to a second embodiment.

FIG. 12 is a cross-sectional view corresponding to FIG. 9, illustrating the screw compressor of the second embodiment.

FIG. 13 is a vertical sectional view corresponding to FIG. 7, illustrating the screw compressor of the second embodiment.

FIG. 14 is a sectional view taken along line XIV-XIV in FIGS. 11 and 12.

FIG. 15 is a cross-sectional view illustrating the vicinity of a compression mechanism of a screw compressor of a third embodiment.

DESCRIPTION OF EMBODIMENTS

[0053] Embodiments of the present disclosure will be described in detail with reference to the drawings.

<<First Embodiment>>

[0054] A screw compressor according to a first embodiment is a single-screw compressor (1) provided in a refrigerant circuit for performing a refrigeration cycle, and compresses a refrigerant (fluid).

[0055] As shown in FIG. 1, in the single-screw compressor (1), a compression mechanism (20) and an electric motor (15) driving the compression mechanism are housed in a single casing (10). The single-screw compressor (1) is configured as a semi-hermetic compressor.

[0056] The casing (10) has an outer wall (17) in the shape of a laterally oriented cylinder. Space inside the casing (10) is divided into a low-pressure space (S1) lo-

cated at one of longitudinal ends of the outer wall (17), and a high-pressure space (S2) located at the other longitudinal end. The casing (10) is provided with a suction pipe connector (11) communicating with the low-pressure space (S1), and a discharge pipe connector (12) communicating with the high-pressure space (S2). Although not shown, a low pressure gas refrigerant flowing from an evaporator of a refrigerant circuit in a refrigeration apparatus, such as a chiller system, flows into the low-pressure space (S1) through the suction pipe connector (11). A compressed, high pressure gas refrigerant discharged from the compression mechanism (20) into the high-pressure space (S2) passes through the discharge pipe connector (12), and is supplied to a condenser of the refrigerant circuit.

[0057] Inside the outer wall (17) of the casing (10), the electric motor (15) is arranged in the low-pressure space (S1), and the compression mechanism (20) is arranged between the low-pressure space (S1) and the high-pressure space (S2). The compression mechanism (20) has a drive shaft (21) coupled to the electric motor (15). The electric motor (15) of the single-screw compressor (1) is connected to a commercial power supply (not shown). The electric motor (15) is supplied with an alternating current from the commercial power supply, and rotates at a predetermined rotational speed.

[0058] Inside the outer wall (17) of the casing (10), an oil separator (16a) is disposed in the high-pressure space (S2). The oil separator (16a) separates a lubricant from the refrigerant discharged from the compression mechanism (20). An oil reservoir chamber (16b) for storing the lubricant (lubricating oil) is formed in the high-pressure space (S2) below the oil separator (16a). The lubricant separated from the refrigerant in the oil separator (16a) flows downward and accumulates in the oil reservoir chamber (16b). The lubricant accumulated in the oil reservoir chamber (16b) has high pressure which is substantially equal to the discharge pressure of the refrigerant.

[0059] As shown in FIGS. 2 and 3, the compression mechanism (20) includes a cylindrical wall (rotor casing) (30), a single screw rotor (a first rotor) (40), and two gate rotors (second rotors) (50) which mesh with the screw rotor (40).

[0060] The cylindrical wall (30) is a cylinder-shaped thick wall, and is integrated with the outer wall (17) to be part of the casing (10). The screw rotor (40) is rotatably housed in the cylindrical wall (30). A bearing holder (35) is fitted in a portion of the cylindrical wall (30) closer to the high-pressure space (S2) of the screw rotor (40).

[0061] A drive shaft (21) arranged coaxially with the screw rotor (40) is inserted through the screw rotor (40). The screw rotor (40) and the drive shaft (21) are connected to each other by a key (22). The screw rotor (40) is driven to rotate in the casing (10) by the electric motor (15) disposed on the suction side of the screw rotor (40). One end of the drive shaft (21) is supported by the bearing holder (35) held by the cylindrical wall (30), via a bearing

(36), and the other end is connected to the electric motor (15).

[0062] As shown in FIG. 4, the screw rotor (40) is a metal member which is substantially in the shape of a cylindrical column. The screw rotor (40) is rotatably fitted in the cylindrical wall (30). The screw rotor (40) has an outer diameter slightly smaller than an inner diameter of the cylindrical wall (30), and has an outer peripheral surface (43) which slides on an inner peripheral surface (30a) of the cylindrical wall (30) with a film of the lubricant present therebetween. That is, the outer peripheral surface (43) of the screw rotor (40) is configured as a sliding surface (3) which slides on the inner peripheral surface (30a) of the cylindrical wall (30). The screw rotor (40) has, on its outer periphery, a plurality of helical grooves (41) (six grooves in this embodiment) helically extending from one axial end of the screw rotor (40) to the other.

[0063] Each of the helical grooves (41) of the screw rotor (40) has a left end in FIG. 4 serving as a starting end, and a right end in FIG. 4 serving as a terminal end. A left end (an end on the suction side) of the screw rotor (40) in FIG. 4 is tapered. In the screw rotor (40) shown in FIG. 4, the starting end of the helical groove (41) is opened at the tapered left end face of the screw rotor (40), while the terminal end of the helical groove (41) is not opened at a right end face of the screw rotor (40). An inner surface (42) of the helical groove (41) includes a lateral face (42a) on the front side in a direction of rotation of the screw rotor (40), a lateral face (42b) on the rear side in the direction of rotation, and a bottom face (42c) connecting the bottom ends of the lateral faces (42a, 42b).

[0064] As shown in FIGS. 3 to 5 and FIGS. 7 to 9, each of the gate rotors (50) is a flat member made of a resin. Each gate rotor (50) has a plurality of (eleven in this embodiment) gates (51), each of which is formed in a rectangular plate shape, and a planar coupling portion (52) coupling base ends of the plurality of gates (51). The gate rotor (50) is in the shape of a gear. The two gate rotors (50) are arranged outside the cylindrical wall (30) to be axially symmetric with respect to the rotation axis of the screw rotor (40). The rotation axis of each gate rotor (50) is in a plane orthogonal to the center axis of the screw rotor (40).

[0065] Each of the gate rotors (50) is attached to a support member (55) made of metal. As shown in FIG. 6, the support member (55) includes a base (56), arms (57), and a shaft (58). The base (56) is in the shape of a relatively thick disk. The arms (57) are provided in the same number (eleven in this embodiment) as the gates (51) of the gate rotor (50), and extend radially outward from an outer peripheral surface of the base (56). Each of the arms (57) abuts on a rear surface of an associated one of the gates (51), thereby supporting the gate (51) from the rear side. The shaft (58) is in a rod shape and coupled to a center portion of the base (56). The shaft (58) has a center axis which coincides with the center axis of the base (56). The shaft (58) penetrates through

the center portion of the gate rotor (50), and is formed to extend forward and rearward of the gate rotor (50). In this embodiment, the shaft (58) has a front shaft portion (58a) which extends forward of the base (56) and is longer than a rear shaft portion (58b) which extends rearward of the base (56).

[0066] The support members (55) to each of which the gate rotor (50) is attached are respectively housed in gate rotor chambers (90) defined inside the casing (10) to be adjacent to the cylindrical wall (30) (see FIG. 3). Each of the gate rotor chambers (90) communicates with the low-pressure space (S1).

[0067] As shown in an enlarged scale in FIGS. 5 and 9, first and second bearing holders (94, 95) formed as an integral part of the casing (10) are provided in each of the gate rotor chambers (90). Each of the first and second bearing holders (94, 95) has a tubular portion (94a, 95a) having a cylindrical shape and a closed bottom, and a flange (94b, 95b) formed around a base end of the tubular portion (94a, 95a). The tubular portion (94a, 95a) of each of the first and second bearing holders (94, 95) is inserted into the gate rotor chamber (90) through an opening formed in the casing (10), and the flange (94b, 95b) is fixed to a portion around the opening of the casing (10). A bearing (92) is held at a distal end of the tubular portion (94a) of the first bearing holder (94), and a bearing (93) is held at a distal end of the tubular portion (95a) of the second bearing holder (95).

[0068] The inside of the tubular portion (94a) of the first bearing holder (94) serves as an oil sump (94c) which stores the lubricant to be supplied to the bearing (92) at the distal end thereof. The inside of the second bearing holder (95) serves as an oil sump (95c) which stores the lubricant to be supplied to the bearing (93) at the distal end thereof. The oil sumps (94c, 95c) communicate with the oil reservoir chamber (16b) formed in the high-pressure space (S2) through a passage (not shown). Each of the oil sumps (94c, 95c) stores the high pressure lubricant supplied from the oil reservoir chamber (16b) through the passage (not shown), and the lubricant reaches a sliding portion of the bearing (93, 94) to lubricate the sliding portion.

[0069] The support member (55) on the right of the screw rotor (40) and the support member (55) on the left of the screw rotor (3) in FIG. 3 are inverted from each other in the vertical direction. Specifically, the support member (55) on the right in FIG. 3 has the front shaft portion (58a) located above the rear shaft portion (58b) (see FIG. 5). The support member (55) on the left in FIG. 3 has the front shaft portion (58a) located below the rear shaft portion (58b) (see FIG. 9). The front shaft portion (58a) of each support member (55) is rotatably supported by the second bearing holder (95) in each gate rotor chamber (90) via the bearing (93), and the rear shaft portion (58b) of each support member (55) is rotatably supported by the first bearing holder (94) in each gate rotor chamber (90) via the bearing (92).

[0070] The casing (10) is provided with an opening (13)

through which an assembly of the gate rotor (50) and the support member (55) can be inserted into the inside of the gate rotor chamber (90) from the outside of the casing (10), and a cover member (14) for covering the opening (13).

[0071] The cylindrical wall (30) has an opening (39) which allows each of the gate rotor chambers (90) to communicate with a screw rotor chamber formed inside the cylindrical wall (30). In each of the gate rotor chambers (90), the assembly of the gate rotor (50) and the support member (55) is disposed at a position where the gate (51) enters the inside of the cylindrical wall (30) through the opening (39) and meshes with the screw rotor (40) (enters the helical groove (41)). An end face of the cylindrical wall (30) forming the opening (39) and facing a front surface (51c) of the gate (51) toward the compression chamber (23) serves as a sealing surface (39a). The sealing surface (39a) is a flat surface extending in the axial direction of the screw rotor (40) along the outer periphery of the screw rotor (40). A distance between each gate rotor (50) and the sealing surface (39a) is set to be very small (e.g., 40 μm or less) so that the leakage of the fluid compressed in the compression chamber (23) to the gate rotor chamber (90) is reduced as much as possible.

[0072] In the compression mechanism (20), a space surrounded by the inner peripheral surface (30a) of the cylindrical wall (30), the inner surface (42) forming the helical groove (41) of the screw rotor (40), and the front surface (51c) of the gate (51) of the gate rotor (50) functions as the compression chamber (23) for compressing the fluid. An end of the helical groove (41) of the screw rotor (40) on the suction side is opened toward the low-pressure space (S1), and this open end serves as a suction port (24) of the compression mechanism (20).

[Unloading Mechanism]

[0073] The single-screw compressor (1) is provided with an unloading mechanism (70, 80) which controls an operating capacity by performing an unloading operation of returning a portion of the gas in the course of the compression to a low pressure side. The unloading mechanism (70, 80) is composed of slide valves (70) and a slide valve driving mechanism (80).

[0074] The slide valves (70) are respectively arranged in slide valve housings (31). As shown in FIG. 2, the slide valve housings (31) are formed at two positions in the circumferential direction of the cylindrical wall (30). Each of the slide valves (70) is configured to be slidable in the axial direction of the cylindrical wall (30), and faces the outer peripheral surface (43) of the screw rotor (40) when the slide valve (70) is inserted into an associated one of the slide valve housings (31). The slide valve (70) is fully opened when it moves to an end toward the discharge side (the right side) in FIG. 2, or fully closed when it moves to an end toward the suction side.

[0075] In the casing (10), communication passages

(32) are formed outside the cylindrical wall (30). The communication passages (32) are formed in one-to-one correspondence with the slide valve housings (31). Each of the communication passages (32) has one end opened in the low-pressure space (S1), and the other end opened at an end on the suction side of the corresponding slide valve housing (31).

[0076] When the slide valves (70) slide toward the high-pressure space (S2) (i.e., to the right when the axial direction of the drive shaft (21) in FIG. 2 is regarded as the lateral direction), axial gaps (G) are formed between end faces of the slide valve housings (31) and end faces of bypass opening degree regulating portions (71) of the slide valves (70). Each axial gap (G) forms, together with an associated one of the communication passages (32), a bypass passage (33) through which the refrigerant in the course of compression in the compression chamber (23) is returned to the low-pressure space (S1). That is to say, the bypass passage (33) has one end communicating to the low-pressure space (S1) corresponding to the suction side of the compression chamber (23), and the other end openable at the inner peripheral surface (30a) of the cylindrical wall (30) where the compression in the compression chamber (23) is in progress. When the slide valves (70) are moved to change the opening degree of the bypass passages (33), a flow rate of the refrigerant returning from the position where the compression is in progress to the low-pressure space varies. As a result, the capacity of the compression mechanism (20) varies.

[0077] Each slide valve (70) includes the bypass opening degree regulating portion (71) for regulating the opening degree of the bypass passage (33), and a discharge opening regulating portion (72) for regulating an opening area of the discharge port (25) which is formed in the cylindrical wall (30) to allow the compression chamber (23) to communicate with the high-pressure space (S2). The slide valves (70) are slidable in the axial direction of the screw rotor (40). The discharge opening regulating portion (72) of the slide valve (70) is configured to vary the opening area of the discharge port (25) in accordance with the change in the position of the slide valve (70).

[0078] The slide valve driving mechanism (80) includes a cylinder tube (81), a piston (82) inserted in the cylinder tube (81), an arm (84) connected to a piston rod (83) of the piston (82), a connecting rod (85) connecting the arm (84) and the slide valve (70), and a spring (86) for biasing the arm (84) to the right in FIG. 2 (in a direction in which the arm (84) is separated from the casing (10)). The cylinder tube (81) and the piston (82) are components forming a hydraulic cylinder (hydropneumatic cylinder) (87). In this embodiment, one of axial end portions of the bearing holder (35) opposite to the screw rotor (40) is configured as the cylinder tube (81). The hydraulic cylinder (87) is disposed across the bearing (36) from the screw rotor (40), and is integrated with the bearing holder (35) holding the bearing (36).

[0079] Inside the bearing holder (35), a partition plate

(38) is provided to define a bearing chamber (C1) where the bearing (36) is held and a cylinder chamber (C2) where the piston (82) of the hydraulic cylinder (87) is housed.

[0080] When the slide valve driving mechanism (80) is in the state shown in FIG. 2, the internal pressure of a space in the cylinder chamber (C2) on the left of the piston (82) (space on the side of the piston (82) toward the screw rotor (40)) is higher than the internal pressure of a space on the right of the piston (82) (space on the side of the piston (82) toward the arm (84)). The slide valve driving mechanism (80) is configured to adjust the position of the slide valves (70) by regulating the internal pressure of the space on the right of the piston (82) (i.e., the gas pressure in the right space). Thus, although not shown, a passage for regulating the pressure in the right space of the piston (82) is formed in the bearing holder (35).

[0081] While the single-screw compressor (1) is in operation, a suction pressure of the compression mechanism (20) acts on one of the axial end faces of each slide valve (70) (i.e., the end face of the bypass opening degree regulating portion (71)), and a discharge pressure of the compression mechanism (20) acts on the other of the axial end faces of each slide valve (70). Consequently, during the operation of the single-screw compressor (1), a force pushing the slide valves (70) toward the low-pressure space (S1) constantly acts on the slide valves (70). Therefore, if the internal pressures of the left and right spaces of the piston (82) of the slide valve driving mechanism (80) vary, the magnitude of a force pulling the slide valves (70) back toward the high-pressure space (S2) varies, which changes the positions of the slide valves (70).

[Oil Supply Mechanism]

[0082] As shown in FIG. 3 and FIGS. 5 to 9, the single-screw compressor (1) is provided with an oil supply mechanism (60) for supplying the lubricant to the side surfaces (51a, 51b) and front surface (51c) of the gate (51) constituting the sliding surface (3) of the gate rotor (50). In this embodiment, the oil supply mechanism (60) is provided for each of the two gate rotors (50). In the following description, the oil supply mechanism (60) which supplies the lubricant to the sliding surface (3) of the gate rotor (50) on the right in FIG. 3, which is enlarged in FIG. 5, will be referred to as a "right oil supply mechanism (60)," and the oil supply mechanism (60) which supplies the lubricant to the sliding surface (3) of the gate rotor (50) on the left in FIG. 3, which is enlarged in FIG. 9, will be referred to as a "left oil supply mechanism (60)." Each of the two oil supply mechanisms (60) has an in-shaft communication passage (61), an oil sump (62), and a plurality of gate-side oil supply passages (63) (oil supply passages (5)).

(Right Oil Supply Mechanism)

[0083] In the right oil supply mechanism (60) shown in FIGS. 5 and 6, the in-shaft communication passage (61) is formed inside the front shaft portion (58a). The in-shaft communication passage (61) includes a longitudinal communication passage (61a) and two lateral communication passages (61b). The longitudinal communication passage (61a) extends straight in the axial direction to pass through the center of the front shaft portion (58a) from one end to the other end thereof. Each of the two lateral communication passages (61b) extends from the other end (an end toward the base (56)) of the longitudinal communication passage (61a) to the outside in a radial direction of the front shaft portion (58a), and is opened at the outer peripheral surface of the front shaft portion (58a).

[0084] The oil sump (62) is formed between the coupling portion (52) coupling base ends of the gates (51) and the base (56), of the support member (55), corresponding to the coupling portion (52). Specifically, a space defined by a groove (62a) formed in the coupling portion (52) of the gate rotor (50) and a groove (62b) formed in the base (56) of the support member (55) is configured as the oil sump (62). The groove (62a) in the gate rotor (50) and the groove (62b) in the support member (55) are formed in an annular shape. As shown in FIG. 6, the groove (62b) formed in the base (56) of the support member (55) is formed in an annular shape to surround the outer periphery of the front shaft portion (58a), and is opened at the front surface of the base (56) facing the gate rotor (50). The two lateral communication passages (61b) of the in-shaft communication passage (61) are opened in the groove (62b). This configuration allows the oil sump (62) to communicate with the oil sump (95c) of the second bearing holder (95) above the front shaft portion (58a) via the in-shaft communication passage (61).

[0085] The gate-side oil supply passages (63) are respectively formed in the gates (51) of the gate rotor (50). In this embodiment, the gate-side oil supply passages (63) are formed in all of the eleven gates (51). Each of the gate-side oil supply passages (63) includes a body (53), a plurality of lateral branches (54), and a front branch (59).

[0086] Specifically, as shown in FIG. 5, grooves (63a) extending in the radial direction of the gate rotor (50) are respectively formed in the rear surfaces of the gates (51). The grooves (63a) are closed by front surfaces of the arms (57) respectively supporting the gates (51) from the rear side. Space in each of the grooves (63a) closed by the front surfaces of the arms (57) constitutes the body (53) of each of the gate-side oil supply passages (63). As shown in FIG. 7, the body (53) of each gate-side oil supply passage (63) extends radially from the base end to distal end of the gate (51). A base end of the body (53) is connected to the oil sump (62) formed between the coupling portion (52) coupling the base ends of the gates

of the gate rotor (50) and the base (56) of the support member (55).

[0087] As shown in FIGS. 7 and 8, the lateral branches (54) are formed by holes extending from the body (53) in the circumferential direction of the gate rotor (50), and are connected to lateral oil supply ports (63b) which are opened at side surfaces (51a, 51b) of the gate (51). The lateral oil supply ports (63b) constitute oil supply ports (4) for supplying the lubricant to the side surfaces (51a, 51b), which are the sliding surfaces (3), of each gate (51). In this embodiment, each of the gates (51) is provided with four lateral branches (54) on the front side, and four lateral branches (54) on the rear side, in the rotation direction thereof. Thus, in this embodiment, four lateral oil supply ports (63b) are opened at the front side surface (51a) in the rotation direction of the gate (51), and four lateral oil supply ports (63b) are opened at the rear side surface (51b). The four lateral oil supply ports (63b) at the front side surface (51a) and the four oil supply ports (63b) at the rear side surface (51b) are provided at positions corresponding to each other. The four lateral oil supply ports (63b) at each side surface (51a, 51b) are arranged at substantially equal intervals from the base end to distal end of the gate (51). The diameter of each of the lateral oil supply ports (63b) and lateral branches (54) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0088] The number of lateral oil supply ports (63b) and lateral branches (54) is not limited to four, but may be less than four, or more than four. In a preferred embodiment, the diameter is changed in accordance with the number so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0089] As shown in FIG. 8, each of the side surfaces (51a, 51b) of the gate (51) which slides on the screw rotor (40) protrudes at a center portion in the thickness direction of the gate. Each of the protruding center portion forms a seal line (L1, L2) which abuts on the corresponding lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40). The lateral oil supply ports (63b) are opened at the side surfaces (51a, 51b) of each gate (51) at a position forward of the seal line (L1, L2), that is, toward the compression chamber (23).

[0090] In this configuration, each of the gate-side oil supply passages (63) is connected to the lateral oil supply ports (63b) opened at the side surfaces (51a, 51b) of the gate (51) which slide on the screw rotor (40).

[0091] As shown in FIGS. 5, 7, and 8, the front branch (59) is a hole which extends in a thickness direction of the gate (51) (a direction parallel to the axial direction of the gate rotor (50)) from the groove (63a) (body (53)) extending in the radial direction of the gate rotor (50) of the gate (51), and is opened at the front surface (51c). The front branch (59) is connected to a front oil supply

port (63c) opened at the front surface (51c) of the gate (51). The front oil supply port (63c) constitutes an oil supply port (4) for supplying the lubricant to the front surface (51c), which is the sliding surface (3), of the gate (51). In this embodiment, the front branch (59) is provided for each of the plurality of gates (51). Thus, in this embodiment, a single front oil supply port (63c) is opened at each of the front surfaces (51c) of the gates (51). In this embodiment, each of the front oil supply ports (63c) is opened at a position further inward than the center of the front surface (51c) of the gate (51) in the radial direction. The diameter of each of the front oil supply ports (63c) and front branches (59) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the front surfaces (51c) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets. The number of front oil supply ports (63c) and front branches (59) is not limited to one, but may be two or more. In a preferred embodiment, the diameter is changed in accordance with the number so that the oil film is formed on the front surfaces (51c) of the gates (51).

[0092] In this configuration, each of the gate-side oil supply passages (63) is connected to the front oil supply port (63c) opened at the front surface (51c) of the gate (51) facing the compression chamber (23).

[0093] As described above, in the right oil supply mechanism (60), the in-shaft communication passage (61), the oil sump (62), and the plurality of gate-side oil supply passages (63), which are formed in the gate rotor (50) and the support member (55), form a lubricant passage which is branched to have two or more outlets. The lubricant passage has an inlet which is opened in the oil sump (95c) of the second bearing holder (95) in which the high pressure lubricant flowing from the oil reservoir chamber (16b) is accumulated. Although some of the plurality of lateral oil supply ports (63b) and the front oil supply port (63c), which are the outlets of the lubricant passage, are opened in the compression chamber (23), most of them are opened in the gate rotor chamber (90) communicating with the low-pressure space (S1). Therefore, due to the pressure difference between the inlet and outlets of the lubricant passage, the high pressure lubricant in the oil sump (95c) enters the lubricant passage, flows toward the outlets, and then flows to the side surfaces (51a, 51b) and front surface (51c) of each gate (51).

(Left Oil Supply Mechanism)

[0094] In the left oil supply mechanism (60) shown in FIG. 9, the in-shaft communication passage (61) is formed inside the rear shaft portion (58b). The in-shaft communication passage (61) includes a longitudinal communication passage (61a) and two lateral communication passages (61b). The longitudinal communication passage (61a) extends straight in the axial direction to pass through the center of the rear shaft portion (58b) from one end to the other end thereof. Each of the two lateral communication passages (61b) extends from the

other end (an end toward the base (56)) of the longitudinal communication passage (61a) to the outside in a radial direction of the rear shaft portion (58b), and is opened at the outer peripheral surface of the rear shaft portion (58b).

[0095] The oil sump (62) is formed between the coupling portion (52) coupling base ends of the gate rotor (50) and the base (56), of the support member (55), corresponding to the coupling portion (52). Specifically, a space defined by a groove (62a) formed in the coupling portion (52) of the gate rotor (50) and a groove (62b) formed in the base (56) of the support member (55) is configured as the oil sump (62). The groove (62a) in the gate rotor (50) and the groove (62b) in the support member (55) are formed in an annular shape. The groove (62b) formed in the base (56) of the support member (55) is in an annular shape to surround the outer periphery of the rear shaft portion (58b), and is opened at the front surface of the base (56) facing the gate rotor (50). The two lateral communication passages (61b) of the in-shaft communication passage (61) are opened in the groove (62b). This configuration allows the oil sump (62) to communicate with the oil sump (94c) of the first bearing holder (94) above the rear shaft portion (58b) via the in-shaft communication passage (61).

[0096] The gate-side oil supply passages (63) are respectively formed in the gates (51) of the gate rotor (50). In this embodiment, the gate-side oil supply passages (63) are formed in all of the eleven gates (51). Each of the gate-side oil supply passages (63) includes a body (53), a plurality of lateral branches (54), and a front branch (59).

[0097] Specifically, as shown in FIG. 9, grooves (63a) extending in the radial direction of the gate rotor (50) are formed in the rear surfaces of the gates (51). The grooves (63a) are closed by front surfaces of the arms (57) respectively supporting the gates (51) from the rear side. Space in each of the grooves (63a) closed by the front surfaces of the arms (57) constitutes the body (53) of each of the gate-side oil supply passages (63). As shown in FIG. 7, the body (53) of each gate-side oil supply passage (63) extends radially from the base end to distal end of the gate (51). A base end of the body (53) is connected to the oil sump (62) formed between the coupling portion (52) coupling the base ends of the gates of the gate rotor (50) and the base (56) of the support member (55).

[0098] As shown in FIGS. 7 and 8, the lateral branches (54) are formed by holes extending from the body (53) of the gate (51) in the circumferential direction of the gate rotor (50), and are connected to lateral oil supply ports (63b) which are opened at the side surfaces (51a, 51b) of the gate (51). The lateral oil supply ports (63b) constitute oil supply ports (4) for supplying the lubricant to the side surfaces (51a, 51b), which are the sliding surfaces (3), of the gate (51). In this embodiment, each of the gates (51) is provided with four lateral branches (54) on the front side, and four lateral branches (54) on the rear side,

in the rotation direction thereof. Thus, in this embodiment, four lateral oil supply ports (63b) are opened at the front side surface (51a) in the rotation direction of the gate (51), and four lateral oil supply ports (63b) are opened at the rear side surface (51b). The four lateral oil supply ports (63b) at the front side surface (51a) and the four oil supply ports (63b) at the rear side surface (51b) are provided at positions corresponding to each other. The four lateral oil supply ports (63b) at each side surface (51a, 51b) are arranged at substantially equal intervals from the base end to distal end of the gate (51). The diameter of each of the lateral oil supply ports (63b) and lateral branches (54) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0099] The number of lateral oil supply ports (63b) and lateral branches (54) is not limited to four, but may be less than four, or more than four. In a preferred embodiment, the diameter is changed in accordance with the number so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0100] As shown in FIG. 8, each of the side surfaces (51a, 51b) of the gate (51) which slides on the screw rotor (40) protrudes at a center portion in the thickness direction of the gate. Each of the protruding center portion forms a seal line (L1, L2) which abuts on the corresponding lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40). The lateral oil supply ports (63b) are opened at the side surfaces (51a, 51b) of each gate (51) at a position forward of the seal line (L1, L2), that is, toward the compression chamber (23).

[0101] In this configuration, each of the gate-side oil supply passages (63) is connected to the lateral oil supply ports (63b) opened at the side surfaces (51a, 51b) of the gate (51) which slide on the screw rotor (40).

[0102] As shown in FIGS. 7 to 9, the front branch (59) is a hole which extends in a thickness direction of the gate (51) (a direction parallel to the axial direction of the gate rotor (50)) from the groove (63a) (body (53)) extending in the radial direction of the gate rotor (50) of the gate (51), and is opened at the front surface (51c). The front branch (59) is connected to a front oil supply port (63c) opened at the front surface (51c) of the gate (51). The front oil supply port (63c) constitutes an oil supply port (4) for supplying the lubricant to the front surface (51c), which is the sliding surface (3), of the gate (51). In this embodiment, the front branch (59) is provided for each of the plurality of gates (51). Thus, in this embodiment, a single front oil supply port (63c) is opened at each of the front surfaces (51c) of the gates (51). In this embodiment, each of the front oil supply ports (63c) is opened at a position further inward than the center of the front surface (51c) of the gate (51) in the radial direction. The diameter of each of the front oil supply ports (63c) and

front branches (59) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the front surfaces (51c) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets. The number of front oil supply ports (63c) and front branches (59) is not limited to one, but may be two or more. In a preferred embodiment, the diameter is changed in accordance with the number so that the oil film is formed on the front surfaces (51c) of the gates (51).

[0103] In this configuration, each of the gate-side oil supply passages (63) is connected to the front oil supply port (63c) opened at the front surface (51c) of the gate (51) facing the compression chamber (23).

[0104] As described above, in the left oil supply mechanism (60), the in-shaft communication passage (61), the oil sump (62), and the plurality of gate-side oil supply passages (63), which are formed in the gate rotor (50) and the support member (55), form a lubricant passage which is branched to have two or more outlets. The lubricant passage has an inlet which is opened in the oil sump (94c) of the first bearing holder (94) in which the high pressure lubricant flowing from the oil reservoir chamber (16b) is accumulated. Although some of the plurality of lateral oil supply ports (63b) and the front oil supply port (63c), which are the outlets of the lubricant passage, are opened in the compression chamber (23), most of them are opened in the gate rotor chamber (90) communicating with the low-pressure space (S1). Therefore, due to the pressure difference between the inlet and outlets of the lubricant passage, the high pressure lubricant in the oil sump (95c) enters the lubricant passage, flows toward the outlets, and then flows to the side surfaces (51a, 51b) and front surface (51c) of each gate (51).

-Operation-

[0105] When the electric motor (15) of the single-screw compressor (1) is actuated, the drive shaft (21) rotates, and the screw rotor (40) rotates as well. As the screw rotor (40) rotates, the gate rotor (50) also rotates, and the compression mechanism (20) repeats a suction phase, a compression phase, and a discharge phase. In the following description, the operation of the screw compressor (1) will be described, focusing on the compression chamber (23) dotted in FIGS. 10A to 10C.

[0106] The compression chamber (23) dotted in FIG. 10A communicates with the low-pressure space (S1). In this state, the gate (51) of the lower gate rotor (50) in FIG. 10A meshes with the corresponding helical groove (41) which defines the compression chamber (23). When the screw rotor (40) rotates, the gate (51) relatively moves within the helical groove (41) toward the terminal end of the helical groove (41), causing the capacity of the compression chamber (23) to gradually increase. As a result, the low pressure gas refrigerant in the low-pressure space (S1) is sucked into the compression chamber (23) through the suction port (24).

[0107] When the screw rotor (40) further rotates, the

operation enters the state of FIG. 10B. The compression chamber (23) dotted in FIG. 10B is fully closed. In this state, the gate (51) of the upper gate rotor (50) in FIG. 10B meshes with the corresponding helical groove (41) which defines the compression chamber (23), and the compression chamber (23) is partitioned from the low-pressure space (S1) by the gate (51). As the screw rotor (40) rotates, the gate (51) relatively moves within the helical groove (41) toward the terminal end of the helical groove (41), causing the capacity of the compression chamber (23) to gradually decrease. As a result, the low pressure gas refrigerant in the compression chamber (23) is gradually compressed.

[0108] When the screw rotor (40) further rotates, the operation enters the state of FIG. 10C. The compression chamber (23) dotted in FIG. 10C communicates with the high-pressure space (S2) through the discharge port (25). In this state, when the gate (51) moves within the helical groove (41) toward the terminal end of the helical groove (41) with the rotation of the screw rotor (40), the compressed, high pressure refrigerant gas (high pressure gas refrigerant) is pushed out of the compression chamber (23) to the high-pressure space (S2).

[0109] When the above operation is performed, the capacity of the compression mechanism (20) is controlled using the slide valve (70). Although not specifically described, when pushed to the leftmost position in FIG. 2, the slide valve (70) comes to the end where the slide valve (70) is fully closed (suction side). In this state, the capacity of the compression mechanism (20) is maximized. When the slide valve (70) moves back to the right in FIG. 3, the tip end face of the slide valve (70) releases the axial gap (G), and the bypass passage (33) opens at the inner peripheral surface of the cylindrical wall (30). Then, a portion of the refrigerant gas sucked into the compression chamber (23) from the low-pressure space (S1) returns to the low-pressure space (S1) from the compression chamber (23) in the course of the compression phase via the bypass passage (33), and the rest of the refrigerant gas is compressed until the end of the compression phase and discharged to the high-pressure space (S2). Thus, the capacity of the compression mechanism (20) decreases.

-Oil Supply Operation-

[0110] In this manner, when the screw rotor (40) and the two gate rotors (50) rotate to compress the refrigerant gas in the compression chamber (23), the two oil supply mechanisms (60) supply the lubricant to the sliding surfaces (3) of the two gate rotors (50) and the screw rotor (40).

[0111] In the two oil supply mechanisms (60), as described above, the pressure difference between the inlet and outlets of the lubricant passage formed by the in-shaft communication passage (61), the oil sump (62), and the plurality of gate-side oil supply passages (63) causes the lubricant supplied to each oil sump (94c, 95c)

from the oil reservoir chamber (16b) to enter the lubricant passage, and flow toward the outlets. Specifically, the lubricant in the oil sump (94c, 95c) flows into the longitudinal communication passage (61a) of the in-shaft communication passage (61) inside the front shaft portion (58a), diverges from the longitudinal communication passage (61a) to the two lateral communication passages (61b), and eventually flows into the oil sump (62) (see FIGS. 5, 6, and 9). The lubricant that has reached the oil sump (62) flows into the plurality of gate-side oil supply passages (63) extending radially from the oil sump (62) by the effect of the driving force caused by the pressure difference described above and the centrifugal force generated by the rotation of the gate rotor (50) and the support member (55), and then flows radially outward in each of the gate-side oil supply passages (63) (see FIGS. 5 and 9). The lubricant flowing through the gate-side oil supply passages (63) flows to the side surfaces (51a, 51b) of the gate (51) from the plurality of lateral oil supply ports (63b), and to the front surface (51c) of the gate (51) from the front oil supply port (63c).

[0112] From the lateral oil supply ports (63b) of each gate (51), the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gate (51). The lubricant that has flowed from the plurality of lateral oil supply ports (63b) is spread radially outward on the side surfaces (51a, 51b) of the gates (51) by the effect of the centrifugal force to form the oil film on each of the side surfaces (51a, 51b).

[0113] As described above, as shown in FIG. 8, the lateral oil supply ports (63b) are opened at each of the side surfaces (51a, 51b) of the gate (51) at a position forward of the seal line (L1, L2) which abuts on the corresponding lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40), that is, further toward the compression chamber (23) than the seal line. Since the lateral oil supply ports (63b) are provided at such positions, the lubricant is supplied to a portion of the side surface (51a, 51b) forward of the seal line (L1, L2) of each gate (51) in the traveling direction of the gate (51) when the gate travels toward the compression chamber (23) in the helical groove (41) of the screw rotor (40). As a result, the lubricant is reliably supplied to the seal line (L1, L2) of each gate (51) which slides on the corresponding lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40). This can lubricate the seal line (L1, L2), and achieve sealing at the seal line. This keeps the gas refrigerant in the high pressure compression chamber (23) from leaking from the gap between the side surface (51a, 51b) of the gate (51) and the lateral face (42a, 42b) of the helical groove (41) of the cylindrical wall (30) to the low pressure compression chamber (23).

[0114] In this manner, the lubricant that has flowed from the lateral oil supply ports (63b) to the side surfaces (51a, 51b) of the gates (51) and supplied to the sliding surfaces (3) of the screw rotor (40) adheres to the screw rotor (40), and is spread to the outer periphery of the screw rotor by the effect of the centrifugal force generated

by the rotation of the screw rotor (40). As a result, an oil film is formed on the outer peripheral surface (43) of the screw rotor (40) between the helical grooves (41), and the outer peripheral surface (43) and the inner peripheral surface (30a) of the cylindrical wall (30) are lubricated and the gap between them is sealed. This keeps the screw rotor (40) from seizing, and blocks the gas refrigerant in the high pressure compression chamber (23) from leaking to the low pressure compression chamber (23) through the gap between the outer peripheral surface (43) of the screw rotor (40) and the inner peripheral surface (30a) of the cylindrical wall (30).

[0115] On the other hand, the lubricant flows from the front oil supply port (63c) of each of the gates (51) in such an amount that allows an oil film to be formed on the front surface (51c) of the gate (51). The lubricant that has flowed from the front oil supply port (63c) is spread radially outward on the front surface (51c) of the gate (51) by the effect of the centrifugal force to form an oil film on the front surface (51c). As described above, each of the front oil supply ports (63c) is opened at a position inward of the center of the front surface (51c) of each gate (51) in the radial direction (see FIG. 7). Therefore, the lubricant that has flowed from the front oil supply port (63c) on the front surface (51c) of the gate (51) is spread widely outward from the radially inward position.

[0116] The rotation of the gate rotor (50) causes each of the gates (51) to come in and out of the cylindrical wall (30) via the opening (39) of the cylindrical wall (30). As described above, the lubricant flowed from the front oil supply port (63c) is widely spread over the front surface (51c) of each gate (51), and is supplied between the front surface (51c) of the gate (51) and the sealing surface (39a) of the cylindrical wall (30) facing each other. Thus, the lubricant lubricates the front surface (51c) of the gate (51) and the sealing surface (39a) of the cylindrical wall (30), which are the sliding surfaces, and seals a gap therebetween. This keeps the gates (51) from seizing, and blocks the gas refrigerant in the high pressure compression chamber (23) from leaking to the gate rotor chamber (90) through the gap between the front surface (51c) of the gate (51) and the sealing surface (39a) of the cylindrical wall (30).

-Advantages of First Embodiment-

[0117] According to the first embodiment, each of the gates (51) of the gate rotor (50) is provided with the gate-side oil supply passage (63) directly supplying the lubricant to the side surfaces (51a, 51b) which slide on the screw rotor (51) and need to be lubricated and sealed by the lubricant. Thus, as compared to the conventional configuration in which the lubricant is injected into the helical groove (41) to be indirectly supplied to the sliding surfaces (3) of the gate rotor (50) and the screw rotor (40), the lubricant can be reliably supplied to the sliding surfaces (3) of the gate (51) and the screw rotor (40) in a smaller amount, thereby lubricating the gate (51) and the screw

rotor (40) and sealing the gap therebetween. Moreover, the lubricant supplied in this manner to the sliding surfaces (3) of the screw rotor (40) and the gate (51) also adheres to the screw rotor (40), and is spread toward the outer periphery of the screw rotor (40) by the effect of the centrifugal force generated by the rotation of the screw rotor (40). Thus, the lubricant can also be supplied to a gap between the screw rotor (40) and the cylindrical wall (30) to seal the gap.

[0118] As described above, in the present embodiment, the efficiency of the compressor is not lowered because it is unnecessary to increase the power for the transport of the lubricant and the power for the rotation of the screw rotor (40), unlike the conventional configuration in which a large amount of lubricant is supplied. Directly supplying the lubricant in a small amount to the sliding surfaces (3) of the gate (51) and the screw rotor (40) makes it possible to lubricate the gate (51) and the screw rotor (40), and the screw rotor (40) and the cylindrical wall (30), and to seal the gap between the gate (51) and the screw rotor (40), and the gap between the screw rotor (40) and the cylindrical wall (30). That is, according to this embodiment, the gate rotor (50) and the screw rotor (40) can be protected from the sliding wear, and a high pressure fluid can be blocked from leaking from the compression chamber, even if the supply amount of the lubricant is reduced. Therefore, in the present embodiment, the supply amount of the lubricant can be reduced without lowering the reliability of the single-screw compressor (1), which can improve the compressor efficiency.

[0119] According to the present embodiment, the gate-side oil supply passage (63) of the gate (51) is provided with not only the lateral oil supply ports (63b) which are opened at the side surfaces (51a, 51b) that slide on the screw rotor (40) of the gate (51), but also the front oil supply port (63c) which is opened at the front surface (51c) of the gate (51). Therefore, in the gate (51) of the gate rotor (50), the lubricant in the gate-side oil supply passage (63) can be supplied not only to the side surfaces (51a, 51b) that slide on the screw rotor (40), but also to the front surface (51c) that faces the compression chamber (23). As a result, the lubricant is supplied between the front surface (51c) of the gate (51) sliding on the surface of the cylindrical wall (30), which lubricates these sliding surfaces, and seals a gap between them. This can keep the seizing caused by the sliding movement of the gate (51), and can block the fluid from leaking from the high pressure compression chamber (23) through the gap between the front surface (51c) of the gate (51) and the cylindrical wall (30) to the low-pressure space outside the cylindrical wall (30) where the gate rotor (50) is disposed.

[0120] Further, in the present embodiment, the oil sump (62) is formed between the support member (55) supporting the gate rotor (50) and the coupling portion (52) of the gate rotor (50) coupling the base ends of the gates, and a base end of the gate-side oil supply passage

(63) in the gate (51) is connected to the oil sump (62). That is, the gate-side oil supply passage (63) extends radially outward from the oil sump (62) along the corresponding gate (51). In this configuration, the gate rotor (50) rotates to generate the centrifugal force, which causes the lubricant in the oil sump (62) to enter and flow radially outward through the gate-side oil supply passage (63) of the gate (51), and flow from the lateral oil supply ports (63b). That is, this simple configuration can supply the lubricant to the sliding surfaces (3) by utilizing the centrifugal force generated by the rotation of the gate rotor (50).

<<Second Embodiment>>

[0121] In a second embodiment, the oil supply mechanism (60) and first and second bearing holders (94, 95) of the single-screw compressor (1) of the first embodiment are partially modified so that the lubricant is supplied intermittently as needed to the sliding surfaces (3) of the gate rotors (50).

[Oil Supply Mechanism]

[0122] Specifically, as shown in FIGS. 11 and 12, the single-screw compressor of the second embodiment has two oil supply mechanisms (60), each of which includes a plurality of in-shaft communication passages (61), a plurality of oil sumps (62), and a plurality of gate-side oil supply passages (63). In the second embodiment, eleven in-shaft communication passages (61), eleven oil sumps (62), and eleven gate-side oil supply passages (63) are provided.

[0123] As shown in FIG. 11, the right oil supply mechanism (60) includes a plurality of in-shaft communication passages (61) formed inside the front shaft portion (58a). As shown in FIG. 12, the left oil supply mechanism (60) includes a plurality of in-shaft communication passages (61) formed inside the rear shaft portion (58b). Each of the in-shaft communication passages (61) includes a longitudinal communication passage (61a) and a lateral communication passage (61b), and is formed in an L-shape.

[0124] As shown in FIG. 11, each of the longitudinal communication passages (61a) in the right oil supply mechanism (60) extends straight in the axial direction to pass through an outer peripheral portion of the front shaft portion (58a) from one end to the other end thereof. As shown in FIG. 12, each of the longitudinal communication passages (61a) in the left oil supply mechanism (60) extends straight in the axial direction to pass through an outer peripheral portion of the rear shaft portion (58b) from one end to the other end thereof.

[0125] As shown in FIG. 11, each of the lateral communication passages (61b) in the right oil supply mechanism (60) extends outward in the radial direction of the front shaft portion (58a) from the other end (an end toward the base (56)) of an associated one of the longitudinal

communication passages (61a), and is opened at an outer peripheral surface of the front shaft portion (58a). As shown in FIG. 12, each of the lateral communication passages (61b) in the left oil supply mechanism (60) extends outward in the radial direction of the rear shaft portion (58b) from the other end (an end toward the base (56)) of an associated one of the longitudinal communication passages (61a), and is opened at an outer peripheral surface of the rear shaft portion (58b).

[0126] Thus, in each of the oil supply mechanisms (60) of the second embodiment, the in-shaft communication passages (61) are formed in the same number (eleven) as the gates (51) to be in one-to-one correspondence with the eleven gates (51). In each oil supply mechanism (60), the eleven in-shaft communication passages (61) are provided at equal intervals in the circumferential direction of the front shaft portion (58a) or the rear shaft portion (58b) so that each of the eleven lateral communication passages (61b) extends in the direction of extension of the corresponding gate (51).

[0127] In each oil supply mechanism (60), the plurality of oil sumps (62) are formed between a coupling portion (52) coupling base ends of the gates of the gate rotor (50) and the base (56), of the support member (55), corresponding to the coupling portion (52). Specifically, a plurality of grooves (62a) formed in the coupling portion (52) of the gate rotor (50) and a plurality of grooves (62b) formed in the base (56) of the support member (55) form a plurality of spaces, which respectively constitute the oil sumps (62). The grooves (62a) of the gate rotor (50) and the grooves (62b) of the support member (55) are formed in the same number (eleven) as the gates (51) to be in one-to-one correspondence with the gates (51).

[0128] As shown in FIGS. 11 and 13, in the right oil supply mechanism (60), the eleven grooves (62b) formed in the base (56) of the support member (55) extend radially outward from the outer peripheral surface of the front shaft portion (58a), and are opened at the front surface of the base (56) facing the gate rotor (50). As shown in FIGS. 12 and 13, in the left oil supply mechanism (60), the eleven grooves (62b) formed in the base (56) of the support member (55) extend radially outward from the outer peripheral surface of the rear shaft portion (58b), and are opened at the front surface of the base (56) facing the gate rotor (50). In each oil supply mechanism (60), each of the eleven lateral communication passages (61b) of the in-shaft communication passage (61) is opened in an associated one of the grooves (62b).

[0129] In each oil supply mechanism (60), the gate-side oil supply passages (63) are respectively formed in the gates (51) of the gate rotor (50). Also in the second embodiment, the gate-side oil supply passages (63) are formed in all of the eleven gates (51). In each of the oil supply mechanisms (60) of the second embodiment, the eleven gate-side oil supply passages (63) are formed in one-to-one correspondence with the eleven oil sumps (62). Each of the gate-side oil supply passages (63) includes a body (53), a plurality of lateral branches (54),

and a front branch (59).

[0130] Specifically, as shown in FIGS. 11 and 12, grooves (63a) extending in the radial direction of each gate rotor (50) are formed in the rear surfaces of the gates (51). The grooves (63a) formed in the gates (51) are formed in one to-one correspondence with the eleven grooves (62a) formed in the coupling portion (52) of the gate rotor (50), and are integrated with the corresponding grooves (62a). The grooves (63a) formed in the gates (51) are closed by front surfaces of the arms (57) respectively supporting the gates (51) from the rear side. Space in each of the grooves (63a) closed by the front surfaces of the arms (57) constitutes the body (53) of each of the gate-side oil supply passages (63). As shown in FIG. 13, the body (53) of each gate-side oil supply passage (63) extends radially from a base end to distal end of the gate (51). A base end of the body (53) is connected to the oil sump (62) formed between the coupling portion (52) coupling the base ends of the gates of the gate rotor (50) and the base (56), of the support member (55), corresponding to the coupling portion (52).

[0131] As shown in FIG. 13, in each of the oil supply mechanisms (60), the lateral branches (54) are formed by holes extending from each body (53) of the gate (51) in the circumferential direction of the gate rotor (50), and are connected to lateral oil supply ports (63b), which are oil supply ports (4) opened at the side surfaces (51a, 51b) of the gate (51). Also in the second embodiment, each of the gates (51) is provided with four lateral branches (54) on the front side, and four lateral branches (54) on the rear side, in the rotation direction thereof. Thus, also in the second embodiment, four lateral oil supply ports (63b) are opened at the front side surface (51a) in the rotation direction of the gate (51), and four lateral oil supply ports (63b) are opened at the rear side surface (51b). The four lateral oil supply ports (63b) at the front side surface (51a) and the four oil supply ports (63b) at the rear side surface (51b) are provided at positions corresponding to each other. The four lateral oil supply ports (63b) at each side surface (51a, 51b) are arranged at substantially equal intervals from the base end to distal end of the gate (51). The diameter of each of the lateral oil supply ports (63b) and lateral branches (54) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0132] The number of lateral oil supply ports (63b) and lateral branches (54) is not limited to four, but may be less than four, or more than four. In a preferred embodiment, the diameter is changed in accordance with the number so that the lubricant flows in such an amount that allows an oil film to be formed on the side surfaces (51a, 51b) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets.

[0133] Also in the second embodiment, as shown in FIG. 8, each of the side surfaces (51a, 51b) of the gate (51) which slides on the screw rotor (40) protrudes at a

center portion in the thickness direction of the gate. Each of the protruding center portion forms a seal line (L1, L2) which abuts on the corresponding lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40). The lateral oil supply ports (63b) are opened at the side surfaces (51a, 51b) of each gate (51) at a position forward of the seal line (L1, L2), that is, toward the compression chamber (23).

[0134] In this configuration of the second embodiment, each of the gate-side oil supply passages (63) in the oil supply mechanisms (60) is connected to the lateral oil supply ports (63b) opened at the side surfaces (51a, 51b) of the gate (51) which slide on the screw rotor (40).

[0135] As shown in FIGS. 11, 12, and 8, the front branch (59) of the second embodiment is a hole which extends in a thickness direction of the gate (51) (a direction parallel to the axial direction of the gate rotor (50)) from the groove (63a) (body (53)) extending in the radial direction of the gate rotor (50) of the gate (51), and is opened at the front surface (51c). The front branch (59) is connected to a front oil supply port (63c) which is the oil supply port (4) opened at the front surface (51c) of the gate (51). Also in the second embodiment, the front branches (59) are respectively provided for the plurality of gates (51), and thus, a single front oil supply port (63c) is opened at each of the front surfaces (51c) of the gates (51). Each of the front oil supply ports (63c) is opened at a position further inward than the center of the front surface (51c) of the gate (51) in the radial direction. Also in the second embodiment, the diameter of each of the front oil supply ports (63c) and front branches (59) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the front surfaces (51c) of the gates (51), and that the lubricant is kept from scattering in the shape of droplets. The number of front oil supply ports (63c) and front branches (59) is not limited to one, but may be two or more. In a preferred embodiment, the diameter is changed in accordance with the number so that the oil film is formed on the front surfaces (51c) of the gates (51).

[0136] In this configuration of the second embodiment, the gate-side oil supply passages (63) in each of the oil supply mechanisms (60) are connected to the front oil supply ports (63c) each of which is opened at the front surface (51c) of the gate (51) facing the compression chamber (23).

[0137] Thus, in each of the oil supply mechanisms (60) of the second embodiment, the plurality of in-shaft communication passages (61), the plurality of oil sumps (62), and the plurality of gate-side oil supply passages (63), which are formed in the gate rotor (50) and the support member (55), form a plurality of lubricant passages.

[Bearing Holder]

[0138] As shown in FIGS. 11 and 12, in the second embodiment, each of the first and second bearing holders (94, 95) has a tubular portion (94a, 95a) having a cylin-

drical shape and a closed bottom, a flange (94b, 95b) formed around a base end of the tubular portion (94a, 95a), and a closing portion (94d, 95d). The tubular portions (94a, 95a) and the flanges (94b, 95b) are configured in the same manner as those of the first embodiment.

[0139] As shown in FIG. 11, in the right oil supply mechanism (60), the closing portion (95d) of the second bearing holder (95) protrudes downward from an inner bottom surface of the tubular portion (95a), and abuts on a top surface of the front shaft portion (58a) of the support member (55) by a lower end thereof, thereby closing inlets of some of the eleven in-shaft communication passages (61) (inlets of the longitudinal communication passages (61a)) formed inside the front shaft portion (58a) of the support member (55). As shown in FIG. 12, in the left oil supply mechanism (60), the closing portion (94d) of the first bearing holder (94) protrudes downward from an inner bottom surface of the tubular portion (94a), and abuts on a top surface of the rear shaft portion (58b) by a lower end thereof, thereby closing inlets of some of the eleven in-shaft communication passages (61) (inlets of the longitudinal communication passages (61a)) formed inside the rear shaft portion (58b) of the support member (55).

[0140] In the second embodiment, as shown in FIG. 14, in each of the oil supply mechanisms (60), the closing portion (94d, 95d) of the bearing holder (94, 95) is configured to keep four of the inlets (61a-1 to 61a-11) of the eleven in-shaft communication passages (61) in the front shaft portion (58a) or the rear shaft portion (58b) closer to the screw rotor (40) open, and close the remaining seven inlets. With the closing portion (94d, 95d) formed in this manner, the oil sump (94c, 95c) formed in each of the first and second bearing holders (94, 95) is formed to have a wider portion on the side closer to the screw rotor (40), and a narrower portion on the other side.

[0141] Note that the front shaft portion (58a) or the rear shaft portion (58b) in which the in-shaft communication passages (58) are formed rotates in accordance with the rotation of the gate rotors (50), but the closing portion (94d, 95d) is fixed and does not rotate. Therefore, the inlets (61a-1 to 61a-11) of the in-shaft communication passages (61) to be closed by the closing portions (94d, 95d) change in accordance with the rotational angle position of the gate rotor (50).

[0142] For example, when the gate rotor (50) is at the rotational angle position shown in FIG. 14, the closing portion (94d, 95d) closes the fifth to eleventh inlets (61a-5 to 61a-11), while keeping the first to fourth inlets (61a-1 to 61a-4) open. Thus, the first to fourth inlets (61a-1 to 61a-4) are opened to the oil sump (94c, 95c). When the gate rotor (50) moves in the direction of the arrow and its rotational angle position changes, the closing portion (94d, 95d) closes the fourth to tenth inlets (61a-4 to 61a-10), while keeping the first to third inlets (61a-1 to 61a-3) and the eleventh inlet (61a-11) open. Thus, the first to third inlets (61a-1 to 61a-3) and the eleventh inlet (61a-11) are opened in the oil sump (94c, 95c). As described

above, in the second embodiment, the inlets (61a-1 to 61a-11) of the in-shaft communication passage (61) to be closed by the closing portion (94d, 95d) sequentially change as the rotational angle position of the gate rotor (50) changes.

[0143] The in-shaft communication passage (61) whose inlet is closed by the closing portion (94d, 95d) is blocked from the oil sump (94c, 95c). Thus, no lubricant flows into this in-shaft communication passage from the oil sump (94c, 95c). Thus, no lubricant flows into the oil sump (62) and the gate-side oil supply passage (63) which are sequentially connected to the in-shaft communication passage (61) whose inlet is closed. That is, the oil sump (94c, 95c), which is the oil supply source supplying the lubricant to the gate-side oil supply passage (63), is blocked from the gate-side oil supply passage (63). This brings the gate-side oil supply passage (63) into the non-supply state in which no lubricant is supplied to the side surfaces (51a, 51b) and front surface (51c) of the gate (51), which are the sliding surfaces (3) of the gate rotor (50). On the other hand, the lubricant in the oil sump (94c, 95c) flows into the in-shaft communication passage (61) whose inlet is not closed by the closing portion (94d, 95d) and is opened in the oil sump (94c, 95c), and also into the oil sump (62) and the gate-side oil supply passage (63) which are sequentially connected to the in-shaft communication passage (61). That is, the oil sump (94c, 95c), which is the oil supply source supplying the lubricant to the gate-side oil supply passage (63), communicates with the gate-side oil supply passage (63). This brings the gate-side oil supply passage (63) into the supply state in which the lubricant is supplied to the side surfaces (51a, 51b) and front surface (51c) of the gate (51), which are the sliding surfaces (3) of the gate rotor (50).

[0144] As can be seen, in the second embodiment, each of the oil supply mechanisms (60) includes the in-shaft communication passages (61) and the oil sumps (62) which are individually connected to the gate-side oil supply passages (63). Further, the closing portion (94d, 95d) is provided to close some of the inlets (61a-1 to 61a-11) of the in-shaft communication passages (61). The inlets (61a-1 to 61a-11) of the in-shaft communication passage (61) to be closed by the closing portion (94d, 95d) are changed in accordance with the rotation of the gate rotor (50). In this configuration, when the rotational angle position of the gate rotor (50) is in a predetermined angular range A1 to A11, the gate-side oil supply passages (63) are in the supply state in which the gate-side oil supply passages (63) communicate with the oil sump (94c, 95c) and supply the lubricant to the sliding surfaces (3). When the rotational angle position of the gate rotor (50) is out of the predetermined angular range A1 to A11, the gate-side oil supply passages (63) are in the non-supply state in which the gate-side oil supply passages (63) are blocked from the oil sump (94c, 95c) and supply no lubricant to the sliding surfaces (3). Thus, in each of the oil supply mechanisms (60) configured in this man-

ner, the plurality of in-shaft communication passages (61), the plurality of oil sumps (62), and the closing portion (94d, 95d) constitute a switching mechanism (6) for switching the gate-side oil supply passages (63) between the supply state and the non-supply state.

-Advantages of Second Embodiment-

[0145] According to the configuration of the second embodiment described above, the gate-side oil supply passages (63) can be switched between the supply state in which the lubricant is supplied from the gate-side oil supply passages (63) to the sliding surfaces (3), and the non-supply state in which no lubricant is supplied from the gate-side oil supply passages (63) to the sliding surfaces (3). Thus, in a situation where the sliding surfaces (3) of the gate rotor (50) (in this embodiment, the side surfaces (51a, 51b) and front surface (51c) of the gate (51)) provided with the lateral oil supply ports (63b) and the front oil supply port (63c), which are the oil supply ports (4), are not configured to slide constantly, the gate-side oil supply passages (63) can be switched to the non-supply state to stop the supply of the lubricant to the sliding surfaces (3) when the sliding surfaces do not slide and require no lubrication. Therefore, according to the second embodiment, the lubricant can be reliably supplied to the sliding surfaces (3) of the gate rotors (50), while reducing the supply amount of the lubricant.

[0146] Specifically, for example, the switching mechanism (6) is configured to switch the gate-side oil supply passage (63) formed in each gate (51) to the supply state when the front surface (51c) of the gate (51) faces the sealing surface (39a) of the cylindrical wall (30) and when the side surfaces (51b, 51c) of the gate (51) face the inner surface (42) of the helical groove of the screw rotor (40), and to switch the gate-side oil supply passage (63) to the non-supply state when the gate (51) does not face the cylindrical wall (30) or the screw rotor (40). In this configuration, when the gate (51) slides on the cylindrical wall (30) and the screw rotor (40), the sliding surfaces (3) can be lubricated. When the gate (51) does not slide on the cylindrical wall (30) and the screw rotor (40) and forms a gap between the gate (51) and the cylindrical wall (30) and the screw rotor (40), the gap can be sealed. On the other hand, when the gate (51) does not face the cylindrical wall (30) or the screw rotor (40), no lubricant is supplied to the sliding surfaces (3) from the gate-side oil supply passages (63). This can reduce the supply amount of the lubricant.

[0147] In the second embodiment, as described above, when the rotational angle position of the gate rotor (50) is in the predetermined angular range A1 to A11, the switching mechanism (6) switches the gate-side oil supply passages (63) to the supply state in which the gate-side oil supply passages (63) communicate with the oil sump (95c, 94c) to supply the lubricant to the sliding surfaces (3). When the rotational angle positions of the gate rotor (50) is out of the predetermined angular range

A1 to A11, the switching mechanism (6) switches the gate-side oil supply passages (63) to the non-supply state in which the gate-side oil supply passages (63) are blocked from the oil sump (95c, 94c) and supply no lubricant to the sliding surfaces (3). Such a simple configuration of the second embodiment makes it possible to automatically switch the gate-side oil supply passages (63) between the supply state and the non-supply state while the gate rotor (50) makes a single rotation.

<<Third Embodiment>>

[0148] In a third embodiment, the single-screw compressor (1) of the first embodiment is modified such that the oil supply mechanism (60) provided for each of the two gate rotors (50) is provided for the screw rotor (40) which meshes with the two gate rotors (50).

[Oil Supply Mechanism]

[0149] Specifically, as shown in FIG. 15, the single-screw compressor of the third embodiment has the oil supply mechanism (60) which is formed inside the screw rotor (40) and includes a plurality of axial passages (65) and a plurality of screw-side oil supply passages (66) (oil supply passages (5)).

[0150] The plurality of axial passages (65) is formed at a position closer to the rotation axis than the bottom faces (42c) of the helical grooves (41) of the screw rotor (40). In the third embodiment, six axial passages (65) are formed, and are arranged at equal intervals on an outer periphery of the rotation axis of the screw rotor (40). Each axial passage (65) is formed by a hole extending in the direction of the rotation axis inside the screw rotor (40). A discharge end (a right end in FIG. 2) of each axial passage (65) is opened at an end face (right end face in FIG. 2) of the screw rotor (40) on the discharge side. A suction end (a left end in FIG. 2) of each axial passage (65) does not reach an end face (a left end face in FIG. 2) of the screw rotor (40). The discharge end of each axial passage (65) is opened in a space where the high pressure lubricant that has lubricated the bearing (36) of the bearing holder (35) for rotatably supporting the drive shaft (21), for example, is accumulated. This configuration causes the high pressure lubricant to flow into the plurality of axial passages (65), and causes the axial passages (65) to serve as oil sumps in which the high pressure lubricant is accumulated.

[0151] The plurality of screw-side oil supply passages (66) is formed such that at least one screw-side oil supply passage (66) extends from an associated one of the axial passages (65) toward the outer periphery of the screw rotor (40). Each of the screw-side oil supply passages (66) includes a body (66a) and a plurality of lateral branches (66b).

[0152] More specifically, as shown in FIG. 15, the body (66a) of each of the screw-side oil supply passages (66) is formed by a hole extending from an associated one of

the axial passages (65) toward the outer periphery of the screw rotor (40). In the third embodiment, the body (66a) of the screw-side oil supply passage (66) extends to an outer peripheral surface (43) which helically extends between the helical grooves (41) of the screw rotor (40), and is opened at the outer peripheral surface (43). That is, the body (66a) of the screw-side oil supply passage (66) is connected to an outer peripheral oil supply port (66c) which is an oil supply port (4) opened at the outer peripheral surface (43) of the screw rotor (40).

[0153] The lateral branches (66b) are formed by holes extending from the body (66a) toward the lateral faces (42a, 42b) of the helical groove (41), and are connected to groove's lateral oil supply ports (66d) (in-groove oil supply ports), which are the oil supply ports (4) opened at the lateral faces (42a, 42b) of the helical grooves (41). In this embodiment, two lateral branches (66b) are connected to a front portion and rear portion in the rotation direction of the body (66a) of each of the screw-side oil supply passages (66). Thus, in this embodiment, at least two groove's lateral oil supply ports (66d) are opened at the front lateral face (42a) of the inner surface (42) of the helical groove (41) of the screw rotor (40) in the rotation direction, and two groove's lateral oil supply ports (66d) are opened at the rear lateral face (42b). The diameter of each of the groove's lateral oil supply ports (66d) and lateral branches (66b) is determined so that the lubricant flows in such an amount that allows an oil film to be formed on the lateral faces (42a, 42b) of the helical groove (41) of the screw rotor (40), and that the lubricant is kept from scattering in the shape of droplets.

[0154] The number of groove's lateral oil supply ports (66d) and lateral branches (66b) is not limited to two, but may be less than two, or more than two. In a preferred embodiment, the diameter is changed in accordance with the number so that the lubricant flows in such an amount that allows an oil film to be formed on the lateral faces (42a, 42b) of the helical groove (41) of the screw rotor (40), and that the lubricant is kept from scattering in the shape of droplets.

[0155] In this configuration, each of the screw-side oil supply passages (66) is connected to the groove's lateral oil supply ports (66d) opened at the lateral faces (42a, 42b) of the helical groove (41) of the screw rotor (40).

[0156] In a preferred embodiment, the screw-side oil supply passages (66) are positioned such that the groove's lateral oil supply ports (66d) are opened in the compression chamber (23) during the suction phase. Alternatively, the screw-side oil supply passages (66) may be positioned such that the groove's lateral oil supply ports (66d) are opened in the compression chamber (23) during the suction phase, and also in the compression chamber (23) during the compression phase and the discharge phase.

[0157] As described above, in the oil supply mechanism (60) formed in the screw rotor (40), the axial passages (65) and the screw-side oil supply passages (66) form a plurality of lubricant passages, each of which is

branched to have two or more outlets. Each of the lubricant passages has an inlet which is opened in a space where the high pressure lubricant that has lubricated the bearing (36), for example, is accumulated, and an outlet which is opened at the outer peripheral surface (43) of the screw rotor (40) and the lateral faces (42a, 42b) of the groove. Therefore, due to the pressure difference between the inlet and outlets of the lubricant passage, the high pressure lubricant near the inlet enters the lubricant passage, flows toward the outlet, and then flows to the outer peripheral surface (43) of the screw rotor (40) and the lateral faces (42a, 42b) of the helical groove (41).

-Operation-

[0158] How the fluid is compressed in the compression mechanism (20) is the same as in the first embodiment, and the description thereof is not repeated. The oil supply operation different from that of the first embodiment will be described below.

-Oil Supply Operation-

[0159] When the screw rotor (40) and the two gate rotors (50) rotate to compress the refrigerant gas in the compression chamber (23), the oil supply mechanism (60) formed in the screw rotor (40) supplies the lubricant to the sliding surfaces (3) of the two gate rotors (50) and the screw rotor (40).

[0160] In the oil supply mechanism (60), as described above, the pressure difference between the inlets and outlets of the lubricant passage formed by the axial passage (65) and the screw-side oil supply passage (66) causes the high pressure lubricant that has lubricated the bearing (36) and has been accumulated in a predetermined space to enter the lubricant passage, and flow toward the outlets. Specifically, the high pressure lubricant flows into the axial passages (65) serving as the oil sumps, flows into the plurality of screw-side oil supply passages (66) extending from the axial passages (65) toward the outer periphery by the effect of the driving force derived from the pressure difference described above and the centrifugal force generated by the rotation of the screw rotor (40), and then flows outward in the screw-side oil supply passages (66) (see FIG. 15). The lubricant flowing through the screw-side oil supply passages (66) flows from the outer peripheral oil supply ports (66c) to the outer peripheral surface (43) of the screw rotor (40), and also flows from the groove's lateral oil supply ports (66d) to the lateral faces (42a, 42b) of the helical grooves (41) of the screw rotor (40).

[0161] The outer peripheral surface (43) of the screw rotor (40) provided with the helical grooves (41) slides on the inner peripheral surface (30a) of the cylindrical wall (30) covering the outer periphery of the screw rotor (40). Thus, lubrication is required to keep the outer peripheral surface (43) of the screw rotor (40) and the inner peripheral surface (30a) of the cylindrical wall (30) from

seizing. On the other hand, when a gap is formed between the outer peripheral surface (43) of the screw rotor (40) and the inner peripheral surface (30a) of the cylindrical wall (30), the gap needs to be sealed so that the high pressure fluid does not leak to the low pressure side.

[0162] In the third embodiment, the screw-side oil supply passages (66) are formed in the screw rotor (40), and are connected to the outer peripheral oil supply ports (66c) opened at the outer peripheral surface (43) of the screw rotor (40) which slides on the cylindrical wall (30). In the screw rotor (40) configured in this manner, the lubricant in the screw-side oil supply passages (66) flows from the outer peripheral oil supply ports (66c) to the outer peripheral surface (43) of the screw rotor (40) which slides on the inner peripheral surface (30a) of the cylindrical wall (30), thereby lubricating the outer peripheral surface (43), or sealing the gap, if any, between the outer peripheral surface (43) and the inner peripheral surface (30a) of the cylindrical wall (30).

[0163] In the third embodiment, unlike the conventional configuration, the outer peripheral oil supply ports (66c), which are the oil supply ports (4), are opened at the outer peripheral surface (43) of the screw rotor (40) that rotates. Therefore, the lubricant that has flowed from the outer peripheral oil supply ports (66c) is rapidly spread over the rotating screw rotor (40), and is quickly supplied to the sliding surfaces (3) other than the outer peripheral surface (43) at which the outer peripheral oil supply ports (66c) are formed. Further, since the screw rotor (40) and the gate rotors (50) mesh with each other and rotate together, the lubricant supplied to the screw rotor (40) is rapidly spread to the gate rotors (50), and is quickly supplied to the sliding surfaces (3) of the gate rotors (50).

[0164] In the third embodiment, the screw-side oil supply passages (66) are formed in the screw rotor (40), and the oil supply passages (5) are connected to the groove's lateral oil supply ports (66d), which are the in-groove oil supply ports opened at the inner surface (42) of the helical groove (41) of the screw rotor (66). In the screw rotor (40) configured in this manner, the lubricant in the screw-side oil supply passages (66) flows from the groove's lateral oil supply ports (66d) to the lateral faces (42a, 42b) of the helical grooves (41) which slide on the gate rotor (50), thereby lubricating the lateral faces (42a, 42b), or sealing the gap, if any, between the lateral faces (42a, 42b) and the gate rotor (50) sliding on the lateral faces. That is, in the third embodiment, unlike in the conventional configuration, the lubricant is directly supplied to the lateral faces (42a, 42b), which are the sliding surfaces (3), from the groove's lateral oil supply ports (66d) opened at the lateral faces (42a, 42b) of the helical grooves of the screw rotor (40).

[0165] Further, in the third embodiment, unlike in the conventional configuration, the groove's lateral oil supply ports (66d), which are the oil supply ports (4), are opened at the lateral faces (42a, 42b) of the helical grooves of the screw rotor (40) that rotates. Therefore, the lubricant which has flowed from the groove's lateral oil supply ports

(66d) is rapidly spread over the rotating screw rotor (40) by the effect of the centrifugal force, and is quickly supplied to the sliding surfaces (3) other than the lateral faces (42a, 42b) of the helical grooves. Further, the lubricant supplied to the lateral faces (42a, 42b) of the helical groove of the screw rotor (40) also adheres to the gate rotors (50) which mesh with and rotate with the screw rotor (40), and is rapidly spread over the gate rotors (50) by the effect of the centrifugal force. Thus, the lubricant is quickly supplied to the sliding surfaces (3) of the gate rotors (50).

-Advantages of Third Embodiment-

[0166] According to the configuration of the third embodiment described above, the screw-side oil supply passages (66) serving as the oil supply passages (5) are formed in the screw rotor (40), which is at least one of the screw rotor (40) and the gate rotors (50) mesh with each other and rotate together, and the screw-side oil supply passages (66) are connected to the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d), which are the oil supply ports (4) opened at the outer peripheral surface (43) and the groove's lateral faces (42a, 42b). As a result, the lubricant is directly supplied from the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d) to the outer peripheral surface (43) and the lateral faces (42a, 42b) of the helical grooves, which are the sliding surfaces (3). Therefore, as compared to the conventional configuration in which the lubricant is injected from the oil supply port formed in the cylindrical wall to be indirectly supplied to the inner surfaces (42) of the helical grooves of the screw rotor (40), the lubricant can be reliably supplied in a smaller amount to the outer peripheral surface (43) and the lateral faces (42a, 42b), which are the sliding surfaces (3) of the screw rotor (40).

[0167] According to the third embodiment, unlike in the conventional configuration in which the lubricant is injected from the oil supply port formed in the cylindrical wall (30) which does not rotate, the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d), which are the oil supply ports (4), are opened at the outer peripheral surface (43) and the lateral faces (42a, 42b) of the helical grooves, which are the sliding surfaces (3) of the screw rotor (40) that rotates, so that the lubricant flows to the sliding surfaces (3) from these oil supply ports. Therefore, the lubricant that has flowed from the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d) is rapidly spread over the rotating screw rotor (40), and can be quickly supplied to the sliding surfaces (3) other than the outer peripheral surface (43) and the lateral faces (42a, 42b) of the helical grooves at both of which the oil supply ports (4) are formed. Since the screw rotor (40) and the gate rotors (50) mesh with each other and rotate together, the lubricant supplied to the screw rotor (40) is rapidly spread to the gate rotors (50), and can be quickly supplied to the

sliding surfaces (3) of the gate rotors (50).

[0168] As described above, in the third embodiment, the efficiency of the compressor is not lowered because it is unnecessary to increase the power for the transport of the lubricant and the power for the rotation of the screw rotor (40), unlike the conventional configuration in which a large amount of lubricant is supplied. Supplying the lubricant in a small amount to at least one of the sliding surface (3) of the screw rotor (40) or the sliding surface (3) of the gate rotor (50) makes it possible to lubricate the sliding surface (3) of each of the screw rotor (40) and the gate rotor (50), or to seal a gap, if any, between the sliding surface (3) and its counterpart sliding surface. That is, according to the third embodiment, the sliding surfaces (3) of the screw rotor (40) and the gate rotor (50) can be kept from seizing, and the high pressure fluid can be blocked from leaking from the compression chamber, even if the supply amount of the lubricant is reduced. Therefore, in the third embodiment, the supply amount of the lubricant can be reduced without lowering the reliability of the screw compressor (1), which can improve the compressor efficiency.

[0169] According to the third embodiment, the axial passages (65) serving as the oil sumps are formed at a position closer to the rotation axis of the screw rotor (40) than the bottom faces (42c) of the helical grooves (41), and base ends of the screw-side oil supply passages (66) are respectively connected to the axial passages (65). That is, the screw-side oil supply passages (66) extend from the axial passages (65) in the screw rotor (40) toward the outer periphery. In this configuration, the screw rotor (40) rotates to generate the centrifugal force, which causes the lubricant to enter the screw-side oil supply passages (66) from the axial passages (65), flows toward the outer periphery of the screw rotor (40), and flows from the oil supply ports (4) (the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d)) to the sliding surfaces (3) of the screw rotor (40) (the outer peripheral surface (43) and lateral faces (42a, 42b) of the helical grooves). That is, this simple configuration can supply the lubricant to the sliding surfaces (3) of the screw rotor (40) (the outer peripheral surface (43) and the lateral faces (42a, 42b) of the helical grooves) by utilizing the centrifugal force generated by the rotation of the screw rotor (40).

<<Other Embodiments>>

[0170] In the first to third embodiments, the single-screw compressor provided in the refrigerant circuit to compress the refrigerant has been described. However, a target to be compressed (fluid) is not limited to the refrigerant, and the compressor is not limited to the single-screw compressor. The compressor may be a twin screw compressor including a male rotor and a female rotor, or a compressor including female rotors provided on both sides of a male rotor.

[0171] The front oil supply ports (63c) that have been

formed in the first and second embodiments may not be formed. Alternatively, the lateral oil supply ports (63b) may be omitted, and the gate-side oil supply passages (63) may be connected only to the front oil supply ports (63c).

[0172] In the first and second embodiments, it has been described that the lateral oil supply ports (63b) of each of the gate-side oil supply passages (63) are opened at the side surfaces (51a, 51b) of the gate (51) on the front and rear sides in the direction of rotation of the gate (51). However, the lateral oil supply ports (63b) may be opened at least at the rear side surface (51b) of the gate (51), and no oil supply port may be opened at the front side surface (51a) of the gate (51). The rear side surface (51b) in the rotation direction of the gate (51) is the sliding surface (3) which reliably slides on the screw rotor (40) and is pressed by the screw rotor (40), and therefore, is probably worn through the sliding movement. However, the lateral oil supply ports (63b) opened at the rear side surface (51b) cause the lubricant to be reliably supplied between the rear side surface (51b) and the lateral face (42a, 42b) of the helical groove (41). This can protect the gate (51) and the screw rotor (40) from the sliding wear.

[0173] Similarly, in the third embodiment, the groove's lateral oil supply ports (66d) of the screw-side oil supply passages (66) are opened at both lateral faces (42a, 42b) of each of the helical grooves (41) of the screw rotor (40) on the front and rear sides in the rotation direction of the screw rotor. However, the groove's lateral oil supply ports (66d) may be opened at least at the lateral face (42b) on the rear side of the helical groove (41) in the rotation direction, and no oil supply port may be opened at the lateral face (42a) on the front side of the helical groove (41) in the rotation direction. The rear lateral face (42b) of the helical groove (41) in the rotation direction is the sliding surface (3) which reliably slides on the gate (51) of the gate rotor (50) and presses the gate (51) of the gate rotor (50), and therefore, is probably worn through the sliding movement. However, the groove's lateral oil supply ports (66d) opened at the rear lateral face (42b) of the helical groove (41) cause the lubricant to be reliably supplied between the rear lateral face (42b) of the helical groove (41) and the gate (51) of the gate rotor (50). This can protect the gate (51) of the gate rotor (50) and the screw rotor (40) from the sliding wear.

[0174] Further, in the first and second embodiments, four lateral oil supply ports (63b) are opened at each of the side surfaces (51a, 51b) of the gate (51) at substantially equal intervals from the base end to distal end of the gate (51). However, it is not always necessary to provide a plurality of lateral oil supply ports (63b) at equal intervals, and at least one lateral oil supply port (63b) may be formed at a position closer to the base end of the gate (51) than the center thereof in the radial direction. The at least one lateral oil supply port (63b) opened at a position closer to the base end of the gate (51) than the center thereof in the radial direction makes it possible to

supply the lubricant to the base end of the side surface (51a, 51b) of the gate (51), and to easily spread the lubricant toward the distal end of the side surface (51a, 51b) of the gate (51) by utilizing the centrifugal force. This configuration can minimize the number of the lateral oil supply ports (63b), and can further reduce the supply amount of the lubricant.

[0175] Similarly, in the third embodiment, two groove's lateral oil supply ports (66d) are opened at each of the lateral faces (42a, 42b) of the helical grooves (41) of the screw rotor (40). However, the two groove's lateral oil supply ports (66d) are not always necessary, and at least one groove's lateral oil supply port may be formed at each lateral face (42a, 42b) of the helical groove (41) at a position closer to the bottom face (42c) of the helical groove (41) than to the outer peripheral surface (43) of the screw rotor (40). The at least one groove's lateral oil supply port (66d) opened at the lateral face (42a, 42b) of the helical groove (41) of the screw rotor (40) at a position closer to the bottom face (42c) of the helical groove (41) than to the outer peripheral surface (43) makes it possible to supply the lubricant to a portion of the lateral face (42a) of the helical groove (41) closer to the rotation axis, and to easily spread the lubricant to a portion of the lateral face (42a, 42b) of the helical groove (41) closer to the outer peripheral surface (43) by utilizing the centrifugal force. This configuration can minimize the number of the groove's lateral oil supply ports (66d), and can further reduce the supply amount of the lubricant.

[0176] Further, in the first and second embodiments, the oil supply mechanism (60) having the gate-side oil supply passage (63) has been provided in each of the two gate rotors (50). However, the oil supply mechanism (60) may be provided in only one of the gate rotors (50). When the oil supply mechanism (60) provided in one of the gate rotors (50) supplies the lubricant to the sliding surfaces (3) of the gate rotor (50) and the screw rotor (40), the lubricant adheres to the lateral faces (42a, 42b) of the helical grooves (41) of the screw rotor (40). Thus, when the amount of the lubricant that adheres to the lateral faces (42a, 42b) of the helical grooves (41) of the screw rotor (40) is controlled, the lubricant can be left in the helical grooves (41) to lubricate the sliding surfaces (3) of the other gate rotor (50) and the screw rotor (40), and to seal the gap between the sliding surfaces (3).

[0177] In the first and second embodiments, the gate-side oil supply passages (63) of the oil supply mechanism (60) are formed in all the gates (51) of the gate rotor (50). However, the gate-side oil supply passage (63) may be formed in at least one of the gates (51), and more preferably, may be formed in the same number as the number of helical grooves (41) in the screw rotor (40) (six in the above-described embodiments) in the gates (51) adjacent to each other. When the amount of lubricant supplied from the gate-side oil supply passages (63) to the sliding surfaces (3) of the gate rotor (50) and the screw rotor (40) is controlled by adjusting the number and diameter of the lateral oil supply ports (63b), the sliding surfaces

(3) of the gate rotor (50) and the screw rotor (40) can be protected from the seizing even if the gate-side oil supply passage (63) is not formed in every gate (51).

[0178] In the first and second embodiments, the right oil supply mechanism (60) in FIG. 3 has the in-shaft communication passage (61) formed inside the front shaft portion (58a), and the left oil supply mechanism (60) has the in-shaft communication passage (61) formed inside the rear shaft portion (58b). However, the position of the in-shaft communication passage (61) is not limited thereto. The right oil supply mechanism (3) in FIG. 3 may have the in-shaft communication passage (61) formed inside the rear shaft portion (58b), and the left oil supply mechanism (60) may have the in-shaft communication passage (61) formed inside the front shaft portion (58a). Alternatively, both of the oil supply mechanisms (60) may have the in-shaft communication passage (61) formed inside the front shaft portion (58a) or the rear shaft portion (58b).

[0179] In the third embodiment, the screw-side oil supply passages (66) are connected to the outer peripheral oil supply ports (66c) opened at the outer peripheral surface (43) of the screw rotor (40) and the groove's lateral oil supply ports (66d) opened at the lateral faces (42a, 42b) of the helical grooves (41). However, the screw-side oil supply passages (66) are not limited to those connected to the outer peripheral oil supply ports (66c) and the groove's lateral oil supply ports (66d). For example, the screw-side oil supply passages (66) may be connected to bottom oil supply ports which are opened at the bottom faces (42c) of the helical grooves (41) of the screw rotor (40). Alternatively, the screw-side oil supply passages (66) may be connected only to the outer peripheral oil supply ports (66c) or the groove's lateral oil supply ports (66d).

[0180] The switching mechanism (6) of the second embodiment is not limited to have the above-described configuration, and may be configured in any way as long as the gate-side oil supply passages (63) can be switched between the supply state and the non-supply state. Further, the switching mechanism (6) of the second embodiment can be applied to the oil supply mechanism (60) formed in the screw rotor (40) as described in the third embodiment. In this case, a closing portion as described in the second embodiment may be provided in a space in which the discharge ends of the plurality of axial passages (65) are opened and the high pressure lubricant is accumulated.

[0181] The embodiments described above are merely exemplary ones in nature, and do not intend to limit the scope of the present invention, applications, or use thereof.

INDUSTRIAL APPLICABILITY

[0182] As can be seen from the foregoing, the present invention is useful for a screw compressor.

DESCRIPTION OF REFERENCE CHARACTERS

[0183]

5	1 Single-Screw Compressor (Screw Compressor)
	3 Sliding Surface
	4 Oil Supply Port
	5 Oil Supply Passage
	6 Switching Mechanism
10	23 Compression Chamber
	30 Cylindrical Wall (Rotor Casing)
	39 Opening
	40 Screw Rotor (First Rotor)
	41 Helical Groove
15	42 Inner surface of Helical Groove (Sliding Surface)
	42a Lateral Face of Helical Groove (Sliding Surface)
	42b Lateral Face of Helical Groove (Sliding Surface)
	43 Outer Peripheral Surface (Sliding Surface)
	50 Gate Rotor (Second Rotor)
20	51 Gate
	51a Front Side Surface (Side Surface, Sliding Surface)
	51b Rear Side Surface (Side Surface, Sliding Surface)
25	51c Front Surface (Sliding Surface)
	52 Coupling Portion
	55 Support Member
	63 Gate-side Oil Supply Passage (Oil Supply Passage)
30	63b Lateral Oil Supply Port (Oil Supply Port)
	63c Front Oil Supply Port (Oil Supply Port)
	65 Axial Passage (Oil Sump)
	66 Screw-side Oil Supply Passage (Oil Supply Passage)
35	66c Outer Peripheral Oil Supply Port (Oil Supply Port)
	66d Groove's Lateral Oil Supply Port (Oil Supply Port, In-Groove Oil Supply Port)

Claims

1. A screw compressor, comprising:

- | | |
|----|---|
| 45 | a first rotor (40) provided with a helical groove (41); |
| | a second rotor (50) which meshes with the first rotor (40) and rotates together with the first rotor (40); |
| 50 | a rotor casing (30) which covers at least an outer periphery of the first rotor (40), and defines a compression chamber (23) in the helical groove (41) together with the first rotor (40) and the second rotor (50), wherein |
| 55 | a fluid is compressed in the compression chamber (23), and |
| | at least one of the first rotor (40) or the second rotor (50) is provided with an oil supply passage |

- (5) which is connected to an oil supply port (4) opened at a sliding surface (3) of the rotor (40, 50) to supply a lubricant to the sliding surface (3).
2. The screw compressor of claim 1, further comprising:
 - a switching mechanism (6) which switches the oil supply passage (5) between a supply state in which the lubricant is supplied to the sliding surface (3) and a non-supply state in which no lubricant is supplied to the sliding surface (3).
 3. The screw compressor of claim 2, wherein the switching mechanism (6) is configured to switch the oil supply passage (5) to the supply state by causing an oil supply source (94c, 95c) for supplying the lubricant to the oil supply passage (5) to communicate with the oil supply passage (5) when a rotational angle position of the rotor (40, 50) provided with the oil supply passage (5) is in a predetermined angular range, and to switch the oil supply passage (5) to the non-supply state by blocking the oil supply source (94c, 95c) from the oil supply passage (5) when the rotational angle position of the rotor (40, 50) is out of the predetermined angular range.
 4. The screw compressor of any one of claims 1 to 3, wherein
 - the first rotor (40) is a screw rotor (40) rotatably housed in a cylindrical wall (30) constituting the rotor casing (30),
 - the second rotor (50) is a gear-shaped gate rotor (50) having a plurality of flat gates (51) and arranged outside the cylindrical wall (30), some of the gates (51) entering a space inside the cylindrical wall (30) via an opening (39) formed in the cylindrical wall (30) and meshing with the screw rotor (40) so that the gate rotor (50) rotates together with the screw rotor (40),
 - the oil supply passage (5) is formed in at least one of the gates (51) of the gate rotor (50), and
 - the oil supply port (4) is a lateral oil supply port (63b) opened at a side surface (51a, 51b) of the at least one gate (51), the side surface (51a, 51b) serving as the sliding surface (3) which slides on the screw rotor (40).
 5. The screw compressor of claim 4, wherein the lateral oil supply port (63b) is opened at least at one of side surfaces (51b), including the side surface (51a, 51b), on a rear side in a direction of rotation of the at least one gate (51).
 6. The screw compressor of claim 4 or 5, wherein the oil supply passage (5) is connected to a front oil supply port (63c) opened at a front surface (51c) of the at least one gate (51) facing the compression chamber (23).
 7. The screw compressor of any one of claims 4 to 6, wherein the lateral oil supply port (63b) includes at least one lateral oil supply port (63b) formed at a position closer to a base end of the at least one gate (51) than a center, of the at least one gate (51), in a radial direction of the gate rotor (50).
 8. The screw compressor of any one of claims 4 to 7, further comprising:
 - a support member (55) supporting the gate rotor (50) from a rear side opposite to the compression chamber (23), wherein
 - an oil sump (62) to which the lubricant is supplied is formed between the support member (55) and a coupling portion (52) of the gate rotor (50) coupling base ends of the plurality of gates (51), and the oil supply passage (5) extends in a radial direction of the gate rotor (50) of the at least one gate (51), and has a base end connected to the oil sump (62).
 9. The screw compressor of any one of claims 1 to 3, wherein
 - the oil supply passage (5) is formed in the first rotor (40), and
 - the oil supply port (4) is an in-groove oil supply port (66d) opened at an inner surface (42) of the helical groove (41) serving as the sliding surface (3) of the first rotor (40) sliding on the second rotor (50).
 10. The screw compressor of any one of claims 1 to 3, wherein
 - the oil supply passage (5) is formed in the first rotor (40), and
 - the oil supply port (4) is an outer peripheral oil supply port (66c) opened at an outer peripheral surface (43) of the first rotor (40) serving as the sliding surface (3) of the first rotor (40) sliding on the rotor casing (30).
 11. The screw compressor of claim 9 or 10, wherein
 - the first rotor (40) has an oil sump (44) to which the lubricant is supplied, the oil sump (44) being formed at a position closer to a rotation axis of the first rotor (40) than a bottom face (42c) of the helical groove (41), and
 - the oil supply passage (5) extends from the oil sump (44) toward an outer periphery of the first rotor (40).

FIG.1

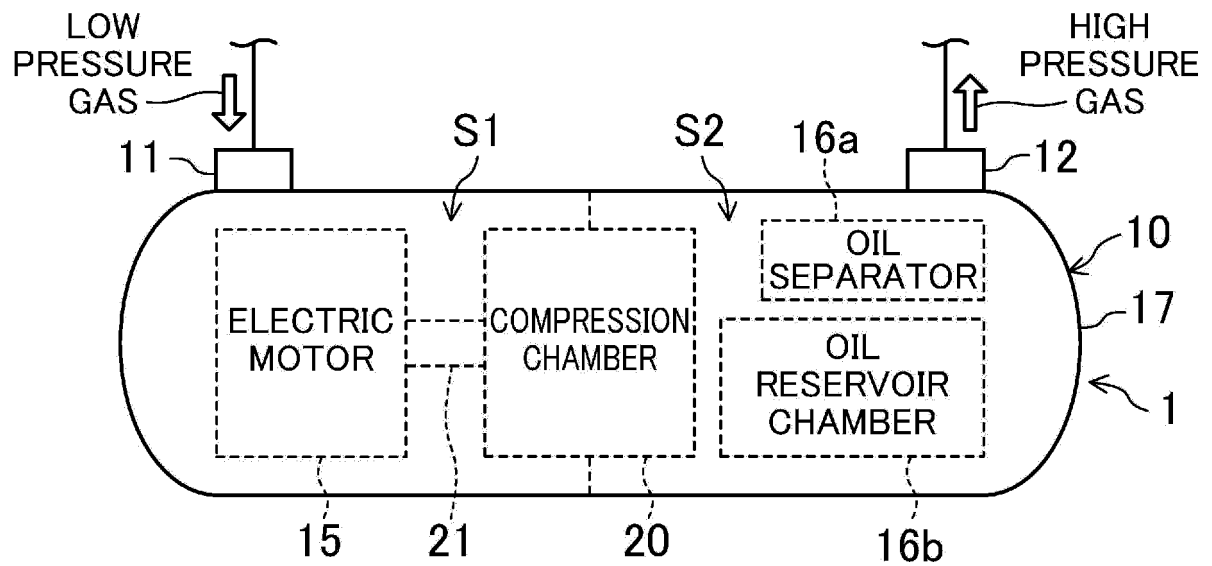


FIG.2

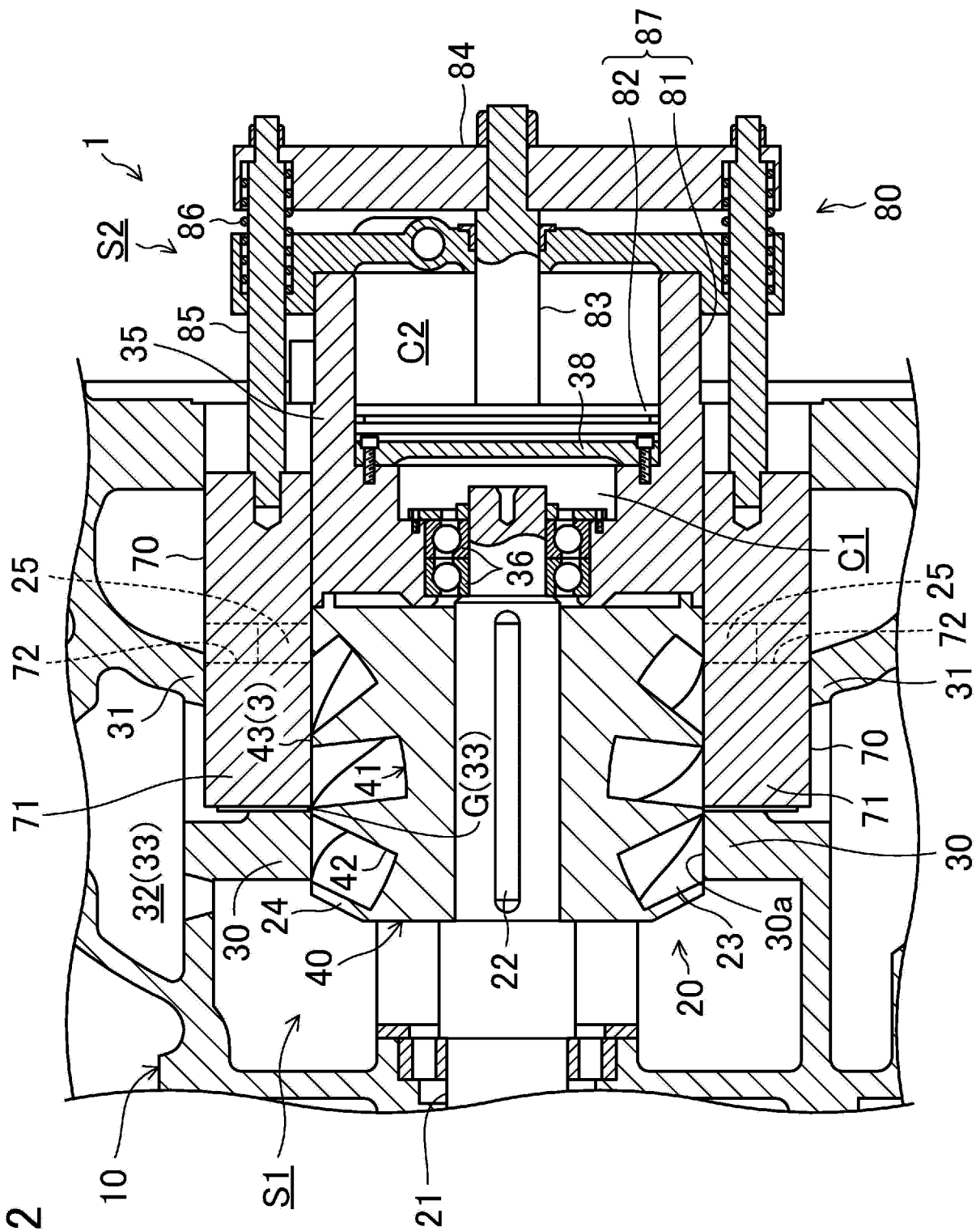


FIG.3

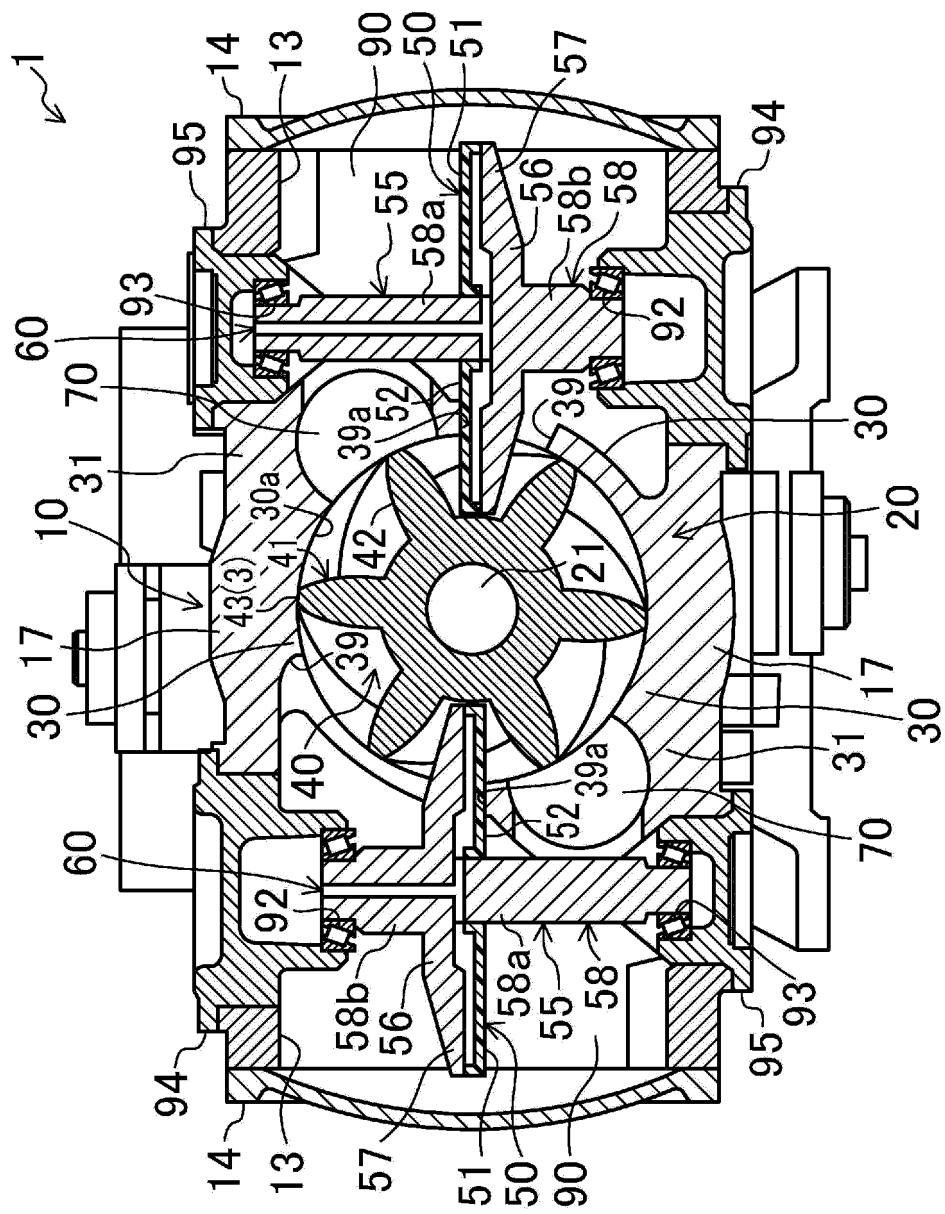


FIG.4

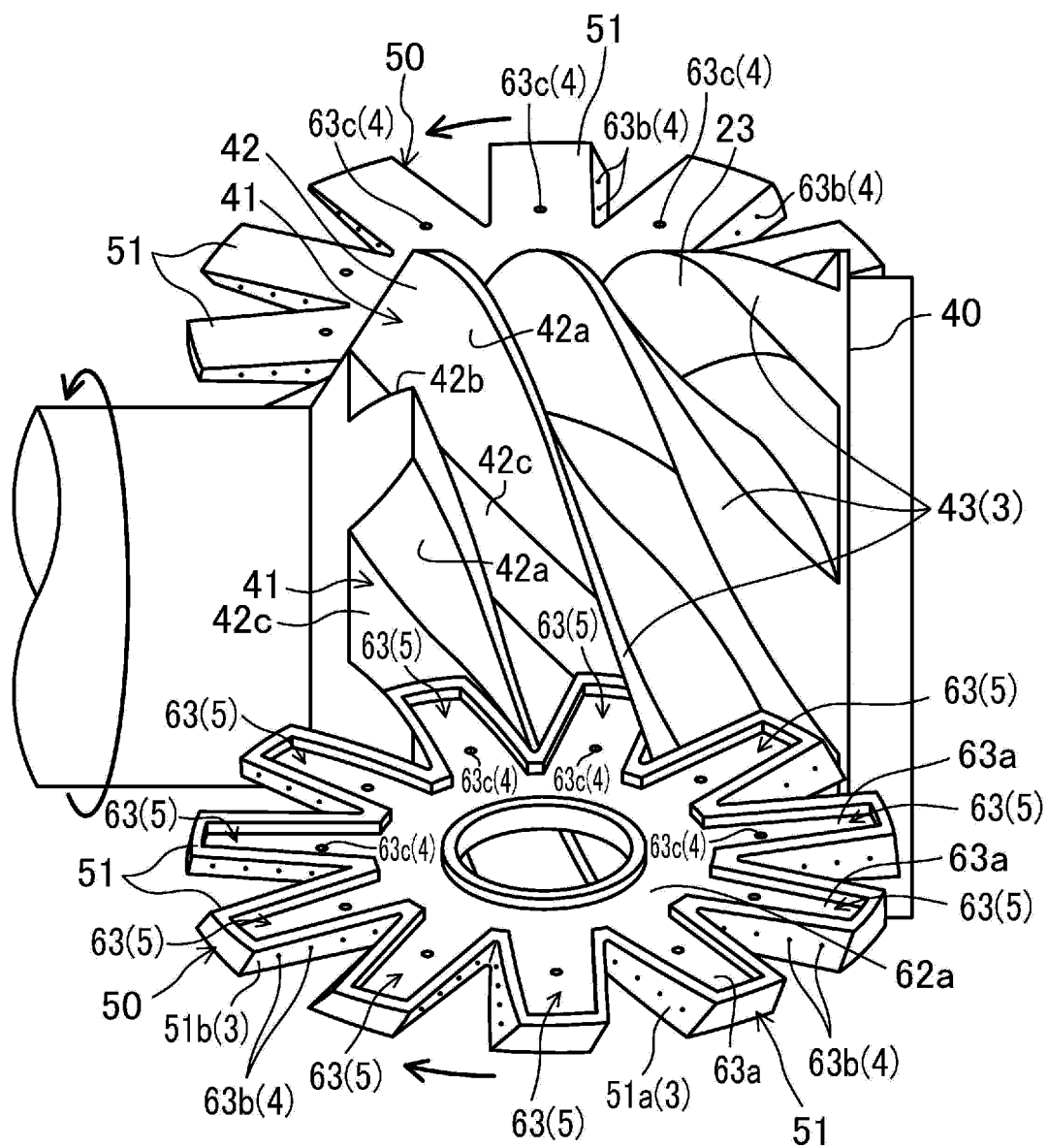


FIG.5

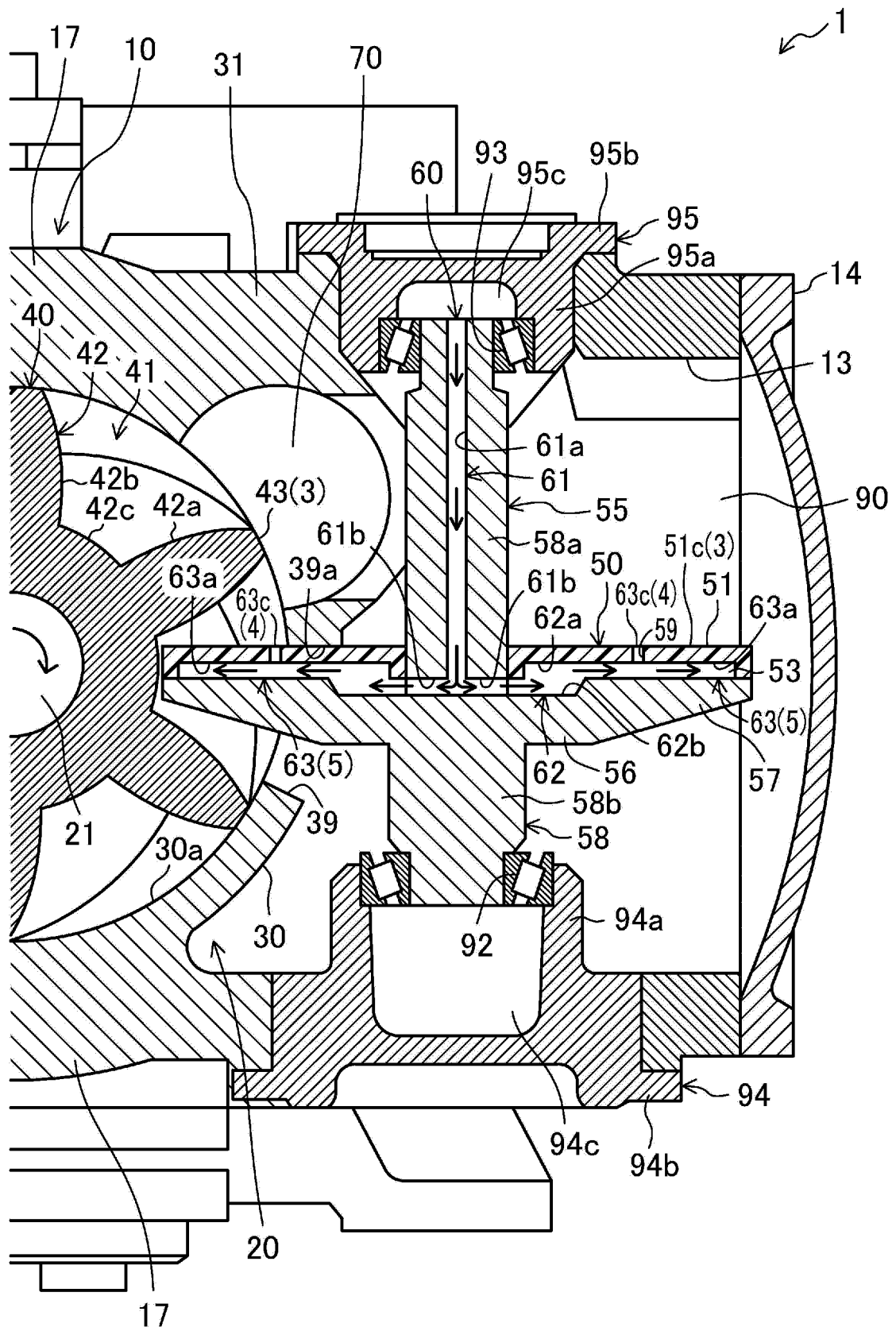


FIG.6

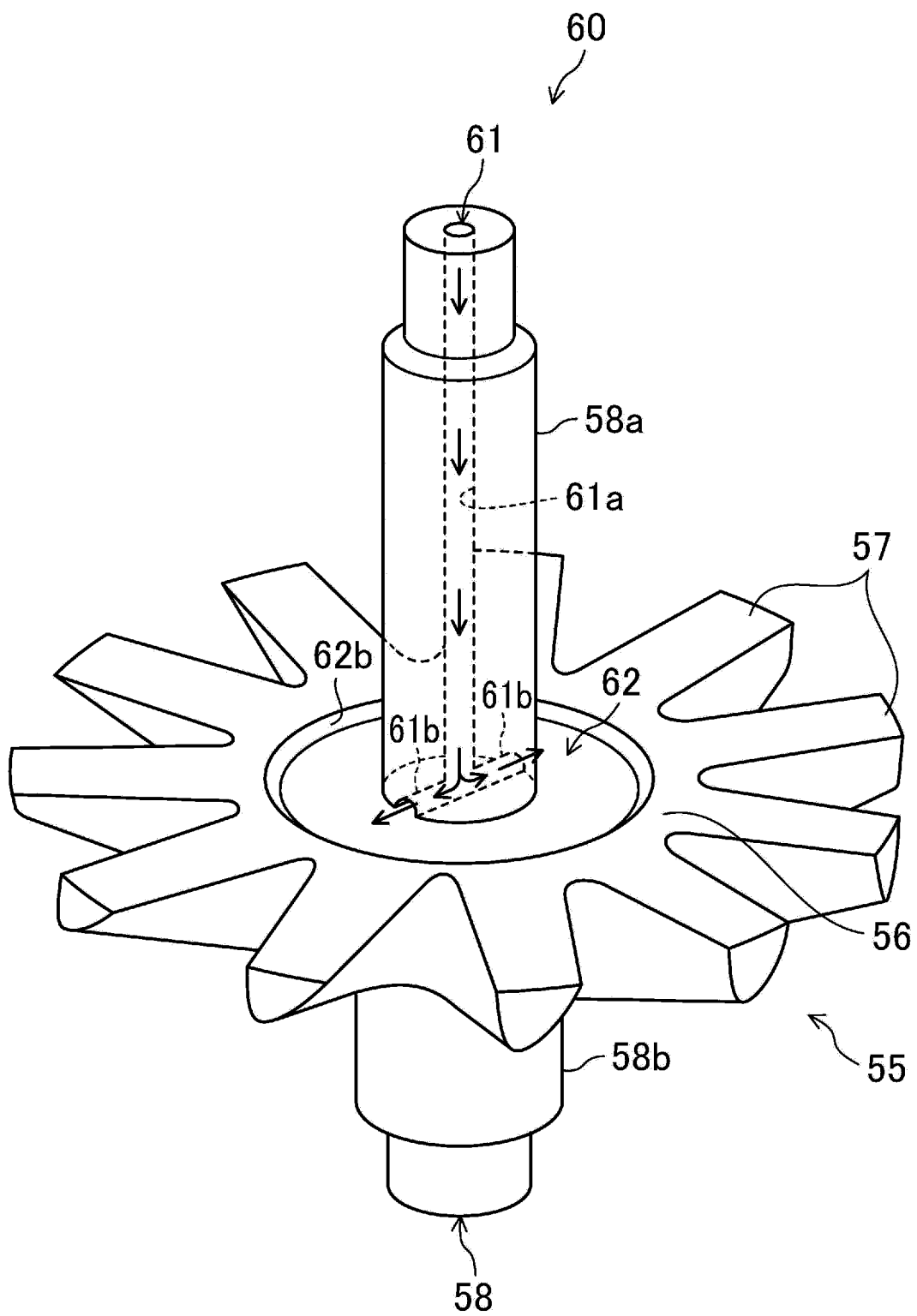


FIG. 7

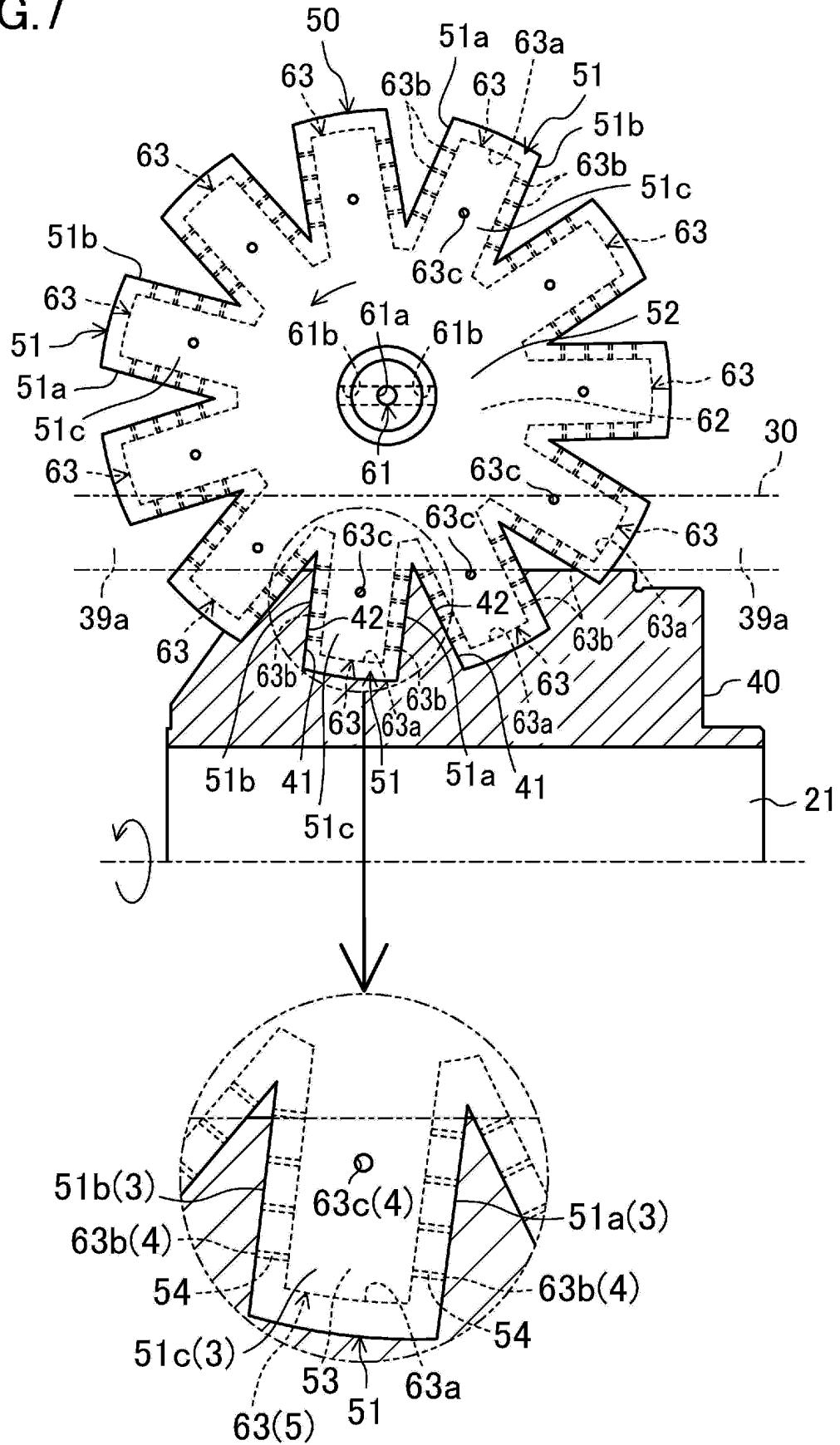


FIG.8

DIRECTION OF ROTATION
OF SCREW ROTOR

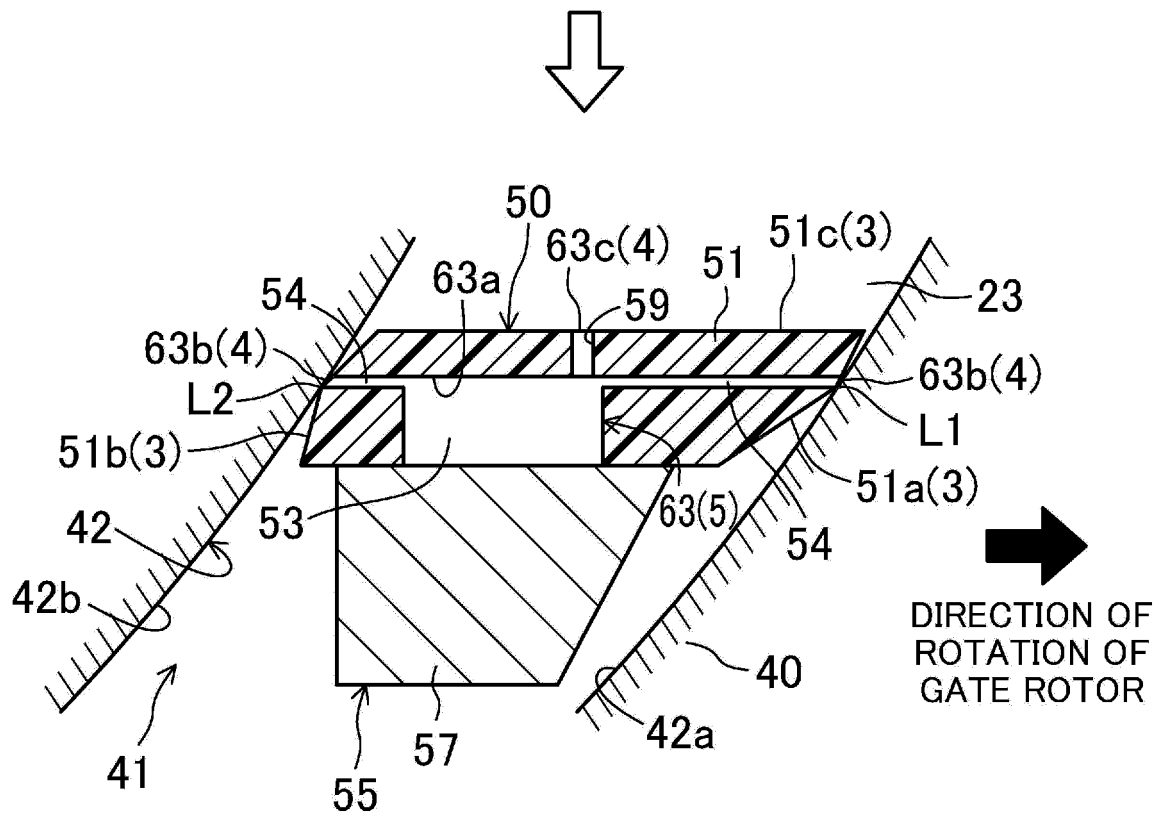


FIG.9

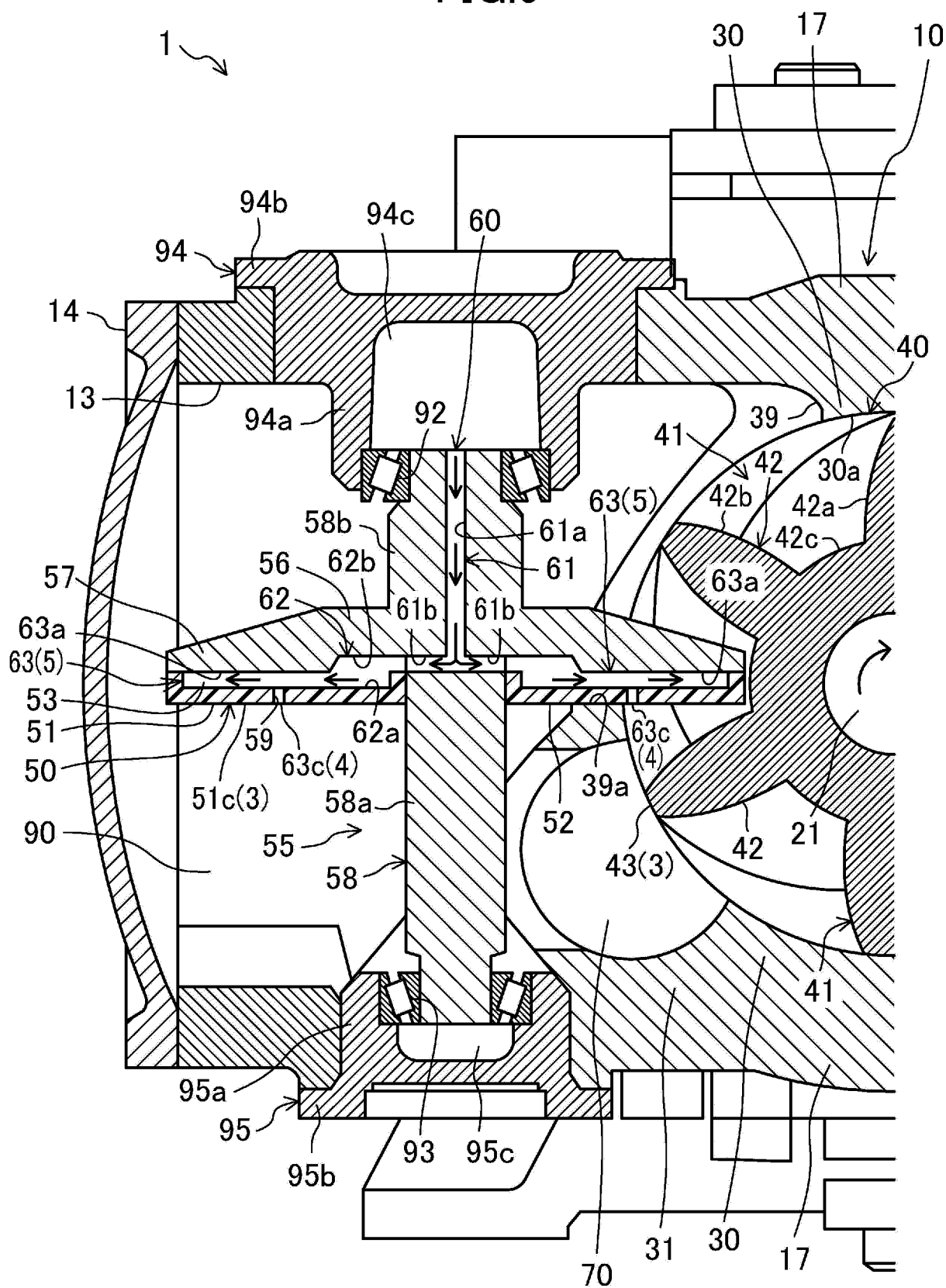


FIG.10

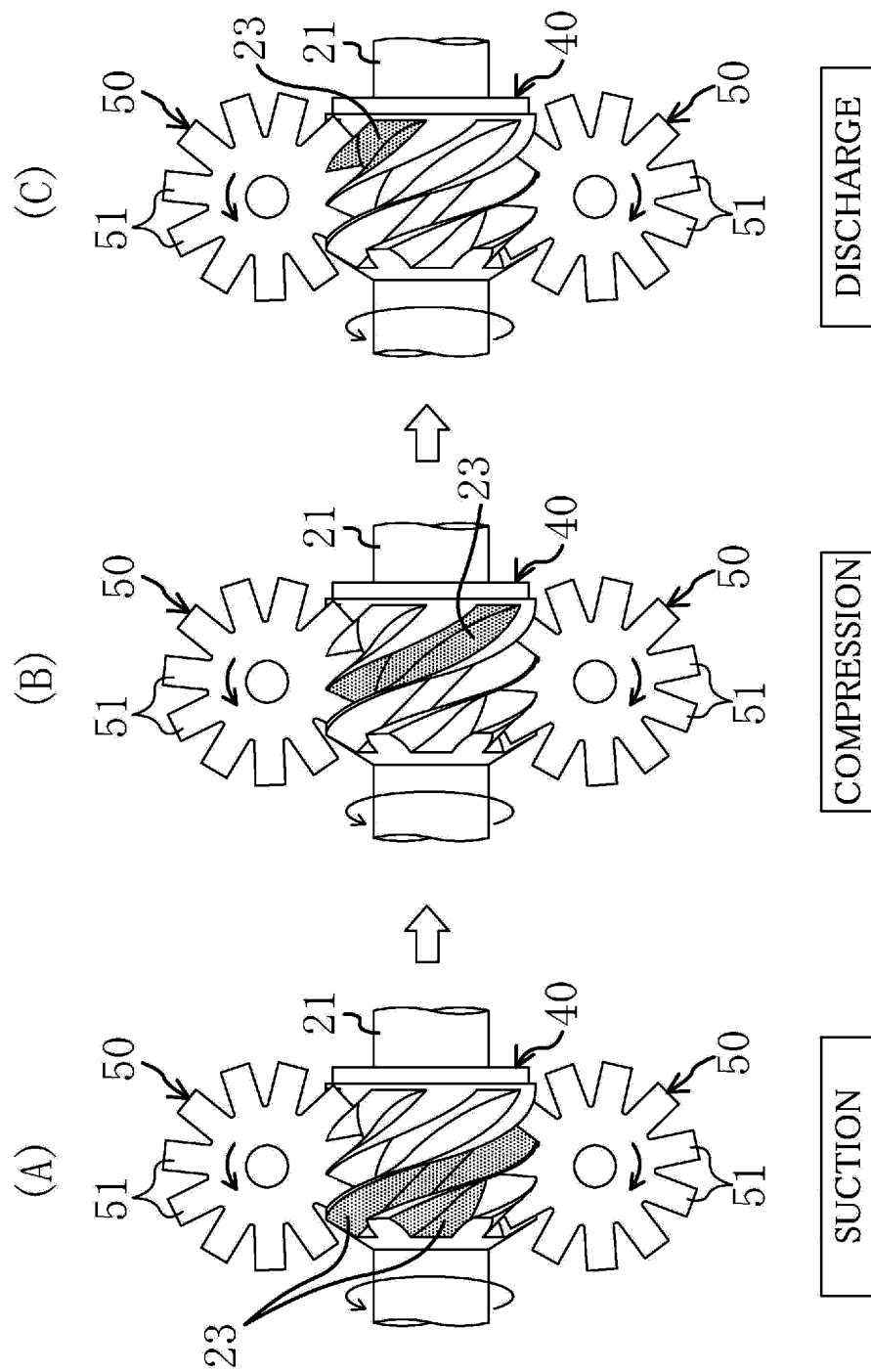


FIG.11

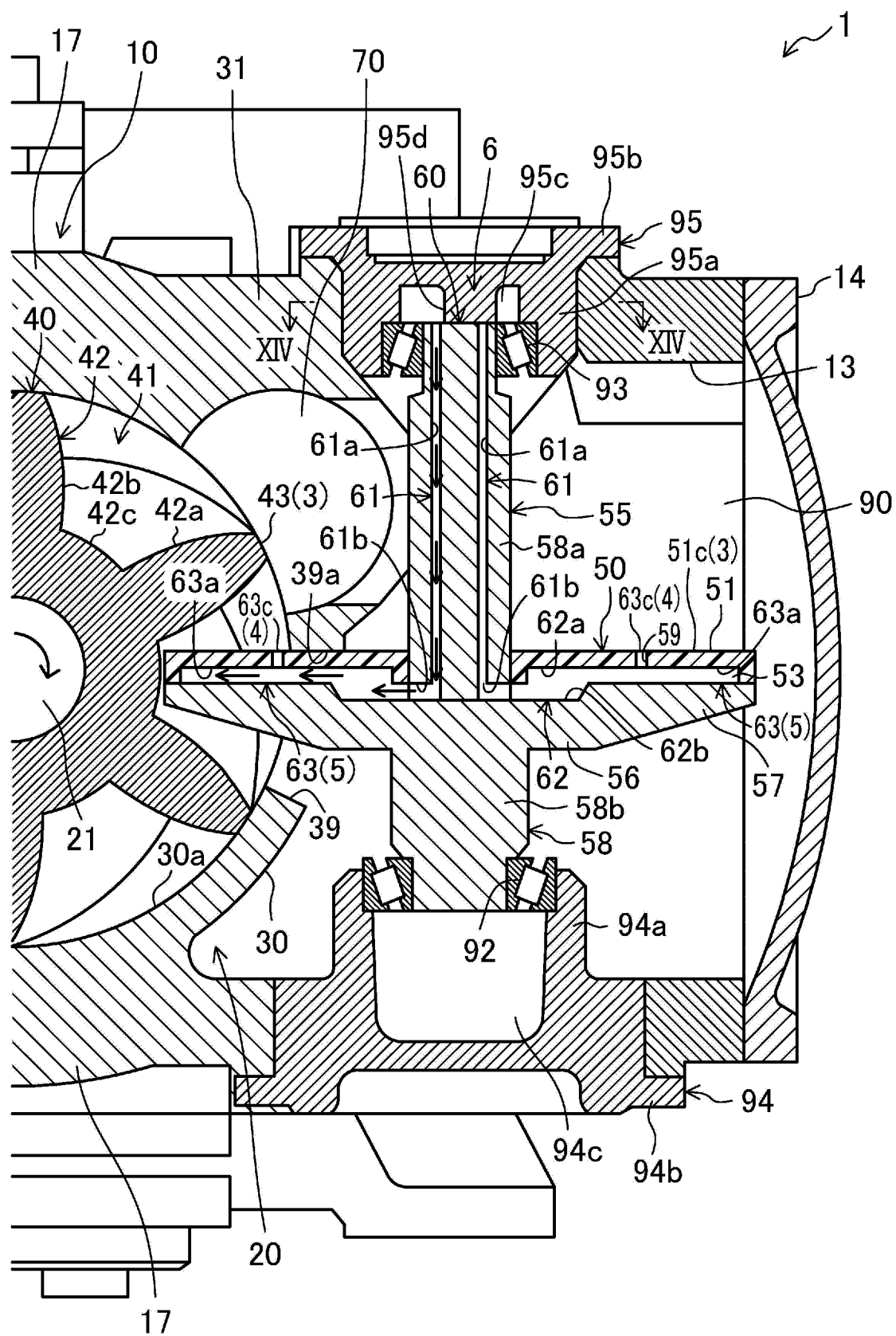


FIG.12

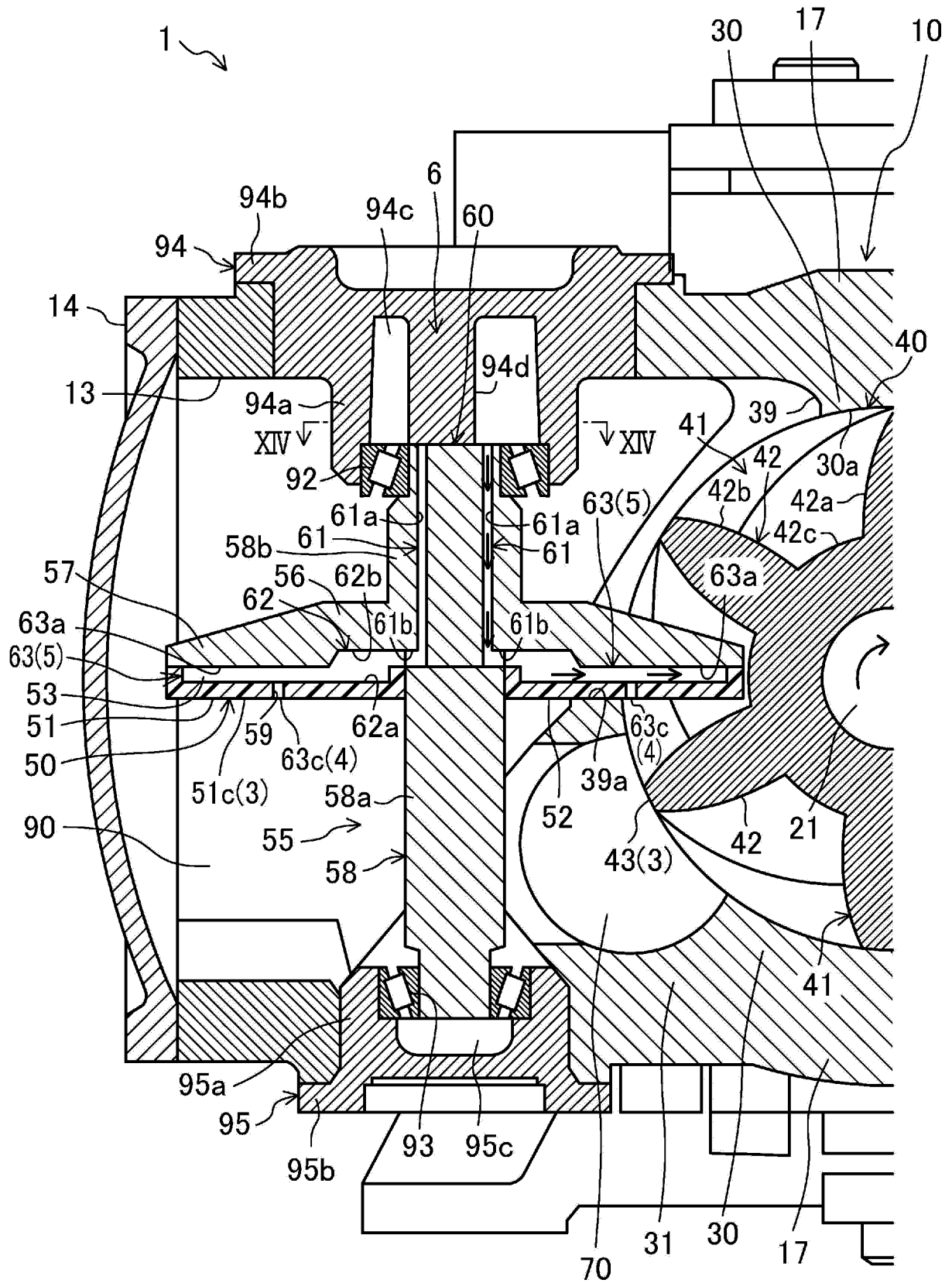


FIG.13

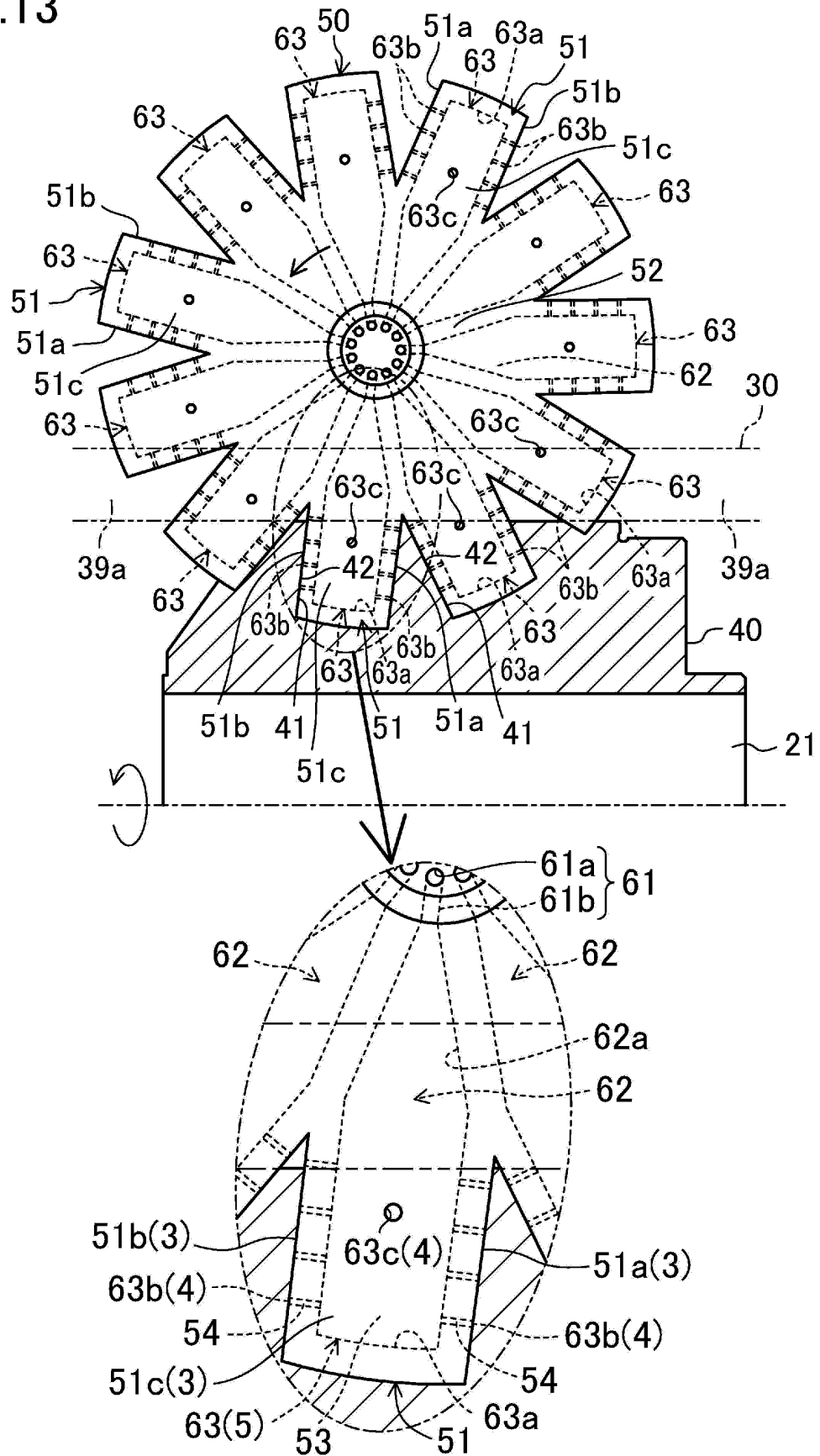


FIG.14

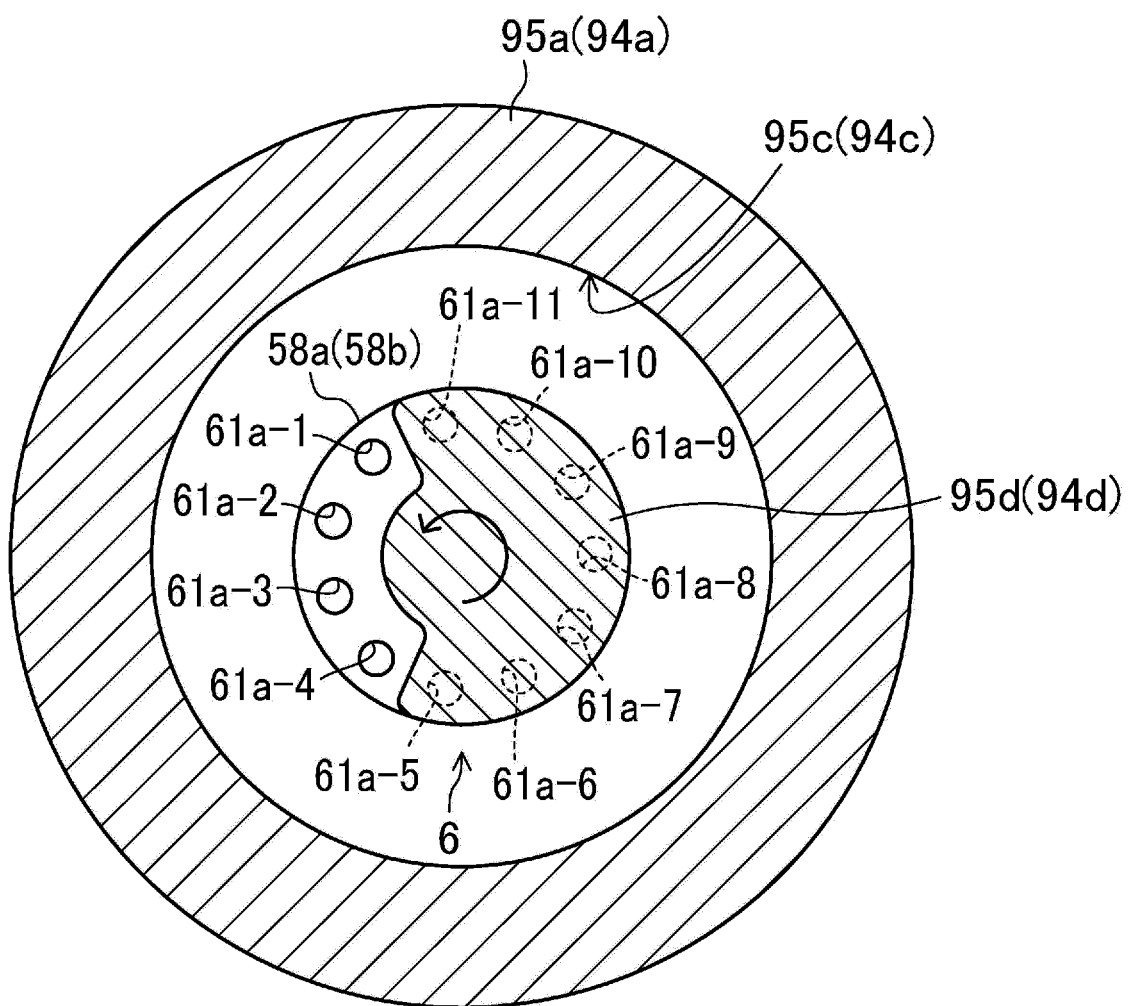
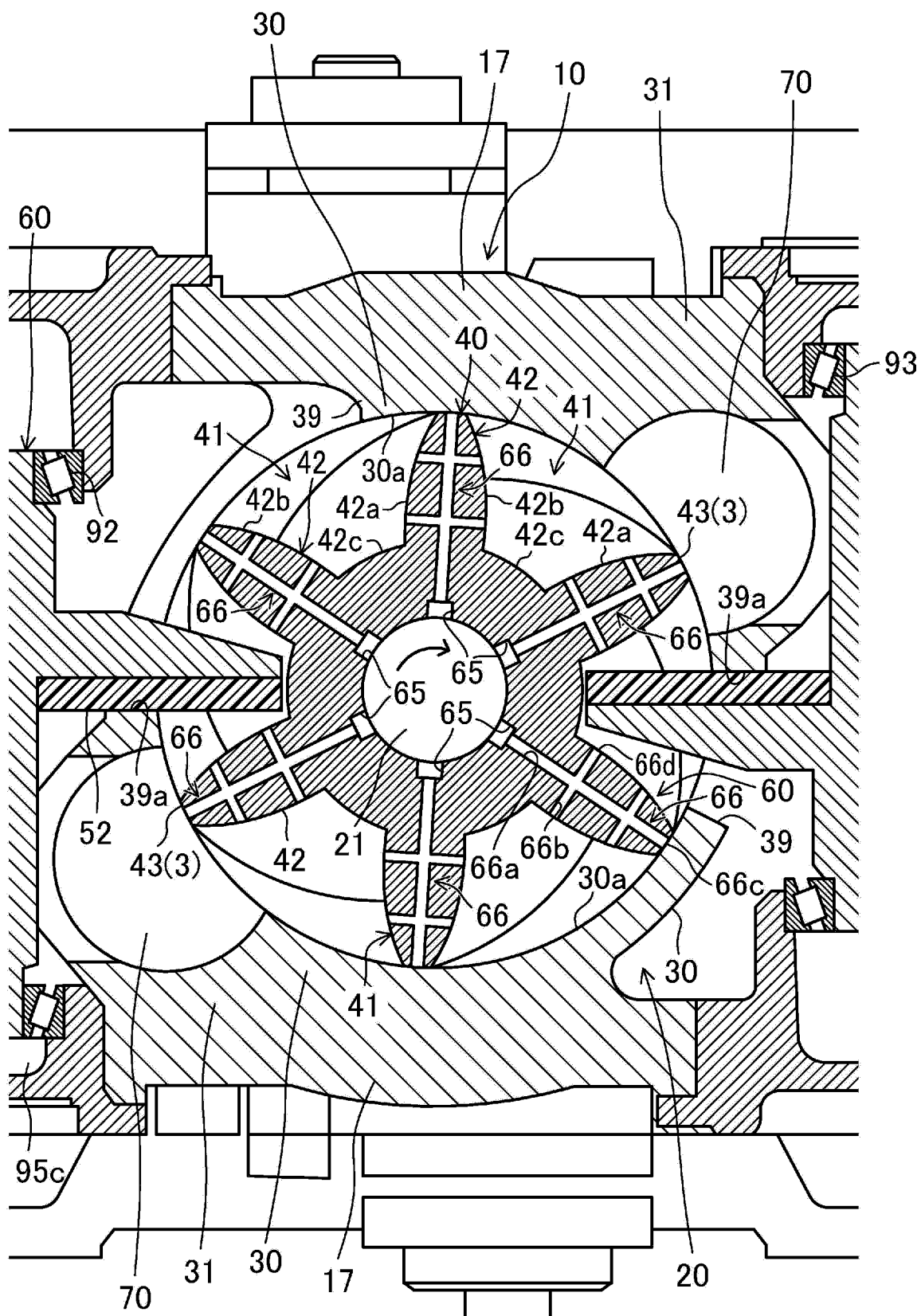


FIG.15



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2018/004747

A. CLASSIFICATION OF SUBJECT MATTER

Int.Cl. F04C18/52 (2006.01) i, F04C18/16 (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)

Int.Cl. F04C18/52, F04C18/16, F04C29/02

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

Published examined utility model applications of Japan 1922-1996

Published unexamined utility model applications of Japan 1971-2018

Registered utility model specifications of Japan 1996-2018

Published registered utility model applications of Japan 1994-2018

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A	JP 2017-15054 A (DAIKIN INDUSTRIES, LTD.) 19 January 2017, paragraphs [0024]-[0075], fig. 1-5 (Family: none)	1-2 3-11
X A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 151500/1979 (Laid-open No. 70186/1981) (HITACHI, LTD.) 10 June 1981, specification, page 4, line 11 to page 7, line 5, fig. 3, 4 (Family: none)	1-3, 9-10 4-8, 11



Further documents are listed in the continuation of Box C.



See patent family annex.

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"&" document member of the same patent family

Date of the actual completion of the international search
25.04.2018Date of mailing of the international search report
15.05.2018Name and mailing address of the ISA/
Japan Patent Office
3-4-3, Kasumigaseki, Chiyoda-ku,
Tokyo 100-8915, Japan

Authorized officer

Telephone No.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2018/004747

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 50-110109 A (HITACHI, LTD.) 29 August 1975, page 1, lower left column, line 4 to page 2, upper right column, line 12, fig. 3, 4 (Family: none)	1-11
A	JP 2009-174524 A (DAIKIN INDUSTRIES, LTD.) 06 August 2009, entire text, all drawings & US 2010/0278677 A1 & WO 2009/084218 A1 & EP 2236834 A1 & CN 101910640 A	1-11
A	JP 2-5778 A (ZIMMERN BERNARD) 10 January 1990, entire text, all drawings & US 4900239 A & FR 2624215 A1	1-11
A	JP 2008-127990 A (HITACHI IND EQUIPMENT SYS) 05 June 2008, entire text, all drawings (Family: none)	1-11
A	GB 923042 A (ZIMMERN, F.) 10 April 1963, entire text, all drawings & DE 1403614 A & FR 1268586 A	1-11

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2009197794 A [0005]